RECENT ADVANCES IN HYDRAULIC POWER DEVELOPMENT

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by

V.R.DAVIES, B.Sc.

ACKNOWLEDGMENT.

Water power engineering at the present day owes much to the liberal minded policy adopted a few years ago by water-wheel manufacturers and designers of publishing the results of their tests, and making known devices which have proved successful, through the medium of the engineering press. Under the stimulus of such a liberal policy a valuable literature upon the development of water power has come into existence, consisting in part of standard reference works, but which is principally to be found in the transactions of the various engineering societies and in the technical press. Upon this literature I have drawn freely for the material for the present discourse; and the only merit I would claim for this composition is, that it represents my impressions of the subject after a careful perusal of the works of many well known authorities upon water power development and allied branches of engineering. Attempts have been made to give credit by means of foot-notes to all sources from which ideas have been gleaned, and to accomplish this end more fully the list of literature at the end of the thesis has been appended.

I wish to express my appreciation of the valuable assistance rendered by the following water-wheel companies, all of whom have supplied valuable data or cuts: Canadian Allis Chalmers,Limited; The Dominion Engineering Company,Limited; The S.Morgan Smith Company; and the Pelton Water-Wheel Company.

Above all I wish to gratefully acknowledge the contributions of Professor Brown,professor of Hydraulics and Applied Mechanics, at McGill University, for helpful hints and suggestions; also for the reading of the manuscript and assistance in the elimination of errors.

Montreal,May 10, 1923.

V. R. Davies

CONTENTS.

PLAN AND SCOPE OF THE THESIS.

The aim throughout the present work has been to represent in the very limited space available the advancement in the art of water power development. It has been traced very briefly and imperfectly from the earliest efforts made by man at the utilization of water power down to the most recent achievements.

A limited amount of space is devoted to the impulse water-wheel, but the major portion is occupied by a discussion of wheels of the reaction type and their accessories. A small part of the underlying theory applying to the impulse wheel has been included;also that applying to the reaction wheel which is generally accepted and much that has been gleaned from various sources which is more or less conjectural and has not been fully tried out. It was felt that these theoretical discussions serve as nothing else can to assist the reader in appreciating the nature of the obstacles overcome and the problems which have been solved in the water power field. The evolution of the water-wheel and its settings is mainly dealt with and many of the achievements in allied branches of engineering which have contributed so much to the general advancement of the art of water power development have been touched upon but sparingly or left out entirely.

Of these, the contributions from the electrical and mechanical engineering are the most noteworthy. There are also many other achievements in the field of hydraulic engineering which have contributed their quota, an account of which could not be given in these pages due to lack of space. Among these may be included the improved designs of intakes, and the successful solution of penstock and surge tank problems, and the increases in the knowledge of hydraulic jump and the flow in open channels.

RECENT ADVANCES

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HISTORICAL SURVEY

In the course of an address entitled "What Canada owes to the Middle Ages" Professor Waugh* stated that the best way to understand and appreciate any great institution, as we have it to-day, is to inquire into its history. To the people of Canada, the history of the development of hydraulic power is a fit subject for such inquiry, for the development of our water power resources has become inseparably linked up with our development as a nation. It is not simply joined with our material progress, but interwoven. It is the warp and the woof of our social and industrial fabric. Certainly every engineer, and almost every other citizen of the Dominion can enumerate briefly the essentials of a hydraulic power scheme; that it consists of a supply of water, conducted through a pipe to a hydraulic turbine which drives a generator for the production of electrical energy.

How far we may make use of the water-falls of this country for the production of electrical power is a matter for adjustment in the public mind. The maximum amount of water which Canada and the United States are allowed to take from the falls of Niagara for power purposes is fixed by treaty between the two countries at comparatively small quantities, so that the scenic beauty of the falls may not be impaired. Who will say that such legislation is not to be highly commended? Mean indeed must be his soul who would advocate the obliteration of such an object of the nation's pride, the Mecca of all Canada. It is a matter for universal regret that the development of many of our water power resources has transformed many a picturesque locality into a barren and ugly waste;† but fortunately, with proper care this state of things can be greatly alleviated. I heard a celebrated artist quite recently deploring the destruction of Shawinigan Falls, and there are few of us who cannot readily sympathize with his feelings. He had visited them in his younger days, and had probably derived from them some of his finest inspirations, only to find, on revisiting the spot years later, that they had disappeared from the face of the landscape, so it seemed to him, forever. Unfortunately for us mortals and our aesthetic propensities, we are compelled by severe economic laws to balance our individual

* Dept. of History, McGill University.
† McCulloh, "Conservation of Water", page 98.

budgets as we go,which laws also apply to governments in the long run,and our various aspirations and desires are continually conflicting with one another. If we over-indulge our love of the beautiful,we are liable to suffer quite severely in a more material way. How far we can afford to sacrifice the objects of the national admiration for the sake of utility,I have not,so far,been able to very well make out,but I am persuaded that the two interests very frequently clash and we are called upon to take our choice of one or the other. There are times when we must exchange the freshness and beauty of the water-fall, exchange that music of the night-time that lulleth to repose, for the helping hand that it can lend us.

Ruskin,who had very strong feelings upon the subordination of art to the material needs of the hour, and who possessed to a remarkable degree that enviable faculty of being able to express how he felt,has expressed himself very forcibly on the marring of England's landscapes by the smoky colliery and factory: but Britain's independence as a nation,indeed,her very existance,has depended more than once on these unlovely objects. The fullest development of our water power resources is justified where the exigences of the case demand, where it is necessary for the welfare of the community, where the benefits derived exceed the losses sustained. The task of the engineer is to make these forces available,and while utilizing,to conserve.

Though the principal advances in the development of power from the energy of flowing water can be traced through comparatively recent historical times,yet water power was used by the ancients,and evidence is available to carry the origin of the utilization of water power back into the remotest antiquity. The knowledge of the science of hydraulics seems to have been quite scanty,but the construction of simple water-wheels was understood and carried out by the Babylonians,and the early Egyptians and Chinese for raising water to irrigate into fertility the scantily watered portions of their countries. In the first century of the Christian era,when the power and grandeur of Rome had reached the zenith, and Roman engineers were carrying out hydraulic engineering works which were never equalled until the dawn of the nineteenth century,even then the laws of the flow of water appear to have been very imperfectly understood. Frontinus attempted to determine the discharge from the aqueducts by computing the combined area of the outlets without taking into account the head. He concluded that the failure of his calculations to check was evidence of graft in the public service.*

The most substantial advances in the evolution of

* Clemens Herschel's translation of the work of Frontinus. Also Taylor & Moody "Hydraulic Turbine in Evolution" Hydro-Electric Conference, Philadelphia,1922.

the water turbine have been made during the nineteenth century, though it is far from correct to state that the discovery of the principles of its operation all belong to that period; Leonardo da Vinci (1452-1519) understood the principles of the formation of a free vortex; Bernouilli had investigated the laws of flow of water before 1750. Many water-wheels came into existance during the seventeenth and eighteenth centuries, which, though not of very great practical importance, involved the application of the principles upon which modern turbines operate.

Taking as our basis the method of deriving energy from flowing water, water-wheels may be divided into three main classes:-

(a) Gravity wheels, in which the weight of the water is the principal actuating force.
(b) Reaction or Pressure wheels, which operate by the pressure of the water due to the head.
(c) Impulse or Velocity wheels, which work by the action of a jet impinging on a series of moving vanes.

Like many classifications in nature, the lines of demarcation are not precise, and no wheel operates on the one principle to the total exclusion of the others. For instance, in the action of the gravity wheel (Fig.1) the effects of impulse are not entirely absent, and this is also true in the case of the reaction or pressure wheel.

Gravity wheels, though they have played a very important part during the last century in the driving of grist mills and small factories, and show very fair efficiencies, have ceased to occupy an important place in the development of modern water powers. This is principally on account of their small capacities, and are not capable of utilizing a head much in excess of their diameters, except by rather cumbersome adaptations, as shown in Fig.1, F and G. Probably one of the most notable gravity wheels in existance is the Laxy over-shot wheel (Fig.2) in the Isle of Man, which is still used for mine drainage.

One of the earliest employments of the reaction principle was in the tub wheel (roue à cuve, Fig.3) in use in France during the seventeenth century. It is of particular interest as being the ancestor of our modern mixed flow turbine. In 1804, Benjamin Tyler, of Lebanon, N.H., patented in the United States a wheel known as the "Wry Fly", a turbine very similar in design to the French tub wheel, but an improvement upon it. A reaction wheel, invented by Barker in 1740, consists of a number of hollow arms connected to a hollow vertical shaft, (Fig.4). It is a true reaction turbine. It is of considerable scientific interest, but it was not built in very large sizes and never attained very great practical importance.

The first highly efficient pressure turbine was, however, a development of Fourneyron (Fig.6) who installed his first turbine at Pont sur l'Ognon (Haute Saône) France, in 1827. It is an outward radial flow reaction turbine, receiving the water axially and discharging it radially outwards. The principal difficulties with this type of wheel, which have to a great extent prevented its general adoption, have been the instability of flow of water through the wheel passages and the difficulties of effective governing. There have, however, been many very notable installations of the Fourneyron turbine down to within a very short time ago. The most notable of these is perhaps the Trenton Falls Plant of the Utica Gas and Electric Company, (Fig.11) installed in 1901. Boyden introduced the Fourneyron turbine into America about 1844, and so improved the original wheel, both mechanically and hydraulically, that his first installation developed an efficiency of 78%. He attached the diffuser (Fig.5) which increased the efficiency of the turbine several per cent by converting the velocity of the outflowing water into effective pressure. The diffuser in its original form was not often used, as it was expensive and increased the size of the turbine.

Shortly after Fourneyron's turbine appeared, Jonval brought out his axial flow reaction turbine (Fig.7) which was an improvement of an earlier type known as the Fountaine turbine. It received and discharged the water in an axial direction and was supplied with guide vanes in a plane parallel to the plane of the runner. The Jonval turbine was equipped with a straight draft-tube and was well adapted to low heads.. It is of particular importance since it contains the germ of the idea which has given rise to the radial flow runner for low head service, as recently developed by Kaplan, Dubs, Moody and Nagler.

The most notable step forward in turbine development about this time was made by J.B.Francis in 1849, who, placing the guide vanes outside the wheel and reversing the direction of flow in the Fourneyron turbine, so as to discharge at the centre, obtained the inward radial flow turbine, which bears his name, (Figs.8,9 & 10). It did not completely oust the Fourneyron at once from its leading position, as we have already seen, but it began to take a leading part in water power development and has received the greater amount of attention from experimenters and inventors. Its advantages over the Fourneyron are fundamental:-

(a) The inlet ports and wheel passages are convergent, which admits of more steadyflow of water through the wheel.

(b) The guide vanes, being placed externally, are more readily accessible and can be built independent of the runner,making possible improvements in regulating devices.

(c) An increase in speed tends to automatically check the flow of water through the runner,due to an increase in centrifugal action,and so tends to assist governing.

(d) The centrifugal pressure of the water tends to balance a portion of the pressure due to the head on the turbine. This is a very important result and one which has assisted in the adaptation of the Francis turbine to very high heads. In practice the velocity of influx is reduced to about one half of the spouting velocity due to the head,the remainder of the head remains as pressure and is absorbed from the water on its passage through the wheel. By this reduction in the velocity of influx of the water the hydraulic losses become less,while the very material reduction in peripheral speed of the wheel allows it to be used for higher heads than other types of pressure turbine.

(e) The outflow velocities are more easily controlled,since there is no tendency to an acceleration of the velocity of the water by passing from the inner circumference outward,as in the case of the Fourneyron.

It was during the last half of the nineteenth century that several noted scientists gave a tremendous impetus to the advancement of science by the introduction of new methods of analysis and investigation, that of experimentation and the deductive system of reasoning. All science went forward under the stimulus of the new ideas. The other important agents of civilization long ago reached their full stature,and many of the finest products of human endeavour,like literature and the fine arts,have been through many centuries the common possession of the race. Even music,the most modern of the arts,is no longer young. But only in the last half century has science reached maturity and revealed its titanic influence in the reconstruction of the surroundings of our lives. The science of hydraulics shared in this general forward movement and the water turbine began to develop along more scientific lines. Possible lines of development were suggested by theoretical analysis and tried out experimentally,and the turbine quickly took on the form not differing widely in arrangement from the mammoth installations of to-day. Fig.12 will serve as an example of the turbine evolved during the fifties. It is one of the Geyelin-Jonval turbines built by the I.P.Morris Company and installed at the Fairmount Water Works,Philadelphia,in 1860. It embodies many of the outstanding features of the modern turbine; the single runner of simple design and unobstructed water passages,the vertical shaft arrangement and a draft-tube symmetrical about the turbine axis discharging radially at the floor of the tail-race. The speed regulation was furnished by a cylinder gate at the discharge from the draft-tube.

A period of relative inactivity succeeded the sixties, during which the turbine made little substantial progress in America. The aim of the majority of manufacturers of wheels appeared to be simply to make a wheel that would go, and little attention was given to the adoption of types to special requirements. A.T.Safford,* writing of this period, has said, "All the scientific teachings of Francis and Boyden were thrown to the winds, and the great god, Cut and Try, came into his own. If a wheel did not come up to expectations, its buckets were chipped back, up or down, or its blades pounded, until it gave something better. Such a period could hardly be avoided, since mathematical analysis and design of turbines were unknown to the majority of early wheel makers. The beginning of the testing system at Lowell, and later, at Holyoke, did much to relieve the situation. Before many years a manufacturer could not avoid a wheel test and sell wheels, with the result that a poor wheel was either improved or abandoned. For a long time a few makers managed to avoid public tests, but gradually they were forced to do so, and by 1890 most of the wheels on the market were more or less satisfactory. During this period, combinations of all kinds were tried and great ingenuity was shown, with the result that by 1873, reported efficiencies of 90% had been reached."

Unscientific as this period appears, it was not devoid of notable achievements. In 1875 Atkins patented his tangential wheel, and judging from the evidence of his application for patent, he had a very clear conception of the principles upon which tangential wheels operate. Hydraulically, the tangential wheel is quite simple and definite and has presented few difficulties, except in the development of a form for the wheel bucket. The most extensive developments of this type of water-wheel have been made by the Pelton Water Wheel Company, and so closely has this company been associated with its evolution that the words Pelton and tangential wheel are almost synonyms. The principal field of employment of the Pelton wheel is for situations having small quantities of water under very high heads, such as are found in the Rockies and in Norway and Switzerland. The Girard turbine was also a product of this period. The outstanding product of this period, was however, the American mixed flow turbine, which is the Francis turbine with its buckets extended so as to give a combination of radial and axial flow. It has not only almost replaced every other type of turbine for medium head service, but it is now invading the domain originally reserved for the Pelton. The most notable installation for high head service of the American mixed flow turbine, is at the Kern River No.3 Station of the Southern California Edison Company, operating under a head of 820 feet.†

* Trans.A.S.C.E.1922.

† Ely C.Hutchinson, Mechanical Engineering, April 1922
Also booklet issued by the Pelton Water Wheel Company.

The advent of electrical power opened up a new field of application of the hydraulic turbine,that of driving electrical generators. Electrical machinery,both motors and generators,are essentially of a high speed nature, since they operate more efficiently at high speed and provide power at a lower first cost. An increase in speed of a turbine also tends very materially to decrease the cost per horse power,both of the turbine and its settings. With these incessant demands for an increase in speed,together with an equally incessant demand for closer regulation of the speed,the hydraulic turbine has enjoyed a period of very active development during the last fifteen or twenty years. Coupled with the two demands as enumerated above,was the requirement of increased capacities,and this,in the vertical shaft turbine, presented a serious problem of supporting the increased weight of the revolving parts,for the thrust-bearing problem has only very recently been solved. In an effort to supply these demands,the vertical shaft arrangement was abandoned in favour of the horizontal,and then two, four,and finally as many as eight runners were placed on a single shaft. These arrangements involved mechanical complications and considerable hydraulic losses. The runners were commonly fed from a common penstock,with a branch leading off to each turbine,where considerable whirling and eddying took place and severe losses resulted. Perhaps the worst features of the earlier horizontal turbines was the proximity of the runners,where two were often made to discharge into a common draft-tube with insufficient distance between them to avoid serious interference. Closer speed regulation of the turbines was sought by the improvement of the gate mechanism. The early cylinder and register gates (Fig.9 & 10) were replaced by the more effective wicket gate (Fig.8),but this greatly increased the number of moving parts. The arrangements adopted were not always of the best,and delicate parts of mechanism were often placed in inaccessible places and even submerged.

A number of notable inventions have recently appeared which have released the turbine from its fetters. The principal of these is probably the solution of the thrust-bearing problem,allowing a return to the vertical arrangement. It was formerly necessary to keep the speeds low, in order to reduce the outflow velocities,which could not be easily regained. The improvement in draft-tubes,making it possible to recover a large percentage of the outflow velocities,has permitted great increases in speed. The improvement of surge tanks,eliminating all serious surges and variations in the pressure,and the improvement of governing mechanisms,have eliminated most of the difficulties of speed control.

The Pelton Wheel. The tangential wheel (Figs.27 & 30) has taken the foremost place among impulse wheels. It is a wheel that works entirely on the impulse principle; the water issuing from a nozzle at the full velocity corresponding to the net head and impinging on a set of buckets attached to the rim of the wheel. There are other types of impulse wheel,such as the Girard turbine (Fig.14) and certain modifications of the Girard which have given very praiseworthy service for small installations, but none have enjoyed the popularity of the tangential or Pelton wheel. Since Atkins' first patents were taken out for his wheel the names of many notable inventors have been associated with the tangential wheel,the principal of which are Knight,Moore,Hesse,Pelton,Hug,Dodd and Doble. Atkins had taken the hurdy-gurdy of the Pacific slope (Fig.13),used to a very great extent in the mining districts of California about the year 1865,and had replaced the flat vanes by hemispherical cups,which improvement,as we shall see in the development of the theory of the tangential wheel,doubled the theoretical efficiency. Tangential wheels have not presented any very great mathematical difficulties in the framing of a correct theory of operation,and since the time of Atkins the principal improvements effected have been in connection with the development of an efficient wheel bucket.

Pelton replaced the simple hemispherical bucket of Atkins by a bucket of rectangular form,having a divide to split the jet and cause discharge outward (Fig.15). This form of bucket resulted in a very material increase in efficiency,since it provided a better clearance of the water from the bucket,and greatly reduced eddying and shock losses. The general principle of Pelton has been retained,but the rectangular form has been superseded by the elliptically shaped bucket (Fig.16) invented by Doble. This form of bucket has now been universally adopted by makers as the absence of sharp corners and abrupt changes of direction of the stream favours the reduction of hydraulic losses.

The Girard Turbine. Between the Pelton or tangential wheel with one or two jets operating under a high head with a small quantity of water,and the Francis or pressure turbine with complete circumferential admission of a large quantity of water under a comparatively low head,we would expect to find a twilight zone at medium heads where partial admission becomes desirable. The Girard turbine was designed to occupy this intermediate position. As might be expected,it varies in construction from a single jet to complete circumferential admission,according as the conditions approach one extreme or the other; the turbine always operating on the impulse principle. It may be constructed as an axial or radial flow machine,and the governing is usually effected by means of a slide (S,Fig.14) which cuts off the inlet passages. The Girard turbine was at one time quite popular for small installations, especially in Europe, but had not been extensively employed on this continent. It is now almost entirely superseded by the Pelton wheel in small sizes, principally owing to the greater facility of governing the latter.

The Theory of Impulse Wheels. It will be interesting to investigate various cases of the correlation of momentum and energy changes of a jet impinging upon various types of vane. However, we need consider only two cases in connection with the operation of impulse wheels. In the first of these, where a jet impinges on a plane surface is involved the principle upon which the hurdy-gurdy operated. The second, where the jet impinges on a hemispherical cup, explains the action of the Pelton wheel and Girard turbine.

When a stream of fluid in steady motion impinges on a solid surface, it presses on the surface with a force equal and opposite to that by which the velocity and direction of motion of the fluid are changed. Generally, in problems on the impact of fluids, it is necessary to neglect the effect of friction between the fluid and the surface on which it moves. Consider a mass of fluid flowing in contact with a solid surface (Fig.17) also in motion, the motion of each being referred to the surface of the earth. The motion of the fluid, then, may be resolved into two parts, one a motion equal to that of the solid, and in the same direction, the other a motion relative to the solid. The motion which the fluid has in common with the solid surface cannot at all be influenced by the contact. The fluid flowing in contact with the surface can only have a relative motion parallel to the surface, while the pressure between the fluid and the solid, if friction is neglected, is normal to the surface. The pressure can therefore only deviate the fluid, without altering the magnitude of the relative velocity; therefore, the relative component of the motion of the fluid can only be altered in direction, but not in magnitude. From this we may infer that during impact, the velocity of the fluid relative to the surface on which it impinges remains unchanged in magnitude. This also depends upon the assumption that there are no shock losses, which requires that the jet be received upon the surface tangentially. The unchanged common component, and combined with it, the deviated relative component, give the resultant final velocity, which may differ greatly in magnitude and direction from the initial velocity.

From the principle of momentum, the impulse of any mass of fluid reaching the surface in any given time is equal to the change in momentum estimated in the same direction. The pressure between the fluid and the surface, in any direction, is equal to the change of momentum in that direction of so much fluid as reaches the surface in one second. If P_a is the pressure in any direction, m the mass of fluid impinging per second, V_a the change of velocity in the direction of P_a, due to impact.

$$\text{Then } P_a = m V_a$$

If V_1 (Fig.17) is the magnitude and direction of motion of a jet before impinging on the solid surface, moving with the velocity u in the direction shown, then the relative velocity is given by compounding these two, and equal to V_r in the direction of the tangent. Since the relative velocity of the fluid and surface is not changed by impact, the fluid will leave the surface with the velocity V_r and in the direction ed. Compounding the relative velocity of the fluid leaving the surface with the velocity of the surface gives the absolute velocity of the fluid equal to V_2 and in the direction ef. We see therefore that the motion of the fluid has been changed by impact from V_1 in the direction ab to V_2 in the direction ef. Combining V_1 in the direction ab and V_2 in the direction ef into the triangle abg, gives V as the total change of motion due to impact. The resultant pressure on the surface is in the direction of V and is equal to V multiplied by the mass impinging per second. That is, putting P for the resultant pressure,

$$P = m V$$

Let P be resolved into two components, N and T, normal and tangential to the direction of motion of the solid surface on which the fluid impinges. Then N is a lateral force producing a pressure on the supports of the solid. T is an effort which does work on the solid. If u is the velocity of the solid, Tu is the work done per second on the moving solid surface.

Let Q be the volume, w the weight of a cubic unit of the fluid, then Qw is the weight of fluid impinging per second; let V_1 be the initial velocity of the fluid before striking the solid surface. Then $Qwv_1^2/2g$ is the original kinetic energy of Q cubic units of the fluid, and the efficiency of the stream, considered as an arrangement for moving the surface is

$$\eta = \frac{Tu}{Qwv_1^2/2g}$$

This may be looked upon as the general statement of the problem. Two special cases arise in connection with the tangential wheels. The first, as in the case of the hurdy-gurdy, where the jet impinges on a flat plate; the second, in the case of the modern Pelton wheel, where the jet impinges on a curved surface. The two cases will be briefly presented and the maximum theoretical efficiencies determined in each case.

(a) Jet impinging on a Plane Surface. Let a jet whose section is A, and whose velocity v, impinge on a plane surface (Fig.18) moving in the same direction with a velocity u. The quantity impinging per second is A(v-u). The momentum of this quantity before impact is $\frac{wA}{g}(v-u)v$. After impact the water still possesses the velocity u in the direction of the jet; and the momentum in that direction of so much water as impinges in one second, after impact, is $\frac{wA}{g}(v-u)u$. The pressure on the plane, which is the change of momentum per second, is the difference of these two quantities,

$$\text{or } P = \frac{wA}{g}(v-u)v - \frac{wA}{g}(v-u)u$$

$$= \frac{wA}{g}(v-u)^2$$

The plane is moving in the same direction as the jet, and the work done per second is therefore

$$P\,u = \frac{wA}{g}(v-u)^2 u \text{ foot-pounds per second.}$$

There issue from the jet Av cubic feet per second, and the energy of this quantity before impact is $\frac{wA}{2g}v^3$ The efficiency of the jet is therefore,

$$\eta = \frac{2(v-u)^2 u}{v^3}$$

The value of u which makes this a maximum is found by differentiating and equating the differential coefficient to zero.

$$\frac{d\eta}{du} = 2(v^2-4vu+3u^2)/v^3 = 0.$$

$$\therefore u = v \quad \text{or} \quad u = \frac{1}{3}v.$$

The former gives a minimum, the latter a maximum efficiency. Putting $u = \frac{1}{3}v$, we have

$$\eta_{max.} = \frac{8}{27}$$

If instead of one plane moving before the jet, a series

of planes is introduced at short intervals at the same point, the quantity of water impinging on the series will be Av instead of A(v-u), and the whole pressure $= \frac{wA}{g}(v-u)v$.
The work done is $= \frac{wA}{g}(v-u)vu$.

$$\text{The efficiency } \eta = \frac{2u(v-u)}{v^2}$$

This becomes a maximum for $\frac{d\eta}{du} = 2(v-2u) = 0$,

$$\text{or } u = \tfrac{1}{2}v, \text{ and then } \eta = \tfrac{1}{2}.$$

Hence, we see that the maximum efficiency attainable with a tangential wheel with flat vanes, like the hurdy-gurdy, is 50%.

(b) Case of a jet impinging on a Concave Cup Vane. (Fig. 18). Let the velocity of the water be v and the velocity of the vane in the same direction u. The weight of water impinging per second is $\frac{wA}{g}(v-u)$. If, however, a series of vanes is introduced in the path of the jet at the same point, the quantity of water impinging per second will be $\frac{wA}{g}v$.
If the cup is hemispherical, the water leaves the cup in a direction parallel to the jet. Its velocity is (v-u) when approaching the cup, and -(v-u) when leaving it. Hence, its absolute velocity when leaving the cup is u - (v-u) = (2u-v). In the case of a series of vanes the change of momentum per second is equal to the pressure on the chain of cups.

$$\text{The pressure } P = \frac{wA}{g} v\{v-(2u-v)\}.$$

$$= 2\frac{wA}{g}(v-u)v.$$

Comparing this with the pressure on a flat vane, it is seen that the pressure on a hemispherical cup is double that on a flat plane.
The work done on the cups

$$= 2\frac{wA}{g}(v-u)vu \text{ foot pounds.}$$

$$\text{The energy of the jet } = \frac{wAv^3}{2g}$$

$$\text{The efficiency } \eta = \frac{4(v-u)u}{v^2}$$

The efficiency is greatest when $\frac{d\eta}{du} = 0$

$$\text{or} \quad \frac{4(v-2u)}{v^2} = 0$$

$$\text{and} \quad u = \tfrac{1}{2} v.$$

This gives the maximum efficiency as unity, or all the energy of the water is expended on the cups. A number of agents operate to reduce the efficiency considerably below the theoretical value. First, complete reversal of the jet cannot be effected and at the same time secure perfect clearance of the water from the moving parts. Second, friction between the jet and vane involves considerable loss, and windage reduces the velocities somewhat below the theoretical. It is found in practice that a value of u equal to 46 or 47 per cent of $\sqrt{2gH}$ gives a maximum efficiency.

Characteristics of the Pelton Wheel. The Pelton wheel is used under heads varying from about 500 feet to 2000 feet, although for small powers it can be used under medium heads of about 100 feet. On the other hand, the extreme high head utilized by the Pelton in a single stage is 5400 feet, at Fully, Switzerland, which plant will be briefly discussed at a later stage. In so far as the runner is concerned, it is the ideal type to use where the supply water is charged with sand or similar matter in suspension, as is not unusual in locations where it is installed. Its simplicity of construction and ease with which the buckets can be replaced when they become damaged or worn gives it a decided advantage over the Girard turbine. This advantage is regarded by some to be more apparent than real, since the buckets wear but slowly and the nozzles (are what) give most trouble in this respect, having to be frequently replaced.* The fact that the speed can be regulated by the deflection of the jet is of the first importance, and gives the Pelton its principal advantage.

The chief characteristic of the impulse wheel is the very great range of load variation during which the efficiency is nearly constant, and in this respect it stands out in marked contrast with reaction wheels, especially those designed to operate at comparatively high speeds under low heads (page 47). The curves from official tests of the Pelton wheels in four recent installations are shown in Figs. 19 and 20. In the first two the efficiency at half load is over 85 per cent, and only falls below 80 per cent when the load is below 30 per cent of the normal. In the third case, Fig. 20(a) the wheels appear to be slightly overrated, the efficiency being a maximum at about three quarters of full load; however, the efficiency is well over

* "Fully Hydro-Electric Power Station"
Engineering, Dec. 15, 1922, page 732.

80 per cent at 40 per cent load. In the last case, Fig.20 (b) the curve of the Fully Pelton wheels,the efficiency is not maintained at such high values during load changes. This unfavorable showing is due to the fact that the peripheral speed was designed for a value equal to from .517 to .550 of $\sqrt{2gH}$, depending on the head variation,in order to increase the security against racing or excessive run-away speeds.

Specific Speed of Water Wheels. The following discussion of the power and speed characteristics is general and applies equally well to impulse or pressure types of water turbines. The direct application of the results, however,might be better appreciated if it had been delayed until after the investigation of the dimensional relations of pressure wheels. The available power from a water-fall depends upon the head and quantity of water. Both of these are subject to very great variations,which call for correspondingly great variations in the design of water-wheels operating under different conditions of head and quantity. High heads are usually combined with small quantities of water,and the energy per pound of water is large; in the case of low heads,the conditions are reversed. For a given head and diameter of wheel, the power would vary with the quantity of water used,the revolutions per minute,and,of course,within certain narrow limits,with the efficiency of the wheel. Hence,if we had two wheels of the same diameter,operating under the same head at the same speed,we could compare them on a power basis; and then,compared with some standard,the power of each wheel might be considered its specific power. The power under such conditions would vary with the quantity of water used,and hence we would expect to find reaction wheels,with complete circumferential admission of water, having greater specific powers than tangential wheels, with one or two jets; and we would expect to find the Girard turbine,or partial admission wheel,occupying an intermediate position. This is actually the case. If the wheels are compared on a basis of speed for a given power,conditions of head and diameter the same,the resulting relations would give their specific speeds. At the beginning,the idea of specific power is perhaps more conveniently grasped,but the two quantities are interdependent and both are derived from the same dimensional relations of the runners. Daniel W.Mead* develops a quantity which he calls the "specific power",which is equivalent to the square of the specific speed as used in practice in the design of water-wheels or centrifugal pumps.

The fundamental laws of hydraulics connecting the head, power,and speed conditions of water-wheels may be stated as follows:-

(a) For different heads,a water-wheel runner will maintain the same characteristics of efficiency if the

* "Water Power Engineering",Daniel W.Mead.

speed is allowed to vary as the square root of the head, i.e. as the water velocities, which vary as $\sqrt{H}$.

(b) The power varies as the three-halves power of the head, since

$$H.P = \frac{QwH}{550} = \frac{wAvH}{550} = \frac{wA\sqrt{2gH}.H}{550}$$

(c) At constant head, the power of a runner varies as the quantity of water it receives, which in turn varies directly as the area of inlet passages; for runners of homologous design, the areas vary as the square of any dimension, and hence will vary as the square of the diameter.

(d) For a given head, the speed varies inversely as the diameter.

In the comparison of runners, the basic units are one foot of head and one horse power.

To compare two runners operating under different conditions of head on the "specific power" basis, compute their power and speed for the same head. This will leave varying power, speed and diameter. Next, recompute their powers on the basis of their being so changed in dimensions as to have the same speed.

Hence, the specific power of a runner may be defined as the power in H.P. of a model of the runner of such diameter that it will make one revolution per minute under one foot head.

Similarly, to compare two runners, operating under different heads, on the specific speed basis, compute as before, their power and speed for the same head. This would again leave varying power, speed and diameter; and the next step would be to recompute their speeds on the basis of the same power. The definition of specific speed may be given as follows:-

The specific speed of a runner is the speed in r.p.m. which a model of that runner would have, if operated under a head of 1 foot, this model to be reduced proportionally in all dimensions from the original until it will develop one horse power under one foot head.*

It is proposed to give a derivation of specific speed from the general laws as stated above. The following nomenclature is used:-

N = speed of the runner, r.p.m.

$H.P$ = brake horse power of the runner.

D = diameter.

H = effective head.

H_u = unit head = 1 ft. (as stated above)

$H.P_u$ = unit power = 1 H.P. (as stated above)

N_1 = speed of runner under unit head, designated unit speed.

* "A New Type of Hydraulic-Turbine Runner" Forrest Nagler, Trans.A.S.M.E., Dec.1919.

$H.P._1$ = power of runner under unit head, designated unit power.

D_u = diameter of homologous runner developing unit power under unit head.

N_s = specific speed.

Then applying the laws given on page 15, we have:-

from (a)

$$N_1 : N = \sqrt{H_u} : \sqrt{H} ; \sqrt{H_u} = 1$$

$$N_1 = \frac{N}{\sqrt{H}} \text{ - - - - - - - - - - - - - - (i)}$$

from (b)

$$H.P._1 : H.P. = H_u^{3/2} : H^{3/2} , \sqrt{H_u} = 1$$

$$\therefore H.P._1 = \frac{H.P}{H^{3/2}} \text{ - - - - - - - - - - - (ii)}$$

from (c)

$$H.P._1 : H.P._u = D^2 : D_u^2 , H.P._u = 1$$

$$D_u = \frac{D}{\sqrt{H.P._1}} \text{ - - - - - - - - - - (iii)}$$

from (d)

$$N_1 : N_s = \frac{D_u}{D} \text{ - - - - - - - - - - - - - (iv)}$$

Substituting equations (i) (ii) and (iii) in equation (iv),

$$\frac{\frac{N}{\sqrt{H}}}{N_s} = \frac{D_u}{D}$$

$$= \frac{D\sqrt{\frac{1}{H.P._1}}}{D}$$

$$\therefore N_s = N\frac{\sqrt{H.P}}{H^{5/4}}$$

It is seen that the speed in r.p.m. of a runner varies directly as the specific speed. It will be interesting to investigate how this quantity, the specific speed, arises out of a discussion of the dimensional relations of turbine runners, and this it is proposed to do for both impulse and reaction wheels. That for the impulse will follow after the tables below, giving the values of the specific speed of the runners of recent installation, and that for the reaction wheel will be given after the theory of the reaction wheel has been briefly discussed. The following tables are of interest as indicating the limitations of the various types of wheel in use to-day.

TABLE No.1 N_s for PROPELLOR TYPE RUNNER.

Power Plant	Horse Power	Speed	Head	N_s	Eff.
Great Falls, Man.	28,000	138.5	56	153	84%

TABLE No.2 N_s for FRANCIS TYPE RUNNERS.

Power Plant	Horse Power	Speed	Head	N_s	Eff.
Cedars Rapids	10,800	55.6	30	82.5	90%
Keokuk, Mississippi River	10,000	57.5	32	76	
Abitibi, Twin Falls	6,000	128.5	56.6	64.1	
Nipigon, ("Ontario Hydro.")	12,500	120	78	57.8	
Shawinigan Falls, (recent)	41,000	138.5	145	55.7	
Niagara (Amer.) Ext.No.3.	37,500	150	213.5	35.6	94
Queenston("Ontario Hydro.")	55,000	187.5	305	34.2	93
Kern River Plant No.3					
60 Cycle Runner	22,500	600	820	20.5	90
50 " " "	22,500	500	820	17.1	90

TABLE No.3 N_s for NEW IMPULSE TYPE of TURBINE*.

Power Plant	Horse Power	Speed	Head	N_s	Eff.
Afton, Calettwr River, United Kingdom	150	725	253.3	8.8	81%

TABLE No.4 N_s of RECENTLY INSTALLED PELTON UNITS.

Power Plant	Horse Power	Speed	Head	N_s	Eff.
Big Creek, Plant No.1(Cal.) Double Runner	20,000	375	1800(var.)	4.52	84%
Kinlochleven H.E Station Double jet	3,300	300	875	3.84	

* "Design and Performance of a New Impulse Water Wheel" Eric Crewdson. Proceedings I.C.E Vol.ccxiii.

TABLE No.4 - Continued.

Power Plant	Horse Power	Speed	Head	N_s	Eff.
Caribou Plant (Cal.) Double Runner	30,000	171.4	1008	3.70	
Fully Hydro-Electric Station, Single jet and Runner	3,000	500	5435	.587	81%

Specific Speed of a Pelton Wheel. The maximum value which the specific speed of a Pelton wheel can have may be determined as follows. Assuming that ~~for~~ a relation of 1/10 for the diameter of jet to pitch diameter of wheel gives and efficiency of 75%, we may work out the horse power for a given head on this basis. It is found in practice that a ratio greater than 1/10 for the relation of jet diameter to diameter of wheel cannot be used without considerable sacrifice of efficiency.

Taking $\eta = .75$, v = vel. of jet, D = diam. of wheel.
d = " " jet.

$$H.P. = \frac{.75\ QwH}{550}$$

$$= .085QH$$

$$N_s = \frac{\frac{60v}{2\pi D}\sqrt{H}\sqrt{.085Q}}{H^{5/4}}$$

$$= 55.8\ \frac{d}{D}$$

Placing $\frac{d}{D} = \frac{1}{10}$

$$N_s = 5.58$$

The maximum power of Pelton units is frequently controlled by the size of jet it is practicable to use. The Kinlochleven wheels have jets of 5.85 inches, and those of the Rjukan Hydro-Electric Station, Norway, are 6.45 inches in diameter. The jets of the recently installed Pelton wheels of the Caribou Plant, California, are 11 inches in diameter from 13 inch nozzle openings.

The Governing of Impulse Water Wheels. The needle nozzle (Fig.23) is now used exclusively with tangential wheels, and most of the governing arrangements provide for regulation by a movement of the needle together with some type of hood or jet deflector,both of which are operated by the oil or water pressure governor of the water wheel. Three systems of regulation have lately been extensively used, namely:-

*(a) By-pass regulation.

(b) Regulation with deflecting nozzle.

(c) Combined spear and deflector regulation.

The first method is not used alone where economy of the water supply is of importance. However,where an approximately constant amount of water must be delivered into the tail race,such as is the case of plants combined with irrigation projects,or where plants below are to be fed from the same operating water,it becomes necessary to adopt the water wasting or by-pass method of regulation The second method was formerly used quite extensively,but it had many disadvantages,the principal of which were the complicated relay mechanism necessary for moving a heavy section of pipe,the disturbance of alignment,and the difficulties of keeping tight a large flexible joint. It has been superseded by a rotating type of nozzle (Figs.21 & 22) which largely does away with these difficulties. The third is becoming the favourite method of regulation,and in large plants,or those having long penstocks,is usually combined with a by-pass arrangement,for wasting the water during the time the needle nozzle is allowed to close slowly,in order to avoid serious pressure surges.

The combined spear and deflector methods give very close regulation and has the further advantage of being simpler and cheaper than most other methods. It is used in one form on the Fully plant (Fig.27) but it has a number of variations (Fig.23),which will be briefly described. Three of the principal arrangements of this method are shown; a fourth,(Fig.24) has recently been brought out by means of which very close regulation is claimed. This last has been used with a small installation of a modified Girard† turbine,built by Gilbert Gilkes & Co., It is proposed to include a brief description of this turbine at a little later stage. In each case shown in Fig.23,when new demands are made upon the plant,the servo -motor of the oil pressure governor operates the spear and deflector simultaneously when opening,but when load is suddenly thrown off the turbine,the servo -motor at once actuates the deflector,which cuts into the jet and diverts the whole or a part of the water from the wheel,while the spear, slowly overcoming the resistance of the dash-pot,moves forward and reduces the quantity to the new requirements of the wheel. The deflector is then brought back to a position just tangential to the reduced jet. In each of the three designs some provision is made for allowing the deflector

*The following description of Pelton Wheel regulating devices is largely taken from a paper by Eric M.Bergstrom,Pros.I.M.E. 1920

†"Design and Performance of a New Impulse Water Wheel" Eric Crewdson, Pros.I.C.E. Vol.ccxiii.

a motion independent of the spear in the forward or closing movement,which is,in fact,the principal distinguishing feature.

The independent movement of the deflector is provided for in design "A" by means of a slot in the end of the spear rod. When a decrease in load occurs,the pin in the end of the lever connected to the piston rod of the governor moves away from the end of the slot,while the deflector is thrust into the jet. The spear,thus released,moves slowly forward,by the pressure of a spring overcoming the resistance of the oil dash-pot,until the end of the slot has regained contact with the pin,the deflector at the same time being drawn out of the jet. In the opening movement,both the spear and the deflector are drawn slowly backward together.

Design "B" is indentical with "A" in operation,except that the free movement is obtained through a displacement of the lever system relative to the point a. In the closing direction the lever bc will follow the deflector movement with b as a fulcrum and take up a position as indicated by the dotted lines on the diagram. The spring, being free to expand,overcomes the dash-pot resistance and moves the spear forward until the lever has again come in contact with the point a,in the position bc. In each of the designs "A" and "B", for both gradual and instantaneous decreases of load,the governor first acts directly upon the deflector,the spear meanwhile remaining stationery; the governor then adjusts the spear to the new load requirements,and brings the deflector out of the jet.

Design "A" is used for the governing of the 3,300 H.P. Pelton wheels of the British Aluminum Co. at Kinlochleven. For these wheels the governor was designed for a closing time of one second,corresponding to a maximum speed variation of approximately 8 per cent when the total load of 3,300 H.P. is instantaneously thrown off.

Better regulation is claimed for design "C",especially for small load changes,where the dash-pot is interposed between the governor servo-motor and the deflector lever. By sudden discharge of load,points a and b will move practically the same distance,consequently the deflector will cut into the jet,and at the same time lever a d will move away from the stop g. Operating similarly to the previous designs,the spring overcomes the dash-pot resistance slowly adjusting the spear to the new position by bringing the lever again into contact with the stop g, at the same time bringing the deflector out to a position tangential to the new jet. If,on the other hand,the load change takes place slowly,the lever will remain in contact with the point g, the governor acting directly on the spear, while the spring,overcoming the resistance of the dash-pot, keeps the deflector out of the jet. This latter feature, where the governor operates the spear direct for small and gradual load changes,gives this design a considerable advantage over the other two. It is this design which is used for regulating the Fully water wheels,and very satisfactory regulation of speed is secured, (page 23).

New Impulse Water Wheel (Modified Girard). The recent demands for closer regulation of the speed of water wheels for driving electrical machinery and the difficulties of efficient governing of the Girard turbine have operated to almost entirely eliminate it as a hydraulic prime mover. Formerly so popular for small installations in mountainous districts,the Girard has given up its place to the Pelton,which admits of closer speed regulation. If this defect can be remedied,there is no doubt it will again become popular for this service on account of its greater specific speed. The new water wheel recently constructed by Gilbert Gilkes & Co. and described by Eric Crewdson,(B.Sc. McGill) Proc.I.C.E.Vol.ccxiii,(Fig.25) appears to have overcome the above difficulty and is of particular interest on account of its having a specific speed intermediate between the highest obtainable with the Pelton and the lowest yet attained with Francis runners (Page 17). This wheel works on purely impulse principles,with free deviation of the jet in an axial direction,and is governed by the combined spear and jet deflection method of regulation. The main features of the installation are given in Table No.2, Page 17 and the characteristics of performance of the wheel are shown by the curves of Fig.26. It will be noted that the efficiency characteristics of the wheel are very similar to those of the Pelton,the efficiency remaining almost constant over a very large variation in load.

The governing mechanism is of recent development (Fig.24) and is well suited for small units. The governor,which operates the valve of the servo-motor,and which also operates the deflector direct,consists essentially of a powerful centrifugal pendulum,mounted on the turbine shaft. For the sake of clearness it is shown on a separate shaft in Fig.24,but it is actually mounted on the turbine shaft and is connected directly by means of a lever 10 to a shaft on which the deflector is mounted. When load is thrown off and the speed tends to rise,the deflector is at once cut into the jet and the spring 34 is compressed,which,slowly overcoming the resistance of the dash-pot,forces the puppet valve 16 against its seating,and closes the hole which extends the full length of the spear for relieving the pressure behind the piston 14. The pressure behind the piston 14 is set up by the full pressure of the penstock communicating through valve 32. When valve 16 is pressed against its seating the pressures on both sides of piston 14 are equal,but the water rushing past the spear head sets up a reaction on it,constantly endeavouring to draw it into the nozzle. While valve 16 is open a state of equilibrium is maintained by a relief of pressure through the hole running through the spear,but as soon as valve 16 is closed, pressure commences to accumulate behind piston 14 and slowly forces the spear forward and reduces the jet area to the new requirements of load; at the same time the deflector is restored to its former position,not tangential to the reduced jet,as in the other devices described,since for adjustment to a given speed,the position of lever 10 is fixed. When load is thrown on,the bell crank lever 20 is drawn away from the pippet valve and the pressure in the cylinder forces it outwards away from its seating. The pressure behind the piston is now relieved and the pressure in front forces the spear back,thus forming a larger jet.

The speed regulation is very efficient. With the whole load suddenly thrown off the generator,the speed rise observed was less than 1 per cent,very efficient regulation as compared with the regulation of the Kinlochleven wheels, or with the Fully wheels.(Fig.29).

Fully Hydro-Electric Station,* Switzerland. In point of size the Fully Hydro-Electric Station is not the most noteworthy installation of the Pelton wheel,but the extraordinarily high head 1645m (5400 ft.) which is utilized in a single stage called for very special skill in the design and construction of the plant,and make it representative of the most modern achievements. The main features of the wheels are listed in Table No.4,page 18 .Lake Fully,at an elevation of 2130 (about 7,000 ft.) was dammed,and together with Lake Sarniot,at a slightly lower level 1990,(6530 ft.) gives a total storage of 3,200,000 cu.metres (700 million gallons),equivalent to about 10 million kw.hours. A singular feature of this development,and one perhaps never before incorporated in a hydraulic power plant,is the provision,at the level of Lake Sarniot,of a pumping station which delivers the water from that lake into the common penstock against the head equal to the difference in level of 140 metres (460 ft.) between the surfaces of the two lakes. The pumping is performed by centrifugal pumps, electrically driven by power derived from the station. Although this arrangement involves some loss,it is preferable to constructing a seperate pipe line to take the water from the lower lake,and to providing special machinery to utilize the lower head.

The plant is provided with four units to deliver 3000 H.P. each when taking 200 litres (44 gals.) per minute at the normal head,or a total of 800 litres for the four units. At this flow the loss in the pipe line,in two sections,an upper 600 m.m.(23.6in) in diameter and 2278 m (7474 ft.) long,and a lower,500 m.m.(19.6) in diameter and 2347.5 m (7701 ft.) long,is 165 metres or about 10 per cent of the total head. On account of the great pressure,very special care was required in the construction of the lower section of the pipe. It was manufactured by the Ehrhardt process in seamless lengths of 3 m (9'-10") of which several were then welded together to form the pipe lengths and the pipe heads or flanges were then welded on also. Bolts and flanges of special manufacture were used to ensure safety at the joints and india-rubber gaskets were used to render the joints water-tight. The elevation of the jet nozzles is 497.33 metres.

* Engineering. Nov.24,and Dec.1 and 15, 1922.

The pitch diameter of the turbine wheels is 3550 m.m. (about 11'-7") (Figs.27 & 28),and the diameter of the jet 38 m.m (1.45"),corresponding to a ratio of 93.5; hence the very low specific speed of 2.7 on the metric system or .587 as given on page 18, on the foot-pound basis. The water velocity is approximately 180 m (590 ft.),varying somewhat due to variations in head,and the peripheral speed of the wheel disc on the pitch diameter is 93 m (305 ft.) per second. The usual practice of making ϕ = .46 or .47 in order to secure maximum efficiency was not followed in the design of these wheels. For the maximum head of 1645m, $\phi = \frac{u}{\sqrt{2gH}}$ = 0.517, and for the maximum head of 1454.2m, ϕ = 0.550. This provision was made in order to limit the racing speed to a value of 750 r.p.m. To have limited the racing speed to 850 r.p.m. would have resulted in a gain of 1 per cent in efficiency. The performance curves were previously given,(Fig.20b).

The turbine wheel,illustrated in Fig.28, acts as a fly-wheel and is a forged Siemens-Martin steel disc,weighing 7100 kg (about 7 tons). At racing speed the stresses in the steel do not exceed 10 kg.per sq.millimetre,(6.35 tons per sq.in.),this giving a factor of safety of from 4 to 5. The wheel carries 54 elliptically shaped buckets,for which, on account of the very high peripheral speed,a very special arrangement is provided for attaching them to the wheel. The runner is provided with a wedge shaped slot,and each bucket has two lugs of corresponding section (Fig.28). Around the periphery special openings were provided for inserting the buckets which are kept in place by 54 axial spacing wedges inserted between the buckets,in Fig.28 ,C. EE are the wedge shaped slots, and CC the transverse wedges. Special precautions had to be taken to prevent these wedges working loose and at the same time to be sure that the stresses set up in the buckets or wheel,by inserting them,were not in excess of those allowable for the materials. Of these wedges,nine of special form were first driven into the disc; they formed starting points for the ring of buckets. The others,of trapizoidal and constant section, were carefully fitted so as to produce a moderate initial tightening. The disc,without the buckets,was then very slowly heated in a specially built electric furnace to a temperature of about 130° C, this producing a considerable expansion of the peripheral length of the ring. The cold buckets were then inserted,together with the axial wedges,and it was possible to introduce between the rim of the disc and nine of the trapizoidal wedges chosen symmetrically around the disc a steel sheet shim of the required predetermined thickness to produce on shrinkage the necessary tightening up. The buckets cannot get free of the disc,even if the tightening wedges were to fail, and the latter eventuality was rendered impossible by heading the wedges over.

The governing of the wheel speed is carried out on the principles of design C,Fig.23,previously described. The action on the spear of the nozzle is very slow,40 seconds, in order not to cause dangerous pressure increases in the pipe line. The action on the deflector is rapid,2 seconds, which provides for very efficient regulation of the speed. The guarantees as to speed regulation were briefly as follows: If full load is instantaneously thrown off the whole plant,the excess pressure in the pipe line is to be less than 5 per cent. If 50 per cent of the load is instantaneously thrown off,the speed variations are not to exceed 3 per cent,nor 7 per cent when 100 per cent of the load is instantaneously thrown off. The operation of the governing mechanism is very interestingly portrayed by the diagram in Fig.29.

The Impulse Turbines of the Caribou,Cal.Plant.* The impulse turbines (Fig.30) of the Caribou Plant,California, are representatives of many notable developments which have recently been carried out in California,and they are the most powerful impulse turbines which up to the present have been built. The installation at present comprises two units each consisting of a 30,000 generator and two 15,000 overhung impulse wheels mounted on the same shaft,operating at 171.4 r.p.m. under a head of 1008 feet. The wheel disc is approximately 11 feet in diameter,7 inches in thickness, and is made of oil tempered forged steel of special high tensile characteristics. The wheel,without the shaft, weighs 25 tons and carries 21 cast steel buckets,each of 1000 lbs.weight and 36 inches in width. Each bucket is balanced as to weight and position of the centre of gravity and is secured to the wheel by means of three large bolts. The wheel is pressed on a cast steel hub, to which it is bolted by reamed bolts,machined to a fit of 50 tons each. The hub is in turn pressed on the shaft with a press fit of 200 tons and is keyed and locked in place.

The shaft,which weighs 26 tons,was forged from a single 50-ton billet. It is 30 feet long,30 inches maximum diameter,turned to $24\frac{3}{4}$ inches at the bearings,and is composed of compression forged open hearth steel,having a tensile strength of 75,000 pounds per square inch. It was given special heat treatment and was hollow bored to insure perfectness of metal. Two bearings,each carrying a load of 200,000 pounds under full load conditions, carry the shaft. They are of the self-aligned babbitted type with heavy oil rings and water cooled shells,and are equipped with force feed which may be used in emergency or put into continuous service if necessity demands. Their bearing length is 72 inches.

The nozzles are of the stationery type,needle controlled, and deliver to the wheel an 11 inch jet from a 13 inch opening. They are equipped with steel and bronze wearing rings and with renewable needle tip and nozzle throat rings,the parts which most quickly wear and most seriously affect the efficiency of the turbines. One nozzle only is provided for each wheel,which has its own needle control,oil pressure governor,pressure regulator and gate valve. The only point in common is that the fly-balls

*Engineering News-Record, March 23,1922.

of the governors are driven from the same shaft. The oil pressure governors are provided with relay for hand operation and remote electrical switchboard control, also with load limiting device to enable carrying constant load irrespective of the load demand, and safety stop which closes the nozzle in case of breakage of the belt driving the speed governor. The speed is regulated by the needle movement, without deflection of the jet.

The Francis Turbine. Formerly water power was developed in small units, in such quantities as could be used locally, and the turbines were usually direct connected to their loads, which consisted of a single piece of machinery or the line shaft of a small factory. The advent of electrical machinery, the generator and motor, and the improvements in transmission of electrical power, have made it possible to place ever increasing loads on the water-wheel. Along with this demand for greater capacities have gone equally insistant demands for greater speeds and closer speed regulation. The efforts to meet these demands were briefly reviewed in the Historical Survey; although several types of turbine were then known, none seemed so capable of expansion to meet the new requirements as the Francis turbine. The power of a wheel at any given head can be increased only by increasing its capacity for taking more water, and this capacity can be increased in one of two ways, either by expanding the wheel radially or axially. To increase the diameter involves a sacrifice of speed, and hence the second expedient has been resorted to in the development of the Francis turbine for low head service. Fig.31 is very interesting as showing the development of reaction wheels, and particularly the Francis. As was previously mentioned, the power of the turbines was often increased without sacrifice of speed by placing two or more runners on one shaft, in this way specific speeds as high as 80 were attained, as in the case of the West Kootenay Power and Light Company's plant, and the turbines of the Pennsylvania Water and Power Company. Referring to the list of plants in Table No.2, page 17, we might infer that the Francis turbine cannot be economically used for heads much below 50 or 60 feet, and in this field it is being rapidly replaced by the newly developed axial and diagonal runners, to be briefly discussed at a later stage, which give much greater specific speeds. Generally, it might be stated that the Francis runner will continue to be the favourite for heads ranging between 100 and 800 ft.

The Francis wheel is invading the domain originally reserved to the Pelton wheel, principally on account of its greater power and efficiency. For high head service it increases in diameter and decreases in its axial dimension, and takes on more and more the characteristics of the impulse wheel. The most notable installation of the Francis runner under a high head is at the Kern River No.3 Plant of the Southern California Edison Company, the main features of which are included in Table No.2, page 17. The efficiency curves are shown in Fig.42; they show high efficiencies at reduced load, a characteristic of impulse wheels. The runner is somewhat of the form in Fig.46-A. We may review briefly the underlying principles which decide the choice between the two types of wheel for high head service. For a fixed head and required capacity, the discharge Q (cu.ft. per sec.) of a turbine becomes fixed. This in turn fixes the total area of the passage ways of the runner. The speed of the turbine fixes the diameter, and (in) this fixes the axial dimension at entrance. For a given quantity, the individual passage areas decrease as the head increases, and hence,a Francis runner designed for such conditions will have narrow,tortuous water passages liable to become choked with litter or ice. Having once decided upon the minimum practical runner area,it at once becomes evident that the higher the head,the larger must be the capacity for a fixed speed,or the higher must be the speed for a fixed capacity. Thus,we approach conditions where the design of a Francis turbine becomes impractical,because of an uncommercially high speed for a given capacity,or because of two great capacity for a commercial speed. Designers not infrequently select the Francis runner,on account of its slight advantage in theoretical efficiency over the Pelton,for locations where the latter would give the more satisfactory service. Though the efficiency of the reaction wheel may be superior at the start,the clearances have to be so fine under such high heads and the water seals so carefully fitted to prevent leakage,that a small amount of wear or corrosion seriously affects the efficiency of the wheel,while the efficiency of impulse wheels is less affected by age,and is more easily restored by the replacement of parts which have become worn. In selecting turbines for the Caribou Plant,previously described,Francis turbines were seriously considered,even for so high a head as 1000 feet. On careful comparison of the advantages of the two wheels,of lower first cost,smaller floor space and higher efficiency offered by the reaction wheel against greater simplicity,smaller liability to wear and greater ease in renewing worn parts in the impulse wheel,the latter type was chosen.

The Theory of the Reaction Turbine. The earlier turbines,notably those of Fourneyron,Jonval,Howd and Francis, were designed so as to provide simple paths of flow of water through the runner,with simple designs of wheel vanes,which were easily subjected to rational calculation, the runners being of either the simple axial or the purely radial inward or outward flow type. In these earlier turbines,the guide vanes discharged the water directly into the runner,with little space between the fixed and revolving vanes in which the water could change in direction. As the demands for greater speed and capacity became increasingly more insistent,and the flow through the wheel took on more of the mixed character,it was found impossible and unnecessary to limit the space between the guide vanes and the wheel vanes to small values,and indeed, experience has demonstrated that it is advantageous to introduce a transition space in which the separate jets discharged from the guides could reunite and form a continuous stream before entering the runner. One result of this provision was the elimination of vibration or noise,due to the passing of the runner vanes through a succession of jets. As the specific speeds increased,the wheel vanes in the Francis runner were cut back still greater amounts

(Fig.31, G.H.), giving a much smaller radius to the runner at the point of entrance of the water than the radius to the ends of the guide vanes in order to avoid excessive velocities, and now a large transition space exists between the guides and the runner. The modern diagonal flow or propellor type of runner furnishes what might be considered the most extreme example of this process (Figs. 43, 50 & 51).

The development of the mixed flow runner made mathematical analysis of the conditions of flow through the runner, and the framing of a rational theory for the design of turbines increasingly more difficult and uncertain. The flow of water in stationary conduits or pipes is not perfectly understood, and when the water passages are in a state of rotation, many further difficulties are added to the problem. It is felt by designers that the principles as laid down in the standard works on hydraulics are not all that is necessary for the design of the turbines of to-day; however, a consideration of the momentum and energy changes of the water as it passes through the wheel cannot fail to furnish a clearer conception of the problem in hand, and hence it is proposed to state briefly the fundamental principles as applied to the inward radial flow turbine. During the last ten years outward flow turbines have received but little attention and outward flow wheels were becoming of little more than historical interest, but the recent developments of the diagonal runners (Fig. 32,) have introduced elements of the outward flow, and it may again become of considerable importance. The fundamental theory is valuable as a means of establishing the dimensional relations of the component parts of a turbine, and without attempting to develop the complete theory, a brief discussion of the momentum and energy changes of the water in its passage through the runner will be given. No special reference will be made to the conditions of exit from the guide vanes, it being assumed that they are so designed as to impart to the water at entrance a velocity in the most advantageous direction, that is with the relative component tangential to the wheel vane, in order to avoid losses by shock.

The following notation will be used in the analysis:-*

at entrance to the wheel,

b_1 = axial dimension of wheel, or breadth.
r_1 = radius of wheel to vane tips.
c_1 = absolute velocity of water.
w_1 = velocity relative to the wheel vanes.
u_1 = peripheral velocity of wheel ar radius r_1.
c_{m1} = radial component of absolute velocity.
c_{u1} = whirl " " " " which we may call the "absolute whirl".

* The notation of M. Lewis F. Moody as adapted from Dr. Camerer "Wasserkraftmachinen."

w_{u1} = whirl component of the relative velocity, which we may call the "relative whirl".

at exit from the wheel.

b_2 = axial dimension of wheel where water is discharged.

r_2 = radius at which water is discharged.

c_2 = absolute velocity of the water.

w_2 = velocity relative to wheel vanes.

u_2 = peripheral velocity of wheel at radius r_2.

c_{m2} = radial component of absolute velocity at exit, also considered as the velocity of flow $= \frac{Q}{A_2}$, A_2 = area at exit.

c_{u2} = whirl component of the absolute velocity at discharge, or "absolute whirl" at discharge.

w_{u2} = whirl component of the relative velocity at discharge, or "relative whirl".

As the water passes through the guide vanes, it has imparted to it a whirling motion, and in this whirling mass of water is placed the runner, which develops a torque by reducing, but usually not totally destroying this whirling component. As previously pointed out, conditions at entrance to the runner are indicated by the subscript 1, and those at exit by subscript 2.

Since it is assumed that the guide vanes are so designed that at the best or most advantageous gate opening, or that at which maximum efficiency is desired, the wheel vanes will receive the water without shock, that is, the direction of the relative velocity is tangential to the wheel vane at entrance, we have, from Fig. 33, :-

$$c_{m1} = c_{u1} \tan \alpha = (c_{u1} - u_1) \tan \beta .$$

$$\therefore u_1 = c_{u2}\left(1 - \frac{\tan \alpha}{\tan \beta}\right)$$

$$c_{m2} = (u_2 - c_{u2}) \tan \gamma \; ; \quad w_1 = c_{m1} \operatorname{cosec} \beta$$

$$w_2 = c_{m2} \operatorname{cosec} \gamma$$

At entrance to the runner the moment of momentum of the whirling mass of water per pound about the axis of rotation is $= \dfrac{c_{u1}\, r_1}{g}$ ft.lbs.

At exit from the runner, the residual moment of momentum is $= \frac{c_{u2}\ r_2}{g}$ ft.lbs.

The change of momentum = the turning moment per second $= \frac{c_{u1}\ r_1 - c_{u2}\ r_2}{g}$ ft.lbs.

∴ work done by this moment $= \left(\frac{c_{u1}\ r_1 - c_{u2}\ r_2}{g}\right)\omega$ ft.lbs.

where ω = angular velocity, radians per second.
from which $r_1 = u_1$; $r_2 = u_2$

∴ work done per second per pound of water $= \frac{c_{u1}\ u_1 - c_{u2}\ u_2}{g}$ ft.lbs.(i)

Evidently this is a maximum when $c_{u2} = o$ and then the work done per pound per second is

$$U = \frac{c_{u1}\ u_1}{g} \text{ ft.lbs.} \text{- - - - - - - - - - - - - -(ii)}$$

If c_{u2} is not zero, we have, on substituting its value $c_{u2} = u_2 - c_{m2}\ \cot\gamma$ in (i)

$$U = \frac{c_{u1}u_1 - u_2^2\ \ u_2\ c_{m2}\ \ \cot\gamma}{g} \text{ ft.lbs.per sec.}$$

When the wheel vanes are correctly proportioned, as pointed out above,

$$u_1 = c_{u1}\left(1 - \frac{\tan\alpha}{\tan\beta}\right) ;$$

with b and b_2 as the effective breadths of the wheel passages at entrance and exit, so that $2\pi b_1 r_1$ and $2\pi b_2 r_2$ are the effective passage areas, we have, since the flow is continuous,

$$c_{m2} = c_{m1}\frac{r_1\ b_1}{r_2\ b_2} \qquad \text{Also } \frac{u_2}{u_1} = \frac{r_2}{r_1}, \text{ so that}$$

from (i) $$U = \frac{1}{g}\left[c_{u1}^2\left(1-\frac{\tan\alpha}{\tan\beta}\right)\left\{1-\left(\frac{r_2}{r_1}\right)^2\left(1-\frac{\tan\alpha}{\tan\beta}+\frac{b_1}{b_2}\ \frac{\tan\alpha}{\tan\beta}\right\}\right] \text{(iii)}$$

If we assume the turbine is so designed that $c_{u2} = o$, it will simplify the expressions for u_1, c_{u1} and η involving the dimensions of the wheel. c_{u2} is approximately equal to zero in many turbines of to-day at the most advantageous gate opening, though less effort is not expended in attempts to reduce it to zero, or even to a very small magnitude, since the new designs of draft tubes are able to regain to a very considerable extent this component of the outflow velocity, making it possible to design turbines of greater specific speed.

With $c_{u2} = o$ equation (iii) becomes

$$U = \frac{c_{u1}^2}{g}\left(1 - \frac{\tan\alpha}{\tan\beta}\right)$$

Neglecting friction losses and changes in level of the water on its passage through the wheel, and assuming H = head available at the entrance to the wheel vanes,

H = head rejected from the turbine. + head used in work.

$$= \frac{c_2^2}{2g} + \frac{c_{u1}^2}{g}\left(1 - \frac{\tan\alpha}{\tan\beta}\right) \quad \text{- - - - - - - - -(iv)}$$

and since $c_{m2} = c_2$ when $c_{u2} = o$ (Fig. 33) we may replace c_2 by $c_{u1}\tan\alpha\cdot\frac{b_1 r_1}{b_2 r_2}$, in (iv) and we have

$$H = \frac{c_{u1}^2}{2g}\left(\tan\alpha\cdot\frac{b_1\ r_1}{b_2\ r_2}\right)^2 + \frac{c_{u1}^2}{g}\cdot\left(1 - \frac{\tan\alpha}{\tan\beta}\right)$$

$$= \frac{c_{u1}^2}{2g}\left\{2 + \left(\tan\alpha\cdot\frac{b_1\ r_1}{b_2\ r_2}\right)^2 - 2\,\frac{\tan\alpha}{\tan\beta}\right\}$$

which gives

$$c_{u1} = \sqrt{\frac{2gH}{2 + \left(\tan\alpha\cdot\frac{b_1 r_1}{b_2 r_2}\right)^2 - 2\,\frac{\tan\alpha}{\tan\beta}}}$$

and since $u_1 = c_{u1}\left(1 - \frac{\tan\alpha}{\tan\beta}\right)$

$$\therefore\ u_1 = \left(1 - \frac{\tan\alpha}{\tan\beta}\right)\sqrt{\frac{2gH}{2 + \tan\alpha\cdot\left(\frac{b_1\ r_1}{b_2\ r_2}\right)^2 - 2\,\frac{\tan\alpha}{\tan\beta}}}$$

$$= \phi\sqrt{2gH} \quad \text{- - - - - - - - - - - - - - - - - (v)}$$

where ϕ is the ratio of the velocity of the vane tip to the theoretical spouting velocity of the water under the head H. It is interesting to note how ϕ depends upon the relation of α and β, as indicated in the following table:-

	values of α				
β	10°	15°	20°	25°	30°
60°	.658	.636	.604	.564	.516
75°	.685	.669	.648	.625	.596
90°	.702	.695	.685	.672	.656
105°	.724	.729	.732	.733	.730

For an increase in α and β, ϕ increases, as shown by the diagonally stepped line across the table. The modern high speed runners have values of ϕ up to unity and even above; this tends to increase the friction losses, due to the greater relative velocities.

The efficiency $\eta = \dfrac{\text{work done per pound}}{\text{work available, per pound}}$

$= \dfrac{c_{u1}\, u_1}{gH}$, on the assumption of $c_{u2} = 0$ and no losses in the wheel.

$$= \frac{2\left(1-\frac{\tan\alpha}{\tan\beta}\right)}{2\left(1-\frac{\tan\alpha}{\tan\beta}\right)+\tan^2\alpha\cdot\frac{r_1^2\, b_1^2}{r_2^2\, b_2^2}}$$

$$= \frac{1}{1+\dfrac{\tan^2\alpha}{2\left(1-\frac{\tan\alpha}{\tan\beta}\right)}\cdot\dfrac{r_1^2\, b_1^2}{r_2^2\, b_2^2}} \qquad \text{-------(vi)}$$

For any designed values of α and β, for pure radial flow, an increase in the area of exit from the wheel, i.e. an increase in the factor $r_2 b_2$, would be accompanied by an increase in efficiency, but such increase is accompanied by divergent flow, which tends toward the increase of hydraulic losses, hence, for high pressure turbines, $r_2\, b_2$ is made about equal to $r_1\, b_1$ in practice.

With $r_2\, b_2 = r_1\, b_1$ and $\beta = 90^\circ$, the value of

$$\eta = \frac{1}{1+\frac{\tan^2\alpha}{2}}$$

A reduction of α effects an increase in the theoretical efficiency,since the whirling component of the water is thereby increased,but at the same time the radial component or velocity of flow is decreased,reducing the power of the turbine.

The conditions as outlined above are applicable only to radial flow,and would require modification for mixed flow. A satisfactory theory does not appear to have been developed for mixed flow so far,but the analysis of Mr.Lewis F. Moody in the solution of the problem of the determination of the maximum specific speed for a given efficiency,or for the maximum efficiency for a given specific speed,involving a comparison of the outflow and relative friction losses seems to throw considerable light upon the problem of turbine design. An account of Mr.Moody's method will be given below,taken mostly verbatim from his paper "The Present Trend of Turbine Development",given before the Engineers Club of Philadelphia,in 1921.

Conditions for Maximum Specific Speed and Efficiency in Reaction Turbines. In the previous discussion of the general theory of the reaction turbine,considerable stress was laid on the importance of keeping the whirl component of the absolute velocity of outflow as small as possible, in order to prevent rejection of energy from the runner in the form of kinetic head,or to prevent undue disturbance in the draft tube. With the new designs of draft tubes, a discussion of which will be given later,it is possible to regain not only a large proportion of the meridian or radial components of the outflow velocity,but the whirl component as well. With improved facilities for regaining these velocities,it is possible to increase them,and hence to increase the rate of flow through the runner,with a corresponding increase in the power of a runner of given diameter,resulting in a greater specific speed. Mr.Moody puts the proposition as follows:-

"With the use of the earlier forms of draft tube any direction of discharge having a whirl component introduced uncertainties and complications,and the problem was not attractive. With the use of a draft tube capable of efficiently regaining the energy of whirl components as well as meridian components of flow,however,it became possible to formulate some simple relations which it will be interesting to investigate. For one thing,we shall no longer have to make the "outflow loss" from the runner dependent upon the direction of discharge,since we can use a draft tube which will handle the whirl components just as efficiently as the meridian components;the"outflow loss" from the runner can therefore be expressed as a function merely of the amount of discharge velocity regardless of its direction. If the draft tube efficiency is e_d,and the coefficient of loss in the draft tube is f_3, the portion of the velocity head of the water discharged from the runner which is lost or dissipated will be

$f_3 \dfrac{c_2^2}{2g}$; c_2 being the absolute velocity of discharge from the runner, and

$$f_3 = 1 - e_d .$$

In runners of high specific speed another important loss is that due to the frictional resistance of the runner vanes, since very high relative velocities between the vanes and water are employed. In recent forms of high speed turbines, (reference to cuts), the rotational speed is high and the torque correspondingly low, so that the runner vanes have but little curvature and deflect the water only small amounts. The loss of head due to the frictional resistance of the runner vanes can be expressed as,

$f_2 \frac{w_2^2}{2g}$, in which w_2 is the relative velocity with which the vanes move through the water, measured at their outflow edges, (but for the reason just mentioned, the relative velocity is nearly the same over the whole vane).

Instead of making any arbitrary assumption regarding the direction of discharge upon which to base our turbine design, it will be useful to find the conditions which will give the minimum value for the sum of the above two losses, - the outflow loss from the runner and the resistance loss in the runner. The problem may be stated as the determination of the conditions for maximum efficiency for a given specific speed, or the maximum specific speed for a given efficiency".

Before proceeding with the problem, the author of the paper makes a few comments upon the two forms in which specific speed is often expressed, i.e., the form used in turbine design,

$N_s = N \frac{\sqrt{H.P.}}{H^{5/4}}$ and the form used in the design of centrifugal pumps, $N_s = N \frac{\sqrt{Q}}{H^{3/4}}$

The two forms are interchangeable, since both are derived from the same dimensional relations; the first is more convenient for use in turbine design, since the output and head are first decided upon, but the second is more useful in pump design, where the head and quantity are first determined. The second is derived from the first by substituting $\frac{QwH}{550}\eta$ for H.P. where η is the efficiency of the runner, and the author distinguishes between the two by designating the second N_{sg}; he then continues.-

"Although in adapting turbines of types already developed to new requirements of head and power the first expression is more convenient, the second has advantages when the problem is the development of a new design or type, since by its use we may design for a desired discharge under a definite head, without having to assume an efficiency in advance. Moreover, if the efficiency should turn out materially different from that expected, the discharge and velocities will not be thrown so far out of agreement with the designed values if taken for a given N_{sg} as they would be if taken for a given N_s.

In the problem in hand we shall gain simplicity by employing N_{sg}, and if we determine the conditions for maximum N_{sg} for a given efficiency we shall at the same time have found those for the maximum N_s, since

$$N_s = \sqrt{\frac{62.4}{550}}\; N_{sg}\sqrt{\eta}$$

and the conditions which will give the best efficiency for a given N_{sg} will also give the best efficiency for a given N_s.

The notation as used in the author's paper was used in the derivation of the general theory and is given on page 28. All the quantities in the following analysis have the same significance as given there, but in addition to indicating the radial component of the outflow velocity c_{m2} may be taken in a more general way to indicate the meridian component of the velocity of discharge.

The development of the relations then continue with the following.-

"Since we are not concerned with any particular value of the head, we can adopt a "specific value" for each velocity, using the device of Thomann*, which is convenient in problems of this kind, and call

$$C_2 = \frac{c_2}{\sqrt{2gH}};\quad W_2 = \frac{w_2}{\sqrt{2gH}};\quad U_2 = \frac{u_2}{\sqrt{2gH}};\ \text{etc.}$$

and since we shall deal throughout with the outflow triangle, we can omit the subscript (2)

The form of the triangle can easily be related to the specific speed by putting $U_2 = \dfrac{\pi D_2 N}{60\sqrt{2gH}}$; and

$C_{m2} = \dfrac{Q}{A\sqrt{2gH}}$ in which A is the area of the runner discharge measured normally to C_{m2}.

Then
$$N_{sg} = \frac{60\sqrt{2gH}}{\pi D_2}\, U_2 \sqrt{\frac{A\; C_{m2}\sqrt{2gH}}{H^{\frac{3}{2}}}}$$

$$= \frac{60(2g)^{\frac{3}{4}}\sqrt{A}}{\pi D_2}\cdot U_2\sqrt{C_{m2}}\,, \quad - - - - (1)$$

*R.Thomann "Die Wasser turbinen" 1908, pp.11-12.

or dropping subscripts(2)

$$k_{sg}\, N_{sg} = U\sqrt{C_m}$$

in which $k_{sg} = \dfrac{\pi D_2}{60(2g)^{3/4}\sqrt{A}}$

Before considering the problem with reference to the amounts of the vane loss and the outflow loss, we can reach an interesting conclusion simply by supposing that the magnitudes of the absolute and relative velocities are kept constant while their directions are changed until the most advantageous shape of outflow triangle is attained. As shown in Fig. 34(a) various shapes of outflow triangle can be drawn with the same values of C and W, and therefore the same losses of head. There will be one of these triangles which will give the highest specific speed, which we have just seen is proportional to $U\sqrt{Cm}$. To find the relation which will thus give the maximum N_{sg} for a given loss, we can put

$$k_{sg}.N_{sg} = U\sqrt{Cm}$$

$$= (\sqrt{C^2 - Cm^2} + \sqrt{W^2 - Cm^2})\sqrt{Cm} \quad \text{---(ii)}$$

and considering C and W constant, we can differentiate ($k_{sg}\ N_{sg}$) with respect to Cm, and equate to zero:

(The author's steps in the development are omitted).

From this we have, simplifying,

$$\sqrt{C^2-Cm^2} + \sqrt{W^2-Cm^2} = 2Cm^2\left(\frac{1}{\sqrt{C^2-Cm^2}} + \frac{1}{\sqrt{W^2-Cm^2}}\right)$$

and substituting Cu for $\sqrt{C^2-Cm^2}$ and Wu for $\sqrt{W^2-Cm^2}$

$$Cu + Wu = 2Cm^2\left(\frac{1}{Cu} + \frac{1}{Wu}\right)$$

$$\text{or } U = 2Cm^2\left(\frac{Wu + Cu}{Cu\ Wu}\right)$$

$$= 2\,\frac{Cm^2\ U}{Cu\ Wu}.$$

Which gives $Cm = \sqrt{\dfrac{Cu\ Wu}{2}}$ (iii) as the most advantageous proportions. That is, the meridian velocity should be chosen as 0.707 multiplied by the mean proportional of the absolute and relative whirl. In calculations applying to the runner as a whole, the velocities corresponding to an average point in the runner may be used, such as the centre of area of a sector of the discharge space*

* To enable us to visualize the meaning of this relation, the diagram, Fig. 35 is shown. The vertex of the outflow

" Now proceeding with the problem and considering the amounts of the losses of head,we can derive some useful relations,as follows:-

From the above expression for specific speed,we see that we can keep the length of the base U of the outflow triangle and its altitude Cm constant,and can change its shape without changing the specific speed. By shifting the vertex parallel to the base,there must be some position which will make the sum of the outflow loss and vane resistance loss a minimum.

Calling the sum of these losses H_L , the loss of head expressed as a fraction of the effective head is

$$h_L = \frac{H_L}{H} = f_2 \frac{w_2^2}{2gH} + f_3 \frac{c_2^2}{2gH} = f_2 W^2 + f_3 C^2 \text{---(iv)}$$

Expressing h_L in terms of U, C_m and C_u

$$h_L = f_2 W_u^2 + f_3 C_u^2 + (f_2 + f_3) C_m^2$$

$$\left[\text{But} \quad W_u = U - C_u\right]$$

$$= f_2 U^2 + f_2 C_u^2 - 2f_2 UC_u + f_3 C_u^2 + (f_2 + f_3) C_m^2$$

$$= f_2 U^2 + (f_2 + f_3) C_u^2 - 2f_2 U C_u + (f_2 + f_3) C_m^2 \text{---(v)}$$

For a given U and C_m we can find the value of C_u which will make h_L a minimum,neglecting the effect of any small variation in f_2 due to a change in direction of W as being of a higher order of small quantities than differences in the losses themselves.

Then,

$$\frac{dh_L}{dC_u} = 2(f_2 + f_3) C_u - 2f_2 U = 0;$$

---

triangle should fall somewhere on the ellipse of major axis U and of minor axis $= 0.707\sqrt{\frac{U}{2} \times \frac{U}{2}} = 0.354$ U,

For example,if a runner has a vane inclination β_2 its probable point of best operation can be found by drawing W at this angle with the base line and completing the triangle with the vertex at the point where the W line intersects the ellipse. According to this diagram,it would evidently never pay,from the standpoint of efficiency alone, to use a meridian velocity greater than 35.4 per cent of the circumferential velocity of the runner. Since the meridian velocity may be somewhat increased,however,before the efficiency is seriously impaired,it will probably be economical to use slightly higher values of C_m than are called for by this relation,in order to reduce the turbine dimensions and cost."

That is,

$$C_u = \frac{f_2}{f_2 + f_3} U \ ; \quad \text{or } f_2 \ (U - C_u) = f_3 C_u$$

From which

$$f_2 \ W_u = f_3 C_u \text{ or } \frac{C_u}{W_u} = \frac{f_2}{f_3} \quad \text{- - - - - - - (vi)}$$

This means that for the best efficiency the absolute whirl should be to the relative whirl as the coefficient of frictional loss to the coefficient of outflow loss.

The above result shows for one thing that in a runner for which f_2 is large, as is the case in runners having a large amount of vane surface-particularly when this surface is increased by an outer band- the best condition of operation will be with comparatively low relative whirl and high absolute whirl, so that such runners will discharge the water at a greater angle to the meridian plane than runners having less vane surface, and correspondingly lower f_2. The numerical value of f_2 can be calculated by figuring the loss of head in the runner buckets considered as rectangular channels. The purpose of this investigation is not so much to take up questions of design and calculation, however, as to establish some of the controlling relations, and we will give less attention to the numerical values of the equations than to the general conclusions which may result."

Upon the assumption that the above condition is to be complied with, the author then proceeds to find the best relation between the meridian component C_m and the velocity of the runner U.

Writing down equation (v) and expressing C_u in terms of U according to the results obtained in the previous determination, namely,

$$C_u = \frac{f_2}{f_2 + f_3} U$$

and substituting for U its value according to the relation

$$U = \frac{k_{sg} \ N_{sg}}{\sqrt{C_m}}$$

he obtains

$$h_L = \frac{f_2 f_3}{f_2 f_3} \cdot \frac{k_{sg}^2 \ N_{sg}^2}{C_m} + (f_2 + f_3) \ C_m^2 \quad \text{- - - - - (vii)}$$

To obtain the conditions for a minimum h_L for a given N_{sg} he then differentiates the above equation with respect to C_m considering N_{sg} a constant, equating the first derivative to zero, replacing U by its value $\frac{k_{sg} N_{sg}}{\sqrt{C_m}}$ and simplifying, he obtains,

$$\frac{C_m}{U} = \frac{\sqrt{\frac{1}{2}(f_2 f_3)}}{(f_2 + f_3)} \text{ - - - - - - - - - - - - (viii)}$$

The same result may be obtained, as the author points out, from equation (iii),

$$C_m = \sqrt{\frac{C_u W_u}{2}} \quad \text{by inserting}$$

$$C_u = \frac{f_2}{f_2 + f_3} U \quad \text{and } W_u = \frac{f_3}{f_2 + f_3} U ,$$

From here, continuing, he states,

"For a minimum loss for a given specific speed, the various outflow velocities from the runner should therefore be related to each other in the proportions shown by the following diagram (Fig 36(b) which is geometrically similar to the outflow diagram as indicated (Fig. 36 (a).

That is, C_u W_u, C_m, and U should be to each other respectively, as f_2, f_3, $\sqrt{\frac{f_2 f_3}{2}}$ and $(f_2 + f_3)$.

From the above triangle it is seen that

$$\tan \alpha_2 = \sqrt{\frac{1}{2} \frac{f_3}{f_2}}, \text{ and } \tan \beta_2 = \sqrt{\frac{1}{2} \frac{f_2}{f_3}}$$

In order to use advantageously small values of β_2, to secure runners of very high specific speed, it therefore becomes necessary to obtain low values of f_2, and thereby to avoid the necessity for high angles of whirl at the runner discharge. If this reduction of vane area is carried too far, however, another source of loss will be introduced due to the permitting of the passage of a considerable portion of the flow between the vanes without its acting upon them."

The above extracts from Mr. Moody's paper have been given in full, because it is felt that they present, better than anything that has yet come to the writer's attention, the underlying principles in the design of high speed runners, not only of the new propellor or diagonal type, but also of Francis runners. In the discussion which it is proposed to give later of high speed runners, the manner in which some of the above principles have been complied with will be pointed out. Regarding the future development of the turbine, the author has to say:

"In looking forward to the future development of the turbine, it can be seen from the foregoing that theoretic considerations call for a conservative proportioning of the various velocities and angles of the turbine, and there is no apparent means of increasing specific speeds merely by using some new and radical proportion between the velocities. Efficiencies are already so high that no startling increase is possible. The prospect of further increase in specific speed therefore lies entirely with the use of higher velocity heads in comparison with the head on the plant, and an avoidance of serious impairment of the efficiency by giving attention to the securing of low coefficients of loss in the runner and draft tube, and by observing the relations between the velocities deduced above.

To get some idea of the possibilities of further increases in specific speed, it may be pointed out that if the best relation between the velocities is adhered to the losses here considered will vary as the four thirds power of the specific speed N_{sg} the functional relation between efficiency and N_{sg} being derived below."

The author takes the equation

$$h_L = \frac{f_2 \quad f_3}{f_2 + f_3} U + (f_2 + f_3) C_m^2$$

and substitutes for U in terms of C_m according to equation (viii) page 39, from which he obtains

$$h_L = 3(f_2 + f_3) C_m^2 \quad \text{- - - - - - - - -} \quad (ix)$$

and from $k_{sg} N_{sg}$

$$= U \sqrt{C_m}$$

$$= \frac{f_2 + f_3}{\sqrt{\dfrac{f_2 \quad f_3}{2}}} C_m^{\frac{3}{2}}$$

$$h_L = \left(\frac{3}{2}\right)^{\frac{2}{3}} \frac{(f_2 f_3)^{\frac{3}{2}}}{(f_2 + f_3)^{\frac{1}{2}}} k_{sg}^{\frac{4}{3}} N_{sg}^{\frac{4}{3}} \quad \text{- - - - - -} \quad (x)$$

and efficiency

$$\eta = 1 - h_L \quad \text{- - - - - - - - - - -} \quad (xi)$$

This considers, as stated by the author, the effect of only the two losses here treated of, and neglects certain other losses. These values of efficiency are therefore merely indications of the limiting values attainable. The two losses mentioned above account for nearly the whole loss in turbines of high specific speed, but for low speed turbines two other losses assume importance; those due to leakage and disc friction. In the paper by Mr. Moody, expressions are given for these losses as fractions of the total quantity and horse power of the turbine. The expression for the leakage loss is interesting, as he makes it inversely proportional to the square of the specific speed N_{sg}; the expression for disc friction loss is developed along the lines similar to those given in the standard works on hydraulics,but by expressing it as a fraction of the total horse power of the turbine,it is conveniently made to vary inversely as the specific speed N_s. The leakage loss is given as:

$$L_L = \frac{201\phi^2}{N_{sg}^2},$$ ϕ having the usual significance.

For present purposes it is sufficiently accurate to neglect the variation of ϕ; and calling $\phi = 0.7$,

$$L_L = \frac{100}{N_{sg}^2} \text{ (approximately) - - - - -(xii)}$$

The disc friction loss is commonly taken as varying as the 5^{th} power of the diameter and the 3^{rd} power of the speed rotation;and hence the loss may be expressed as

$h.p. = KD^5 N^3$,* h.p. = horse power loss due to disc friction. $K = 0.000,000,000,413$;

Expressed as a fraction,the disc friction loss

$$L_D = \frac{h.p.}{H.P.}; \text{ but } H.P. = \frac{N_s^2 H^{\frac{5}{2}}}{N^2}$$

$$\therefore \quad L_D = K \left(\frac{60\sqrt{2g}}{\pi}\right)^5 \frac{\phi^5}{N_s^2}$$

$$= \frac{35\,\phi}{N_s^2} \text{ - - - - - - - - - - - - - - -(xiii)}$$

On account of the high power of ϕ its variation should be considered; but for high speed runners the crown does not continue to increase in diameter and in recent developments the band or shroud is omitted,so that for a close enough approximation we can call ϕ = a constant, say 0.7 and

$$L_D = \frac{6}{N_s^2} \text{ (approximately) - - - - - (xiv)}$$

The Specific Speed of Francis Runners. The following will contain a discussion of the dimensional relations of the Francis runner as affecting its specific speed. Fewer guide vanes and runner vanes are now used in turbine design than were formerly thought necessary, the great runners of the Cedars Rapids turbines having but sixteen vanes;the

* See Results of Experiments,Gibson "Hydraulics" Sec.62 also,F.H.Rogers,"The modern Hydraulic Turbine", A Symposium on Hydro-Electric Development,Engineers Club,Philadelphia, 1922.

inlet area of the runner is therefore less obstructed by vanes, and will be almost equal to the mean diameter of the runner, D multiplied by the breadth at entrance b_1, that is,

$$A = k_a \, b_1 \, D_1 \quad \text{- - - - - - - - - -(i)}$$

where k_a = a constant of area, almost equal to unity.

The velocity of flow or radial component of velocity, will equal some function of the spouting velocity of the water, say = $k_v \sqrt{2gH}$. Also the breadth, b_1, will bear some relation to the mean diameter of the runner, D, or $b_1 = k_b D$, so that the quantity

$$Q = \pi k_a \, k_v \, k_b \, D^2 \sqrt{2gH}$$

$$= k_g D^2 \sqrt{H} \quad \text{- - - - - - - - - - -(ii)}$$

$$\text{and} \quad D = \sqrt{\frac{Q}{k_g \sqrt{H}}} \quad \text{- - - - - - - - - - -(ii)a}$$

The horse power of the runner

$$H.P. = \frac{62.4}{550} \, Q \, H \, \eta \quad \text{- - - - - -(iii)}$$

and the peripheral speed

$$u_1 = \phi \sqrt{2gH} \quad \text{- - - - - - - -(iv)}$$

The speed in r.p.m.

$$N = \frac{60 \sqrt{2gH}}{\pi D} \cdot \phi \quad \text{- - - - - - - - - - -(v)}$$

Substituting in equation (v) in terms of horse power, H.P., and head H., and collecting all the constants into one and calling it N_s, we have

$$N = \frac{N_s \, H \sqrt[4]{H}}{\sqrt{H.P.}} \quad \text{- - - - - - - - - -(vi)}$$

$$\therefore \; N_s = \frac{N \sqrt{H.P.}}{H^{\frac{5}{4}}} \quad \text{- - - - - - - - -(vii)}$$

A knowledge of the constants entering into the above value of the specific speed is necessary in order to predict what the performance of the runner will be. For reasons already stated k_a will be almost unity. The upper limit of k_b is about 0.52 for runners of high specific speed; the value of ϕ does not ordinarily exceed 0.8 in Francis runners, while k_v varies with the gate opening, having a maximum value of about 0.28. Under these conditions the efficiency would be about 90 per cent and the specific speed

80, in the foot-pound system of units.

Following this discussion of turbine design, brief descriptions will be given of three notable installations of Francis runners, the Cedars Rapids Plant, the Queenston Plant of the Ontario Hydro-Electric Commission, and the Kern River No.3 Plant of the Southern California Edison Company, as being representative of installations for low, medium and high head service, respectively.

The Cedars Rapids Hydro-Electric Power Station. The main features of this plant are given in Table No.2, page 17. The Cedars Rapids turbines (Fig. 37) are of particular interest for several reasons. It is one of the lowest heads utilized for a large development, the turbines are near the higher limits of specific speed for Francis turbines, and they have the largest runners ever built, weighing over 80 tons. In view of the recent developments in high speed runners for low head service, it is exceedingly improbable that there will ever be another installation of the Cedars Rapids type of turbines. The immediate predecessor, so to speak, of the Cedars Rapids plant was the plant of the Mississipi Power Co., at Keokuk, (See Table 2, page 17) partly completed in 1913. One improvement of the Cedars Rapids turbines over those installed at Keokuk was in the construction of the foundation ring, or so called "speed ring", which, instead of being cast in two rings, one upper and one lower, and connected by means of stay-bolts, were cast with vanes connecting the two rings, the vanes being of fin-shape section and set approximately parallel to the direction of flow to reduce the hydraulic losses. This type of construction for the speed ring has been very extensively employed, and is preferable as giving a solid connection between the two rings. A new form of speed ring, however, has been developed for the Queenston units, to be described later.

The first ten units were completed in 1914 and were furnished with curved draft tubes and Kingsbury thrust-bearings to take the weight of the revolving parts, a load totalling 550,000 pounds. A description of the three types of thrustbearing, the Kingsbury, the Reist Spring bearing and the Gibbs, will be given later. Units No.11 and No.12 were installed during 1918; unit No.13 was put in service in 1921. These last three units are supplied with the Reist spring type of thrustbearing.

The Queenston Hydro-Electric Power Plant.* The turbines installed at the Queenston Plant of the Hydro-Electric Power Commission of Ontario not only exceed in power by almost 50 per cent the most powerful turbines hitherto built, but also embody many of the latest improvements in turbine design. The present installation consists of five 55,000 horse power units, operating under a 305 foot head at 187.5 r.p.m. and having a specific speed of 34.2 (page 17). Two of the generators are being furnished by the Canadian General Electric Company and three by the Canadian Westinghouse Company. Two of the turbines are of Wellman-Seaver-Morgan Company's manufacture and three are by the William Cramp and Sons, Ship and Engine Building Company, who also furnish the governing mechanisms for the five units. The

* F.H.Rogers, "Queenston-Chippawa Development of the Ontario Hydro-Electric Power Commission." Hydro-Electric Conference, Phil. 1922.

main valves are furnished by the Larner-Johnson Valve and Engineering Company. All the five units are very similar in design,Fig. 38 ,being a section through one of the units with I.P.Morris turbine of the Cramp Company. Three of the units are equipped with Kingsbury thrust bearings and two with the Reist spring thrust bearings. The thrust load of the revolving parts,including allowance for the hydraulic pressure,amounts to almost 1,000,000 pounds.

A new type of casing was developed especially for this installation,called the Taylor sectional spiral casing, after the inventor H.B.Taylor. Due to the great size of the units combined with the comparatively high head under which they operate,the usual form of speed ring was not considered adequate for supporting the weight of the units, since it does not provide for a very uniform distribution of stresses in the connecting bolts,and hence permits considerable leakage. For plants of moderate size the usual practice has been to cast the speed ring with its stay vanes separate from the scroll casing,and to fit the latter around it,securing it in position by means of bolts in horizontal flanges. This type of construction results in a long unsupported section across the throat at the radial flanges,causing excessive strain on the bolts and consequent leakage. Fig. 39 ,shows a comparison of the old and new types of speed ring and scroll casing. In the new design the speed ring is,in effect,cast integral with the scroll case,the radial flanges being extended to the inner bore of the casing,the stay vanes being cast in the segments;the turbine gates or guide vanes are then carried on two rings,a lower and an upper,bolted direct to the scroll casing. The unsupported section is thus decreased while making it possible to increase the number of bolts and distribute the stress uniformly to them. The tendency of the joints to open is reduced and gives a casing free from leakage.

The runner,outside diameter $10'\text{-}4\frac{7}{8}''$, weight 42,000 pounds,are very similar in design to the 37,500 horse power runners of Extension No.3 to the Niagara Falls Power Company's Plant. The efficiency characteristics are almost identical,giving a maximum of 93 per cent and a half load efficiency of 90 per cent. The labyrinth seals (Fig. 40) used for the 37,500 horse power turbines, were employed and are very efficient in reducing the loss from leakage. In its path past the runner,the water is compelled to enter successively three compartments,at which the velocity head is completely destroyed. The head producing leakage is thus reduced to about one third the value of the total head exsisting at the guide vanes. It is estimated that at full load the leakage at the seals of the runner amounts to about 14 cu.feet per second, an equivalent to 7/8 of 1 per cent of the discharge of the runner. Under similar conditions the loss from leakage with seals of the usual design would amount to about 1.5 per cent of full load quantity.

The I.P.Morris turbines are provided with the Moody spreading draft tube,and also the turbine of Unit No.2, built by the Wellman Seaver-Morgan Company; the other unit, No.1,the turbine of which is by this company,is provided with a curved draft tube designed by them (Fig. 41). This latter tube does not appear to differ materially from the curved draft tube of the Cedars Rapids units. As the Units No.1 and No.2 are identical throughout with the exception of the draft tubes,the results of the official tests for efficiency will be awaited with more than usual interest as indicating the difference in the efficiency of the two designs of draft tube.

The governor liquid is water furnished from a central pumping station,consisting of two motor driven pumps,each capable of delivering 700 gallons per minute at a pressure of 200 pounds per square inch. Each of these pumps is designed to take care of five units,the second pump being held as a spare. The discharged water from the governors is returned to the suction of the pumps,in order to prevent waste,as the water is treated with potassium bichromate to prevent rusting or corrosion of the governor valves and piping.*

Kern River Plant No.3,Hydro-Electric Power Station.† The main features of this power station are enumerated in Table No.2,page 17 . The water for operating the turbines is obtained from the Kern River by means of a concrete diversion dam of ogee form. The Kern River is one of the swift mountain streams of the Pacific slope,where little storage for the water can be obtained,and it was therefore found necessary to construct a sand-settling basin to remove the fine sand from the water to prevent injury to the turbines. The head,820 feet,is the highest so far utilized by Francis Turbines in a single stage and a few novel features are embodied in their design. One of these special features is the design of the water seal rings between the rotating runner and the casing. Straight-forced concentric clearance rings of rolled steel were attached to the runners,and bronze matching-rings were placed on the casing covers. All the sealing rings are made removable and renewable,both on the runners and on the casing cover on account of the necessity of keeping the clearances small on high head turbines to prevent great leakage loss. A second special feature is the provision of brakes,which are necessary for closing down Francis turbines when operating under very high heads. It takes very little water at such a high head to develop sufficient power to keep the turbine rotating at full speed without load. It is quite conceivable that,if the guide vanes become worn or slightly bent through closing on some obstruction,sufficient water might escape to prevent closing the unit down with the governor alone. A third feature peculiar to units of the Francis type for such high head service is the necessity for providing a stuffing box around

* See paper by F.H.Rogers,"Selection of Auxiliaries for Hydro-Electric Power Station". Power,May 16-1922.

† Ely C.Hutchinson,"Kern River No.3 Plant of the Southern California Edison Company." Mechanical Engineering,April 1922, and Booklet by Pelton Water-Wheel Co.

the turbine shaft which will prevent leakage and at the same time neither absorb an undue amount of power nor run hot. This was accomplished by fitting a water-jacketed, renewable wearing-sleeve over the shaft,and arranging to pack the stuffing-box only loosely with specially designed metallic packing-rings,any possible water seepage being drained off to tail-water by means of an automatic ejector.

From the sand-box to the surge-chamber,located at the top of the final descent to the power house,the water is conducted mainly in tunnel ,alternating with short stretches of concrete flume,the total distance being about $11\frac{1}{2}$ miles. A section of this flume,not far back from the surge-chamber, is provided with a syphon spillway. The surge-chamber is of reinforced concrete and is designed to equalize the flow between the tunnel section and the supply to the turbines. Unlike the Fully Development,the head is not of sufficient magnitude to require penstocks of special design. They consist of two steel pipes,84 inches in diameter at the surge-chamber,tapering to 60 inches at the power house,one leading to each of the two units. Each penstock is 2520 feet long,the upper 795 feet being 3/8-inch riveted construction,and the remainder forge-lap welded,the thickness increasing with the head to a maximum of $1\frac{1}{16}$ inches. A recording Venturi meter is placed in each penstock for measuring the water; a practice,the advisability of which has been much discussed of late,but not often followed,due to the loss in head of the water when passing the meter. We are not informed by the author of the paper from which this information is taken,just why the meters are installed, but since this plant is combined with an irrigation scheme, it may be that a knowledge of the quantity passing is necessary for irrigation purposes. A Johnson needle valve is placed in each penstock for the main water control to the turbines,and bursting plates are provided to relieve the penstocks of serious pressure surges.

The turbine runners are of bronze,and each turbine is provided with an interchangeable runner,for delivering power at 50 or 60 cycles,a provision made necessary by the nature of the service. The performance of the runners is shown by the efficiency curves,Fig. 42. The governor fluid is oil,the governor acting upon the guide vanes in the usual manner for small load variations. However,when a large drop in load occurs,the governor opens a relief valve, which is of the Pelton needle-nozzle type,and allows a portion of the water to run to waste,while closing slowly it avoids dangerous rises of pressure in the penstock. The turbines are equipped with curved draft tubes,and Reist spring thrust bearings take the weight of the revolving parts.

The Characteristics of Runners of High Specific Speed.

It has already been pointed out that the water turbine has enjoyed its recent active development due to the requirements for more speed and greater capacities for driving electrical generators. It was mentioned in the historical survey how efforts were made to effect increases in speed by means of belts and gears,or by placing several runners on a single shaft; but all these expedients were accompanied by mechanical complications and loss in efficiency. Efforts were made to eliminate all this complication,but simplicity was secured at a sacrifice of speed. All attention was centered on the mixed flow runner of the Francis type and its specific speed was pushed higher and still higher until specific speeds of well over 80 were attained. The turbines of the Cedars Rapids plant represent the highest development of the Francis type runner along these lines for low head service on a large scale. The direct connected units eliminate all the old complications, and the wheels operate at very high efficiencies,but these desirable features are secured at a great sacrifice of speed,necessitating the use of generators of very large diameter.

The practical limits of specific speed having been reached with the Francis type runner,attention was directed to the development of a new type,and several noted hydraulic engineers have been associated with its development, the principal of whom being Moody and Nagler in America, Kaplan in Austria, and Dubs in Switzerland. The greater amount of attention will here be given to the runners of the first two; some information was available concerning the Kaplan runner,but none was available concerning the runner by Dubs,apart from the section shown in Fig. 43 showing its general form.

A runner of high specific speed has several inherent characteristics which may be discovered by a close study of the part of Mr.Moody's paper,already given. These characteristics are briefly, first,high relative velocity between the water and runner vane; second,small vane curvatures and slight deflection of the water from its path of flow; third,outflow velocities of comparatively great magnitude. The first of these requires that the areas of the runner vanes be kept small, the second results in the third,which in turn necessitates some efficient means of regaining the energy of the water at discharge from the runner. This last is accomplished by means of the new designs of draft tube. These large outflow velocities created a very puzzling problem in the design and selection of turbines for operation under various heads,and one which has only recently been correctly solved. Referring to Fig.44 , it is seen that the total draft head

$$= S + \frac{v_1^2}{2g} - \frac{v_2^2}{2g} - \text{losses.} \quad \text{- - - - -(i)}$$

Assuming that by the use of an efficient draft tube the losses can be prevented from becoming larger, as v_1 or the velocity head at the throat of the runner increases, S must decrease if cavitation is not to occur, since the draft head cannot have a theoretical value in excess of 34 feet of water, and is not depended on for more than about 26 feet. To quote an example given by Mr. Rogers* : "Suppose a runner, similar to B, Fig. 46, $N_s = 80$, to have a diameter of 5 feet and to operate under a head of 30 feet, it would run at 132 r.p.m. and develop 1800 horse power. At this load the actual velocity at the throat of the runner D_2 would be about 27 feet per second, equivalent to a velocity head of 11.3 feet. Subtracting the losses in the tube and the outflow loss at the discharge of the tube would leave 7.5 feet as regained velocity head which must be added to the height of the runner above tailwater to obtain the total draft head. In the case assumed, the runner would be located 17 or 18 feet above tailwater. If the head were increased to 60 feet this runner would develop 5100 horse power, and the corresponding regained velocity head would be 15 feet, so that the runner would be located not more than 9 or 10 feet above tailwater. It is thus seen that as the head on the runner is increased, the position of the runner with respect to the tailwater elevation must be lowered."

The above principles were for a long time overlooked, and to quote from a recent paper. "it was common practice to select the turbine for any installation by taking the value from a so called "experience curve" † which was plotted between the variables: head and specific speed. Thus, it was assumed that for any given head available for a projected plant, the specific speed adopted should not be higher than the limit shown by the curve. No sooner would such a curve be plotted through the highest points obtained up to any time than someone would build a successful installation corresponding to a point considerably above the curve. On the other hand, there were plenty of installations which would plot well below the curve where serious trouble had developed in actual operation."

The explanation of these apparently paradoxical results was simple, namely, that the specific speed permissible for any installation is not a function solely of the head on the plant independently of anything else, but that it is also dependent upon the elevation of the turbine runner with respect to the surface of the tailwater.

The curves in Fig. 47, taken from the paper by Mr. Rogers, referred to previously, show the relation of the elevation of the runner above tailwater elevation to the specific speed, plotted in the lower part of the diagram. In the upper part of the diagram the specific speed is plotted against velocity head at the discharge or throat of the

* F.H. Rogers, "The Modern Hydraulic Turbine" A Sympossium on Hydro-Electric Development; Engineers Club, Philadelphia, 1922.
† Taylor and Moody "The Hydraulic Turbine in Evolution" Hydro-Electric Conference, Philadelphia, 1922.

runner expressed as a percentage of the total head on the runner. These curves show the total amount of energy discharged by the runner,which,in the case of the Francis type of runner,begins to increase very rapidly with an increase in the specific speed in the neighbourhood of 80. It will be noted from the similar curve of the diagonal or propellor type of runner,that it enjoys a much greater range in specific speed before the outflow velocities assume such a large proportion of the total head. Installations of the new type runner having specific speeds of almost 160 have been made and experimental work has been done on some with specific speed much in excess of this value with satisfactory results. From the above considerations it is seen how the specific speed of a runner affects its elevation above the surface of the tailwater. Where these principles have been disregarded and the runner has been placed too high there is a failure to secure sufficient absolute pressure at the top of the draft tube, resulting in objectionable vibration of the turbine,and serious corrosion of the runner due to cavitation and the liberation of corrosive gases from the water,principally free oxygen.

As indicated by the curves in the lower part of the diagram in Fig. 47,as the head on the runner and its specific speed increase the elevation of the runner with respect to the tailwater elevation must be lowered,and for certain values the runner should even be placed below tailwater elevation. There is no reason why the runner should not be so placed except that such an arrangement would involve very expensive excavation to accomodate the draft tube and would require some provision for removing the water from the turbine casing,such as a gate in the discharge channel. A proposal by Mr.B.H.Taylor to invert the turbine (Fig. 48) has recently been receiving considerable attention. Tests have confirmed the expectations that the performance of the turbine in this position would be identical with the usual arrangement and it is probable that installations of this type will soon be undertaken.

When discussing the theory of the reaction turbine, reference was made to the increasing importance of the elements of outward flow (page 28) in the development of the new turbine runners of high specific speed (Fig. 43). In these new types of runner the whirling mass of water is conducted as close as possible to the turbine shaft,where its angular velocity is accelerated as it approaches the axis of rotation,as will be observed below. The water leaves the runner with a very great whirling velocity which is most conveniently regained by conducting the water away from the centre as quickly as possible. This is one of the essentials of high speed development,the problem not arising with the low speed types of runner,where the water imparts its energy to the runner blades at a comparatively great distance from the turbine shaft. The nature of the problem may be more conveniently understood by a consideration of the laws of a free vortex. In a free vortex the velocity varies inversely as the distance from the axis of rotation. Fig.45 ,represents a vortex formed in an open basin. The two types of vortex which may be taken to represent the flow in a hydraulic turbine are the Free Spiral Vortex and the Free Cylindrical Vortex. The first is formed when water escapes freely through an orifice in the bottom of a vessel,in which the water moves spirally towards the centre of the vortex,its velocity having a whirl component and also a meridian component downward. If,while the mass of fluid is rotating,the orifice be suddenly closed,the meridian component of velocity will be arrested,and the motion becomes one of simple rotation in horizontal planes,and forms the Free Cylindrical Vortex. To find an equation to the surface of this latter type, referring to Fig.45 ,since the velocity varies inversely as the distance from the axis of rotation*,we have.-

$$v = \frac{k}{r} \quad \text{- - - - - - - - - - (ii)}$$

and putting $\frac{p}{w}$ constant in Bernouilli's equation, we get the equation to the surface of equal pressure,that is,

$$\frac{v^2}{2g} + z = C \text{ (a constant) - - - (iii)}$$

Combining equations (ii) and (iii) we have,

$$C-z = \frac{K}{r^2} \quad \text{- - - - - - - - - - -(iv)}$$

the equation to a hyperbolic curve of the nature $y\,x^2 = a$, which is asymptotic to the axis of rotation and to a horizontal line through $z = c$.

In discussing the application of the above principles as applied to draft tube design Mr.Moody† considers the Free Spiral Vortex and neglecting the meridian component of the velocity,derives equation (iv) as above,which,as he states,approximates very closely to the surface of a free vortex.

A very interesting and instructive discussion of cavitation and the formation of vortexes in the flow of water through the turbine was given my Messrs.Taylor and Moody in a recent paper,†† The portion of the paper dealing with this phase of the subject will be given in full, on account of the light it casts upon the causes of vibration and corrosion of the runner.

"Since points of low pressure are conductive to cavitation,or the formation of voids in the water stream, with tendencies toward eddy formation,the formation of

*Gibson "Hydraulics". Sect.38, page 97.

†"The Present Trend of Turbine Development." Symposium on Hydro-Electric Development. Philadelphia, 1921.

†† Taylor and Moody,"The Hydraulic Turbine in Evolution." Hydro-Electric Conference,Philadelphia, 1922.

air pockets,and danger of corrosion and vibration,it is of course,desirable to avoid any such low pressure regions. An undue reduction in pressure in advance of the runner can be avoided by the admission of the water to the runner before it has approached too closely to the axis,the further reduction in pressure which the water must experience on its way to the draft tube then taking place within the runner during the process of transferring energy from the water to the runner. A reduction of pressure in advance of the runner is aggrevated when the turbiine is operating at part gate if it is equipped with the usual wicket gates, since at part gate the water leaves the guide vanes with increased velocity in a more tangential direction and with lower pressure; the pressure starting at a lower value near the guide vanes drops more rapidly toward the axis. It may easily happen,particularly at part gate,that pressures may exsist in the inner portion of the entrance space to the runner which are equal to,or even less than the pressure in the draft tube. The new diagonal form of runner blade is adapted to prevent an undue pressure reduction in advance of the runner blades by providing a considerable diameter for the inflow edge of the blade where it joins the runner hub.

Another consideration which leads to the adoption of a diagonal flow runner in preference to the purely axial flow type is that as the stream elements at the upper portion of the guide vanes approach the axis,the velocity of the water increases in inverse proportion to the radius. The velocity of the entrance edges of the runner blades, however,decreases in direct proportion to the radius,so that if the flow is carried too near the axis it may readily occur that the water will be travelling faster than the runner,requiring a backward angle of the runner blade and leading to complicated curvature and inefficient action. There is no necessity for the close approach to the axis and by carrying the runner blades diagonally across the entrance passage an efficient runner has been produced and one having great mechanical strength due to the length and form of the blade section where it joins the runner hub.

Since the primary whirl persists,although in reduced amount,at discharge from the runner,it is desirable when structurally feasible,to provide in the draft tube a central core continuous with the runner hub. The provision of this core avoids the tendency toward the formation of a central cavity within the flowing stream with the resulting production of eddies,turbulance and unsteady flow.

The second kind of whirl in the turbine occurs in meridian planes,that is,planes containing the turbine axis, and constitutes a rotation about axes perpendicular to these meridian planes. This rotation sets up an increase in pressure toward the upper and inner surface of the entrance space to the runner,and an increase in velocity and decrease in pressure of the lower distributor plate and runner tips. This "secondary" whirl,therefore,reduces the velocity of the stream elements leaving the upper part of the guide vanes and correspondingly reduces the velocity attained at entrance to the runner by these elements which enter the runner nearest to the axis;the stream elements leaving the lower end of the guide vanes being increased in velocity by this secondary whirl,but as these elements move toward the turbine axis only a small distance,their velocity is not further increased by any great amount.

The primary and secondary whirls,therefore,tend to some extent to neutralize each other in their effects of causing unequal distribution of the velocities and pressures in different portions of the stream. After turning the water from an inward direction to the axial,it then becomes desirable to conduct it away from the axis as an excellent means for regaining as much as possible of the energy of with which it leaves the runner,while at the same time regaining the energy corresponding to the meridian component of velocity. The deflections from radial inward flow to axial flow and from axial to radial outward flow can very well be made continuous in a turbine designed to occupy small space in an axial direction. *Fig. 32 shows such a design in which the secondary whirl is maintained and gradually reduced in magnitude as the flow attains the outward direction,without any abrupt changes in curvature of path.

The third kind of whirl is a local whirl having a fixed relation to each blade of the runner and carried along with the blade,the axis of this whirl being transverse to the general direction of flow and lying approximately in a revolving meridian plane. Consider the flow behind a blade,Fig. 49 . The water adjacent to the rear surface of a blade will follow a curved path,so that it may change in direction to correspond with the deflection produced by the blades and thus develop torque on the runner. The curvature of the path of successive stream elements will decrease at points receding from the blade,and at some distance from the blade curvature will disappear. The flow behind each blade therefore contains a whirling motion approximating in some degree to a rotation about an axis normal to the plane of the figure. This axis may be taken as the centre of an arc coinciding with the rear surface of the blade. If we apply to this local whirl the laws of a free vortex and suppose the velocity relative to the blade to vary inversely as the radial distance from the axis of rotation, we will have a picture of the flow sufficiently close to what probably occurs behind a blade to furnish us some useful conclusions. Such a local vortex at each blade would result in a local reduction of pressure on the back surface of the blade and we can investigate the probable amount of such local pressure reduction as follows: Calling r the radius from the centre of curvature of the blade surface to any point in the stream; w the

*Not the same figure referred to, but similar.

relative velocity of the water with respect to the blade and h_p the pressure head at any such point; P the pressure in pounds per square foot; W the weight of water per cubic foot, we can readily determine the effect of centrifugal force by considering an elementary particle of the stream of radial thickness dr and cross sectional area a. The differential pressure difference across such an element will be equal to the centrifugal force of the element divided by a: or,

$$d\,h_p = \frac{d\,P}{W} = \frac{W a dr}{W a g} \cdot \frac{w^2}{r} = \frac{w^2\,dr}{gr}$$

In a vortex the velocity varies inversely as the radius, so that

$$w = \frac{W_b\,r_b}{r}$$

Hence

$$h_p - h_{pb} = \int_{r_b}^{r} \frac{w_b^2\,r_b^2\,dr}{gr^2} = \frac{w_b^2\,r_b^2}{2g}\left(\frac{1}{r_b^2} - \frac{1}{r^2}\right)$$

$$= \frac{w_b^2}{2g}\left(1 - \frac{r_b^2}{r^2}\right)^* \text{ - - - (v)}$$

in which h_{pb}, w_b and r_b are the pressure head, velocity and radius at the back surface of the blade. From the figure it would appear reasonable to assume that the whirl ceases where the water directed by the next blade tends to follow a straight path. As an approximate method of figuring the local drop in pressure behind each blade, it is therefore proposed to perform the above intedration between the radius r_b at the blade surface and the radius r_o representing a perpendicular dropped upon the centre line of the preceding blade, using for the relative velocity at the rear surface of the blade the amount w_2 corresponding to the outflow triangle of the runner. We then have

$$h_{p2} - h_{pb} = \frac{w_2^2}{2g}\left(1 - \frac{r_b^2}{r_o^2}\right) \text{ - - - - - - - - - (v)a}$$

This represents the amount by which the local pressure on the surface of each blade falls below the average at the top of the draft tube. The actual amount of this reduction will vary greatly with the form, length and spacing of the blades in any particular runner, and the above method of estimating it can only be regarded as roughly approximate. It does, however, show that a local pressure reduction is to be expected, and gives an idea of some of the factors affecting its value.

As any tendency of the water to leave the vane surface

*See Gibson "Hydraulics", Sect.38, page 97.

is believed to be conducive to cavitation with resulting risk of corrosion and vibration,it is considered highly important to provide enough margin of pressure in the draft tube below the runner to take care of this local pressure drop,so that an adequate intensity of absolute pressure will remain at the vane surface. This consideration requires that the runner be placed at an elevation above tailwater not exceeding a value consistent with the Bermouilli formula applied to the draft tube as follows:

$$h_{p2} + \frac{c_{m2}^2}{2g} + \frac{c_{u2}^2}{2g} + H_2 - f_d \left(\frac{c_m^2}{2g} + \frac{c_{u2}^2}{2g}\right) = h_a \; ; \; - -(vi)$$

in which c_{m2} and c_{u2} are the meridian and whirl components of the discharge velocity from the runner; H_2 the elevation of runner above tailwater; f_d the proportion of velocity head lost in the draft tube and at final discharge; and h_a the pressure head of the atmosphere.

The average pressure at the top of the draft tube will equal

$$h_{p2} = h_a - H_2 - c_d \left(\frac{c_{m2}^2 + c_{u2}^2}{2g}\right) - - - - - -(vii)$$

in which c_d is the efficiency of the draft tube; and

$$h_{pb} = h_a - H_2 - c_d\left(\frac{c_{m2}^2 + c_{u2}^2}{2g}\right) - \frac{Kw_2^2}{2g} - - - -(viii)$$

in which K is some factor corresponding to the above formula for local pressure drop,or otherwise determined. One useful method of determining the local pressure reduction is by referring to marine propellor practice. Conservative practice in marine propellor design dictates that the tip speed of the propellor should be kept below a fairly definite limit based on experience with actual propellors which have developed serious vibration or corrosion. If we express the local pressure drop in terms of velocity head corresponding to tip speed rather than to relative velocity,these velocities being not greatly different in propellors and high speed runners,and if we assume that vibration or corrosion are due to cavitation produced when the local pressure at some point in the propellor or runner has been reduced to an extent sufficient to overcome the static pressure of the atmosphere and the depth of immersion of the top of the wheel,we can obtain an expression of the form

$$\text{local pressure drop} = (\text{a constant}) \times \frac{(\text{tip speed})^2}{2g}$$

in which the coefficient is determined by inserting in this formula the total static pressure at the top of a propellor and the corresponding tip speed which has been found to give safe protection against cavitation for ordinary forms of blades.

Although the allowance to be made in any given installation depends on many factors, as already pointed out, the above will give a general idea of the effect of the local whirl or vortex at each blade.

One of the things to be avoided is an undue reduction in blade surface particularly for turbines operating with inadequate margin of pressure in the draft tube. Insufficient blade area is not only conducive to cavitation at normal load, but gives rise to unsteady or unstable operation at part load. Evidently the driving pressure is derived from the pressure difference between face and back of blade, resolved into the tangential direction. If the angle of inclination of the blades is small, a high intensity of pressure increase is required on the blade face, and high intensity of decrease on the back. A small number of narrow blades therefore magnifies the problem outlined above."

The authors continue the paper by describing the new form of runner, giving performance curves and pointing out wherein the new runners resemble, and where they differ from, a ship's propellor.

The Moody Diagonal High Speed Runner. The theoretical considerations underlying the development of high speed runners are contained fairly completely in the foregoing comments on the characteristics on high speed runners, and in the analysis on pages 33 to 41, taken from Mr. Moody's paper of 1921. There remains therefore to describe the general form and performance of the runner as developed by Moody and to reproduce some of the curves giving the efficiency and other characteristics. Fig.50* shows a runner of this new type as designed for the Great Falls Development in Manitoba, it is designed to deliver 28000 horse power at 138.5 r.p.m. under a head of 56 feet. The specific speed is 153. Fig 52, taken from the paper by Messrs. Taylor and Moody is a cross section of the unit as installed. Note the draft tube is provided with central core, previously referred to, and of which more will be said under the heading of draft tubes. One of the Cedars Rapids runners is also shown in Fig.50, by way of comparison. In the new runner, the outer band is eliminated to reduce frictional losses at the high relative velocities; also the number of runner blades is reduced to six, effecting a great reduction in area as compared with the Cedars Rapids runner, with its sixteen blades. This also means a much wider space between the blades and much less danger of clogging from ice or debris. The intake racks may also be spaced wider in proportion. The runner blades have very little curvature; this, it will be observed by reference to Fig.50, and to equation (v)a, page 53, has a tendency to make the factor $\frac{r_b^2}{r_c^2}$ approach unity, and therefore to reduce the amount of local pressure reduction behind each blade, reducing the liability to vibration and corrosion.

Fig. 53 is interesting as giving the results of tests

*See paper by S.H. Van Patter. Journal E.I.C. Sept. 1922.

of models of the Great Falls and Cedars Rapids runners. As stated by Mr.Van Patter,* "These curves are plotted from the test results of the 15 inch experimental (Moody) turbines,and so the efficiencies shown are considerably less than obtained at Holyoke,for which official results are not yet available. The dotted curves are for the Cedars model runner with curved and Moody draft tubes. This runner gave a maximum efficiency of 90 per cent on Holyoke test. It will be noted that the Moody tube gives a considerable increase in efficiency at part gates. The Moody tube also gave a considerable increase in maximum power, showing about 7 per cent over the curved tube in this test. The full curves are for Moody high speed runners at specific speeds of 110, 125 and 153. As the speed increases the part gate efficiencies fall off appreciably. This,of course,is a big disadvantage for installations with a small number of units when the load is variable. However,with the increasing tendency to link up such small stations with others in the vicinity,this point becomes of small importance. Also,by going to a somewhat lower speed,very good part of gate efficiencies may be obtained, as shown for the curves for model No.81 and a considerable saving in cost will be effected as compared with the Francis type turbine. This model,No.81,gave an efficiency of about 91 per cent at Holyoke."

It was stated,page 51 ,that diagonal flow is preferable to pure axial flow,and that a very great reduction of blade area is to be avoided,page 55 ,since pure axial flow and small blade area tend toward very unsteady flow and loss of efficiency at part gates. Fig.54 ,shows a comparison of two runners,a four bladed axial type runner with a six bladed diagonal runner. †"It will be noted that the axial runner shows marked instability at small gates,while the diagonal runner exhibits smooth curves of normal form." It is generally conceded that high speed runners involve a considerable sacrifice of efficiency,even under the most favourable conditions of operation,but especially so at part gate. In the two types of runner being compared by the curves of Fig.55 ,we may take it that the difference in their performance is due to the manner in which they behave with respect to the three types of whirl as mentioned on pages 51 and 52. The primary whirl,we may expect,and also the tertiary whirl causing the local pressure reduction behind each blade,will probably be the same in each case,and produce approximately the same effects. We are led,therefore,to conclude that the principal difference in their performance is due to the behaviour of the secondary whirl. It appears desirable,especially at part gate,to have the water which is admitted at the upper extremities of the guide vanes enter the runner before its velocity is greatly reduced by this secondary whirl. This is a point of considerable interest,as it marks the principal difference between the two types of high speed runner being developed on this continent. The part gate efficiencies of the high speed runner as shown by the curves of

*Journal E.I.C. Sept.1922.

†See paper by Taylor and Moody,referred to on page 48.

Fig. 53 are greater than those shown by the curves of Fig. 56; and probably the behaviour of the secondary whirl accounts for much of the difference.

There is a tendency for eddies to form after each blade in a high speed runner which lowers the efficiency and this tendency is no doubt increased at port gate. A secondary blade placed at the back and slightly in advance of the main blade has been proposed as a means of reducing such disturbance. This suggestion was made by Messrs. Taylor and Moody, as follows:

"As it is believed that there is a tendency in a narrow bladed runner for an eddy to form behind each blade at part gate and sometimes even at normal gate,one of the writers has proposed the use of secondary blades close to the main blades but slightly in advance of them,for the purpose of directing the flow more efficiently along the back of the main blade,and by this means eliminating instability and vibration and improving the part gate efficiency, etc. Fig.57 shows a four bladed axial runner equipped with such secondary blades,the two blades being in contact in the position shown. The secondary blade is of diagonal flow form. It was found by tests that the addition of the secondary blades or "intervanes", reduced the specific speed,but increased the efficiency by eight per cent and that it prevented unstable flow and vibration and improved the part gate efficiency. It was found that the best angular position of these blades was about ten degrees in advance of their position of contact with the main blades. The field of usefulness of the idea is yet to be determined; it is merely presented here as a suggestion of interest."

<u>The Nagler Propellor Type Runner.</u> The runner originated by Mr.Nagler probably represents the extreme development of high specific speed. Concerning high speed development, the inventor of this runner has to say* "The primary essential to high characteristic speed (125 to 200) is a reduction in hydraulic friction and harmful centrifugal forces. Neglecting friction and possibly blade thickness, there are no mathematical or hydraulic laws that will prevent doubling or quadrupling any particular characteristic speed by simply flattening the blade angles. A direct analogy to this is the well known illustration of relative velocities evidenced in the sail of an ice boat. The practical effect of so doing with a given profile is to lengthen the blade so that the friction on the increased wetted surface reduces efficiency and speed or results in constriction of passage. In the author's design these effects are counteracted by cutting out blades,the effect of which is not manifested in the reduction of power that might be expected. On the contrary the discharge is increased without reduction of efficiency."

Flattening the blade angle results in greater pressure differences between the face and back of blade,as already pointed out,page 55 ,which is sometimes attended with certain undesirable features.

*Forrest Nagler,"A New Type of Hydraulic Turbine Runner." Proc. A.S.Mech.Engineers. 1919.

Fig.56 ,is a curve plotted from the log sheet of a test on a model of the Nagler runner,taken from the paper referred to above. Fig.54 ,contains the curves showing the performance of these runners as installed at the plants named.* A cut of the runner for the Green Island Development is shown in Fig.51 ,and a crosssection of the same plant is given in Fig.58 .Like all high speed runners,it sacrifices efficiency at part gate,but when the load factor is high or the plant comprises a number of units,this feature is not a serious drawback.

The Kaplan High Speed Runner. A paper recently published by C.Reindl, Muenchen,in Zeitschrift deutscher Ingenieure, describes the development of the high speed turbine runner as originated by Dr.Kaplan,of Austria. The paper also gives many valuable references to works published upon the relation of specific speed to frictional losses as investigated in the extracts from Mr.Moody's paper, pages 33 to 41 . The general principles of design of Dr.Kaplan's turbine are,of course,identical with those already discussed,and he claims,with much evidence to support him,that the greater part of the pioneer work on the high speed runner was done upon the other side of the atlantic. The idea of cutting back the runner vanes,as first practiced with the Francis runner,originated in Sweden. As stated by Reindl: "The principle of the Kaplan turbine is,to make a long story short,to shorten as far as possible the length of the blades of the runner to obtain as small losses in friction and bending of the current as opportune and to widen the distance between the guide vanes and the runner blades so that the natural flow of the water would be least disturbed;this entirely contrary to conventional theories regarding water turbines, just as other theories are set aside by the new proposal of the Kaplan turbine."

A complete discussion of the Kaplan turbine would mean a recapitulation of much that has already been said concerning high speed runners; the abandonment of the idea that complete guidance of the water is necessary; a whirling mass of water leaving the guide vanes and approaching the centre of the turbine with its velocity increasing according to the relation $\omega w = c_u/r$; and the utilization of the energy of the water at a small distance from the axis of rotation and the necessity for conducting it away again from the central axis without disturbance and loss.

A cut of the Kaplan turbine is shown in Fig. 59 B, Fig. 59 A shows the early Kaplan and Fig. 59 B the Kaplan turbine of 1913, while Fig. 59 C shows one of later development. Dr.Kaplan originated and took out patents (1913) for a turbine with movable blades; it being possible to change the angle of the blades according to the demands made upon the turbine and the quantity of water available. (Fig. 60). Another model of turbine has been developed in which the maximum torque can be predetermined and fixed

*Louis E.Ayres,"Developing the Low Head Plant." Electrical World. Dec.25,1920.

at such a value as not to endanger the generator to which it is connected. The specific speeds developed range from 400 to 1500 in the c.g.s. system or from 90 to 336, in foot-pound units.

Quoting from the paper referred to above,the following are some results of a test on a model runner of 183 m/m diameter,and three propellor blades,April 1921.

Gate Opening.. %	Specific Speed. N_s Metric	Ft-pounds	Efficiency. %
75	1095	246	77
75	1120	252	78
100	1150	258	78
100	1210	272	77
120	1350	304	73

These results were obtained on a wheel which had been demensioned as to surface of blades in accordance with the accepted theory.

The average values of efficiency for the Kaplan type of turbine as ordinarily constructed are given as follows :

With N_s = 700 to 800, measured efficiency = 85%
(157 to 179)

" N_s = 1200 " " = 77%
(270)

It will be noted that for moderately high specific speeds upon the authority given above,the efficiency of the Kaplan turbine compares very favourably with the turbines of Moody and Nagler,and that speeds much in excess of anything attempted by the latter two types are obtainable.

Advantages of the High Speed Runner. The advantages of the new form of turbine are many,and mark a distinct step forward in turbine development,despite the disparaging remarks of A.T.Safford:* "Recently there has been placed on the market a propellor type runner,for which many advantages are claimed,chief of which is a high specific speed. It is proposed to demonstrate again the truth of the old adage 'there is nothing new under the sun' to show wheels almost exactly the same were used many years ago,and that their high speed was appreciated. It is claimed that the modern propellor wheel is axial flow. Exception is taken to this,as it is difficult to understand how,in the matter of direction of flow,the propellor wheel can be different from any modern high speed reaction wheel. The propellor wheel,as commercially installed,is set in a wheel case of the Francis type,the water being admitted through the usual wicket gates, in a direction approaching the tangential."

*Proc. A.S.C.E. April 1922.

It is, of course, impossible to get away from the fundamental forms, but when comparing the Jonval and other axial flow turbines originally developed with the modern high speed types it should be recalled that the former had specific speeds ranging from 10 to 20. To show the transformation in turbine practice made possible by the high specific speed turbines, Fig. 46 , has been reproduced,* showing a comparison of three runners of equal power but of different type, all drawn to the same scale. Runner A is of the low speed type used in the 30,000 horse power turbines at Big Creek Station, No.8 of the Southern California Edison Company, for a head of 680 feet; figure B shows the Francis high speed type used for the Cedars Rapids and many other moderate and low head plants; figure C shows the new diagonal propellor type, as developed by Moody. Some of the adavntages of the high speed runners have already been mentioned; these and others may be briefly summarized as follows:

(a) The increased speed makes possible a considerable reduction in the generator diameter; the outside diameters of the Cedars Rapids generators are 37 feet 4 inches, whereas the same dimensions of those for the Great Falls Development, Manitoba, do not exceed 24 feet for over double the capacity. A corresponding reduction is possible in the structures for supporting both the revolving and stationery parts and a saving can be effected in the overhead cranes and super-structure.

(b) Higher generator efficiency, since a generator operates more efficiently at high speed.

(c) It is claimed that the mechanical strength of the propellor type runner can be made greater than that of the Francis type, since the overhang of the blade from the crown of the runner is less, and the connection between the runner hub and the blades is conaiderably longer, resulting in smaller bending stresses in the blades.

(d) The runner blades being fewer, the spaces between them are correspondingly increased, reducing the danger of clogging from ice or floating debris, and permitting the head racks to be spaced at greater distances.

(e) The propellor type runner can be constructed with the runner blades bolted to a central hub, thus removable and renewable. In the Francis type, where the runner blades must be cast integrally with the hub, damage to one blade requires renewal of the entire piece.

The Ejector Turbine. The flow of many of our larger rivers which furnish most of the low head water powers is subject to very great seasonal variation, and this characteristic of their flow is one of the principal obstacles to their economic utilization. It is not, in general, possible to provide sufficient storage facilities on such streams to empound the entire flow of a flood period; in fact it is frequently impracticable to provide storage for more than sufficient water to carry the load over the daily peak, the greater portion of the flow being wasted over the spillway of the dam. During a flood period the increased flow raises the tailwater elevation in the

*F.H.Rogers, "Developments in High Speed Runners for Hydraulic Turbines." Power, April, 25-1922.

majority of streams providing low heads,thus reducing the available head and reducing the power in the ratio of the head to the three-halves power. Again,for most efficient operation,the turbine speed should vary as the square root of the head,but since the majority of plants must operate at constant speed,this reduction in head is also attended with a reduction in efficiency of the turbine. The combined effect of the two losses may be such as to seriously curtail the plant output.

Much thought has been devoted to this subject both on this continent and in Europe. To provide additional units to maintain the plant output during flood periods means an increase in the initial outlay and in the capital charges. The possibility of maintaining the output by designing the turbines so as to give maximum efficiency at a gate-opening somewhat in excess of normal; say 125%, would mean that the turbine would have to operate at reduced efficiency the greater part of the time,but it is desirable,when the flow is least,to have the efficiency a maximum. Above all,this principle could not be applied to the modern high speed runners of the propellor type, for which the efficiency is maintained over only a very small range of gate-openings. The most hopeful proposal for the solution of this problem is the ejector turbine (Fig.63).

To increase the fall and power,Clemens Herschel* in 1908 devised and tested at Holyoke the plan of connecting the lower end of the turbine draft tube to a chamber wherein a partial vacuum is produced by causing part of the waste water to flow through a tube shaped like a Venturi meter,suitable connection being made between the specially designed throat of the tube and the vacuum chamber. This device,called the "fall increaser" gives greater available power at high water stages,since the vacuum head H_1 is added to the head H between the upper and lower water levels,and since the discharge through the turbine is also increased.

Using this principle for increasing the head upon the turbine,Lewis F.Moody originated the Ejector Turbine (Fig.63); using a runner of the standard types,the setting is so constructed that water may be by-passed into the draft tube,increasing the head and quantity of water applied to the turbine. This turbine is quite fully described in an article by S.Logan Kerr,appearing in Mechanical Engineering,April 1922. In addition to discussing the need for such a turbine and its design, the author includes the results of tests,and also gives descriptions of appartus and methods used in testing, making the paper of considerable value to anyone undertaking similar work. Fig.62 is taken from this article and gives the performance curves for one of two runners tested,using the ejector setting.† The results of the tests on one runner only,No.55,are shown (Fig.62). This

*Merriman,"A Treatise on Hydraulics" 9th edition,page 477.

†Tests made at the I.P.Morris Experimental Laboratory of the William Cramp & Sons Ship and Engine Building Company, Philadelphia.

runner is of the mixed flow type,16 inch throat diameter, and has a specific speed of 84.4 (foot-pound system) at best ϕ (ϕ having the usual significance). The first series of curves (Fig.62 A) was drawn using efficiencies as ordinates and horse power at 1 ft.head,reduced to 1 ft.throat diameter,as abscissae . The envelope of these curves (dotted) was taken as the performance of the turbine and ejector combined. From these curves it may be observed that a considerable increase in power is obtained over the rated output by the use of the ejector. Based on a rating of 0.300 H.P. at 1 ft.head and 1 ft.throat diameter,for runner No.55 this increase amounted to 36.7 per cent; on a basis of maximum power it amounted to 34.5 per cent.

A second set of curves was plotted (Fig. 62,B) showing the performance of the turbine and ejector under varying heads,each curve consisting of two parts,the normal performance curve and the envelope of the several ejector performance curves. Curves are plotted for 70,80,90,100 and 110 per cent of normal head,the latter being taken as 1 ft.to have a standard for a basis of comparison. Along with these curves also have been plotted curves of horse power against quantity.

The curves in Fig.62,C were plotted using vane and ejector openings as ordinates against horse power as abscissae. It may be noted that with runner No.55 normal rated horse power can be mainatined under 80 per cent of normal head.

A similar test was made and a similar set of curves were plotted for the second runner,No.65, which was of the diagonal type,16.25 inches in diameter and having a specific speed of 100. The increase in power with the ejector was considerable,but not as great as in the case of the Francis type runner,being 30 per cent over rating and 17 per cent on a basis of maximum power. With the diagonal type runner,83 per cent of normal head was sufficient to maintain the rated power,when using the ejector.

When operating at normal ϕ the water coming off a runner of the Francis type has a slight forward whirl in the direction of the runner;in the high speed runners this whirl is of considerable magnitude,and is also forward. In order to create as little disturbance of the flow in the draft tube as possible,the ejector is provided with guide vanes which give the water an initial whirl as it enters the draft tube,the angle of whirl varying with the requirements of operation of the turbine. So far,no commercial installations of this type of turbine have been made.

The Principles of Draft Tube Design. Draft tubes are attached only to reaction turbines,since continuity of flow is required for their successful operation. Certain attempts have been made to increase the effective head on impluse wheels by enclosing the runner in an air-tight casing and allowing the discharge water to form a column in a tube leading to tailwater; the weight of the column creating a suction. The simplest principles of a draft tube are applied in certain cases,but the arrangement has not proven generally popular. Draft tubes were first attached to reaction turbines with the object of allowing them to be raised above tailwater elevation without loss of head,the tubes being straight,and little attention being given to the possibilities of regaining the velocity head of discharge by a tube of increasing section. The Boyden diffuser was designed to regain this kinetic energy of the discharge water, but it was not extensively used. We are not to infer from these remarks,however,that the possibilities of the spreading draft tube were overlooked by the early hydraulic engineers; for spreading draft tubes were constructed about the sixties and seventies,and in some cases a central cone was inserted for conducting the water away from the axis of the tube in order to reduce its rate of whirl and increase the pressure. When one carefully follows what has been previously given concerning the design of turbines,and particularly the discussion of the magnitude and direction of the outflow velocities in a development of high speed runners,one is led,however,to conclude that much credit is due to recent designers in developing draft tubes which have greatly improved the efficiencies of the turbines in many recent installations; and this,not withstanding the remarks of Mr.A.T.Safford* ;-After giving as the reasons for the general neglect of draft tubes that they were not considered a good investment from a financial point of view, he states;

"Apparently the pendulum has swung to the other extreme, and there are now on the market various kinds of patent draft tubes,each of which,like the old patent medicine,is promised to be the cure-all for every hydraulic trouble. The basis of one of these tubes is an inverted cone placed beneath the discharge opening of the draft tube,with its axis coincident with that of the runner. The object is to change smoothly and without shock from a vertical to a horizontal direction.

The placing of a cone beneath the draft tube is almost as old as the turbine itself. Some of the early Jonval-Koechlin turbines had it. Almost every German textbook on water-wheels from 1860 to Dr.Camerer's"Wasserkraftmachinen" of 1914, shows one or more settings with cones below the draft tube."

The earlier designers may have developed a tube of the form similar to that of to-day,but their draft tube problems were in no way similar to those of the present. For one thing,they had not the runners of such high specific speed and again,they had not the problem of regaining the high whirl components at outflow that exist with the high speed runners of recent development. The object is not

*"The American Mixed Flow Turbine and its Setting." Proc. A.S.C.E. April,1922.

now merely "to change from a vertical to a horizontal direction," but to conduct a whirling mass of water, which is made to pass through the runner comparatively close to the axis, a sufficient distance from the axis of rotation to reduce its velocity of whirl and convert the kinetic energy of the water into useful pressure head. The problem is complicated by the fact that the magnitude and direction of the whirl changes with the turbine gate-opening. In the Francis type turbine, if the gate opening is less than normal, the water whirls in the direction of rotation of the runner. When the gate-opening is above normal, the direction of whirl is opposite to the direction of rotation. In the modern high speed runners, the absolute whirl is always in a direction corresponding with the direction of rotation of the runner (see note, page 51 and Fig. 34), but it varies greatly in its magnitude and inclination to the meridian plane with a change in gate-opening. For the Francis type there is evidently a position of the gates where no whirl will exist, or only a slight forward whirl, and the water will leave the runner with almost pure meridian flow. This is often spoken of as the most advantageous gate-opening, and is the condition with which the ordinary types of straight or curved draft tube are capable of most efficiently dealing. The changing conditions of whirl in the new high speed types of runner for a reduced gate-opening place a greater burden upon the draft tube of regaining the outflow velocities; a task which could not be accomplished with the old forms of tube (Fig. 64). These changing conditions of whirl are somewhat analogous to the action of a jet impinging on a Pelton runner bucket. If the bucket is moving at a velocity below that for best efficiency, the water will have an absolute velocity backward; if the velocity is that of best efficiency, the water will leave the bucket with its absolute velocity reduced almost to zero; if the speed of the bucket be above that for best efficiency, the water will leave with a forward absolute velocity.

The whirling mass of water leaving the runner behaves somewhat like a gyroscope, resisting any effort to divert it from its general direction of motion. A curved draft tube, such as shown in Fig. , a type formally quite popular, is a device ill-fitted to handle efficiently such a whirling mass. An attempt to conduct the water from a high speed turbine through a tube of this form results in much disturbance, areas of reduced pressure and backward flow, giving rise to the boiling appearance of the tailwater leaving installations equipped with curved tubes. This whirling mass of water as it leaves the runner resembles an inverted vortex, and a reversal of the conditions of flow in the vortex suggests the most logical method of regaining its energy. No mention is made of such a process in any of the standard works on hydraulics which the writer has at hand* ; it is therefore believed that credit

*Among these are included Gibson, "Hydraulics and its Applications;" Merriman, "Treatise On Hydraulics." 10th ed; Mead, "Water Power Engineering"; von Schon, "Hydro-Electric practice"; Morley A. Parker, "Control of Water "; and Bovey, "A Treatise on Hydraulics".

should go for the suggestion to the originators of the "Patent" forms, Messrs.White and Moody. From the evidence contained in much of the recent literature on draft tubes it would appear that Mr.Moody was the first on this continent to approach the subject of draft tubes design in the light of a knowledge of the conditions of discharge from modern high speed runners; though a form of draft tube simultaneously originated by Mr.W.M.White had its origin in a series of experiments commenced as early as 1913. An article appearing in Mechanical Engineering April,1922,by W.K.Ramsey gives a general description of the two tubes. The same issue of Mechanical Engineering also contains the results of a series of tests conducted on a number of straight conical tubes by Mr.E.G.Lyon. His method of dealing with the results is instructive and interesting,but since the flow in all cases was pure meridian flow without any element of whirl,the results cannot be considered as applying to the draft tubes of high speed turbines. A long straight draft tube with a small angle of diffusion,like the tubes tested by Mr.Lyon is probably the most efficient design,but for large units the construction of such draft tubes would involve very costly excavation,especially with runners of high specific speed,which often have to be located very close to tailwater elevation. To overcome this difficulty,the new forms of draft tube were proposed. Those by Moody and White will be very briefly described, together with the form of draft tube invented by Dr.Kaplan, which it is claimed,was the first type originated involving the principles upon which the other two forms also operate *.

The Moody Spreading Draft Tube. Assuming that the water leaving a high speed runner behaves to a very great extent like a free vortex,we may find the laws applicable to its flow by a consideration of equation (iv) page 50 , and this equation is used by Mr.Moody in the discussion of the principles upon which the spreading draft tube operates. Taking Fig. 67 to represent a free vortex formed in an open basin,neglecting the components of velocity in the meridian plane and assuming for an approximation that the velocity is entirely in the circumferential direction we shall have†

$$v = \frac{K_1}{r} \text{ - - - - - - - - - - - - - - -(i)}$$

since in a free vortex the velocity varies inversely as the radius, r being the radius and K a constant. If z is the ordinate to the same point,then since the pressure at every poiny is atmospheric,and since therefore the velocity head must increase by the same amount that the elevation head decreases,we shall have

$$z = \frac{v^2}{2g} \text{ - - - - - - - - - - -(ii)}$$

Combining (i) and (ii), we have

$$r^2 z = K \text{ (a constant) - - -(iii)}$$

*See paper by C.Reindl,referred to on page 58 .
†Discussions taken mostly from Moody,"The Present Trend of Turbine Development", Engineers Club,Philadelphia, 1921.

This is the equation of a third degree hyperbola, the same as equation (iv) page 50, which is the equation to the surface curve of a free cylindrical vortex and which agrees very closely with that of a free spiral vortex. The whirl component of velocity in the water creates a surface of discontinuity, maintaining an open space through the centre of the vortex; equilibrium being maintained between the increased centrifugal force and the increasing pressure. This is maintained since if the flow approach the axis more closely the velocity would be required by the relation above to assume infinite values. When the space within the surface of discontinuity is filled with water it is likely that this water is set into an eddying condition without partaking of the general flow of the surrounding stream. The principle just explained suggests a useful method of regaining the kinetic energy of the whirling component of flow in the water discharged from a turbine runner, or a pump impeller. For example, if the flow is turned into an axial direction, away from the axis, the velocity of whirl will diminish in inverse proportion to the increasing radius, and the corresponding velocity head will diminish inversely as the square of the radius, so that it is merely necessary to lead the water a moderate distance away from the axis of rotation to obtain the conversion of a large proportion of the velocity head of whirl into pressure head. This principle is used in the spreading draft tube. The above is the inventor's explanation of the action of the spreading draft tube ; one of these tubes is shown attached to the Queenston turbine, shown in Fig. 38. Another form of the tube is shown in Fig. 52, attached to the high speed turbine of the Great Falls Plant, Manitoba. This form is provided with the central core, to prevent a too rapid increase in the area of the draft tube section so as not to disturb the stability of the flow and also to prevent the formation of a surface of discontinuity, and the presence of a region of eddies. While providing for the recovery of the whirl components of velocity head, the tube must be provided with a gradual increase in area so as to regain as much as possible of the meridian component of velocity. In the discharge from high speed runners there is a tendency for a region of low pressure to form at the centre of the runner, and tests have revealed the fact that reverse flow up the centre of a tube actually occurs in certain conditions. The central cone is designed to prevent this reverse flow and the attendant disturbance created. The principal advantage of this draft tube is that it helps to maintain the efficiency of the turbine at part gate, (Fig. 72).

The White Hydraucone Regainer. In a very excellent paper* recently published, Mr. W. M. White gives what he believes to be the principles upon which the draft tube (Fig. 71) originated by him, works. This paper also gives much valuable information concerning other forms of draft tube, pointing out the defects of the ordinary curved type, and also contains the results of much experimental work. The following is, in brief, the principle given for the operation of this type of draft tube:

*W. M. White, "The Hydraucone Regainer, Its Development and Application in Hydro-Electric Plants." Proc. A.S.M.E. May, 1921.

"The new method of regaining pressure from velocity of fluids in motion consists in causing the stream flow to impinge upon some definite shape,either flat,conical or concave,thus changing its direction,and then placing an envelope around the shape so formed upon the particular base used,which envelope conforms to the shape of the fluid at entrance and gradually recedes from what would be the normal or free shape of the non-enclosed fluid impinging upon the particular base used; the effect of this gradually diverging envelope being to change the velocity head of fluids flowing at high velocity into its entrance into pressure and low velocity at exit."

This change from high velocity to pressure is brought about by what the inventor of this form of draft tube refers to as the "hydraucone action" of water, and this is explained as follows: "By the "hydraucone action" of water is meant that action which occurs when an unenclosed stream impinges against and is deflected along a surface" (Fig. 68) and again,when discussing this action he says -- "It will be noted that the water at the centre of the jet slows up as it reaches the plate and delivers its velocity head into pressure head at the centre of the plate. This pressure head at the centre is transformed into velocity as this stream flow passes from the centre of the point of impingement to the point beyond the radius of curvature. Referring to Fig. 68 ,the radius of curvature of the water in making its change in direction is a definite curve and is the minimum which can be used in regaining devices and yet obtain the reactions within the point of impingement which will give smooth stream flow of discharge along the plate and consequently without eddies and attendant losses."

A model of a water-wheel was tested,using the new device, and better efficiencies were obtained than those possible with the curved models of tube. Devices of the form shown in Fig. 65 and Fig.66,were tested, in an arrangement as shown in Fig. 69 . The flow through the tubes in these tests was in all cases pure meridian flow,having no whirl component. The results of one of the tests,among the best given,on a model as shown in Fig.69 ,hydraucone radius $2\frac{3}{4}$" are as follows:

Distance between plates,ins.	1/2	17/32	9/16	19/32	5/8	3/4	7/8
Efficiency,per cent.	76.7	77	78.5	78.7	78	74	68

A series of tests using what was termed an "equal area cone" hydraucone radius $2\frac{3}{4}$" Fig. 70, gave the following,

Distance between plates, ins.	1/2	9/16	5/8	11/16	3/4	7/8
Efficiency,per cent	71.3	72.4	72.8	72.4	70	64.7

Such efficiencies of regain might reasonably be expected, since G.E.Lyon * ,from his tests on straight conical tubes obtains the efficiencies ranging above 90 per cent.

The tube gives better efficiencies in actual operation than can be secured with curved draft tubes or straight tubes of similar length,but the inventor's explanation of

*Mechanical Engineering, April,1922.

its action is no doubt incorrect. His so-called hydraucone action of water has been long understood and may be explained by the application of the principles of relative velocities of water impinging on a flat plate, which was discussed in the early part of the present work. (See also Gibson "Hydraulics", second edition, sect. 106, page 374)." The phenomenon of the variation in efficiency as the distance between the plates is varied may be satisfactorily explained by a consideration of Sect. 32, p. 79 of Gibson's Hydraulics, where the case of a jet impinging on a flat surface is considered when the plate is brought gradually closer to the orifice until the escaping jet touches both the plate and the nozzle. In the writer's opinion the action of the hydraucone regainer is not fundamentally different in principle from the spreading draft tube by Moody, the plate at the bottom of the tube preventing the counter currents in the tube which have such disturbing effects in draft tubes handling water with very large components of whirl velocity. A cone of water is then formed above this plate corresponding to the central cone of the other design of tube. Mr. Chester W. Larner's discussion of Mr. White's paper helps to throw much light upon this very difficult subject; he gives, in part, the following:

"The author claims to use as the basis of his draft tube design the shape of a free jet impinging on a plate and he says it is necessary to take into consideration the action of this free jet as it strikes the plate if the maximum recovery of velocity head is to be expected. This statement, in one form or another, is reiterated frequently throughout the paper and it appears obvious that the author regards the analogy to a free jet as of fundamental importance. This statement is open to criticism for the following reasons;

1. A free jet presupposes straight-line flow, which does not occur in a draft tube. If whirl is introduced into a jet, the jet is instantly dispersed into spray.

2. The object of a draft tube is recovery of velocity head and there is no recovery of velocity head in a free jet. Fig. 68, in which the velocities before and after impingement are alleged to be the same, indicates that the author appreciates this fact and yet, at the same time, he says that this illustration discloses the principle of the new method of regaining pressure from velocity.

3. The author does not adhere to the free jet principle in working out his own design. He begins to taper his draft tube at the top (See Fig. 71) and diverges more and more from the shape of the free jet as he progresses down the draft tube and around the turn. In so doing he utilizes the diffuser principle which is, after all, the only abstract principle actually involved in good draft tube design. The problem of design are manifold, but efficient recovery of velocity head is purely a question of reducing the velocity with as little loss as possible. The ideal draft tube is one in which the areas normal to the flow are gradually and progressively enlarged and so shaped and disposed that the velocity and distribution of flow conform to the area provided."

There appears to be no great differences in the efficiencies obtainable with the two forms of draft tube as discussed above when operating at normal load,and this fact is very well brought out in a paper recently published by Mr. N.R.Gibson* when testing out his method for the measurement of the flow of water in pipes in determining the efficiencies of the three new units at extension No.3 of the Niagara Falls Power Company. Unit No.16 has a White Hydraucone Regainer; units Nos.17 & 18 are equipped with the Moody Spreading Draft Tube. It would appear,however,that at part gate the spreading tube with the central cone gives greater efficiencies than the regainer,as described here. This is probably due to the former's capacity for more efficiently taking care of the whirling components of velocity. Mr.H.B.Taylor,discussing Mr.White's paper has to say "The part gate efficiencies of units 17 & 18,which are equipped with Moody's tubes, are considerably higher than the corresponding efficiencies of unit 16,equipped with the White tube,the difference at part gate amounting to five per cent.,and although a part of this difference can be accounted for by the fact that the runner used in unit 16 was designed for a somewhat higher power capacity than the runners in units 17 & 18,this difference may be allowed for by a relative shifting of the efficiency curves (Fig. 72) bringing the results together at high loads and still leaving a difference in favour of the Moody tube of approximately three per cent. at half load."

Referring to the curves given in the paper by Mr.N.R. Gibson,there is a decided advantage in efficiency in favour of units 17 & 18,over unit 16 at part gate but the latter possesses the advantage at full load; it is difficult to say whether the difference in part gate efficiency is due entirely to the form of draft tube used.

The Kaplan Draft Tube. The draft tube originated by Dr.Kaplan is discussed in the paper by C.Reindle (See page 58); the objects sought as therein stated are identical with those already stated in the preceding pages,namely,to let the discharge current from the runner adjust itself in a natural way along the line of least resistnace,free from external coercion; to let it spread at a bend,filling perfectly the section without detachment from the walls,as this so often happens with conventional forms of draft tube, despite the care taken in design and dimensioning. Figs.61 and61-A show the general form taken by the Kaplan type of draft tube,being the form adopted at the power station of Velm. The explanation of the action of this tube is given as follows:

"The water in the vertical part,which is of rectangular section,is bent into the horizontal part,which Kaplan calls the diffuser,according to the priciple that the radius of the bend on the outside must be smaller than the radius on the inside in order to force the water to spread over the enlarged section and compeletely fill the section. The same considerations hold good for the change in direction from the runner casing to the entrance of the draft tube,this is

*N.R.Gibson,"The Application of a New Method of Water Measurement in the Efficiency Tests of the 37500 H.P.turbines of the Niagara Falls Power Company" Engineers Club of Philadelphia,1921.

characteristic of the draft tube of the Velm plant (Fig.61)."

The preceding is a general statement explaining the action of the Kaplan draft tube; however,nothing is said about the hydraulic principles involved and no analysis is attempted. To follow the water in its course,we first have the water as it leaves the turbine casing (Fig.61) impinging against the flat outside wall of the elbow and spreading out; the water then follows the vertical portion of the tube downward and again impinges on the bottom of the horizontal portion and then spreads horizontally to fill this portion as it moves horizontally outward. No test results are given,or other information concerning the efficiency of this form of draft tube,but it is stated that,when measurements were made upon the discharge from the draft tube of the Velm plant,it was found the water emerged from the end of the tube with a velocity which was uniform over the whole section.

Other Forms of Draft Tube. When discussing the Queenston Hydraulic Power Development,the curved draft tube (Fig.41) supplied to unit No.1,was mentioned. No information concerning this form of tube is available,but it is a matter of considerable interest that a tube very closely resembling the recently despised and rejected form, should have been honoured by use in so important a plant; we may readily percieve that its simplicity in construction would recommend it for use,but high efficiency is one of the main considerations in the design of the Queenston plant and hence it must give promise of good efficiency also.

A draft tube provided with steel fins on the inside, projecting about eight inches into the tube and extending downward six or eight feet below the runner is mentioned by Mr.G.S.Williams in his discussion of the paper on draft tubes by Mr.W.M.White. The object of these fins is to check the whirl as the water leaves the runner;and it is claimed that a gain in efficiency of about 2 per cent was effected over that secured with the form known as the hydraucone regainer. Up to the present the idea does not appear to have been further developed.

Overcoming the Vertical Thrust in Hydraulic Turbines.

The solution of the thrust problem in hydraulic turbines has contributed fully as much as any single factor in the development of large capacity units of high efficiency; in fact,on looking into the problem for a little,it would strike one that these large units could not exsist without the present day devices for taking the weight of the revolving parts of the generator and turbine. Upon this point we may refer to the opinions expressed by several well known hydraulic engineers and turbine designers. Mr.Van Patter, in the Engineering Journal,Sept.1922,p.461, made this statement; "As most of you are aware,the tendency in the hydro-electric field during recent years has been towards larger and larger units of the vertical shaft,single runner type. This has been made possible by the development of the direct connected hydro-electric unit and the satisfactory solution of the thrust bearing problem."

When describing Extension No.3 to the plant of the Niagara Falls Power Company, Mr.Geo.R.Shepherd,of that Company,stated, "The experience of the power Company in former developments led to the adoption in 1906 of horizontal units for that part of Station No.3 which was then under construction. But in 1916 when most of the preliminary engineering on the present extension was undertaken thrust bearings had reached such a state of perfection that eliminated the engineering problem in connection with the supporting of the weight of the rotating parts of the unit. These mechanical difficulties having been eliminated,the efficiency of the units became the deciding factor,and the vertical units were chosen as offering the best opportunity for the development of a high efficiency."

Mr.Eric Crewdson,discussing turbine design (see paper referred to,page 21) has said,"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."

The solution of the thrust bearing problem has been made possible by the extension of the knowledge of the theory of lubrication; and this knowledge has had a gradual growth extending over the latter part of the last century. Beauchamps Tower carried out an extensive series of experiments on lubrication and friction during the early eighties for the Institution of Mechanical Engineers. These experiments were performed with a well fitted cylindrical car-axle and brass bearing,flooded with oil,and it was found that there was practically no wear,that the total friction was nearly independent of the load,and that the mean coefficient of friction was very low,in some cases not exceeding 0.001. He further found that there was oil between the journal and the brass under pressure which varied from point to point on the bearing surface,the maximum pressure being roughly twice the mean pressure,and the integrated pressure being equal to the total load on the bearing.

Professor Osborne Reynolds* was led to make a theoretical investigation of the principles of lubrication from a consideration of the results obtained by Tower,basing his theoretical work on elementary physical data,including the journal diameter,its speed of rotation,the dimensions of the bearing surface of the brass,the viscosity of the oil and the load on the bearing. He was able to calculate the friction and what the pressure should be at any point in the oil film,and his calculated results agreed fairly well with Tower's experimental ones. He showed that the oil, because of its adhesion to the journal,and because of its viscosity or resistance to flow,is,by the rotation of the journal,dragged into a wedge-shaped space between it and the bearing. This action sets up the pressure in the oil film,which,in turn supports the load on the brass bearing. It completely separates the bearing surfaces,so there is no metallic contact. This wedge-shaped film of oil was shown by Reynolds to be the absolutely essential feature of effective automatic lubrication. Without it no great load can be borne on the bearing except with the accompaniment of high friction. The object sought in bearing design is to provide for this protective film of oil; and so long as dirt or grit are absent the bearing will be very unlikely to fail if the oil film is not squeezed out by the development of very high pressures on very restricted surfaces. Upon this point Mr.H.G.Reist† has to say: "The pressure necessary to accomplish this (squeeze out the oil film) is much greater than is generally known. In one case that came to the writer's attention a pressure of 5000 pounds per square inch was carried for several hours and the babbitted surface was absolutely free from damage. In another case a pressure of 2000 pounds per square inch at a rubbing speed of 4500 feet per minute was successfully carried. The consideration of such experience led to the conclusion that damage to properly lubricated bearings was due to failure of the oil film on so small a part of the total surface that the unit pressures on these surfaces exceed the values just mentioned."

Based upon Reynolds' wedge-shaped film theory of lubrication there have been three thrust bearings brought out in recent years for application to hydro-electric units. These are the Kingsbury (operating on the Michell principle)thrust bearing of the segmental block type; the Reist spring thrust bearing; and the plain groved plate bearing developed by Gibbs. Each of these bearings will be described briefly in turn.

<u>The Kingsbury Thrust Bearing.</u> There has been much controversy as to whom the honour for the invention of the segmental block thrust bearing should go; whether to Kingsbury,in America,or to Michell,an Australian. The facts,however,seem to be about as follows: Based upon

*Phil.Trans.,1886.

†H.G.Reist "A Self-Adjusting Spring Thrust Bearing",Proc. A.S.M.E. June 1918.

a study of Osborne Reynolds' thesis of 1886, Michell developed his thrust bearing during the years 1902-3-4 and he published an account of it in the German periodical Zeitschrift fur Mathematik und Physik, in 1905. He patented his device in Great Britain and Australia in January, 1905 but took out no foreign patents, on account of insufficient financial means. Kingsbury filed his application for patent two years after Michell; and although Kingsbury had been working on the subject, the scientific development of the thrust block with the tilting pieces (Figs. 73 and 74) was first worked out by Michell.* Kingsbury's claim to priority rested on the fact that under the United States law a citizen of the United States was allowed two years priority over any foreigner, provided he could prove that he had been experimenting on his invention, and Kingsbury was able to show that he had done so.

The principle of operation of the vertical thrust bearing of this type may be studied by the aid of Figs. 74 and 75 ; a plain collar forms one bearing member and pivoted segments form the other. The arrangement of the Kingsbury bearing and of the Michell bearing is practically identical, the object being to induce a film of the lubricant to pass between the plane collar and each of the segmental shoes, thickest where it entered between the shoe and the block at the front, and tapering off like a wedge to the back or trailing end of the shoe. One difference between the two forms is that the Kingsbury is sometimes constructed upon a spherically shaped adjusting collar to secure perfect alignment. Another difference is that in the Michell bearing the supporting pivot is placed behind the centre of the segmental shoe, about one third from the trailing end, while that of the Kingsbury is placed in the centre. It is stated by certain authorities that this distinction makes no practical difference to the formation of the wedge-shaped film of oil. The blocks are pivoted so as to be free to tilt both radially and tangentially so as to adjust themselves to operating conditions. The Michell thrust bearing was adopted by the British Admiralty for use in the Imperial Navy and has revolutionized boiler room arrangements. It is now extensively used in Marine work. The Kingsbury thrust bearing has been extensively used for taking the weight of the revolving load in hydro-electric units, some noted installations having already been mentioned. Fig. 76 illustrates the thrust bearings of Kingsbury manufacture supplied for three of the 55,000 horse power units of the Queenston development of the Ontario Hydro-Electric Commission. The thrust load is almost 1000000 pounds and the power consumed in the thrust bearing 85 horse power or o.16 of 1 per cent of the power per unit. Two of the

*In addition to the reference on page 72, the following contain the principles of operation of the Michell thrust bearing: "Theory of Michell Thrust Bearing", Engineering Feb.20, 1920; "Michell Bearing, Lubrication Tests," H.T.Newbigin, Engineering, Sept.1, 1922; "The Correct Interpretation of Michell Thrust Bearing Experiments," Engineering, Jan.5, 1923.

Queenston units have Reist spring thrust bearings; the last two units of the Cedars Rapids plant are also supplied with the Reist type of thrust bearing. The first ten units of the Cedars Rapids plant were supplied with Kingsbury thrust bearings;the thrust load is 550,000 pounds,and the power consumption in the thrust bearing is about 10 horse power per unit,being about one tenth of one per cent. of the output of the generator. In this installation it was found necessary to insulate the thrust bearings to protect them from the destructive effects of circulating current that was found to be present in these machines. The thrust bearings are located above the generator directly in the path of this current. Many notable hydro-electric power stations are supplied with this type of thrust bearing,and the power consumption in the bearing is in all cases very low.

The Reist Spring Thrust Bearing. It is extremely difficult to make a perfectly plane surface without slight irregularities,and also very difficult to form two surfaces which will turn upon one another and be in perfect contact. Such surfaces can be satisfactorily constructed for small bearings for moderate loads,but as the sizes and the loads to be carried increase the difficulties of construction and operation increase very rapidly,since the deflections are liable to be greater in the surfaces,and if there is not perfect conformity,the local pressures developed are liable to be very intense. Though the bearing members of thrust bearings are usually scraped to each other to avoid dangerously high spots,yet the thickness of an oil film is very small,in the order of 0.0002 to 0.0003 inch ,and therefore the irregularities must be less than these values. Such careful fitting must be done without load,and when the load comes on,the surfaces will be deflected somewhat,no matter how rigid,and perfect contact will not be secured.

It was to overcome these difficulties that Mr.Reist proposed a thrust bearing consisting of a solid plate and a flexible plate supported by springs (Fig. 77)*. The provision of the flexible plate avoids the occurance of unduly high pressures at any point with the attendant destruction of the oil film. Oil groves are provided in one of the members and sometimes in both; the oil from the oil bath entering these groves is induced to form a wedge-shaped film as the motion of the rotating member drags the oil around in between the plates. The stationary ring resting on the springs is of steel and has a babbitted surface; the rotating ring is made of a special grade of cast iron. Very high unit pressures are used in the design of these bearings,being in the neighbourhood of 300 to 400 pounds per square inch; as contrasted with the 70 pounds per square inch permissible for the ordinary collar type of thrust bearing. This type of bearing is becoming very popular; it was used for Units Nos.11 and 12 of the Cedars Rapids development,† for two of the 55,000 horse power units of

*H.G.Reist,"A Self-Adjusting Spring Thrust Bearing," Proc.A.S.M.E. June,1918.

† See General Electric Review,Nov.1919.

the Queenston plant,and many other important plants.

The Gibbs Solid Groved-Block Thrust Bearing. The thrust bearing as developed by Mr.Gibbs (Fig.78) consists of three principal elements; namely, a rotor ring, a stator ring and a levelling ring,enclosed in a casing and submerged in oil. It operates on the principle of the wedge,in the following manner: The stationary ring or stator has (depending on the size of the ring) four or more radial groves across the bearing surface dividing it into a corresponding number of segmental sectors. Each sector face has a definite portion flat,and the remaining part of the sector has a gradual taper or level to the radial groves. The circumferential width of the face and the depth of the taper face depends on the unit pressure on the bearing face and the speed of the rotor.

The stator ring,for low and medium pressures up to 300 pounds per square inch,is made of close grain cast iron, and is generally made in one peice,except in some cases where it is necessary to make it in halves so that it can be removed without disturbing the shaft or other parts attached to the shaft. The bottom face of this ring is made spherical to fit the spherical seat of the levelling ring,and is connected to the levelling ring by means of a dowel pin,so as to allow the stator ring to have a limited amount of adjustment.

The revolving ring,or rotor,is made of cast iron,on to which is placed a soft metal face,(babbitt) and is perfectly flat. When the rings are placed in normal position (the rotor ring on the stator ring) there will be a series of flat faces with alternating wedge surfaces,which,when the bearing is at rest,are filled with oil.

When the rotor ring is rotated,it pulls in the oil,by adhesion,from the radial groove,up the wedge surface,it also carries the oil across the flat surface of the stator ring. In drawing the oil up the wedge surface,the rotor ring automatically develops a pressure between the rotor and stator rings which equals the total load on the bearing. In operation this form of bearing has been found to be extremely satisfactory,the spherically shaped seating of the stator ring takes care of any adjustment necessary to secure perfect alignment,and the bearing has been successfully operated at speeds up to 5000 feet per minute at the periphery of the rotor without any detrimental effects to the bearing. This is an extremely simple and durable bearing,well adapted for medium thrust loads,up to about 220000 pounds; and it will probably always be confined to thrust loads of this magnitude,on account of the difficulties arising with the use of solid bearing members of very large size,as pointed out on page 74 .

Eugene U.Gibbs, "Thrust Bearings," Canadian Engineer, December 26,1918.

OPERATION AND PROTECTION.

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It has been attempted in the preceding pages to trace the advancement in the development of water power from its earliest beginnings down to the most recent achievements; and also to pass under review certain proposals which may become useful in solving difficulties yet unsurmounted, proposals which have not yet been extensively tried out or used in practice. In addition to the improvements in the water-wheel itself and its settings there has been marked progress in all the allied branches of engineering,improving the facilities for power plant operation and protection. These improved facilities,some of which it is proposed to review in this closing chapter,are as follows:-

(a) Improvements in the governing devices of the impulse and reaction water-wheels; in the case of the reaction wheel this has been made possible by the invention of the differential surge tank,eliminating dangerous and disturbing surges which develop in penstocks of any considerable length. This improvement in the speed regulation has made it possible to deliver electrical energy at a more uniform voltage and has facilitated the tying in and synchronizing of several power plants in a vicinity,improving the load factor and eliminating waste of power.

(b) The improvement of protective devices for transmission lines and the perfection of insulating materials, making possible great increases in the transmission line voltages. Line voltages of 220,000 volts are now quite common and the General Electric Company have successfully transmitted current at 1,000,000 volts in their laboratories. These increased voltages effect a very great reduction in the line losses.

(c) The introduction of devices for the measurement of the discharge of a water-wheel facilitating the testing and rating of the wheels.

(d) The protection of power plants from the action of anchor and frazil ice.

Speed Regulation of Reaction Water-Wheels. When describing impulse water-wheels,the governing devices for these wheels were discussed at some length,but those of reaction water-turbines have not hitherto been mentioned. However,on account of the general uniformity in type of the governing devices for the latter type of wheel they will not require much space here. The essential parts are the fly-balls,the valve arrangement for controlling the supply of governor liquid to the pressure cylinders, the pressure cylinders for applying the energy for operating the turbine gates,and some means of supplying the governor liquid at the desired pressure. The governor liquid may be either water or oil; if water is employed it is sometimes taken direct at the penstock pressure and delivered to tailwater after use,but this method is not extensively used, on account of the corrosive action on the metal parts with which the natural water comes in contact. The more usual practice is to supply the waterby means of pumps at the desired pressure,the water being treated with potassium bichromate and delivered back to the suctions of the pumps after use to prevent waste. There has been considerable discussion upon the relative merits of oil and water as a governing fluid,but it would appear,however,that oil is much more satisfactory in operation,and is also more economical in the long run than is water,except for the very largest installations.

In the governors for smaller units,the fly-balls are usually driven by a belt from the turbine shaft,but for units of any considerable size the fly-balls are invariably mounted directly upon the turbine shaft; and for all units except those of quite small capacities the pressure cylinders are mounted so as to operate directly upon the shifting ring of the turbine gates,wicket gates being almost universally employed.

The problem of hydraulic turbine regulation involves not only a consideration of the pressure rise in the penstock as load is suddenly thrown off the turbine and water is rejected by it,but also a consideration of the pressure drop in the penstock when load is suddenly applied to the turbine and its requirements of water increase. For getting rid of the excess pressure in the penstock due to a sudden reduction of load,there are two possible arrangements:-

(a) The synchronous by-pass. This is a valve which is capable of discharging the full capacity of the water-wheel and which is so connected to the gate mechanism as to open when the wheel gates close and discharge the water rejected by the wheel or vice-versa. With this device the conduit velocity is constant regardless of the load and hence instantaneous load changes can occur without affecting the pressure in the conduit. The synchronous by-pass is rarely used because it wastes water at all times except when the wheel is carrying ~~at~~ full load.

(b) The surge tank. This is a tank or reservoir rising above forebay level and connected to the conduit at a point as close as possible to the plant. The tank is elevated on a tower,built on the ground,or excavated in the ground,depending on the elevation of the site relative to the forebay level. If the ground adjacent to the power house does not rise abruptly,it may be necessary to use a closed tank located at the power house with air pressure on top of the water. The surge tank receives the rejected flow when load is thrown off the plant and supplies the additional demand when load is thrown on.

When discussing the speed regulation of hydraulic turbines Mr.Raymond D.Johnson summed up the function of a surge tank as follows:-*

"The function of a properly designed surge tank is complex. It may be said to have six distinct duties:

(1) To regulate the pressure,preventing undue rise

*"Speed Regulation of Hydraulic Turbines," Raymond D. Johnson, Symposium at Engineers Club of Philadelphia,1921.

or fall following sudden motion of the wheel gates.

(2) To act as a reservoir furnishing water promptly to the wheel when demand is made for more, thus taking care of the time element in which the water in the long conduit may be accelerated.

(3) To lengthen the period of oscillation of a surge so that the governor may prove fast enough to follow it with the gate motion, in order to keep the power output of the unit at constant value.

(4) To damp out this surge vibration in spite of the augmenting effect of the governor action, so that the vibration of pressure will not continue to increase after having once been started in oscillation.

(5) To furnish sufficient internal resistance to accomplish this damping effect without depending upon excessive friction in the water system itself which would also assist in this duty.

(6) To conserve water which would otherwise be wasted in overflow or through a by-pass."

The practice was formerly to employ a tank of the form of a simple stand-pipe, the size of the tank varying with the capacity of the unit; with units of moderate size very large tanks were necessary in order to reduce the amplitude of the surges in the water and in even the largest tanks dangerous surges could not be avoided if the load changes became approximately synchronous with the period of oscillation of the surge. To overcome these difficulties the differential surge tank was brought out; a tank provided with an interior pipe or riser, the water being forced to pass by properly proportioned ports on entering or leaving the main tank, thus damping out surges which tend to develop. A very full discussion of surge tank design is contained in the papers included under the heading "Literature" on page 83.

<u>Devices for the Measurement of the Flow in Penstocks.</u> There are at present two methods for measuring the flow of water in penstocks for the determination of the efficiency of hydraulic turbines which are receiving considerable attention; these are known as the Gibson method and the Allen method, after the names of the originators. The Gibson method makes use of the recorded pressure rise in the penstock as the turbine gates are gradually closed as a means of estimating the mean velocity of flow; the pressure rise taking place in accordance with the well known laws of momentum and energy changes. The mean change of pressure recorded when a body of water in motion is brought to rest or its velocity reduced is a precise measure of the velocity change. The apparatus consists of a U-tube containing mercury, which records the existing pressure at any instant in the pipe to which it is connected, it being possible to have the top of the mercury column move in any desired ratio to the change in pressure in the pipe. A photographic apparatus records continuously the elevation of the top of the mercury column and at the same

time keeps a record of the oscillations of a seconds pendulum. The resulting diagram shows to exact scale the complete record of pressure changes and time. The use of the apparatus in the determination of the turbines of Extension No.3 of the Niagara Falls Power Company is described quite fully by Mr.Gibson in the paper given before the Engineers Club of Philadelphia,in April 1921.

The operation of the Allen method depends upon the change of the electrical conductivity of water due to the presence of a dissolved salt. Electrical leads are extended into the pipe in which the water is flowing,and the salt solution is then introduced in the upper end of the pipe. The arrival of the solution at the leads is indicated by an increase in the current flowing; a traverse of the pipe being made in this way with the observed times required for the solution to travel the distance between the point of admission and detection enables the mean velocity of flow in the pipe to be determined.

Protection of Power Plants from Anchor and Frazil Ice.

Almost everyone is familiar with the sheet of surface ice which forms on all bodies of water in clod climates; this kind of ice does not,however,create a serious problem in the operation pf water power plants. It is likely to become a menace only during the spring break-up and the plant can always be protected from it by the construction of booms, either of the floating or fixed type. The design of the intake for the Queenston* development represents the latest advances in the protection of power plants from ice in this form.

There are,however,two kinds of ice which do not always float upon the surface; they are called,(a) Anchor ice,and (b) Frazil. These two kinds of ice create the most difficult problems in the protection of power plants.

A very large amount of our present knowledge of the formation is due to the patient study and investigations of Dr.Barnes,late professor of Physics at McGill University. In his book entitled "Ice formation" he discusses the relation of temperature to the formation of ice in the streams of northern latitudes where winter conditions prevail unbroken from fall to spring; how the water in a flowing stream always remains exactly at the freezing point,at 32° F or 0° C. By means of a microthermometer he was able to measure the change in temperature at 32° F which would result in changing water to ice. In this connection he states:

"It only requires a few thousandths of a degree change in temperature in a stream to stop the operation of the largest water power development,or to dam a mighty river and divert it from its course.

Nowhere can one find a more wonderful example of the delicate poising of the forces of nature,since within the limits of so small a fluctuation such tremendous physical effects are produced."

When water is at this critical temperature we see that the slightest displacement of the state of equilibrium

*T.H.Hogg "General Features of Design,Queenston-Chippawa Power Development." Paper presented at Hydro-Electric Conference,Philadelphia,1922.

results in the formation throughout the body of water of minute spicules of ice,known as frazil,which become suspended in the water in a fine feathery condition. The tendency of this feathery mass is to float,but it is so light that it is carried in any direction by the slightest currents in the water. When it comes in contact with solid bodies which are slightly below 32° F it clings to them and a very large accumulation of ice very rapidly results,rising the crests of dams and blocking sluce gates causing floods; and also blocking the intakes of water works and the head gates, wheels and penstocks of water power plants.

When the water of a stream or other body of water is at the critical temperature,ice,known as anchor ice,forms at the bottom upon dark objects,such as boulders,which have lost their heat by radiation,it being a well known physical fact that dark bodies receive and deliver up their heat by radiation more readily than do bright and slimy ones. The anchor ice remains attached to the bottom of the stream until the sun's rays become of sufficient strength to melt it by radiction or the water flowing in contact with it enjoys an increase in temperature. Agitation increases the disposition of water at the critical temperature to turn to ice.

From a study of Dr.Barnes' experiments Mr.John Murphy * conceived a plan for protecting engineering works from the action of ice by raising the temperature of bodies with which the water had to come in contact just a minute fraction of a degree above the freezing point. In an address before the Ottawa branch of the Engineering Institute of Canada† Mr.Murphy gave some interesting facts concerning protection from ice,and he told how in the early days of his work on ice prevention he had been derided for proposing to melt ice and warm water for running water power plants. No ice melting or water warming is required in the process,it is merely necessary to keep the parts which come into contact with the water one - thousandth of a degree above freezing to prevent ice adhering to them. He said "One ton of coal has prevented a 3,000 horse power plant from being shut down, while it required 150 tons of coal per day to keep a steam plant of the same size running."

Writing in Electrical World, April 2,1921, Mr.Murphy made this statement, "Between the continuous operation of a hydro-electric plant and its shut-down because of frazil ice formation,there lies only one-thousandth of a degree of heat."

Various expedients are employed in practice for applying the principles as outlined above. Metal articles to be protected are often made part of an electrical circuit, the resistance of the metal causing sufficient heat to be generated in order to raise their temperature the required fraction of a degree. Jets of steam discharged near the object often suffice. In the gate house of the new 41,000 horse power turbines at Shawinigan Falls the racks are protected simply by a blast of warm air extracted from the

* Electrical Engineer,Dept.of Railways and Canals,Ottawa.
† April 1920.

generator room by means of a fan,warmed in a heater to a temperature of 150° F and blown down against the racks. In an "Ice Album" presented to the Engineering Institute of Canada Library,Mr.Murphy gives an account of his experience with plants having wooden and metal racks in the head race. It was found that those having metal racks suffered more severely from ice-attacks, than those equipped with racks of wood,due to the greater heat conductivity of metal,resulting in the surrounding air,at temperatures below freezing,extracting the heat from the metal bars and conducting it away.

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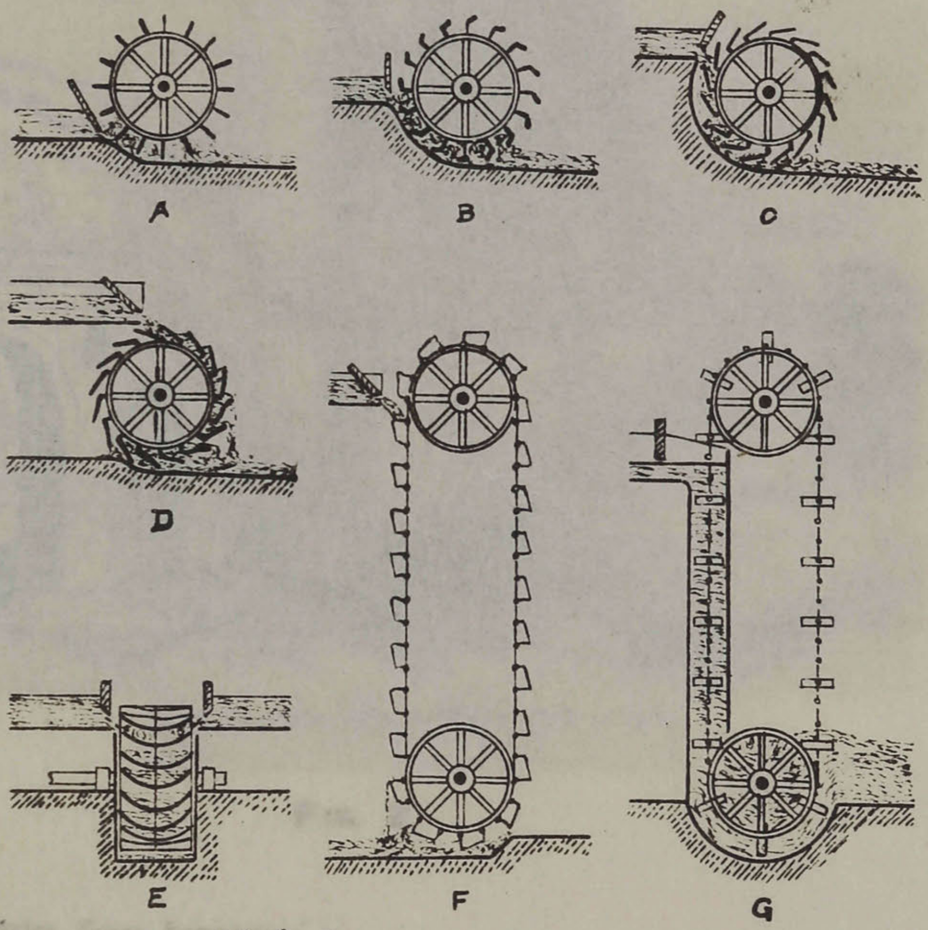
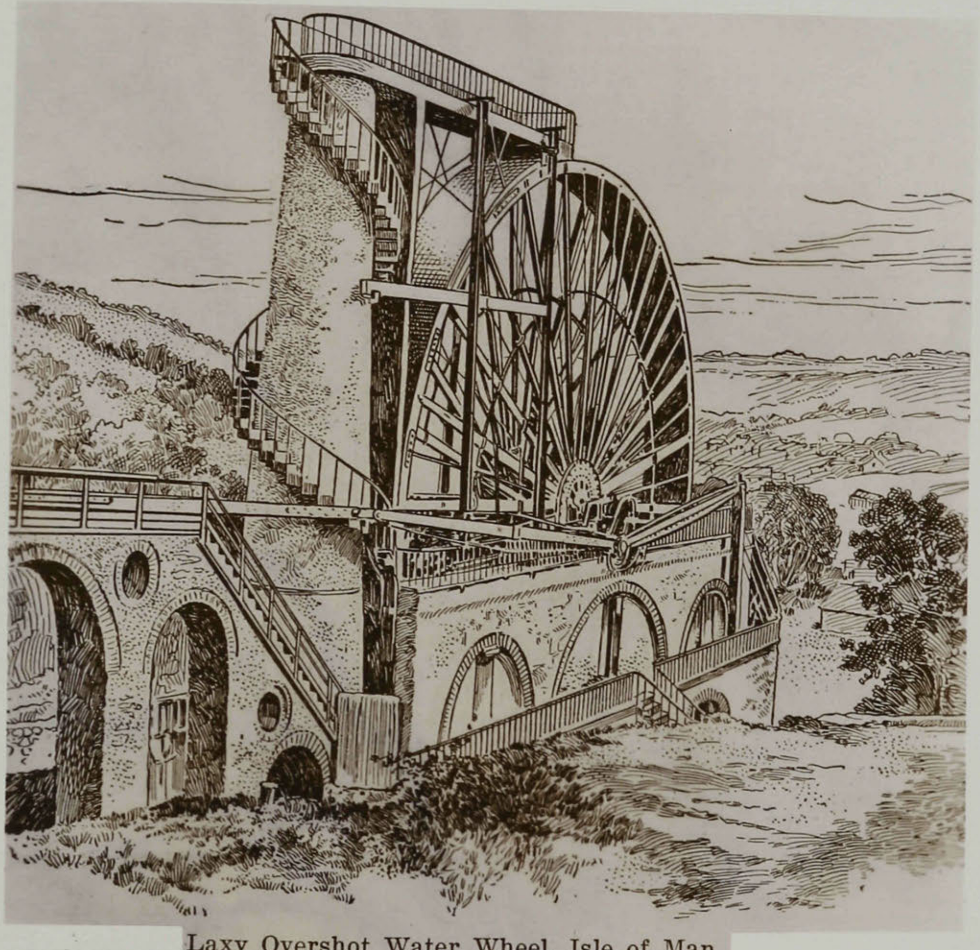


Fig 1. DIAGRAMS OF GRAVITY WHEELS.

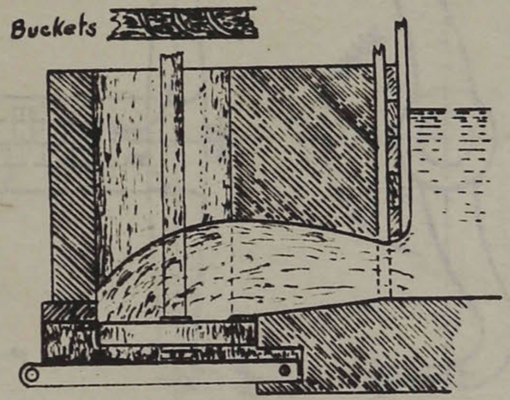
Mead "Water Power Engineering."



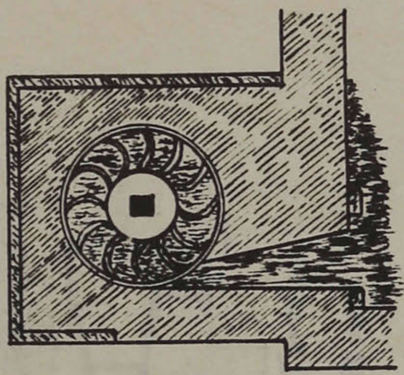
Laxy Overshot Water Wheel, Isle of Man

FIG. 2.

From Mead "Water Power Engineering"



Elevation.



Plan.

Fig 3. TUB WHEEL
(ROUE À CUVE)

A.T. SAFFORD "TRANS. A.S.C.E, 1922."

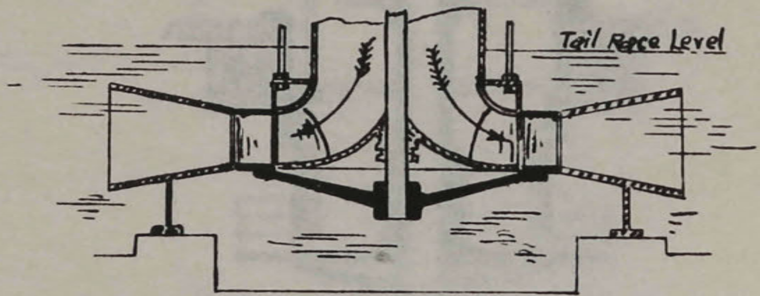


Fig. 5 Outward Radial Flow Turbine with Boyden Diffuser.

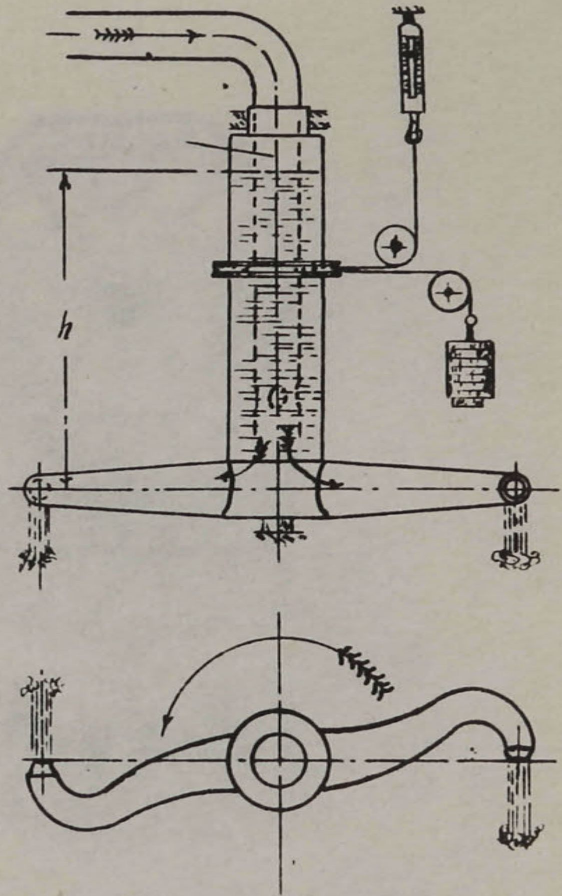


Fig. 4. BARKER'S MILL.

GIBSON "HYDRAULICS."

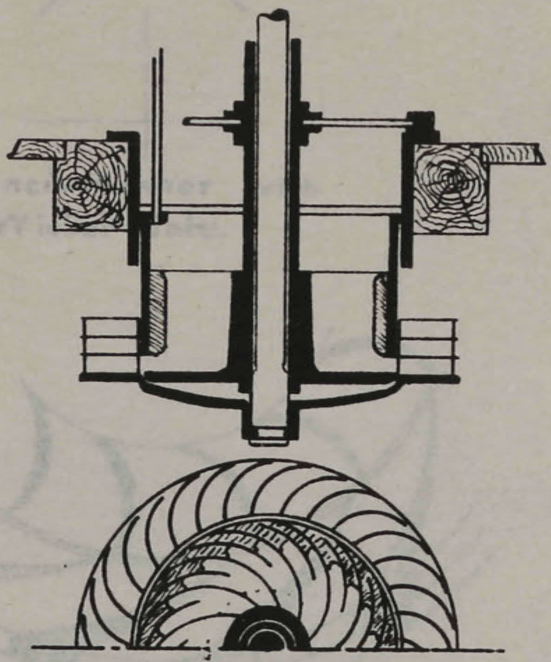


Fig. 6. FOURNEYRON TURBINE 1827.

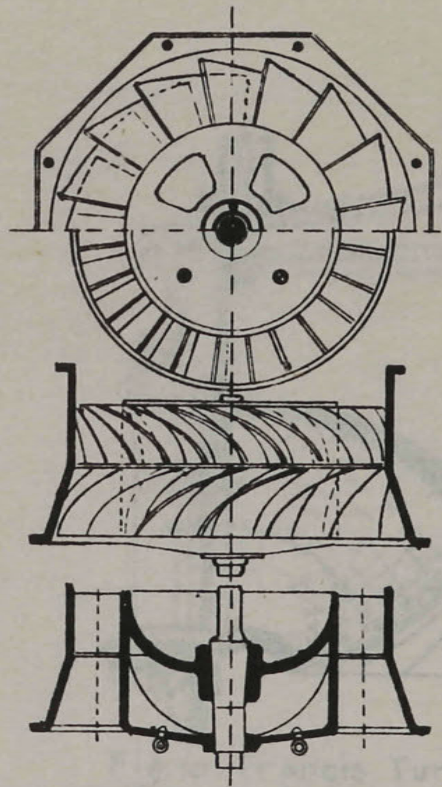


Fig. 7. JONVAL TURBINE, 1841.

A. T. SAFFORD, "TRANS. A.S.C.E. 1922."

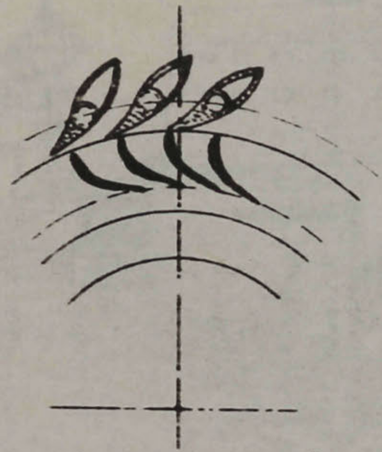


Fig.8. Francis Runner with Wicket Gate.

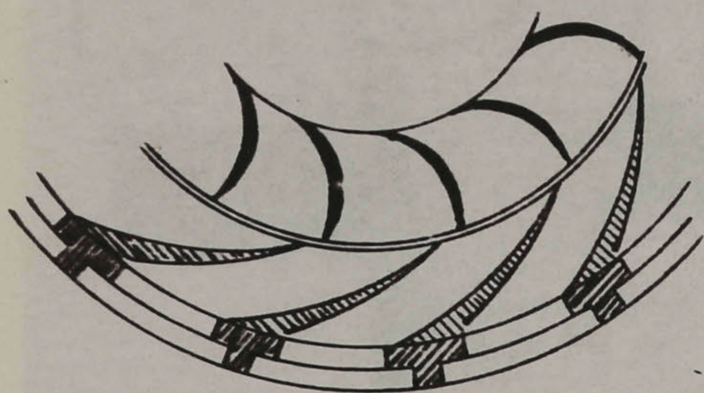


Fig.9. Francis Runner with Outside Register Gate.

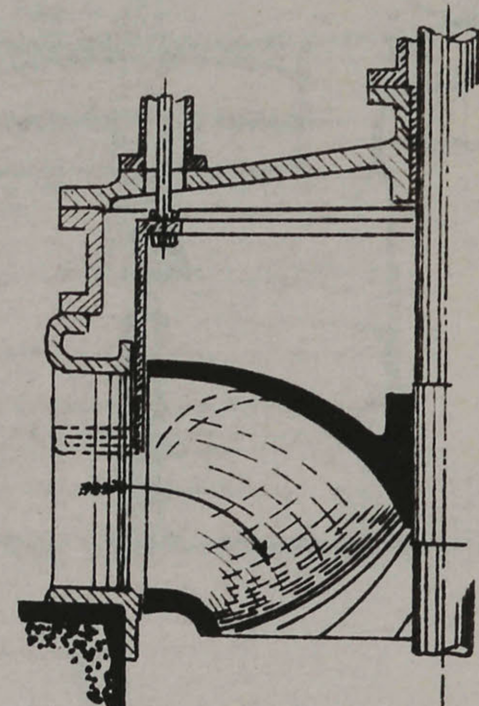
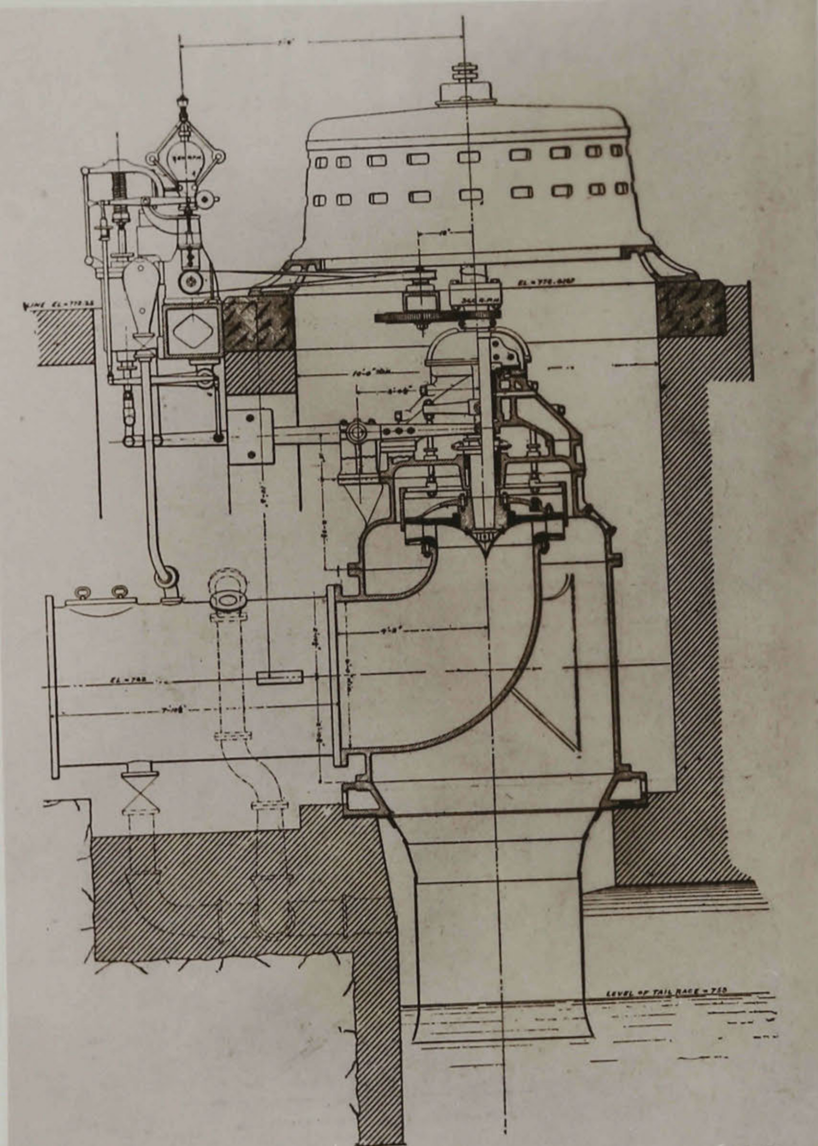


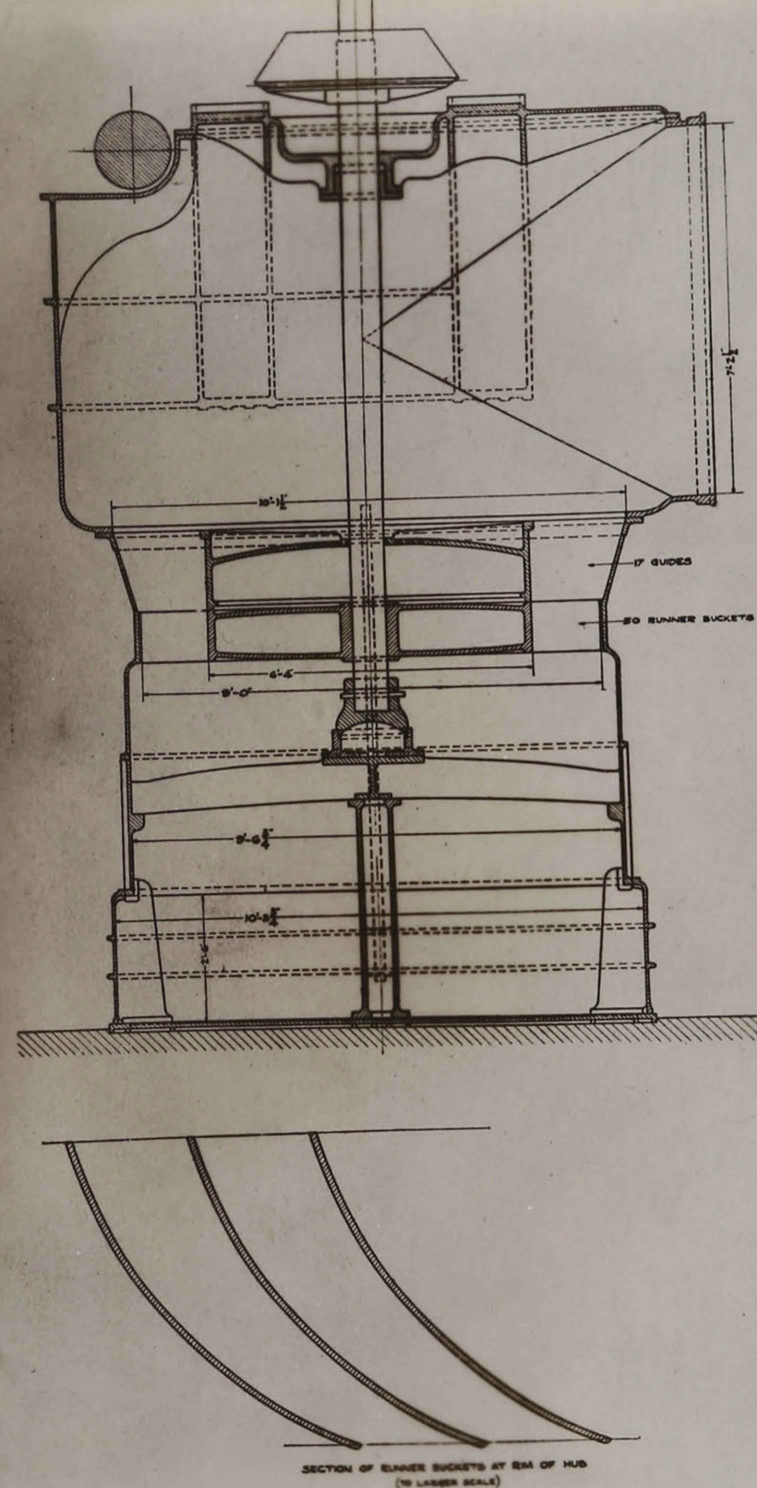
Fig.10. Francis Turbine with Inside Cylinder Gate.

Gesen "Hydraulics"



TURBINE FOR TRENTON FALLS PLANT OF UTICA GAS AND ELECTRIC COMPANY, 1901—SECTIONAL ELEVATION.

FIG. 11.



I. P. MORRIS TURBINES FOR FAIRMOUNT WATER WORKS, 1860.

FIG. 12.

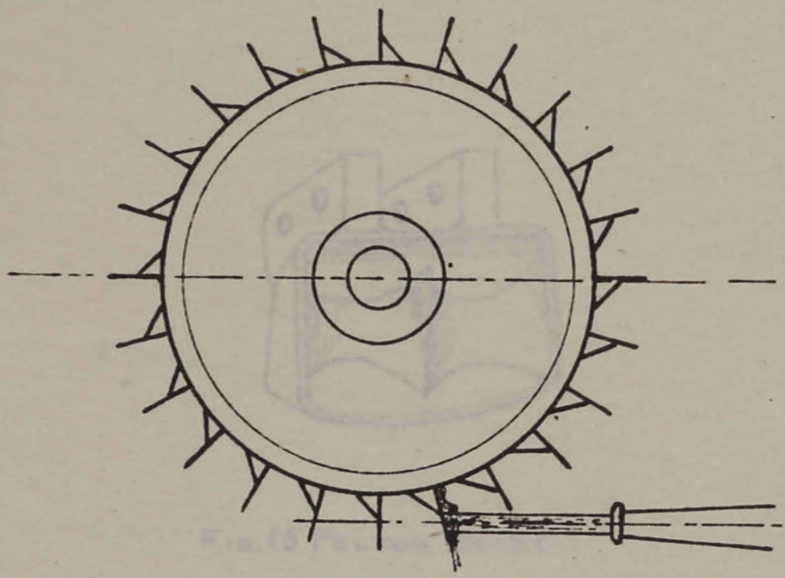


Fig. 13. HURDY GURDY

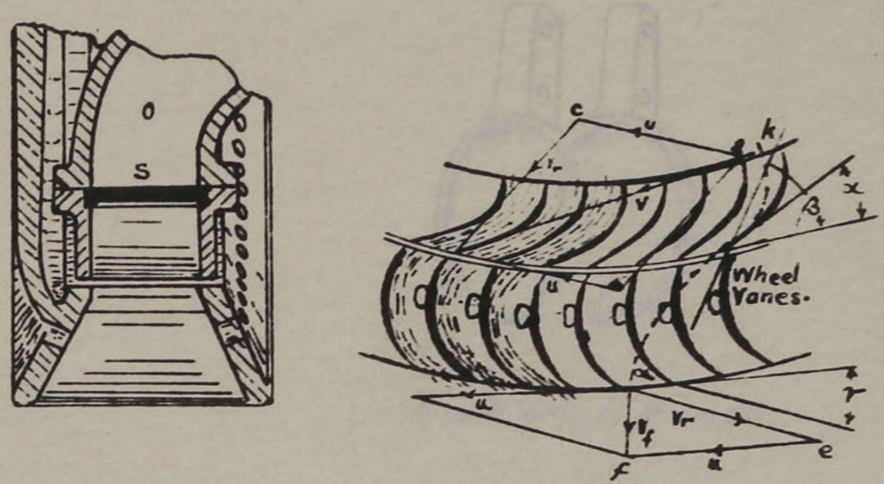


Fig. 14. GIRARD TURBINE WITH OUTWARD RADIAL FLOW.

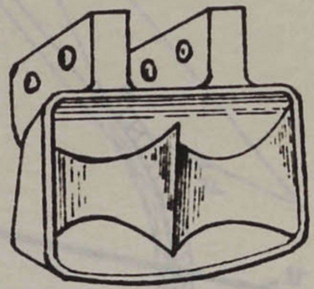


Fig. 15 Pelton Bucket.

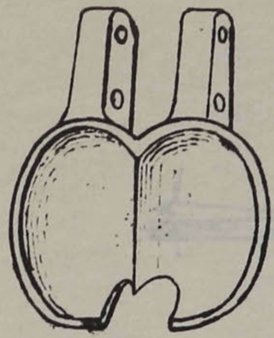


Fig. 16. Doble Bucket.

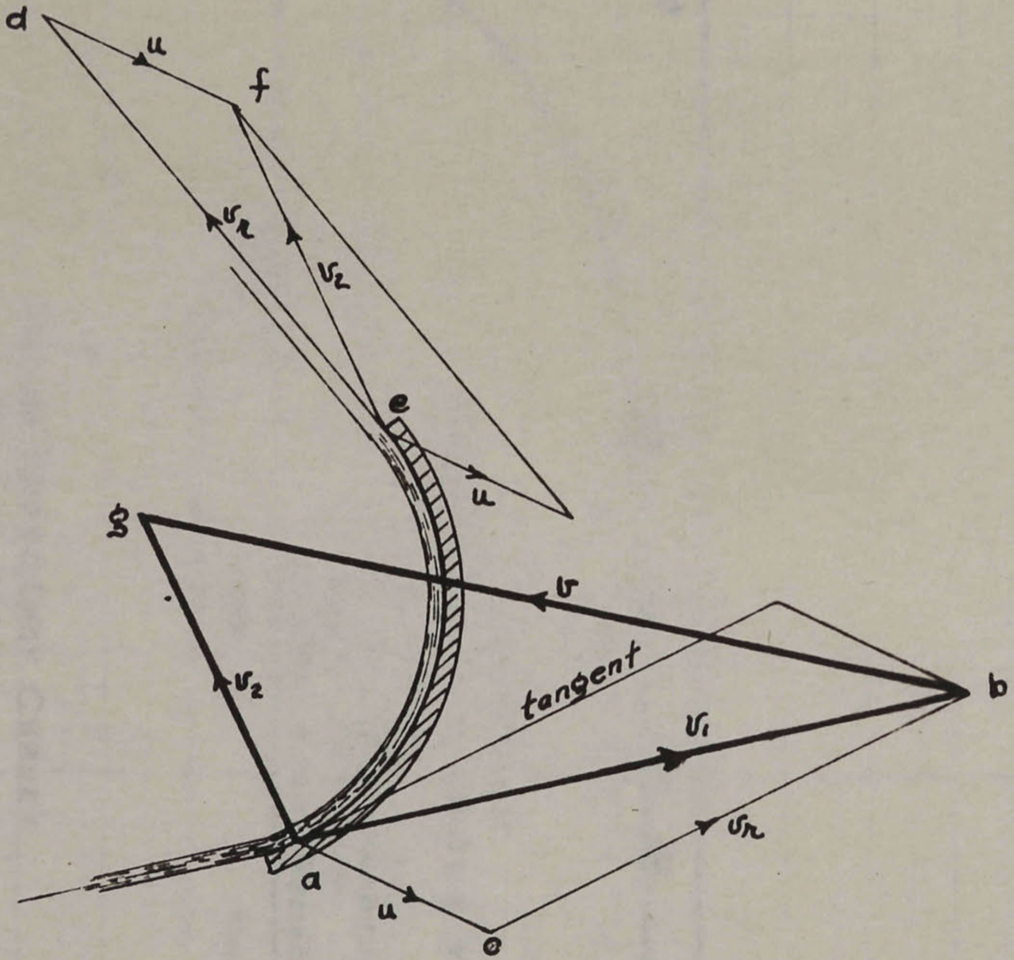


FIG. 17. IMPACT OF A JET.

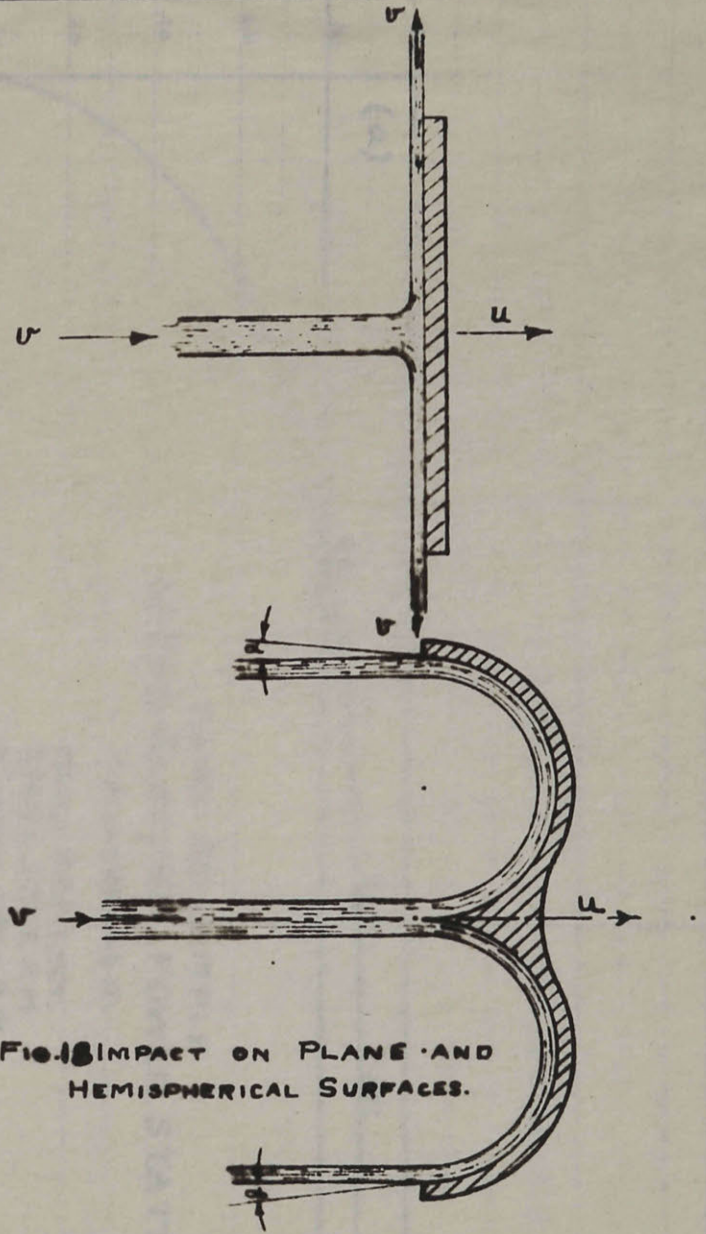


FIG. 18 IMPACT ON PLANE AND HEMISPHERICAL SURFACES.

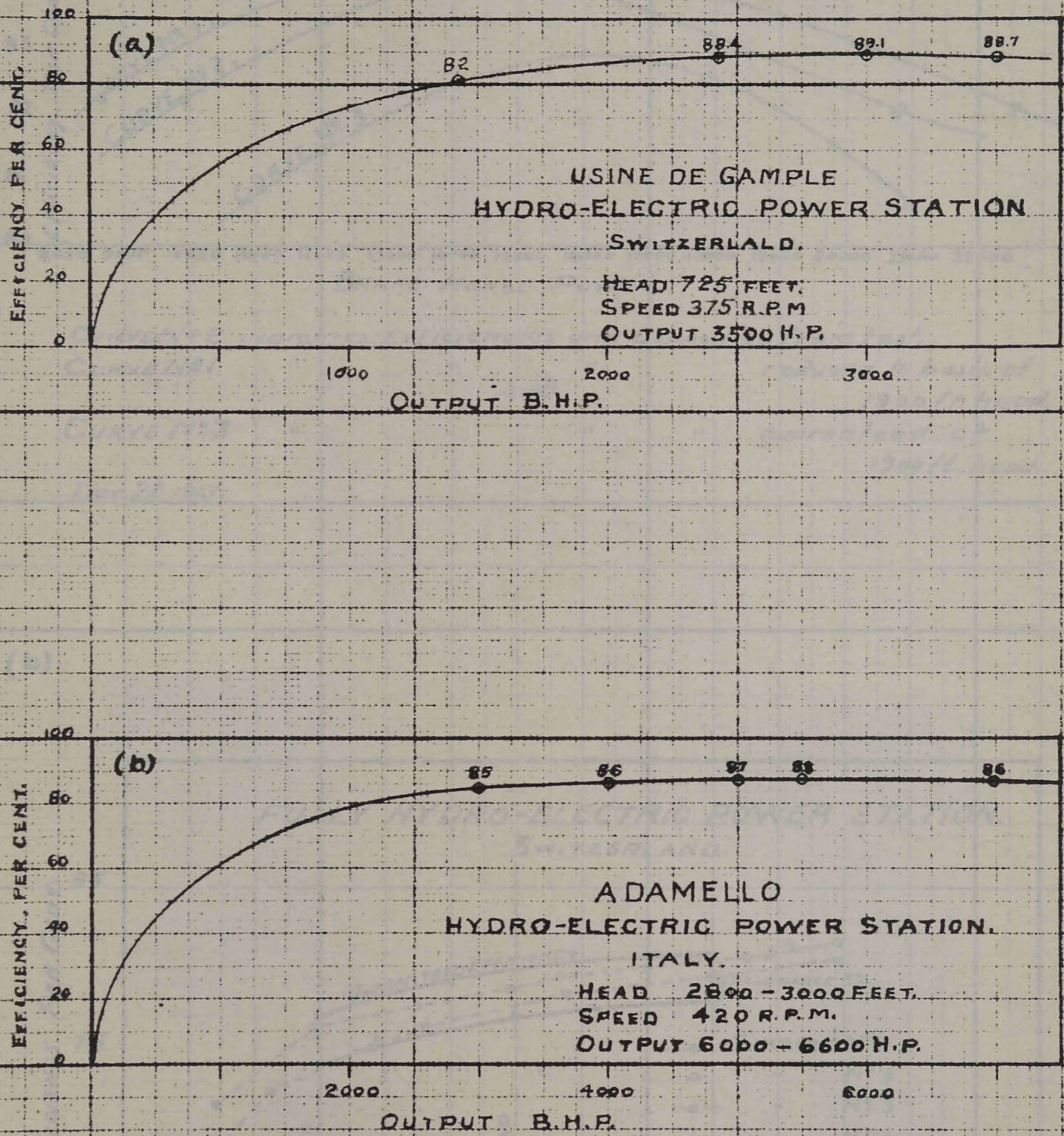


FIG. 19. EFFICIENCY CURVES OF PELTON WHEELS.

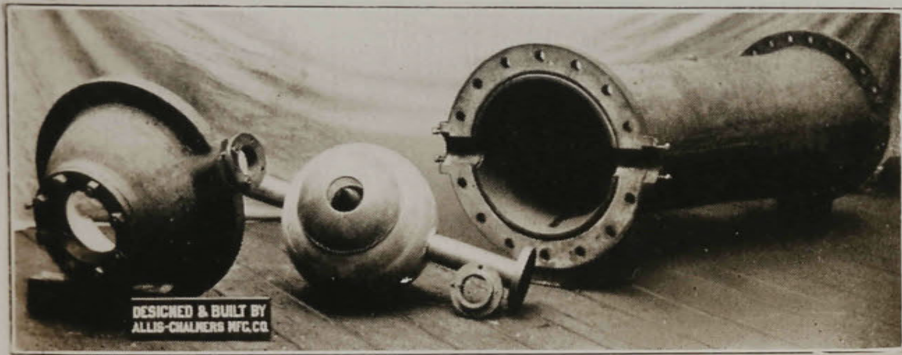


FIG. 21.

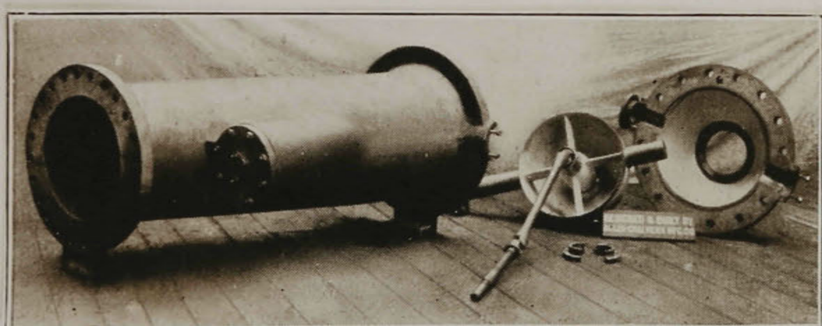


FIG. 22.

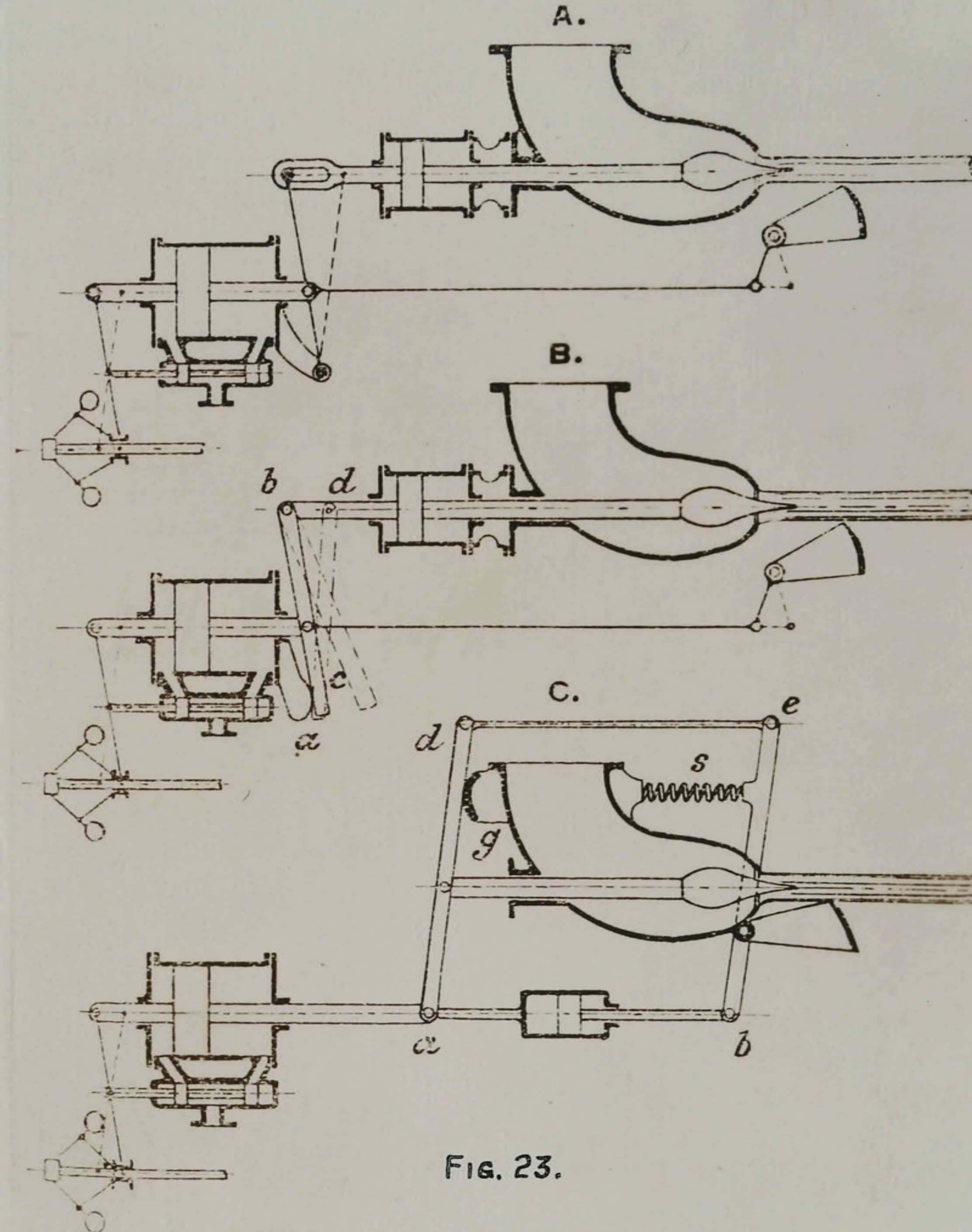


FIG. 23.

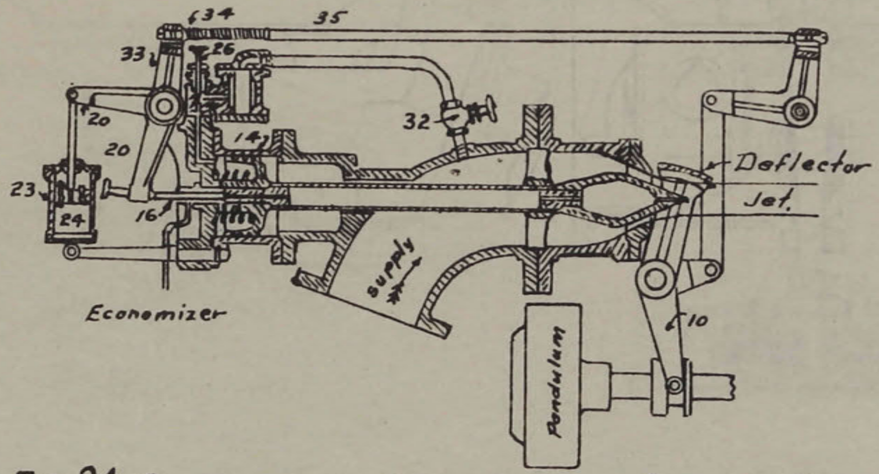
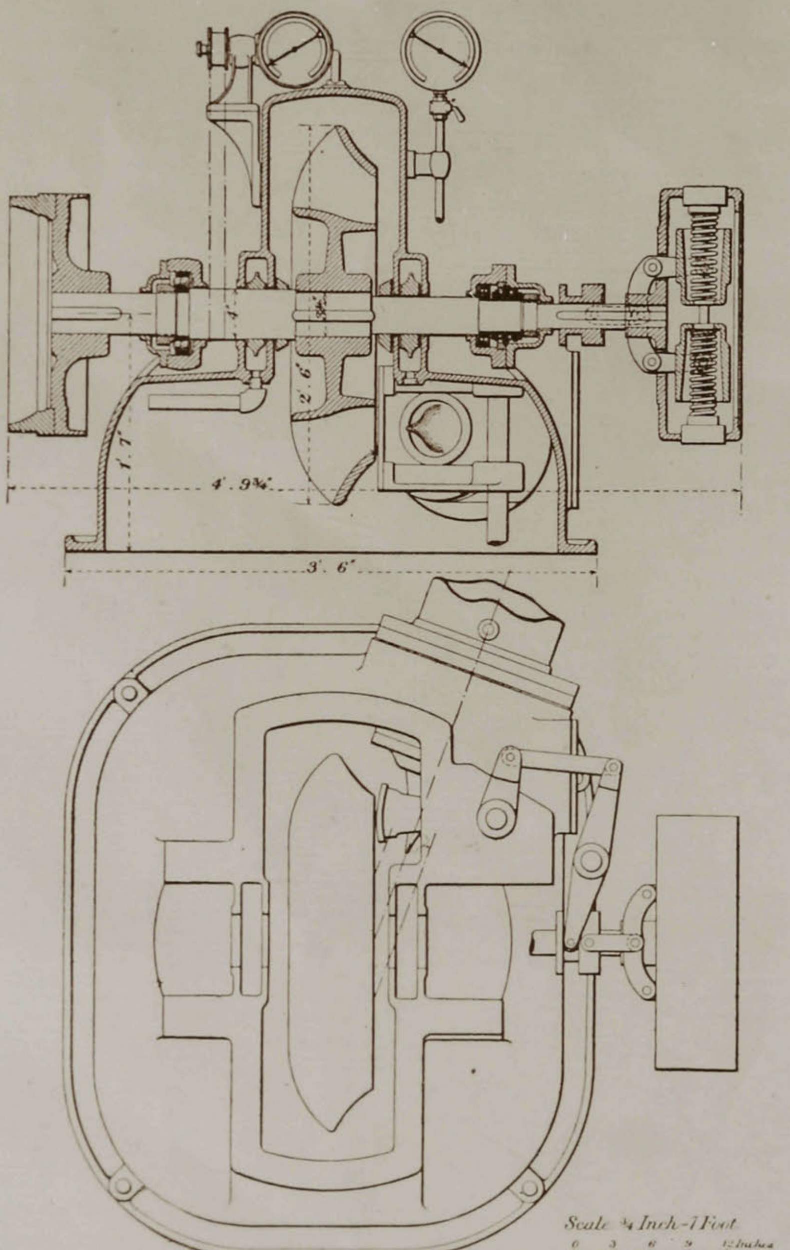


FIG. 24. DIAGRAMMATIC ARRANGEMENT OF GOVERNOR MECHANISM.

A NEW IMPULSE WATER-TURBINE.



Elevation and Plan of Turbine.

Fig. 25.

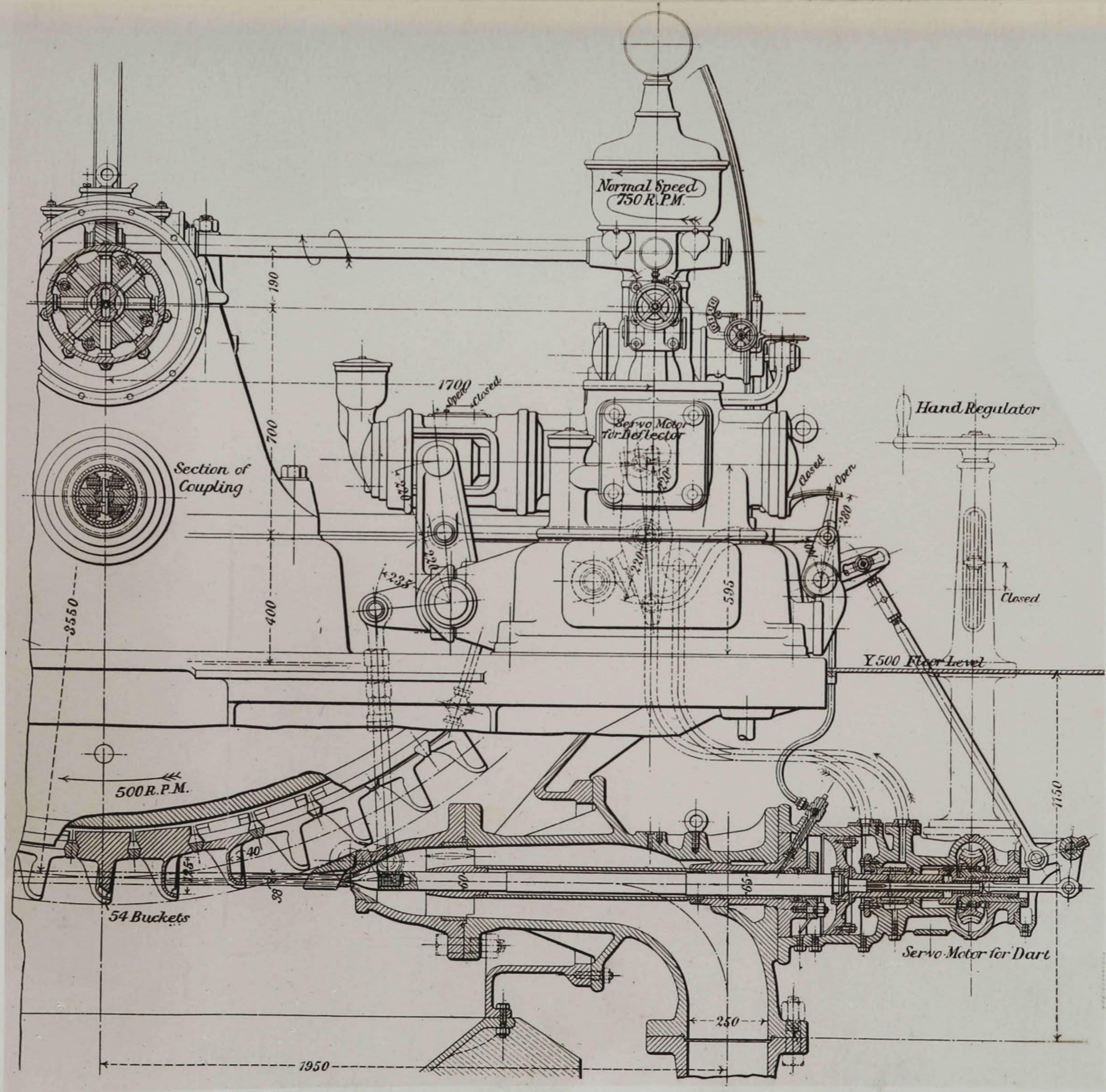
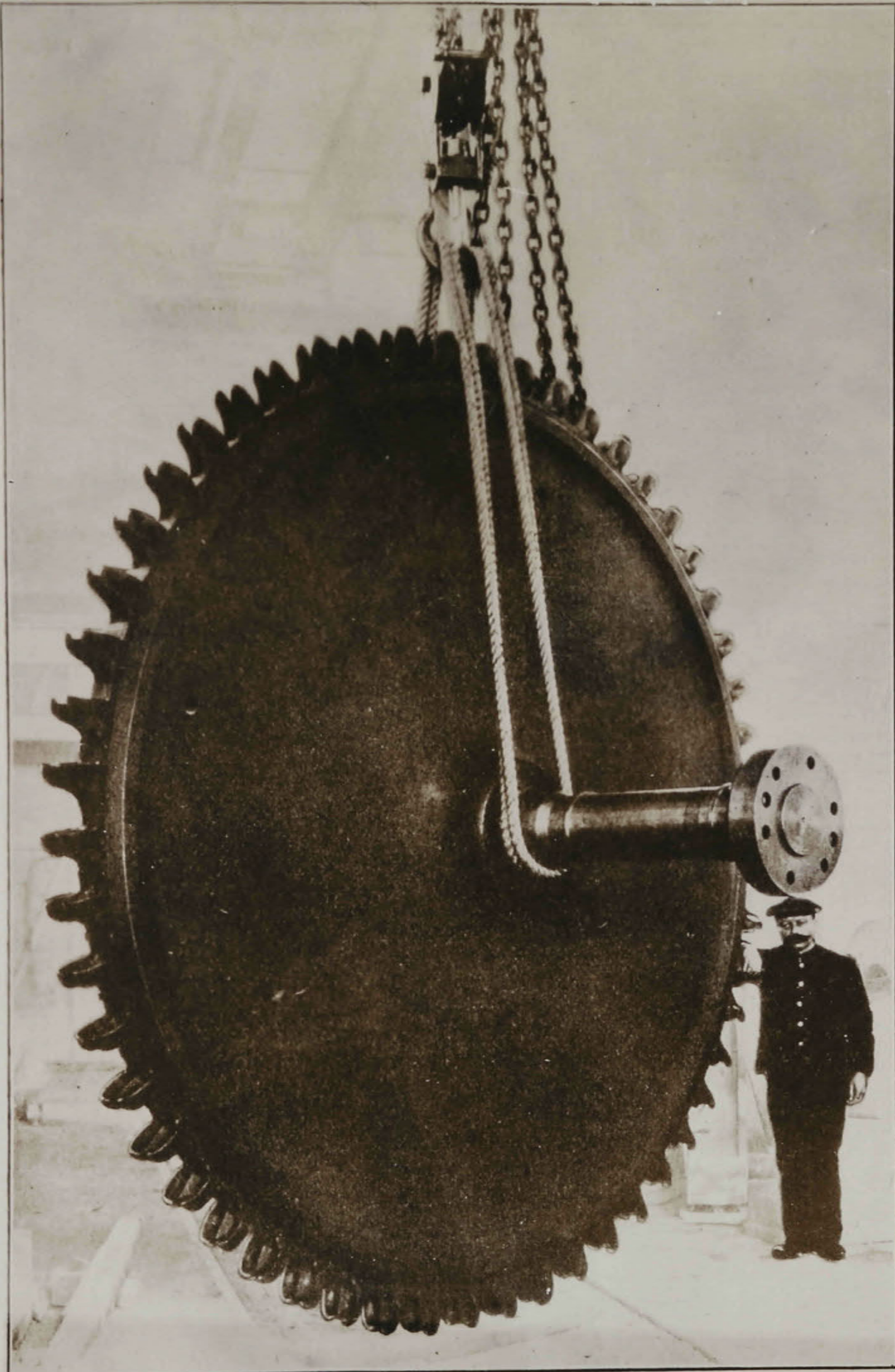
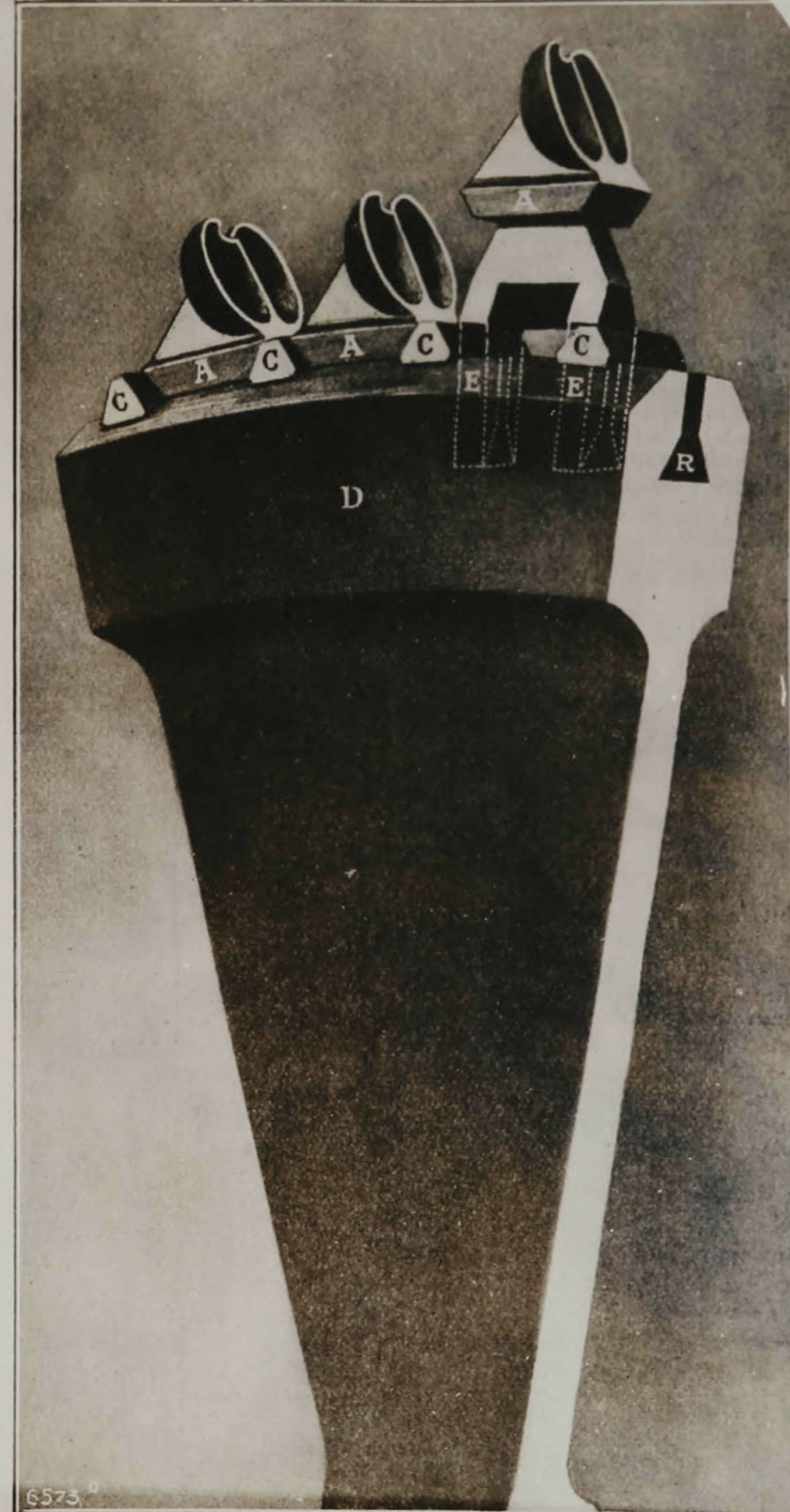


FIG. 27. NOZZLE AND GOVERNOR, FULLY WATER-WHEELS.

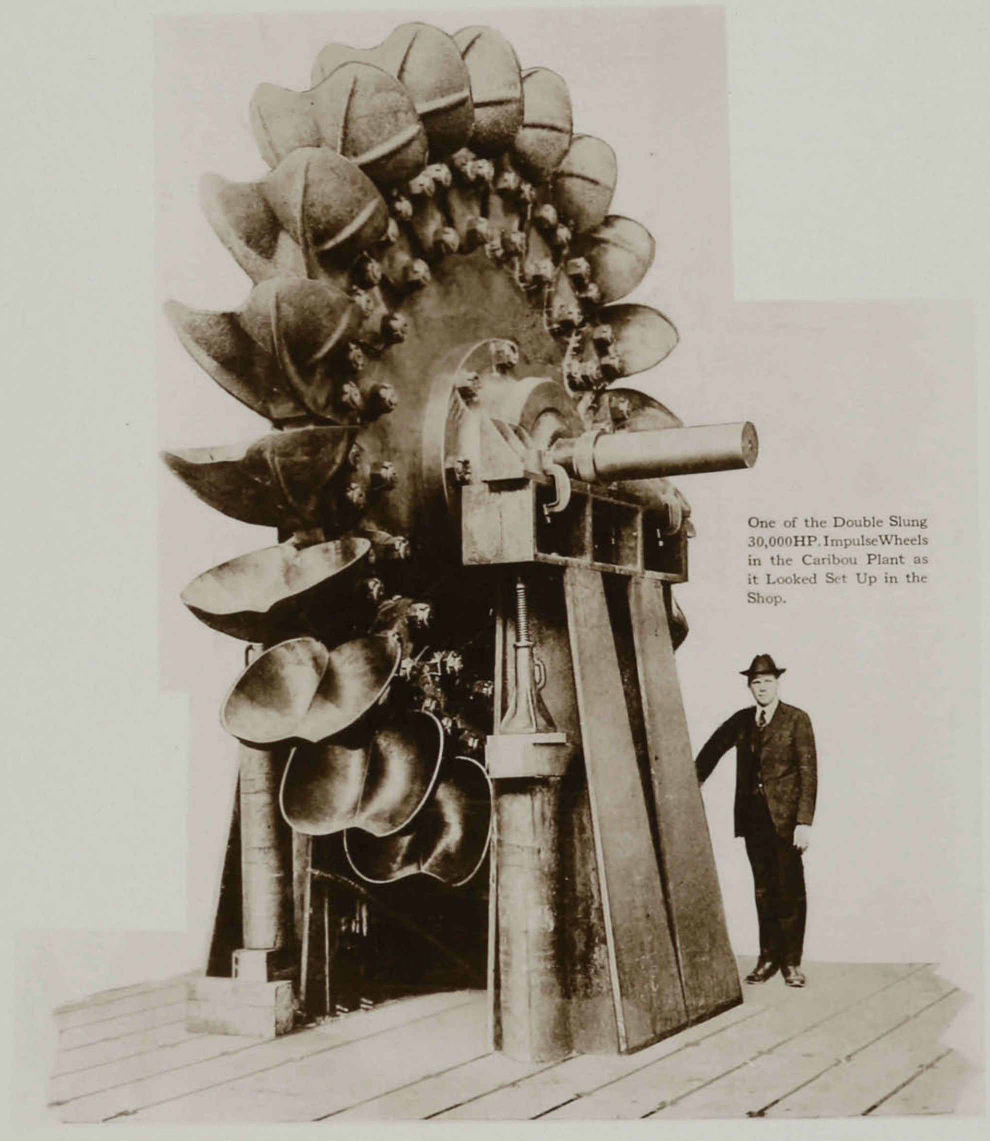


Disc of Pelton Wheel.



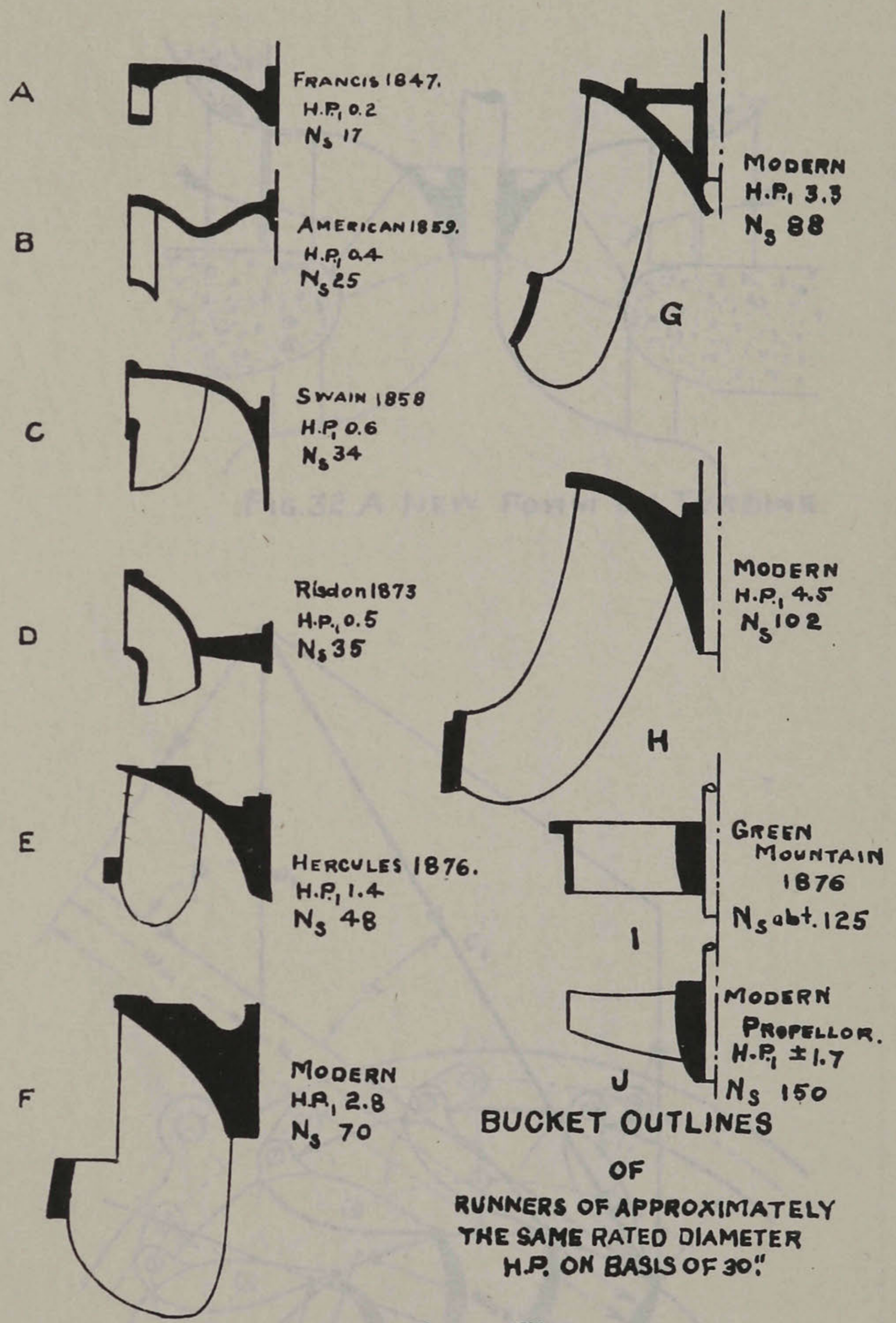
Method of Fixing Buckets.

Fig. 28. Fully Water Wheels.



One of the Double Slung 30,000HP. ImpulseWheels in the Caribou Plant as it Looked Set Up in the Shop.

Fig. 30



BUCKET OUTLINES

OF

RUNNERS OF APPROXIMATELY
THE SAME RATED DIAMETER
H.P. ON BASIS OF 30"

FIG. 31

A. T. Safford. Procs. A.S.C.E. April 1922.

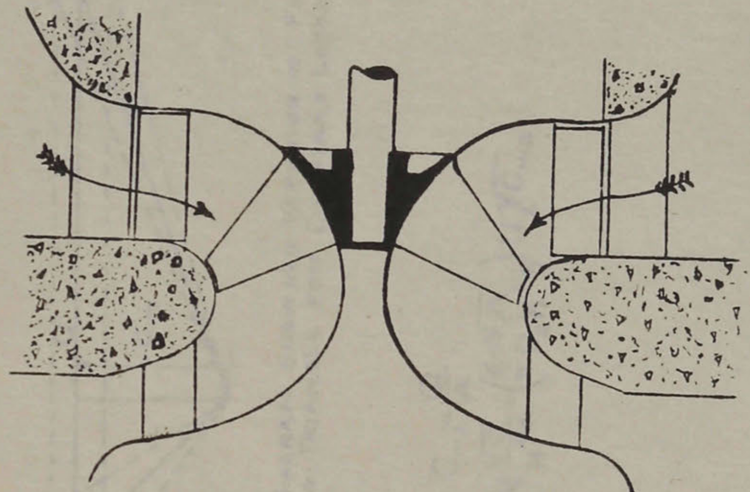


FIG. 32. A NEW FORM OF TURBINE.

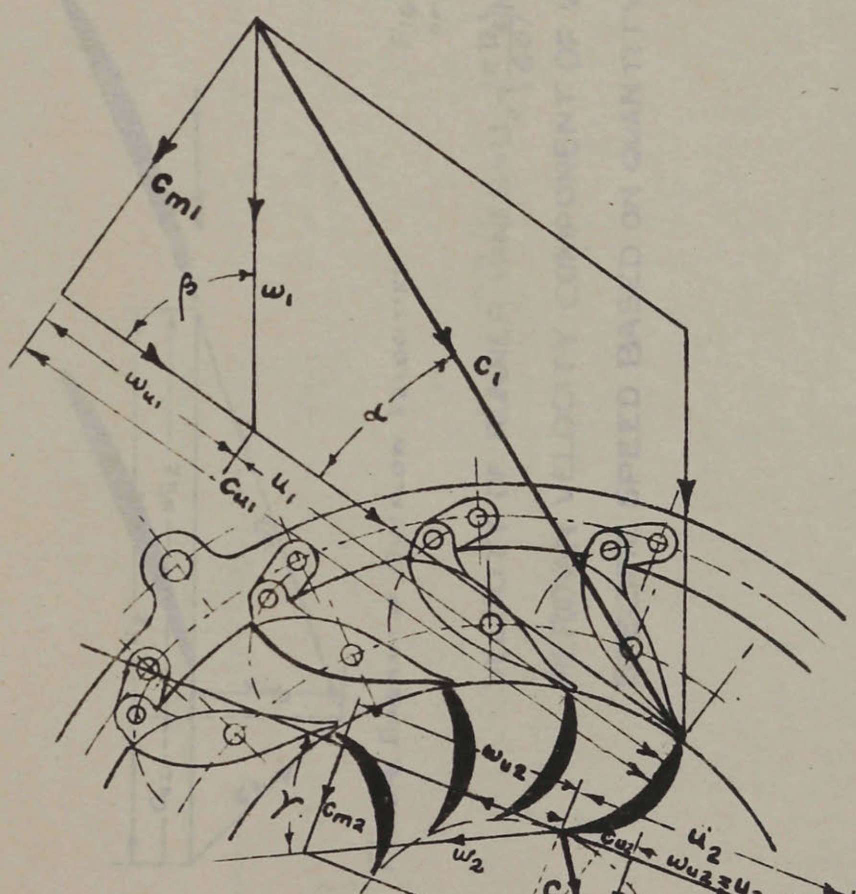


FIG. 33. VELOCITY DIAGRAMS
INWARD RADIAL FLOW.

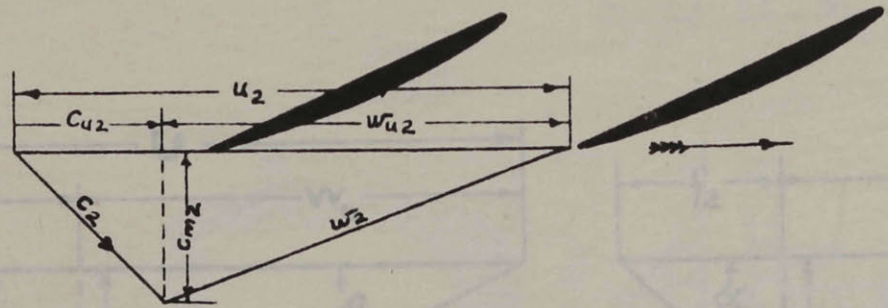


FIG. 34. DIAGRAM OF OUTFLOW VELOCITIES.

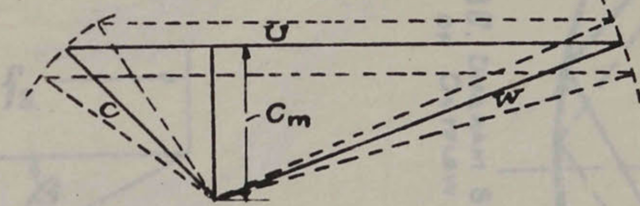


FIG. 34(a) DIAGRAM SHOWING VARIATION IN FORM OF OUTFLOW TRIANGLE FOR CONSTANT LOSSES.

VELOCITY OF RUNNER VANES $= U_2 = \left(\frac{\pi D_2}{60}\right) N$

MERIDIAN VELOCITY COMPONENT OF WATER $= C_{m2} = \frac{Q}{A}$

SPECIFIC SPEED BASED ON QUANTITY $= N_{sq} = N\frac{\sqrt{Q}}{H^{3/4}} = \left(\frac{60\sqrt{A}}{\pi D_2 H^{3/4}}\right) U_2\sqrt{C_{m2}}$

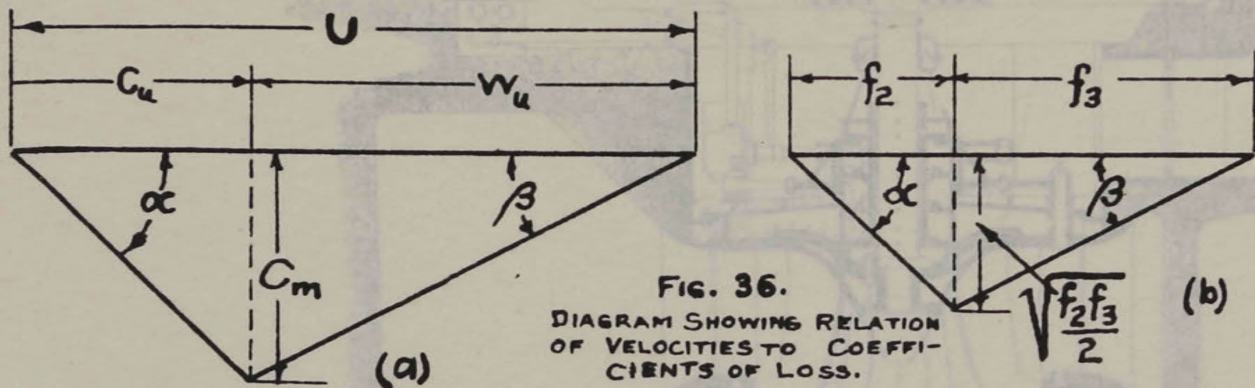


FIG. 36.
DIAGRAM SHOWING RELATION OF VELOCITIES TO COEFFICIENTS OF LOSS.

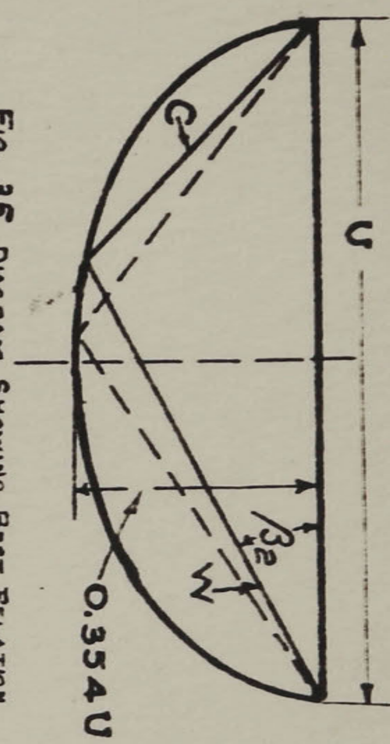


FIG. 35. DIAGRAM SHOWING BEST RELATION OF OUTFLOW VELOCITIES.

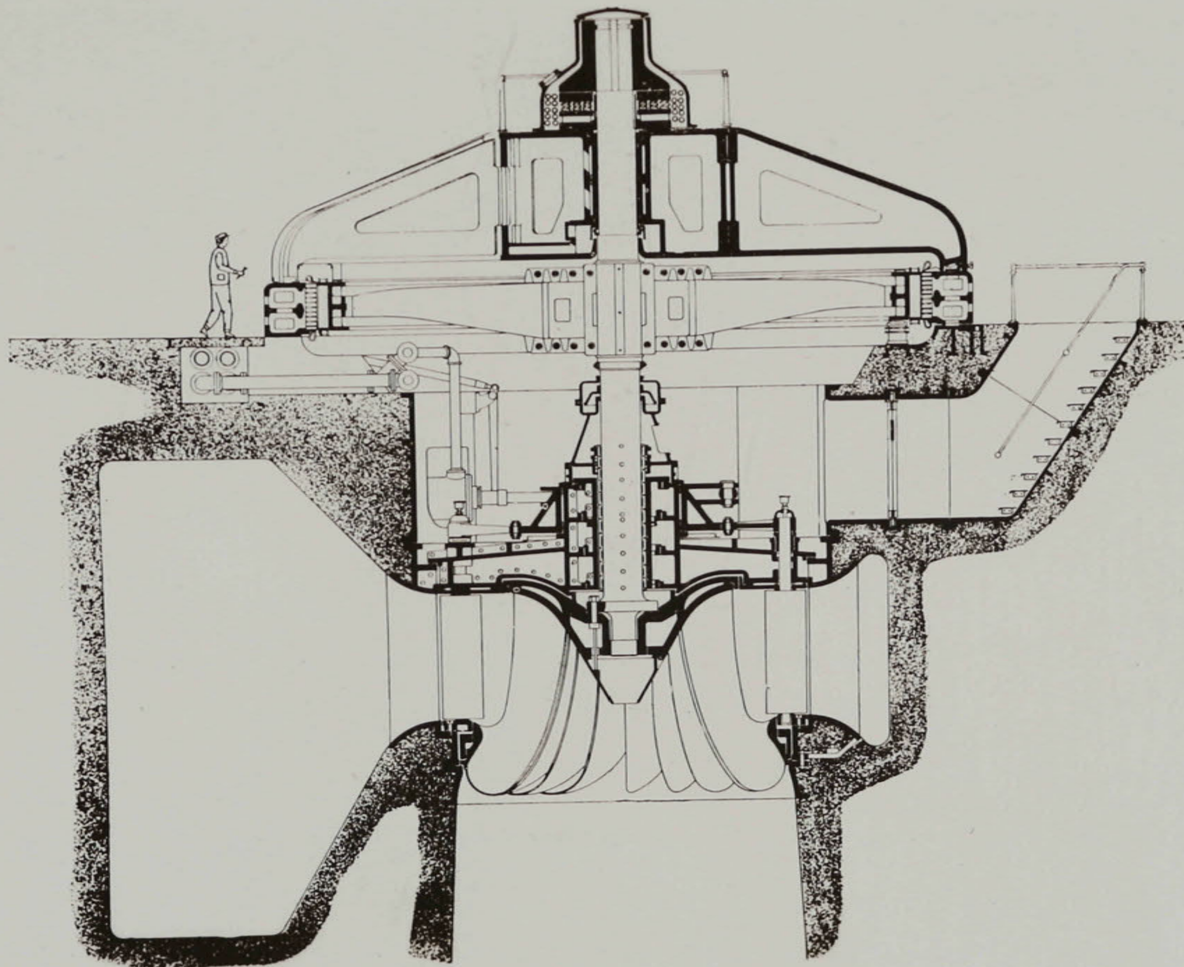


FIG. 37. ONE OF THE LATEST UNITS
OF THE CEDARS RAPIDS POWER PLANT.

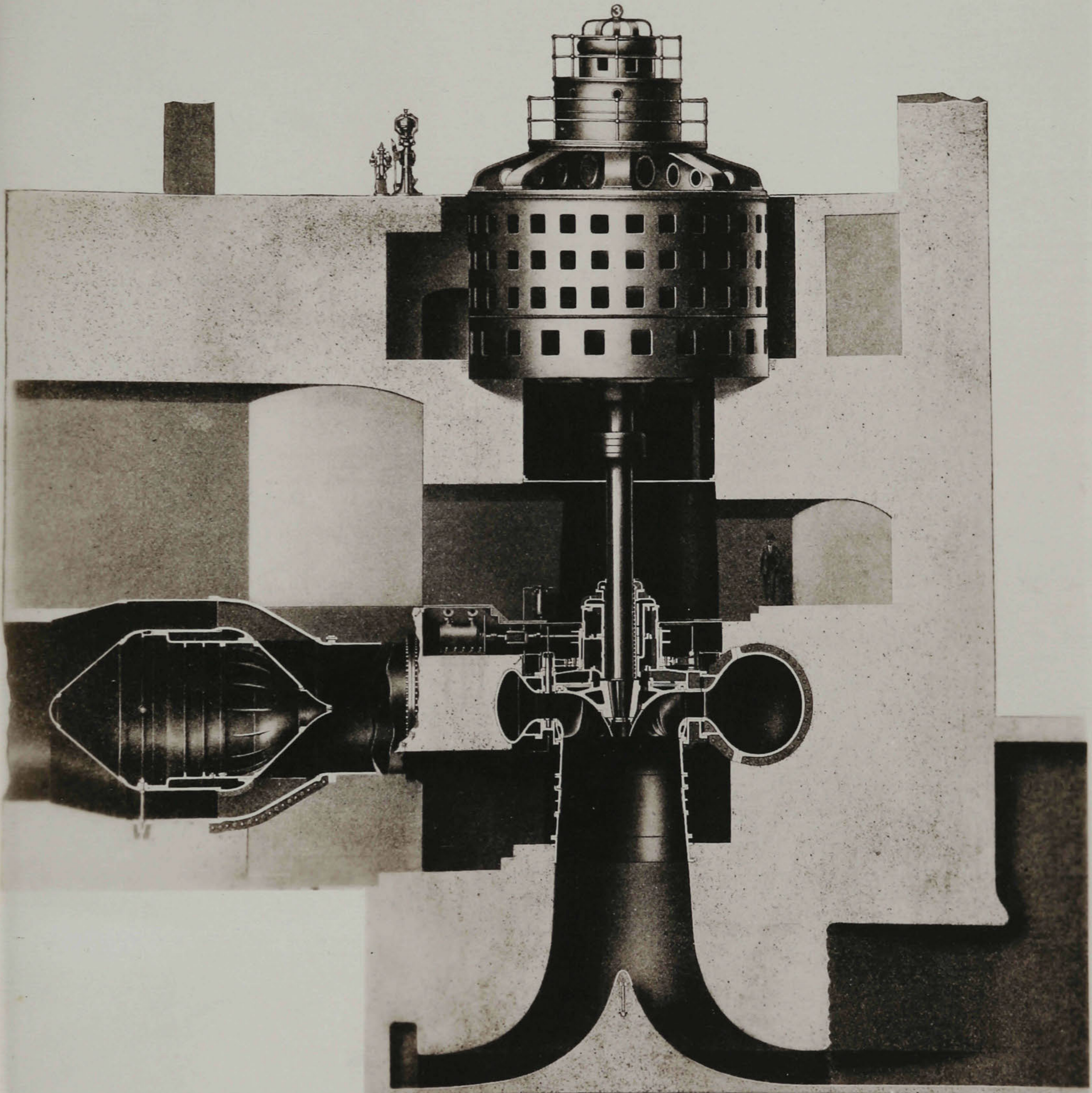
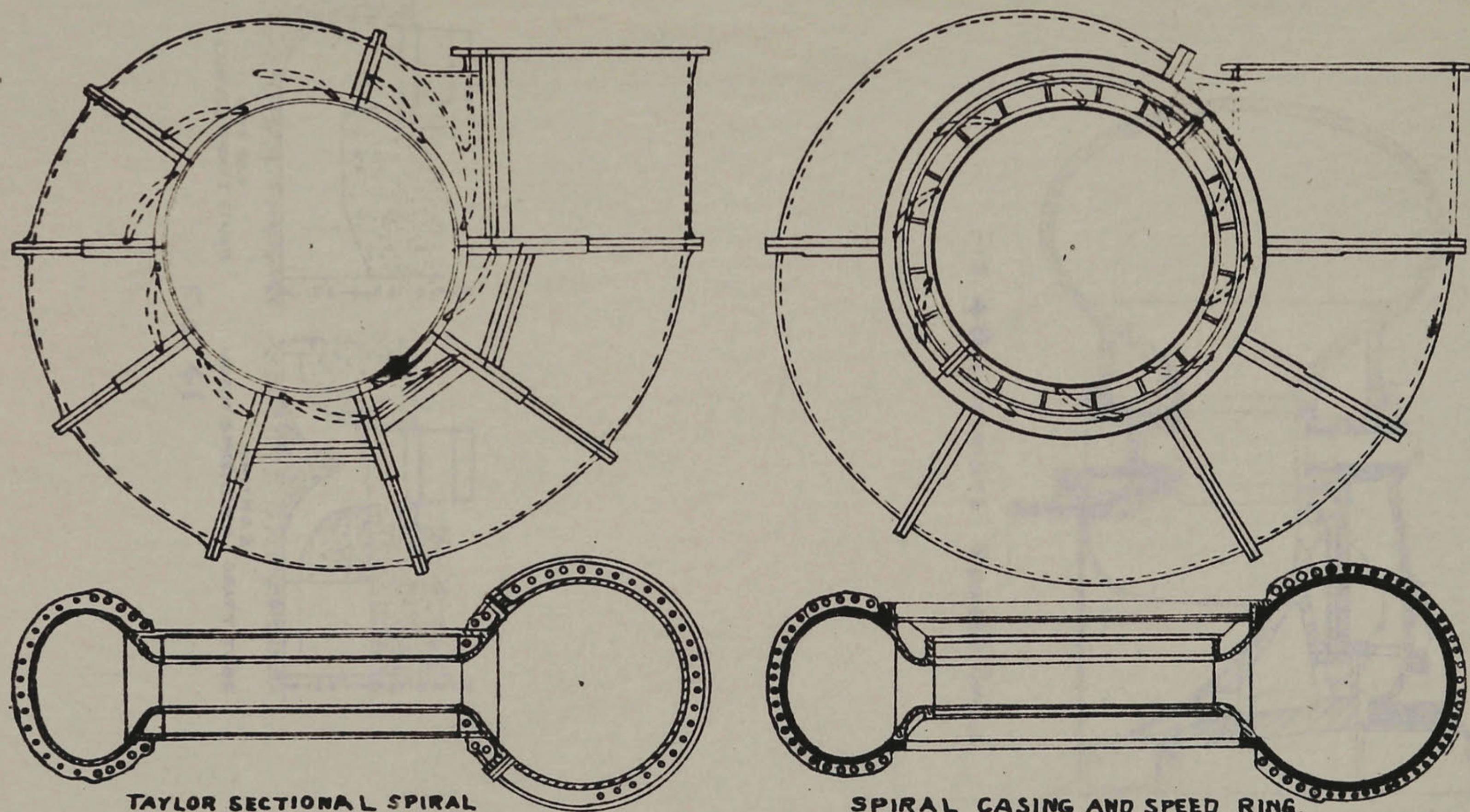


Figure 38 Sectional Elevation of Complete Unit
QUEENSTON-CHIPPAWA DEVELOPMENT.



TAYLOR SECTIONAL SPIRAL CASING.

SPIRAL CASING AND SPEED RING.

FIG. 39.

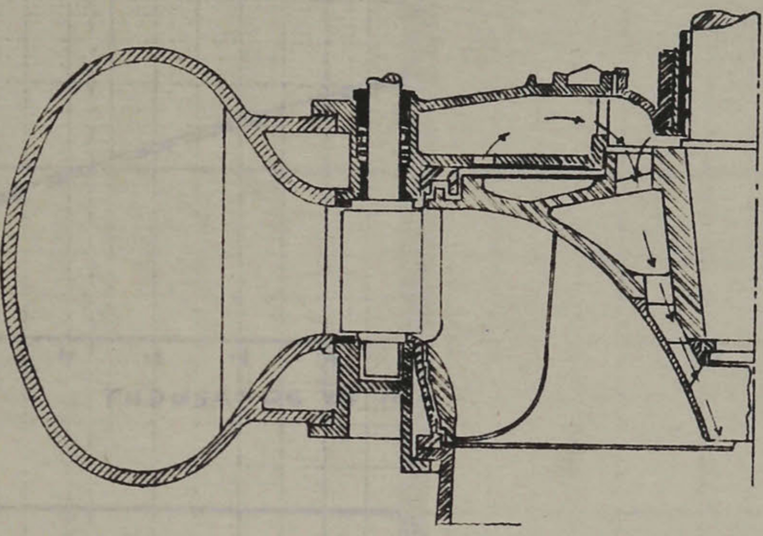
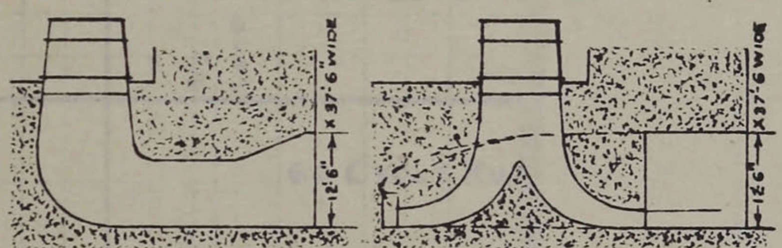


FIG. 40 LABYRINTH RUNNER SEALS.



UNIT No. 1
CURVED DRAFT TUBE

UNIT No. 2
MOODY SPREADING DRAFT TUBE.

FIG. 41

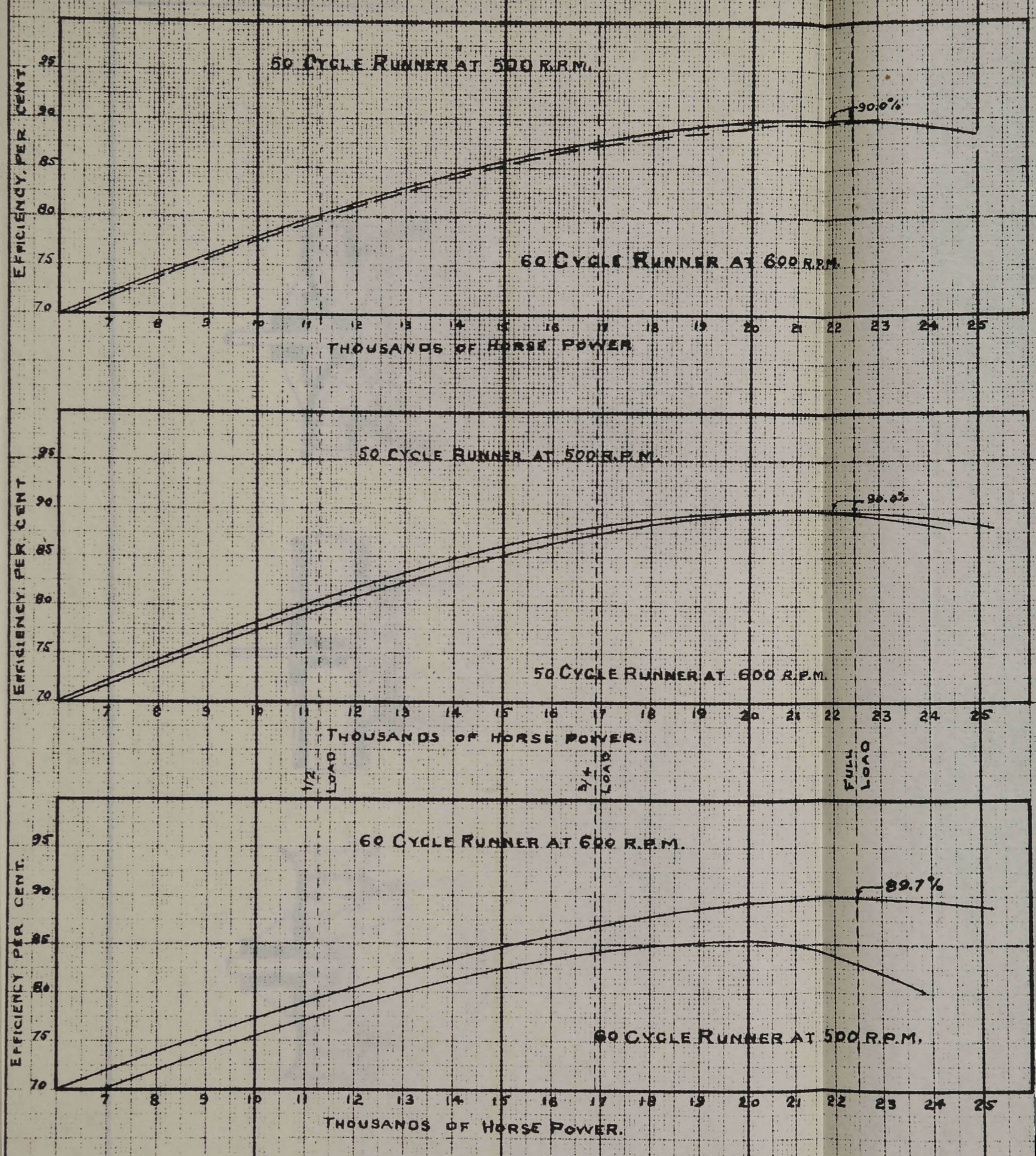


Fig. 42. Efficiency Curves of the High Head Francis Runners at Kern River No. 3 Plant.

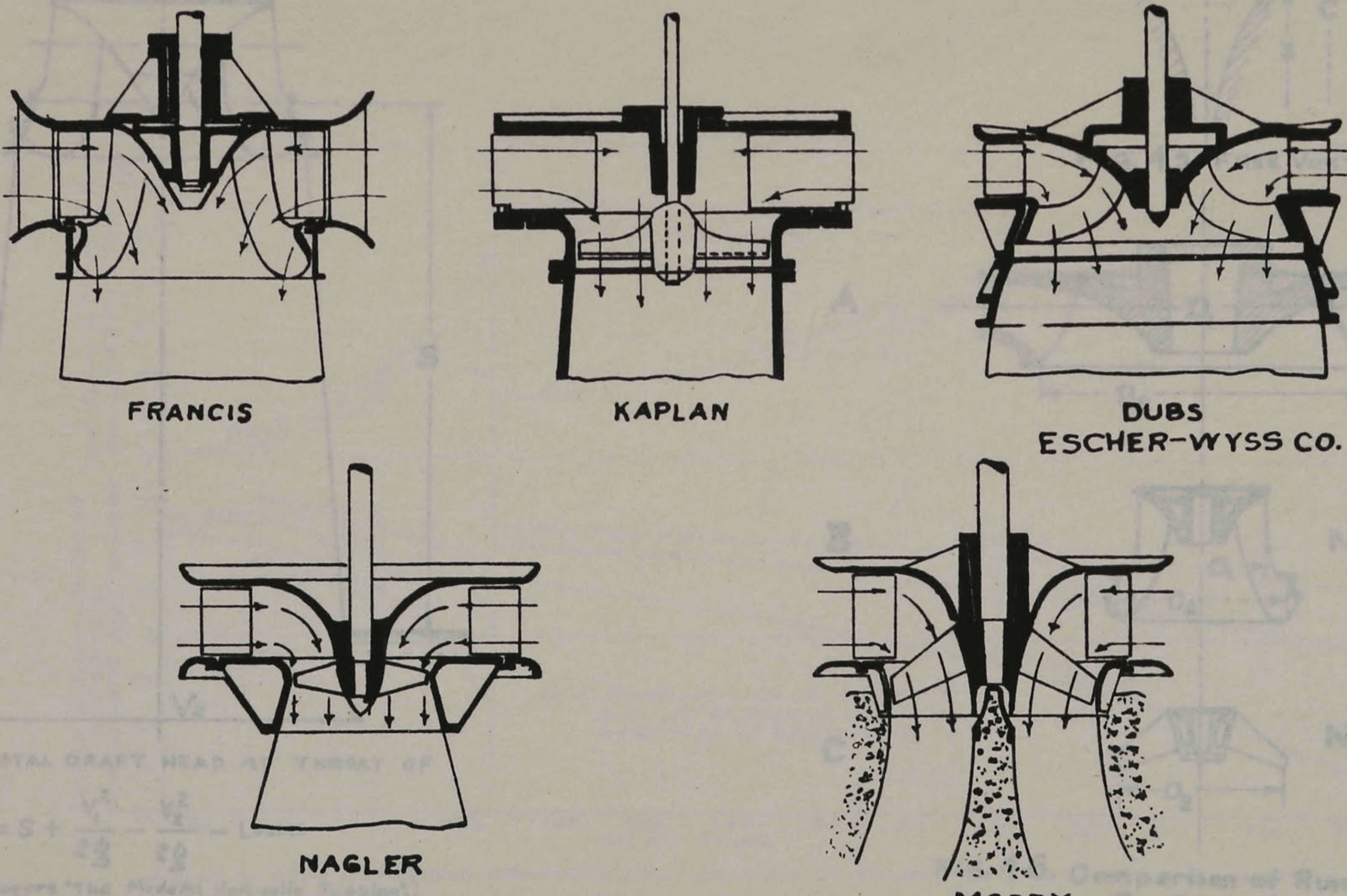


FIG. 43. HIGH SPEED TURBINE TYPES.

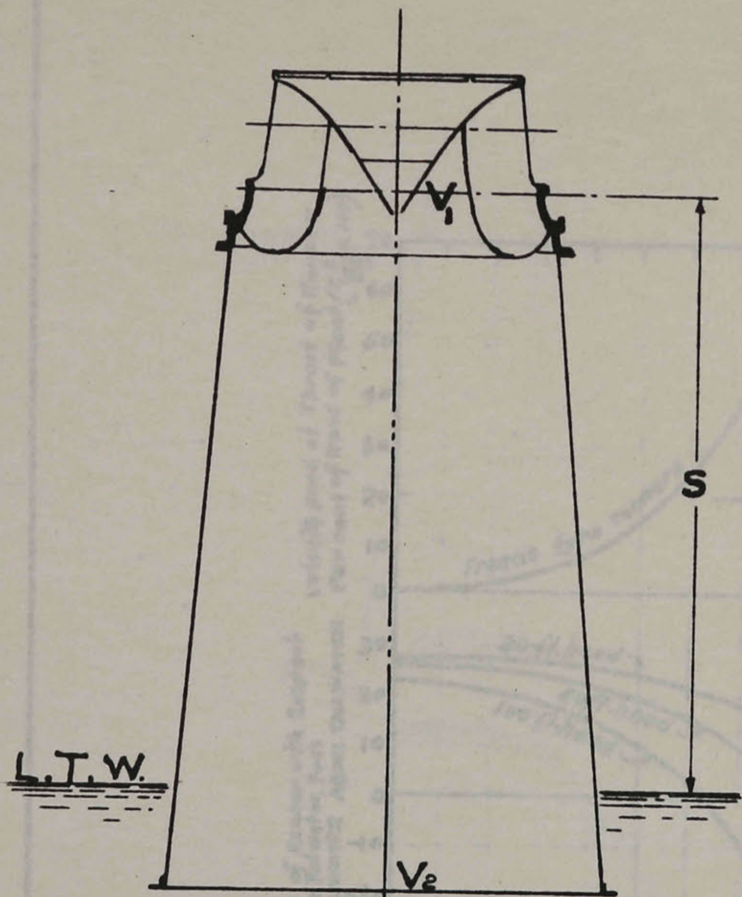
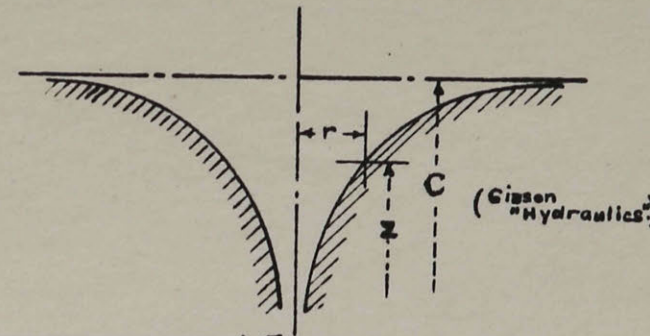


FIG. 44 TOTAL DRAFT HEAD AT THROAT OF RUNNER $= S + \frac{V_1^2}{2g} - \frac{V_2^2}{2g}$ — LOSSES

(Rogers "The Modern Hydraulic Turbine".)



(Gibson "Hydraulics".)

FIG. 45. FREE VORTEX.

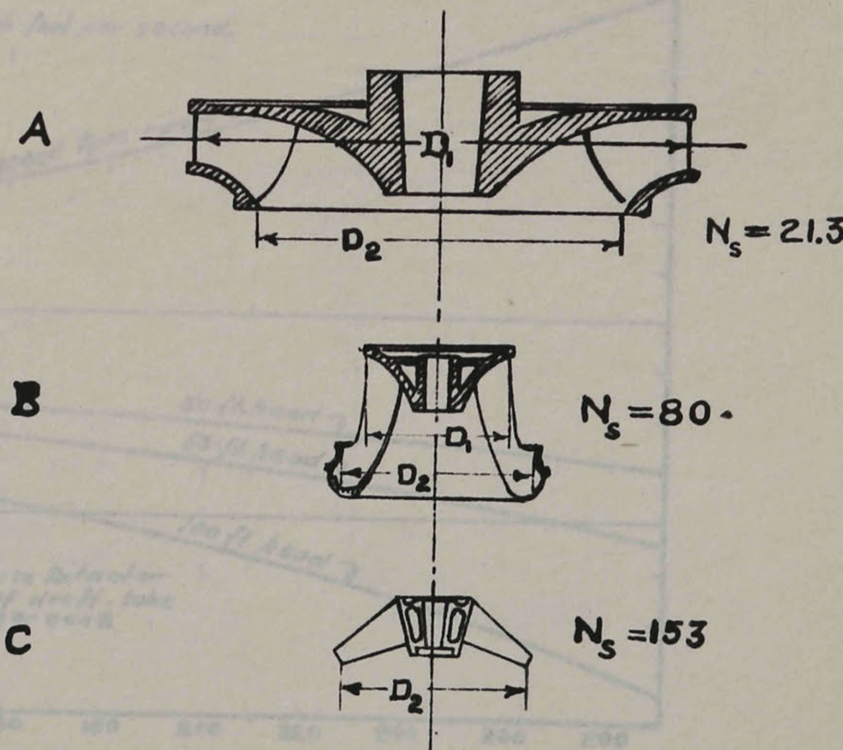


FIG. 46. Comparison of Runners of Equal Power but Different Types.

(Rogers "High Speed Runners")

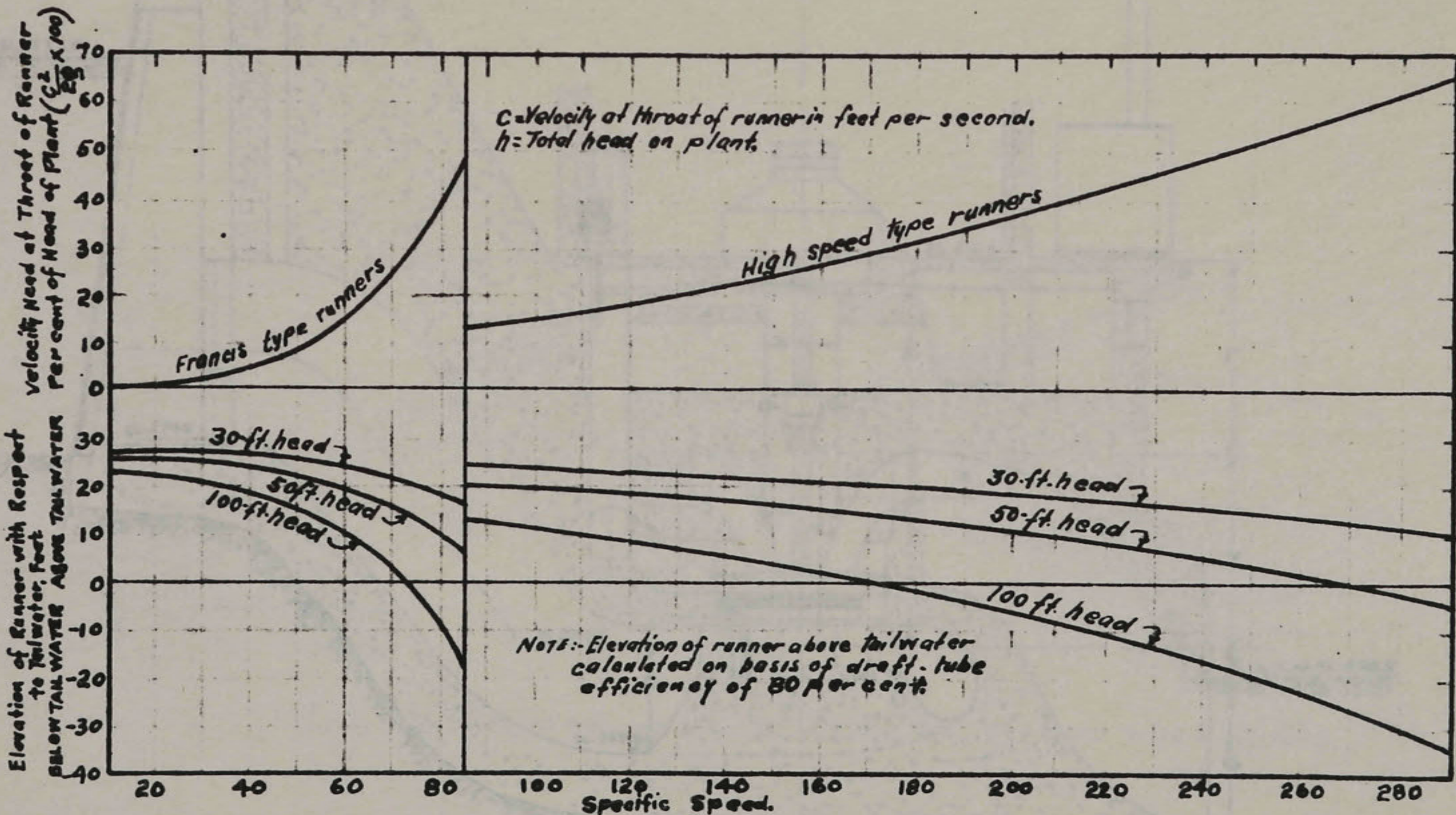


FIG. 47. ENERGY AT RUNNER DISCHARGE AND LOCATION OF RUNNER WITH RESPECT TO TAILWATER.

The curves in the upper half of the figure show the total amount of energy discharged by the runner, while those on the lower half show the possible elevation of the runner with respect to tailwater for different heads and different specific speeds.

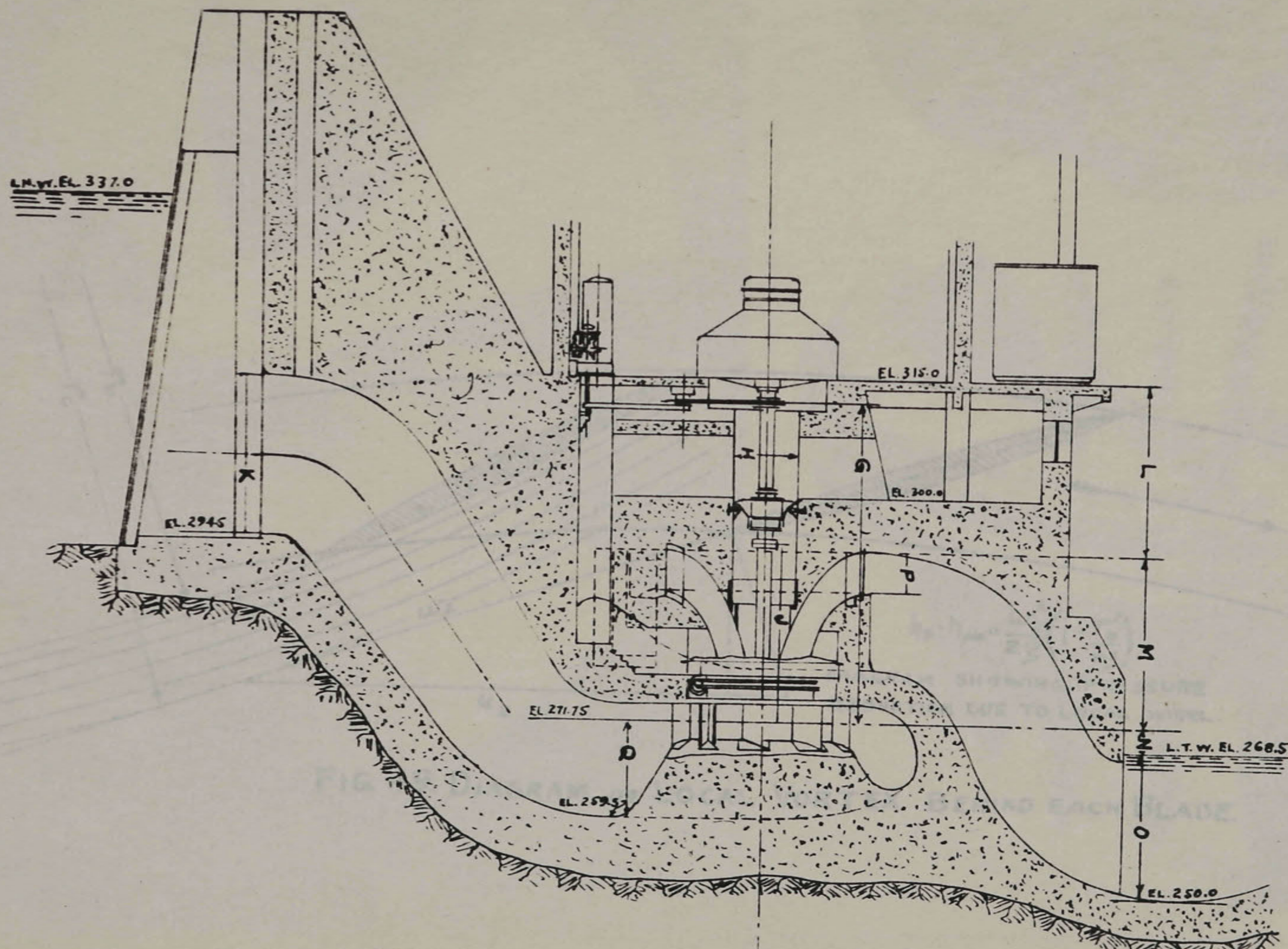


FIG 48 PROPOSED ARRANGEMENT OF TAYLOR INVERTED TURBINE.

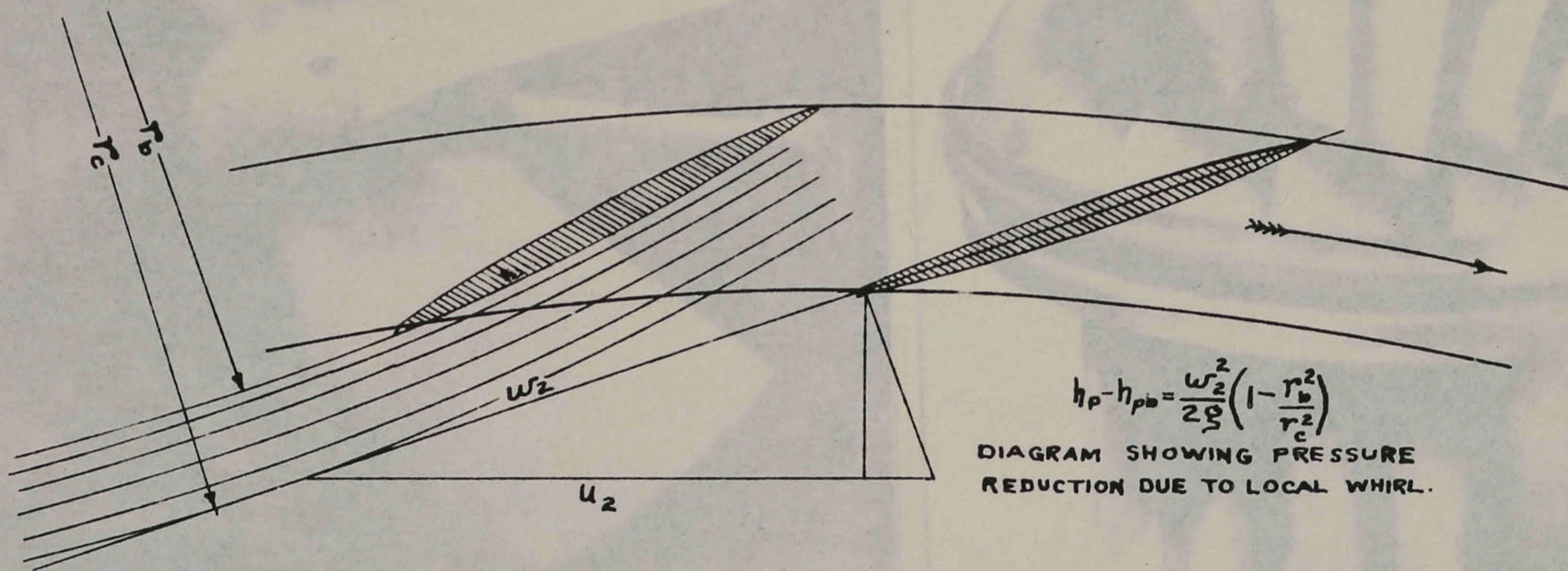
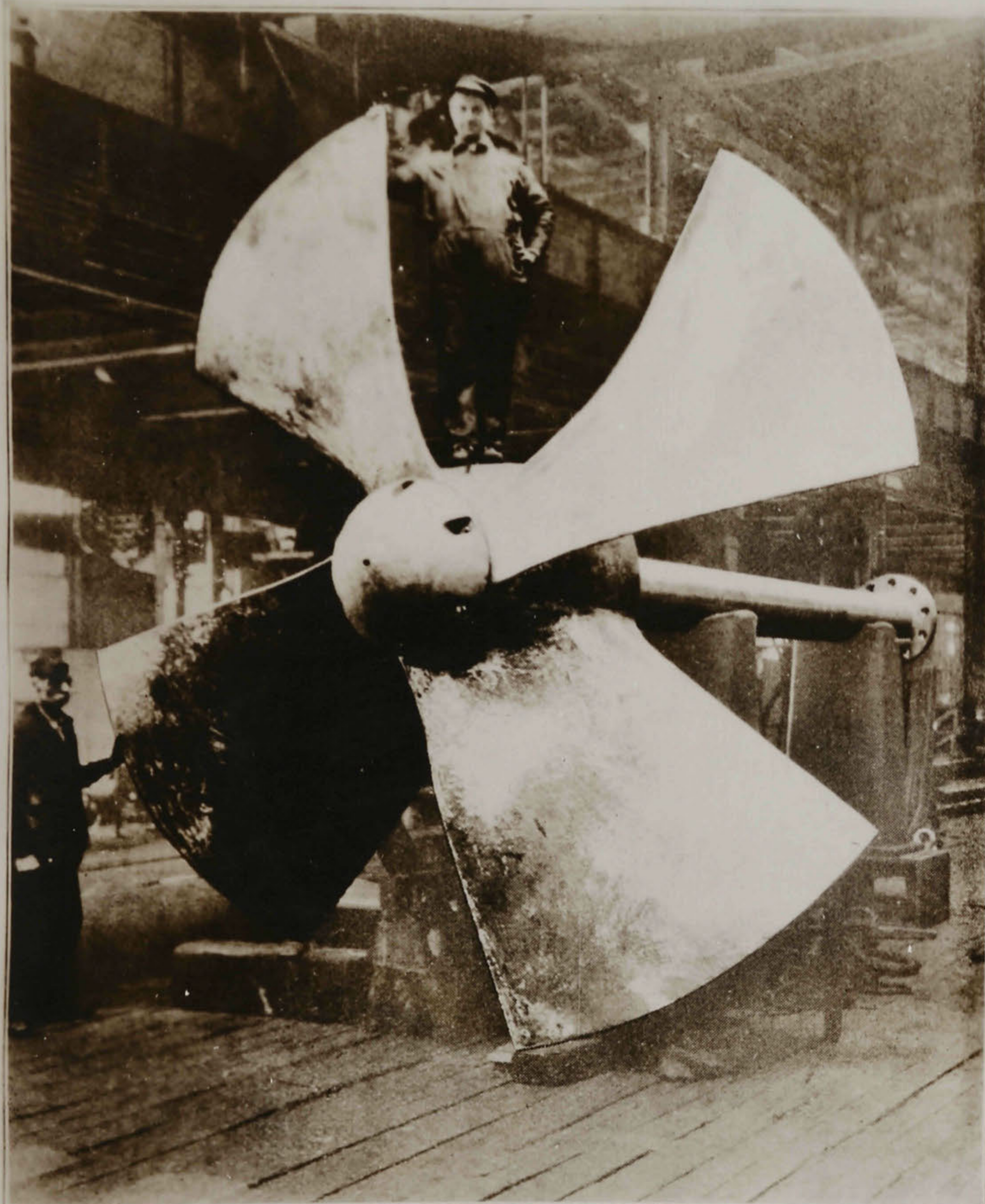


FIG. 49. DIAGRAM OF LOCAL VORTEX BEHIND EACH BLADE.



13'-0"

FIG. 51 156" dia., 2200 hp., four blade cast steel Nagler Runner, cast in four sections. Weight 17,000 lbs., head 13', speed 80 r.p.m., Henry Ford & Son, Inc., Green Island Development. The discharge capacity of this runner is about 85 per cent of that of the runners at Keokuk, each of which weighs approximately 179,000 lbs., and measures over 17' outside diameter.

FIG. 50 Comparison between Propeller and Francis Types of Turbine Runners.

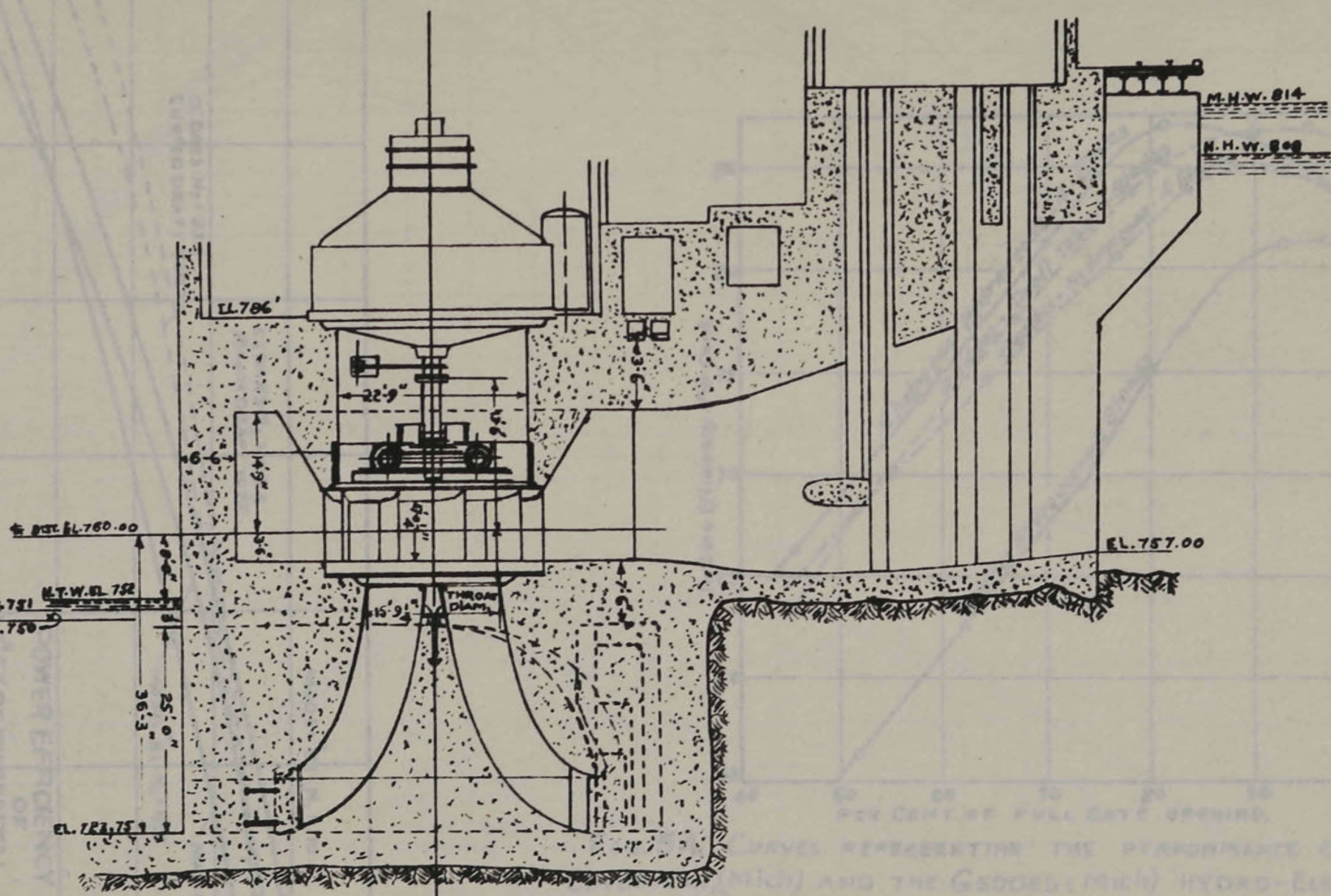


FIG. 52. POWER HOUSE ARRANGEMENT OF DIAGONAL PROPELLOR-TYPE TURBINE.

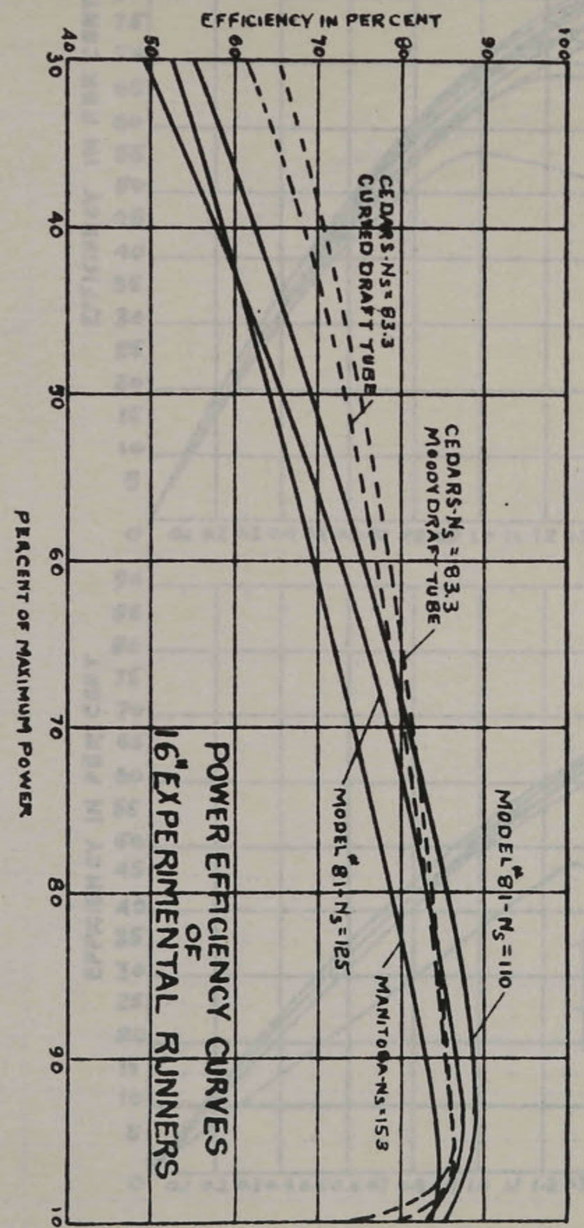


FIG. 53.

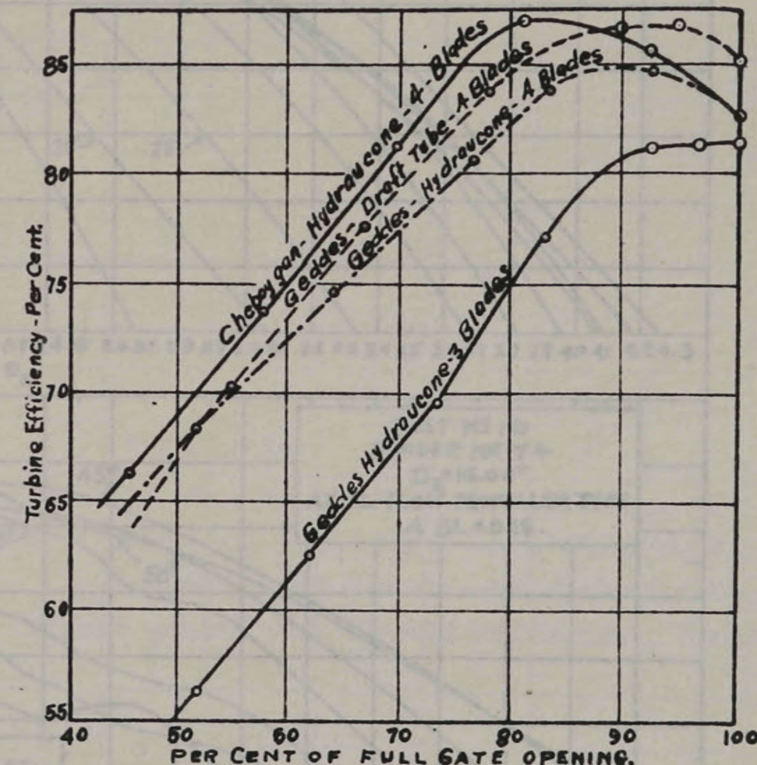


FIG. 54. CURVES REPRESENTING THE PERFORMANCE OF THE CHEBOYGAN, (Mich) AND THE GEDDES (Mich) HYDRO-ELECTRIC POWER PLANTS, WHICH ARE EQUIPPED WITH RUNNERS OF THE NAGLER TYPE.

CHEBOYGAN PLANT, HEAD 17'-6"; CAPACITY 500 K.v.a.

GEDDES PLANT, HEAD 15 FT. 600 H.P. AT 200 R.P.M.

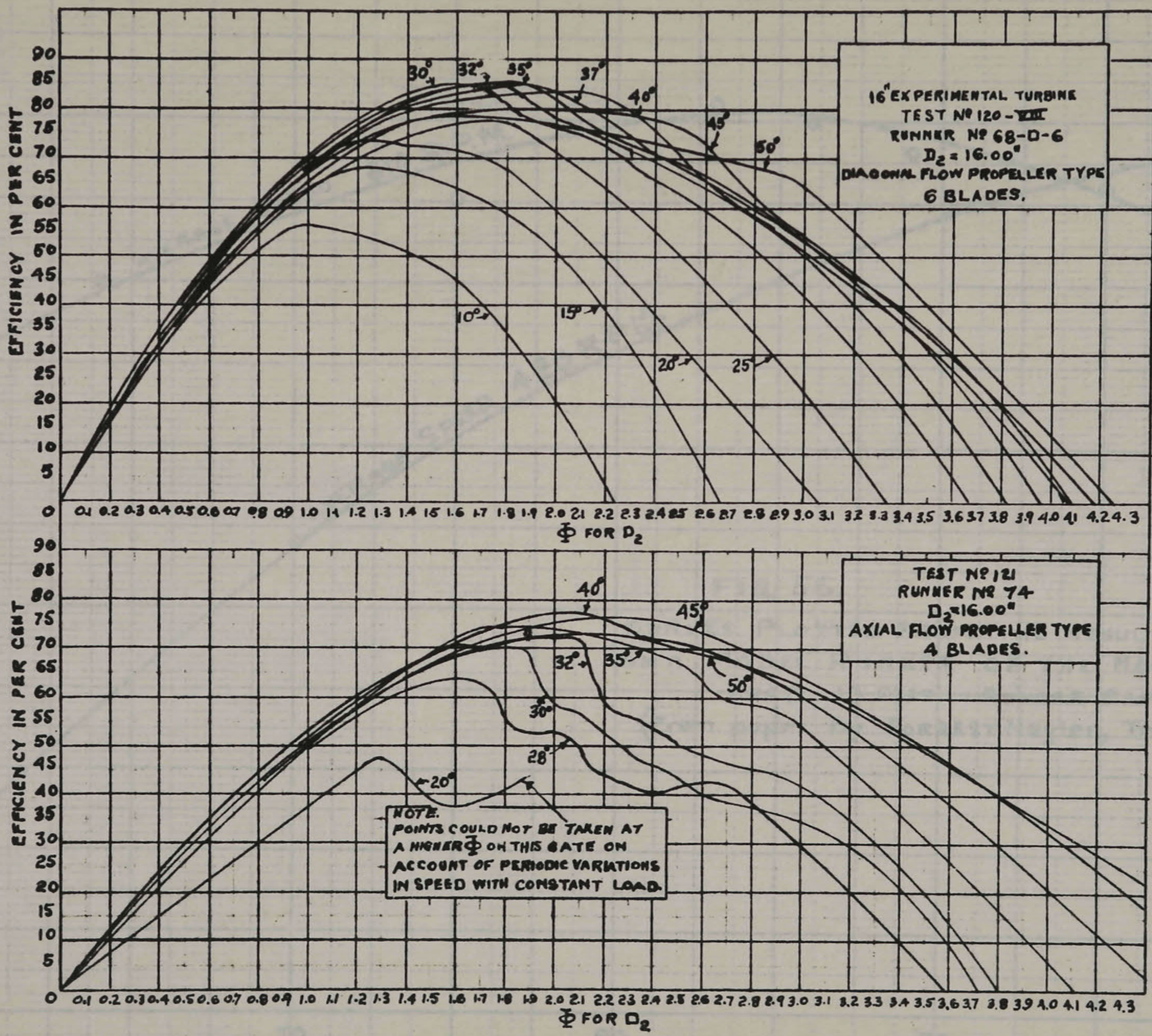


FIG. 55. Comparative Test Curves. Six Bladed Diagonal Propeller-Type Runner Compared With Four Bladed Axial-Type Runner.

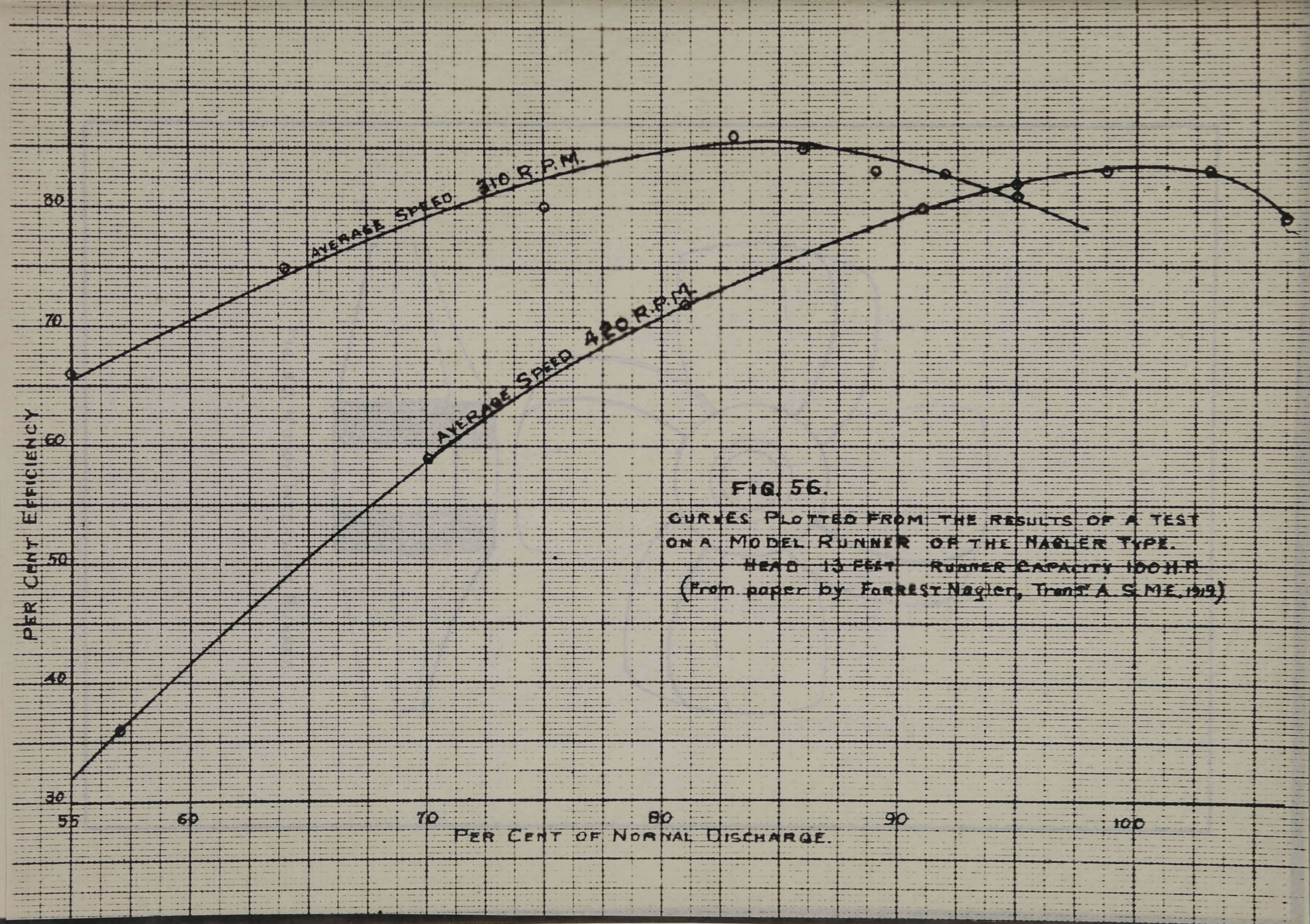


Fig. 56.

Curves Plotted from the Results of a Test on a Model Runner of the Nagler Type.
Head 13 Feet   Runner Capacity 100 H.P.
(From paper by Forrest Nagler, Trans. A. S. M. E. 1919)

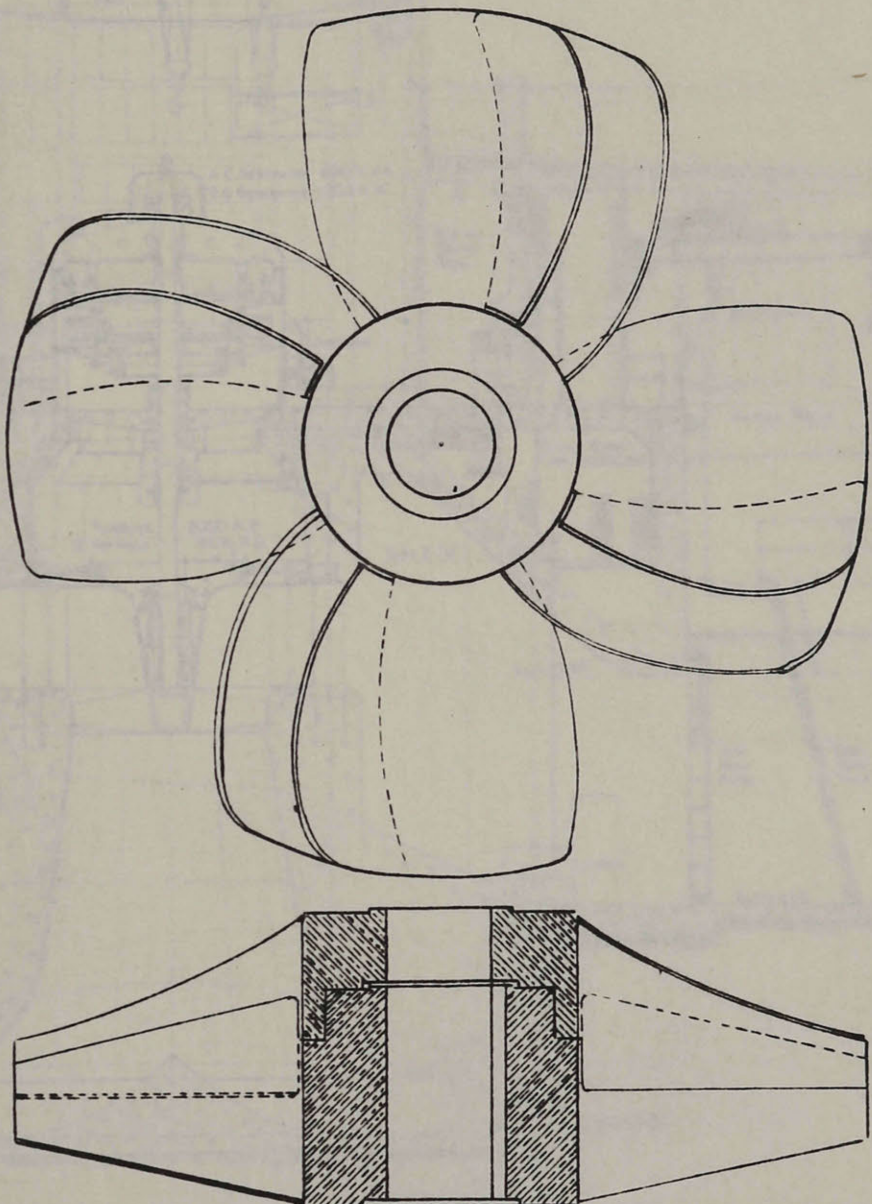


FIG. 57. MOODY INTERVANE RUNNER.

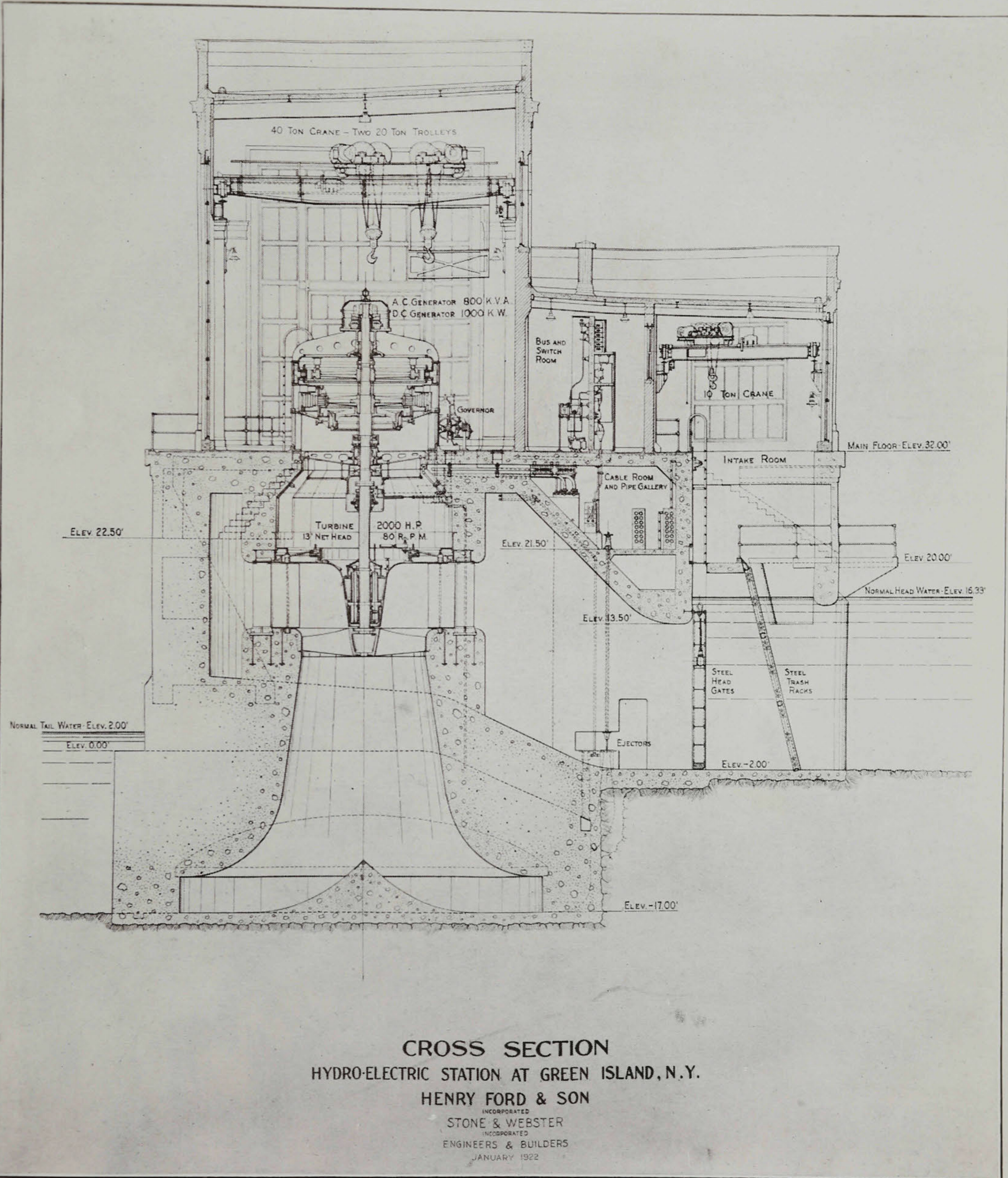
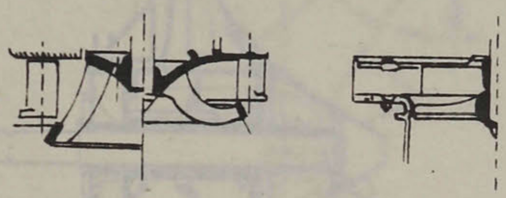


FIG. 58. Four 2200 hp., 13′ head, 80 r.p.m. Combined units placed in commercial operation June 1, 1922.

INSTALLATION OF NAGLER RUNNERS.



A - Early Kaplan Turbine
B - Kaplan Turbine of 1913.
C - " " " 1919.

Fig 59. Kaplan Turbines

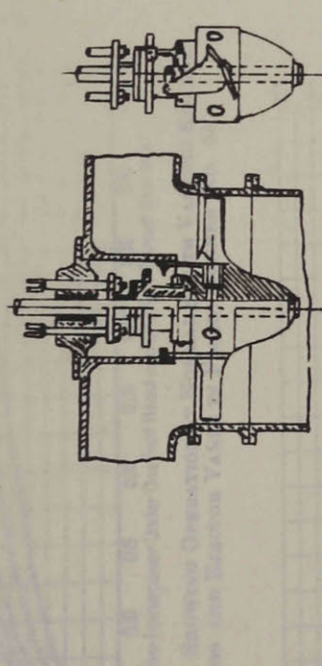


Fig. 60. Kaplan Movable Blade Turbine.

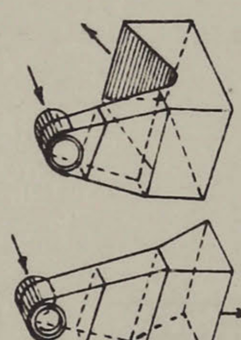


Fig. 61-A

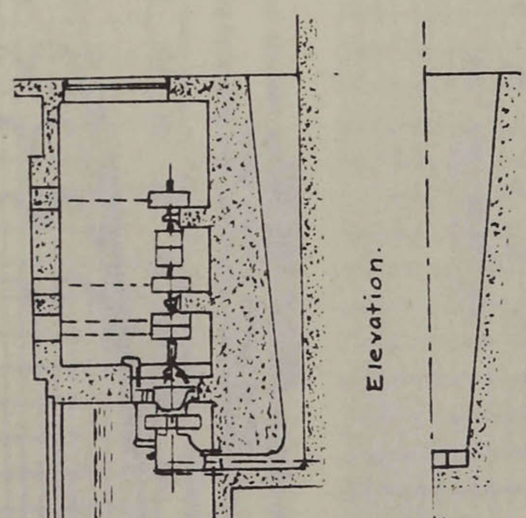
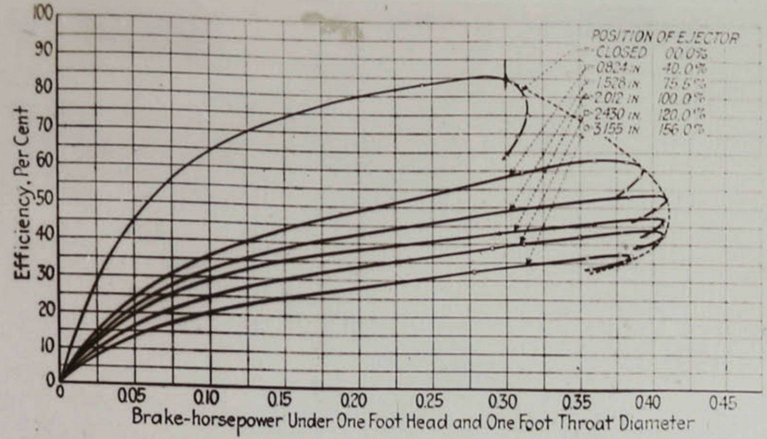
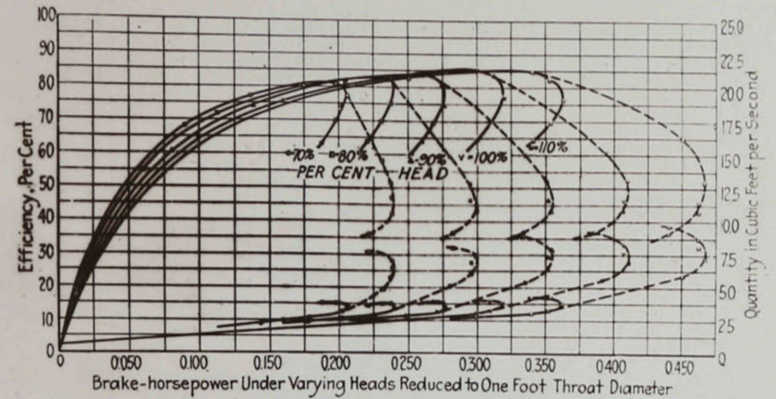


Fig. 61. Kaplan Draft Tube.



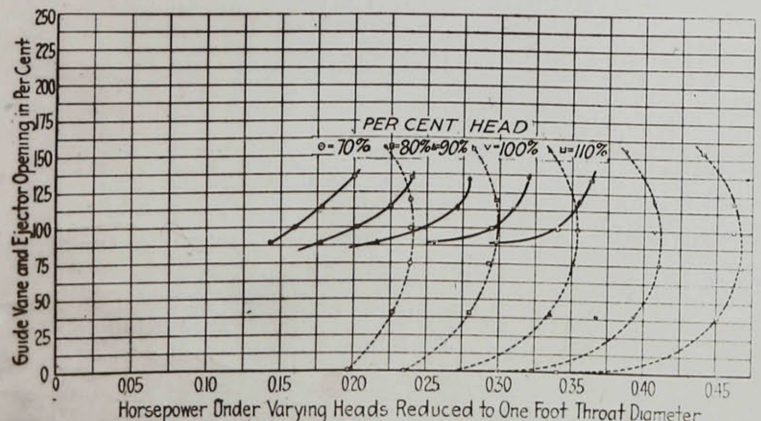
A

CURVES SHOWING OPERATION AT NORMAL ϕ WITH VARYING EJECTOR OPENINGS AND EJECTOR VANES IN PLACE; RUNNER NO. 55



B

CURVES OF HORSEPOWER, EFFICIENCY AND QUANTITY FOR RUNNER NO. 55 UNDER VARYING HEADS
Normal Operation Indicated by Solid Lines and Operation with Ejector by Broken Lines



C

CURVES OF HORSEPOWER FOR VARYING HEADS FOR DIFFERENT GATE (SOLID LINES) AND EJECTOR (BROKEN) OPENINGS, RUNNER NO. 55

FIG. 62.

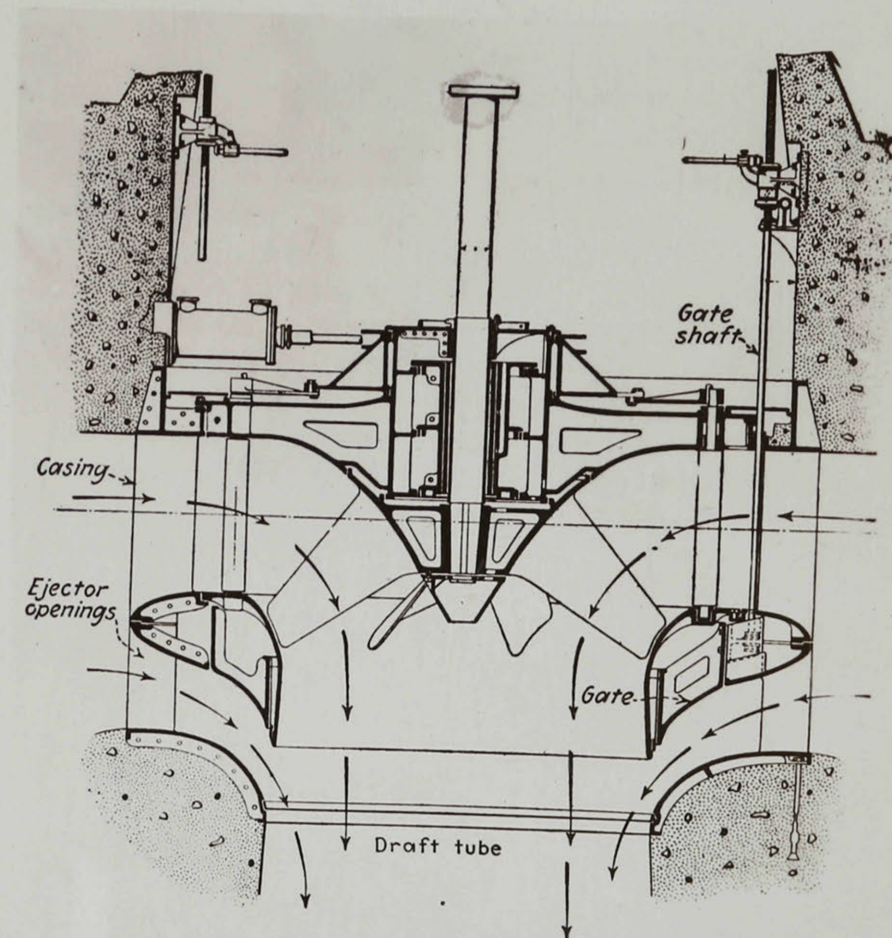


FIG. 63. VERTICAL SECTION THROUGH EJECTOR TURBINE

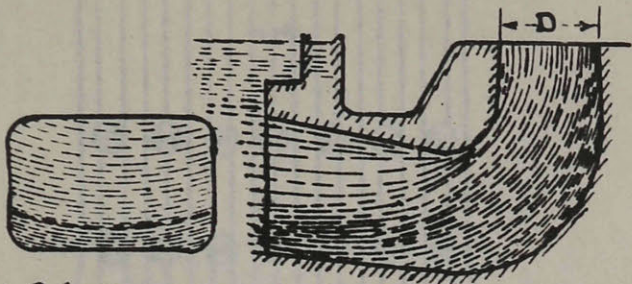


FIG. 64. REPRESENTING THE CONDITIONS OF FLOW IN CURVED DRAFT TUBES.

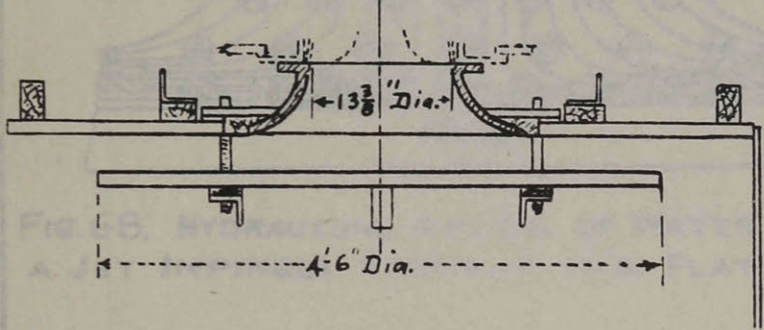


FIG. 65. HYDRAUCONE REGAINER WITH FLAT PLATE.

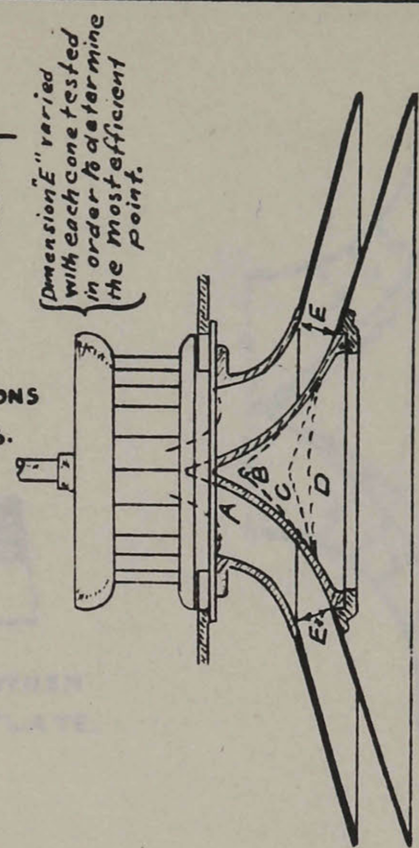


FIG. 66. HYDRAUCONE REGAINERS WITH CONE CENTRES AND CONICAL PLATES.

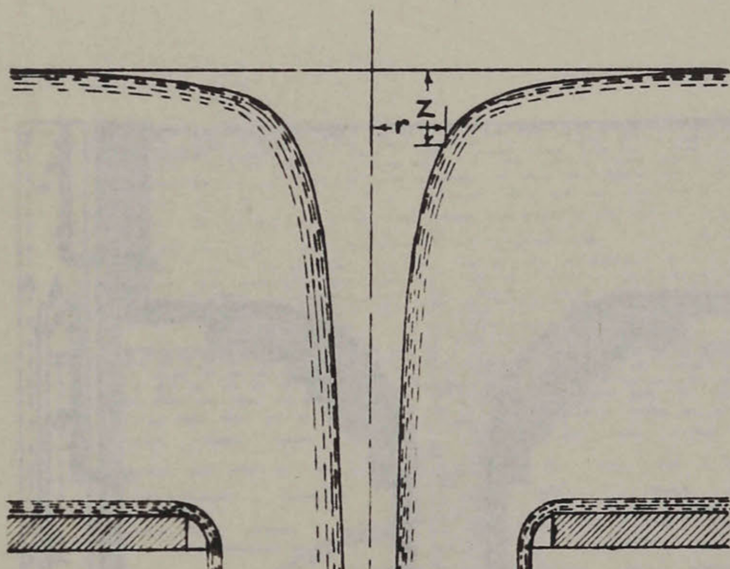


FIG. 67. DIAGRAM OF FREE VORTEX.

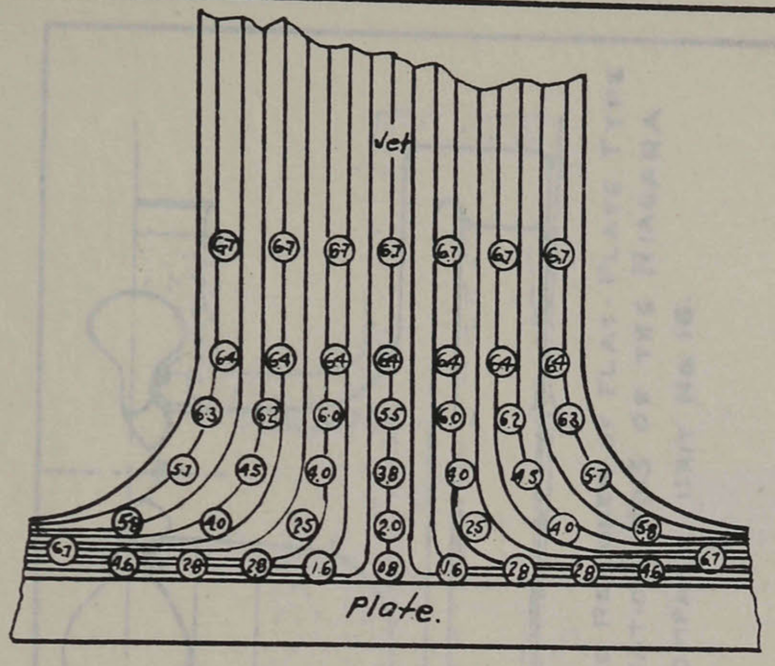


FIG. 68. HYDRAUCONE ACTION OF WATER WHEN A JET IMPINGES NORMALLY ON A FLAT PLATE.

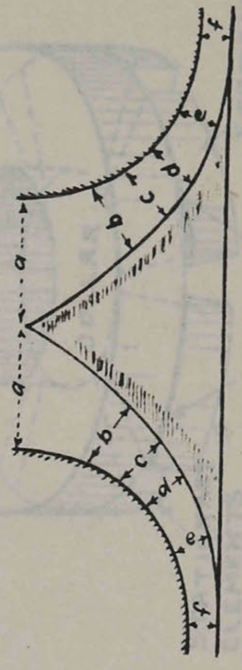


FIG. 70. EQUAL-AREA CONE.

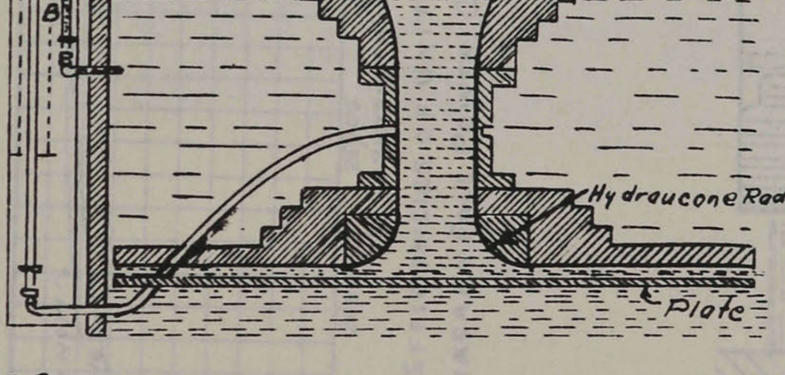


FIG. 69. APPARATUS USED IN TESTING SMALL MODEL DRAFT TUBES AND HYDRACONE REGAINERS.

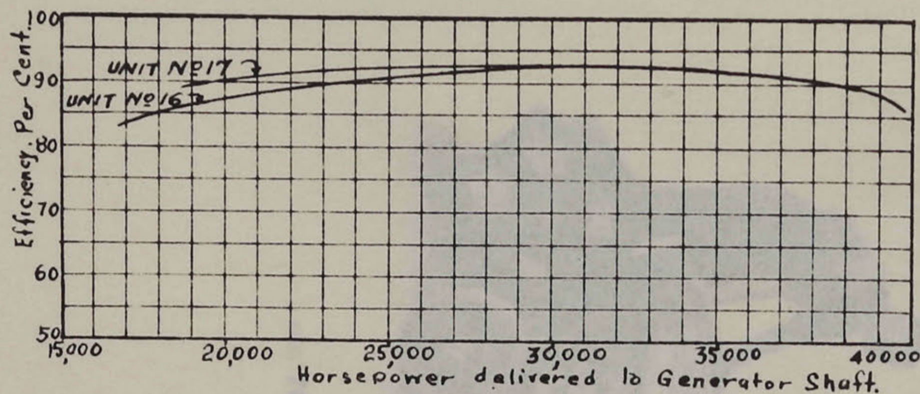


FIG. 72. EFFICIENCIES OF UNITS NO. 16 AND NO. 17. NIAGARA FALL POWER COMPANY.

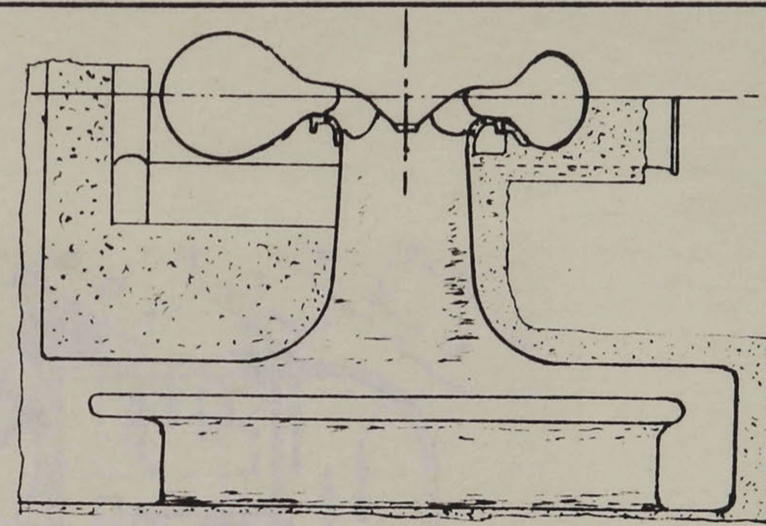


FIG. 71. HYDRAUCONE REGAINER OF FLAT-PLATE TYPE INSTALLED IN STATION NO. 3 OF THE NIAGARA FALLS POWER COMPANY. UNIT NO. 18.

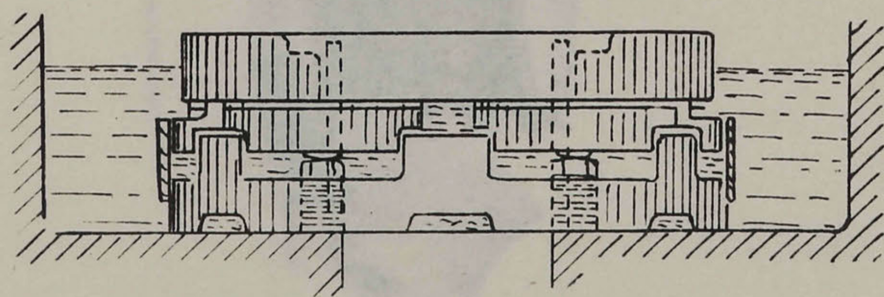


FIG. 74. ADJUSTABLE MOUNTING KINGSBURY THRUST BEARING.

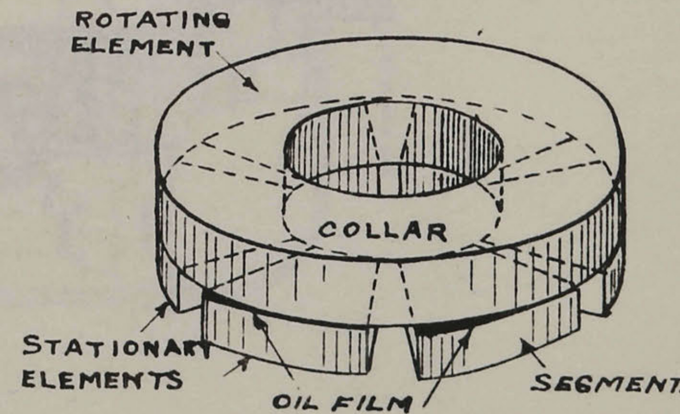


FIG. 73. THRUST BEARING OF THE SEGMENTAL BLOCK TYPE (KINGSBURY).

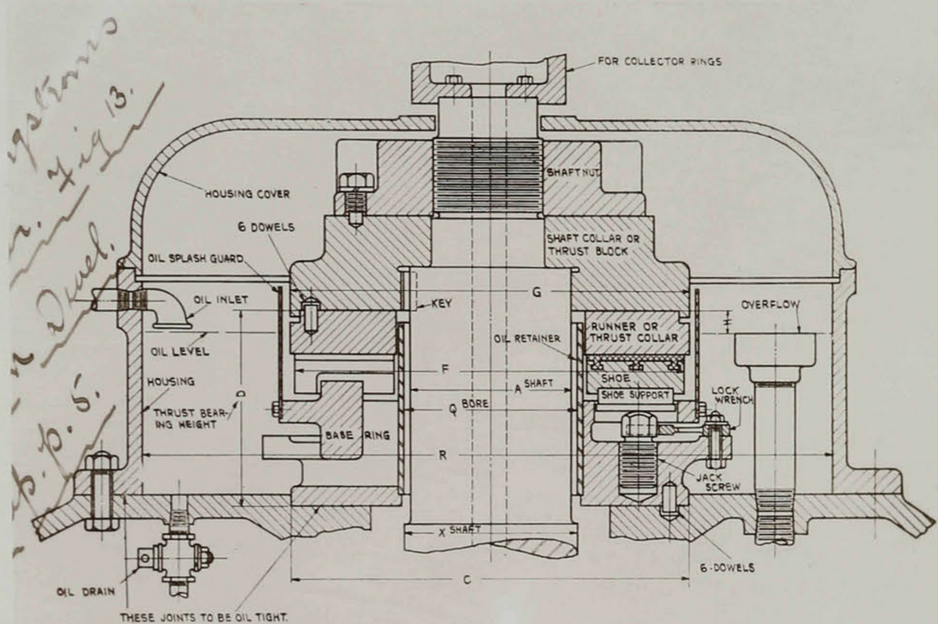
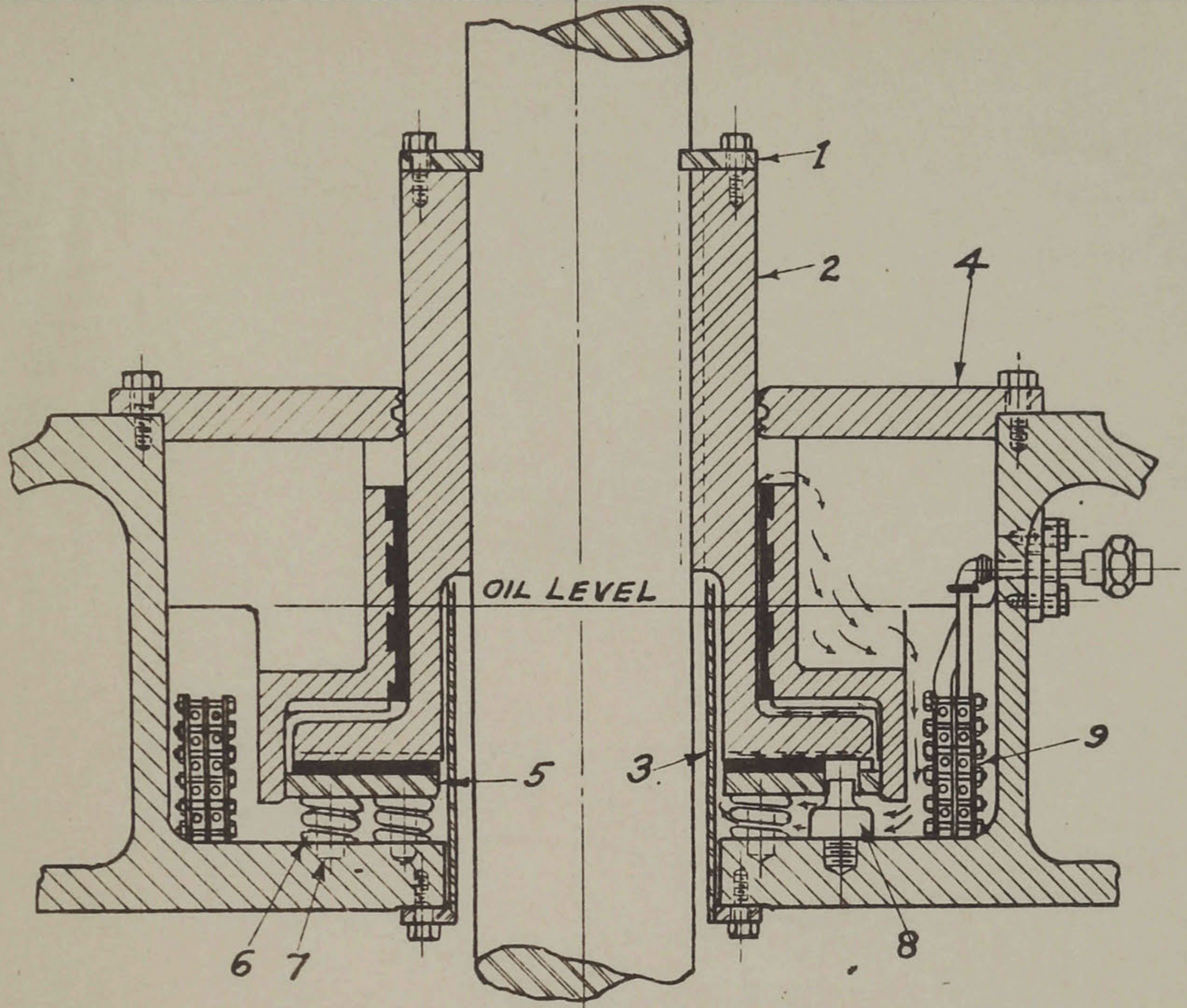


FIG. 75

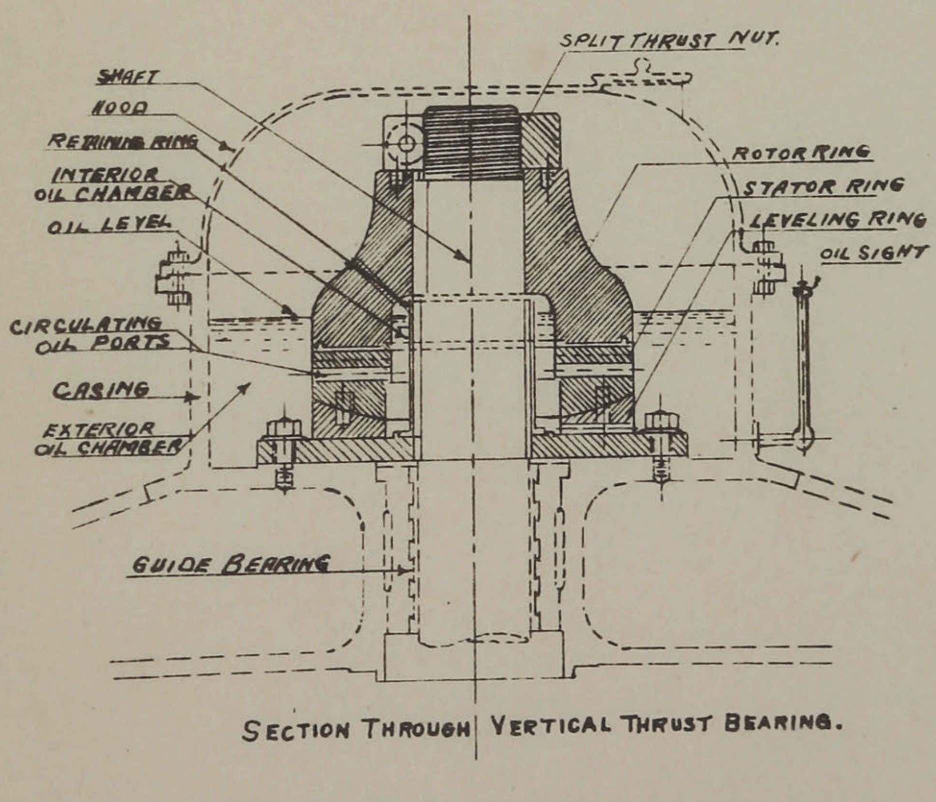
FIG. 76–THRUST BEARING FOR QUEENSTON UNIT.



CROSS SECTION OF COMBINED GUIDE AND THRUST BEARING, (Vertical).

1. Retaining Ring. 4. Guide Bearing. 7. Centre Pins for Springs
2. Thrust Collar. 5. Stationary Ring. 8. Dowel Pin
3. Oil Well Tube. 6. Springs. 9. Coding Coil.

FIG. 77. REIST SPRING THRUST BEARING.



SECTION THROUGH VERTICAL THRUST BEARING.

FIG. 78. GIBBS THRUST BEARING.