HYDRAULIC TURBINES

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THEIR DESIGN AND INSTALLATION

BY

VIKTOR GELPKE

CHIEF ENGINEER, TURBINE DEPARTMENT, AMME, GIESECKE AND KONEGAN A-G

AND

A. H. VAN CLEVE

MEM. AM. SOC. C.E.; MEM. AM. SOC. TESTING MATERIALS;
FORMER CONSULTING ENGINEER, NIAGARA FALLS
POWER COMPANY

William R. Fey

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PREFACE

IT is only within the last few years that either the engineer or the economist has appreciated the right of water powers—either latent or developed—to be included among great national resources. In fact, until electric transmission became an assured success this attitude of indifference was largely justified, for while certain manufacturing cities owed their prosperity to cheap power obtained from the waters of the rivers on which they were situated, yet these cases were exceptional. The majority of the water powers of the world were at points illy adapted for the location of industrial concerns. But with the development of electrical generation and transmission all this was changed. While the age of steam had not passed, water power development had come into its own again. So rapidly has the advance been made that it is now hard to realize that when in 1891 The Niagara Falls Power Company adopted the use of the alternation current, it was in opposition to the advice of some of the world's greatest scientists. But now that electric transmission enables the distant water power to be utilized for industrial and transportation purposes, even the non-technical portion of the community has suddenly awakened to the fact that in the potential power of water under a head we have an inexhaustible and constantly renewed source of energy that constitutes one of the greatest of national resources. While there still exists in this country a popular delusion—not absent in high places—that a water-power, like coal and iron, is conserved by non-use, yet in the end the truth will be realized that the energy of falling water unused is a national income gone forever, and the true view will obtain that such energy should be utilized as rapidly as a market can be obtained for the power developed. Only thus may the national resources in the form of combustible materials be truly conserved. It is, however, of great importance that water-powers should be developed with the greatest possible efficiency, and it is the hope of the authors that this book on "Hydraulic Turbines: Their Design and Installation," may be a timely and useful contribution to this end.

Electric generators have already reached such a high degree of efficiency that no marked improvement can be expected in that direction, and it is to the improved design of the turbine and its accessories that we must look for increased efficiency of development and economy of operation. While various treatises upon turbine design have appeared abroad, yet it is believed that this is the first book in English which covers the entire subject of the correct mathematical design of all classes of turbine and impulse wheels. The German works have been available to but a few and this book should prove useful not only to the engineering student who sees in water-power development a promising field for his life work, but also to the engineer practicing in this department of his profession. Many engineers in charge of important water-power developments have been compelled to leave the selection of the type of turbine and its design to the engineers of the concerns manufacturing this apparatus. While too much credit cannot be given to these able designers who have so advanced the art of turbine building in this country, yet it is certainly desirable that the man responsible for the success of the entire undertaking should be able to wisely choose that type and design which will suit conditions of which he has a more intimate knowledge than anyone else. But, as the authors know from personal experience, construction engineers are busy men, and it is hoped that this book will give them the necessary information without having to read a treatise in a foreign tongue.

The designer and engineering student may use this work with assurance, because hundreds of turbines have been designed and built in accordance therewith. These turbines have given high efficiencies and have proven to be correctly rated.

The book now issued is to a large extent a translation of a German work entitled "Turbinen und Turbinen Anlagen," written by Mr. Viktor Gelpke and published by Julius Springer.

Certain portions of the earlier work have, however, been rewritten, the mathematical analysis has been simplified, the book rearranged and Part III has been added. As the success and efficiency of a hydraulic development depends not only on the correct design of the turbines, but also upon the proper arrangement and proportions of the several water channels, the control of the water, the bearings of the shafts, etc., there will be found in Part I a description of various details employed as turbine accessories in water-power development, the essential principles governing the same and a classification and description of the several classes of turbines. Some portions of Part I are fundamental, but while they may appear superfluous to the experienced hydraulic engineer yet it is believed that they will be of value to the student and general practitioner.

No attempt has been made to cover the entire field of water-power engineering, but only those portions thereof which pertain directly to turbines and their accessories. Indeed, the subject of governors has been largely reserved for a second edition, except in so far as certain forms have been described in detail in Part III.

In Part II appears the mathematical design of turbines of all types, and attention is here called to the several numerical examples which are completely worked out to illustrate the several steps in the design and the use of the proper formula.

Part III has been added to give the reader detailed descriptions of important and representative hydro-electric developments. These plants have been selected not only for their magnitude, but also because they are representative of various forms of development. Thus McCall Ferry is a low-head plant with a long dam and open tailrace; the Canadian Niagara Power Company utilize the water under a moderate head and employ a long tunnel tailrace; the Kern River Plant No. 1 contains large impulse wheels operating under a high head, and so on. The descriptions are detailed and the numerous illustrations have in most cases not hitherto appeared in any book or technical paper. The authors trust that this portion of the work will not only be useful to the student but suggestive to engineers facing similar problems.

ACKNOWLEDGMENTS:

The authors desire especially to express their appreciation of the faithful and efficient services rendered by Mr. R. V. Rose, Assoc. Mem. Am. Soc. C. E., in connection with the work of translation. They are also indebted to the various individuals and corporations through whose courtesy they have been enabled to obtain the information and illustrations necessary for the preparation of the descriptions of various plants. Everywhere encouragement and hearty co-operation have been extended. Among those to whom the authors are thus indebted are the following: Amme, Giesecke & Konegan, A. G. Brunswick, Mr. Hugh L. Cooper, Consulting Engineer, New York City; I. P. Morris Company, Philadelphia, Canadian Niagara Power Company, Niagara Falls, N. Y.; The Niagara Falls Hydraulic Power and Manufacturing Company, Niagara Falls, N. Y.; and Mr. G. H. Bishop of The Southern California Edison Company, Los Angeles.

Thanks are also due to the firms who so kindly furnished the material necessary for the German work, i.e., Escher-Wyss & Co., Zurich; and Thos. Bell & Co., Kriens.

V. G.

December, 1910. A. H. V. C.

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TURBINES AND THEIR ACCESSORIES

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TURBINES AND THEIR ACCESSORIES

In planning an installation for the purpose of utilizing a water power, consideration must be given to the following constructive, mechanical, and technical problems:

A. The method of regulating the water supply;

B. The means of conducting the water to and from the power station;

C. The removal from the water supply of injurious floating and suspended matter;

D. The means of controlling the water supply to the penstocks or other forms of intake;

E. The design of machinery in the power station;

F. The design of foundations and superstructure of the power station;

G. The local use or the transmission of the power developed.

Within the scope of this chapter it will be possible to consider in any detail items *B*, *C*, *D*, and *E* only, and then only where they are closely associated with the design of turbine machinery. The remaining items, *A*, *F*, and *G*, belong to another branch of water-power engineering and will be only briefly alluded to here.

A. THE METHOD OF REGULATING THE WATER SUPPLY

In connection with the regulation of the water used for operating the power plant there must be considered all of the construction work necessary at the head works of the canal or penstocks. Among the objects to be attained by such head works may be mentioned the following, viz.:

(1) The securing of the greatest practicable head;

(2) The protection of the power plant against ice, driftwood, and injurious suspended matter.

The head works for low-head installations will therefore comprise one or more of the following forms of construction: stationary dams, movable dams, stationary weirs, movable weirs, timber shoots, fishways, spillways, flushing gates, apparatus for removing sand and pebbles, machinery for operation of movable dams, etc. In connection with medium and high-head installations the following may be mentioned: dams of various forms and materials, masonry intake to pipe lines, tunnels, or canals; pipe lines, tunnels, or canals to forebay of power house; together with forebay construction of various designs.

B. THE MEANS OF CONDUCTING WATER TO AND FROM THE POWER STATION

The character and quantity of the water supply, the operating head, the topography of the country, and other controlling features will lead the engineer to adopt for conveying the motor water to and from the power house either an open conduit, a closed conduit, a pipe line, or in some cases a combination of all three.

Open Conduits.—An open conduit is usually a canal having a rectangular or trapezoidal cross-section, but wooden flumes are extensively used, especially for plants located in the Western States of America, where suitable lumber is cheap and abundant. The wetted perimeter of a canal may consist of the material through which the canal is excavated, or the canal may be lined with wood or masonry, suitably supported.

Closed Conduits.—A closed conduit usually consists of a concrete tube, or a tunnel, lined or unlined. Such a form of construction may be employed to advantage where the water must be conveyed through hills or over a rolling country. Tunnels are of two classes: gravity tunnels and pressure tunnels. As a rule, and especially in the older tunnels, water flows through a tunnel without pressure, owing to the low tensile strength of the material with which the tunnel is lined, the tunnel, consequently, being only partially filled. Such a tunnel is known as a gravity tunnel, while a tunnel used for water under a head is known as a pressure tunnel. Within recent years it has been possible to construct a tunnel lining so that water may be safely transmitted under pressure, such pressure varying with the character of the material through which the tunnel is excavated, and the quality of the material employed for the lining. In considering the use of a pressure tunnel it must be remembered that while it has an advantage in increasing the power available and in providing for quick automatic regulation, yet it must be considered that in the course of such regulation the tunnel will be subjected to an increase in pressure which may at times exceed the allowable pressure on the lining unless a surge tank or a standpipe is provided to partially overcome this difficulty. To entirely overcome excessive pressures an automatic pressure regulator or a correctly designed governor (Fig. 138) is required, but these are expensive where large quantities of water are to be handled.

As a rule, concrete tubes, while comparatively inexpensive, are used only for low pressure and small volume of flow, but the use of reinforced concrete has widened their application, and a concrete pressure tube 18 feet in diameter is now being used for one of the largest installations in the Province of Ontario, Canada.

Cast-Iron, Steel, and Wooden Pipes, with necessary anchorages, bearing points and expansion joints.

A more detailed discussion will be given to the above-named forms of pipes for transmitting water under pressure.

Cast-Iron Pipe.—The use of cast iron in high-pressure pipes of large diameter should be regarded with caution on account of the porosity of the material, its low tensile strength and elasticity, together with the inequalities of thickness of the pipe and the unequal expansion and other defects of manufacture. On the other hand the advantages of cast iron must not be overlooked, viz.: cheapness, adaptability for specials of complex shape, low oxidation, and stability. For pipes of a diameter of 6 inches to 20 inches subject to a pressure not exceeding 200 pounds per square inch, and for pipes of larger diameter operating under a pressure not exceeding 125 pounds per square inch, standard bell and spigot or flanged cast-iron pipe may sometimes

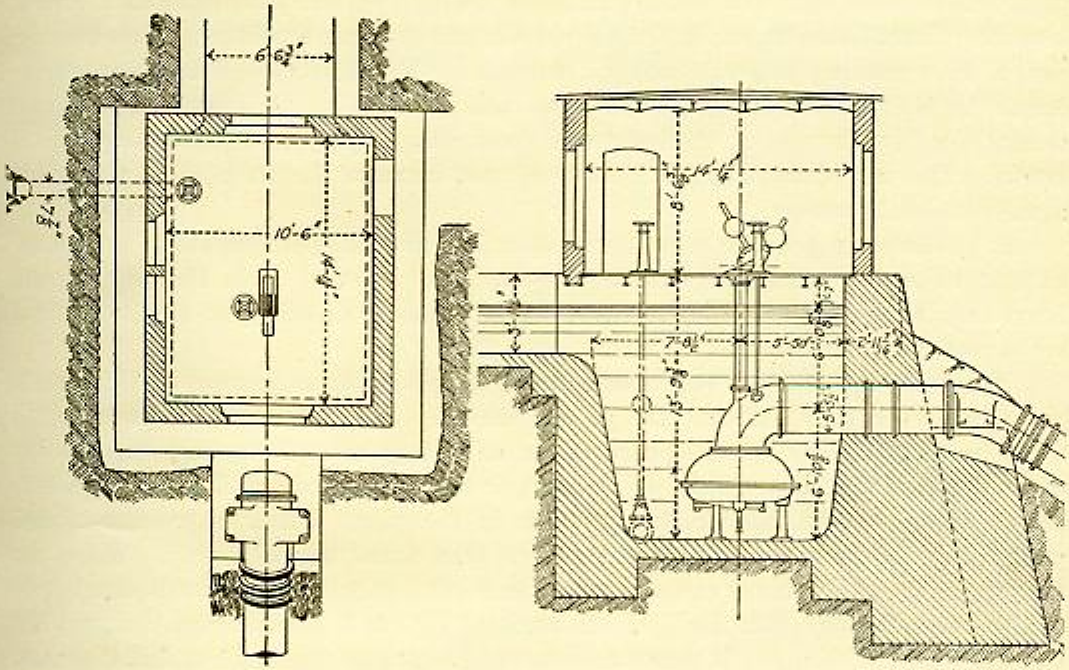


FIG. 1.—Automatic Cup Valve at Upper End of Penstock.

be used to advantage. The working stress in the material should not exceed 2200 pounds and 1500 pounds per square inch respectively in the above cases.

The usual formula for determining the thickness of pipes is as follows:

$t = \frac{PD}{2T}$, where

t = thickness in inches;

P = working pressure in pounds per square inch.

D = diameter in inches.

T = allowable working stress in pounds per square inch of material.

But it must be remembered that this formula is only applicable when the pipe is truly circular, the material is uniform, and where the shell is of uniform thickness throughout; in other words, when the workmanship is perfect. When the conditions differ from those enumerated, or when specials are to be designed, the thickness must be increased above that determined by the above formula. The weaker portions of the castings must also be reinforced by ribs and fillets. On account of the low elasticity of cast iron it must not be assumed that the stresses will be uniformly distributed over the entire cross-section. In the design of the castings great care must be exercised to prevent the reduction of strength by sharp corners and improper location of bolt holes, especially in the vicinity of the weaker points. As an illustration of strengthening properly designed castings see Fig. 3, where the manhole is reinforced by numerous ribs. (Fig. 55.) The concave side of an elbow may best be strengthened by the use of sickle-formed cross ribs to provide against longitudinal cracks. The use of longitudinal ribs would not increase the strength of the elbow against internal pressure.

It is the usual practice to test cast-iron pipe and flanged specials to a pressure at least twice the anticipated working pressure, and in many cases the test pressure bears even a higher proportion to the working pressure, especially if large variations are expected in the latter.

Flanges.—It is safe to use smooth-faced flanges up to a pressure of 185 pounds per square inch provided that the pipe is carefully manufactured, and is inspected when being laid. Smooth flanges have the advantage of enabling easy replacement of broken castings; but, on the other hand, it must be remembered that the gaskets are subjected to the full internal pressure in the pipe. Under a pressure of 285 pounds per square inch rubber gaskets have been found unsatisfactory. Where the flanges are smoothly faced and the pressure is not excessive, red lead is extensively used between flanges. On the right-hand side of Fig. 2*a* is shown a well-designed but expensive form of flange. It possesses a turned recess into which the gasket is pressed and the gasket is thus prevented from being forced out.

In designing both bell-and-spigot and flanged cast-iron pipe to be used under a working pressure of 150 pounds per square inch and less it is advisable to adhere to the standard dimensions used by the manufacturers of such pipe, as the cost of the castings is thereby decreased.

Among the best materials for gaskets are rubber, both flat and round, and corrugated copper sheets. There are many patented gaskets on the market, one of the best of which consists of concentric rings of copper and lead. Under high pressures, and especially in cases where the flanges are not absolutely parallel, lead or type-metal gaskets are often employed. In order to insure the tightness of the joint, the lead should be caulked from the inside of the pipe in such cases. If a round rubber gasket is used a groove having a triangular cross-section should be turned in the flanges as shown in Fig. 2*a*. Concentric circular grooves should be turned in the face of the flange when flat rubber is used under high pressures.

Steel Pipe.—For pressures of over 200 pounds per square inch, riveted or welded pipe composed of steel sheets is frequently used where the diameter exceeds 16 inches.

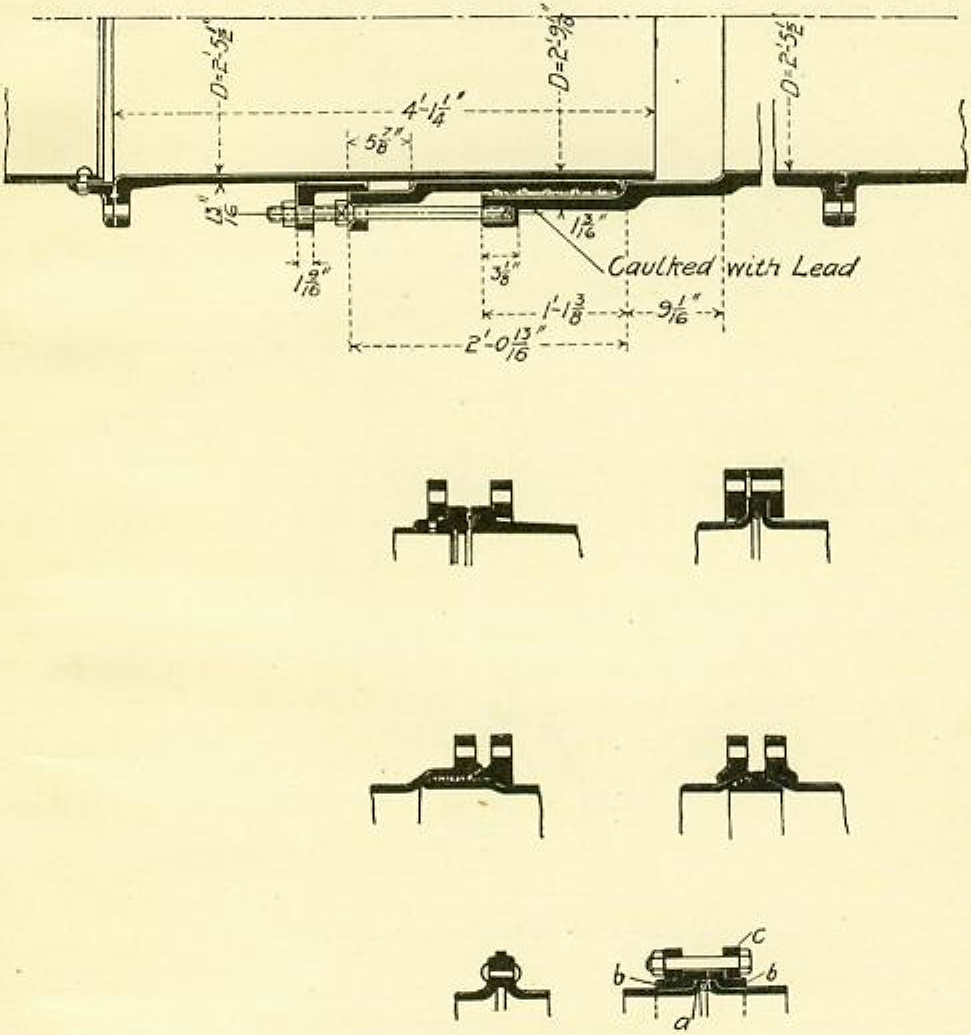


Fig. 2.—Various Forms of Flanged Connections.

For lower pressures spiral-riveted steel pipe is often employed. For large-size pipes used as penstocks the flanges are usually riveted on, such flanges being welded rolled steel angles, or castings, preferably of steel. For the smaller size of pipe, say up to 24 inches diameter, the best practice requires the flanges to be welded to the welded pipe. Seamless-drawn pipe, with flanges welded or screwed on, are also used in the smaller sizes for high pressures. Loose flanges held in place by bands passing around the pipes and bolted together are also used.

Steel is more homogeneous than cast iron, and has a far higher modulus of elasticity. It is, therefore, more reliable than cast iron to resist unlooked-for pressures. Steel pipe is cheaper than flanged cast-iron pipe in the larger sizes. It,

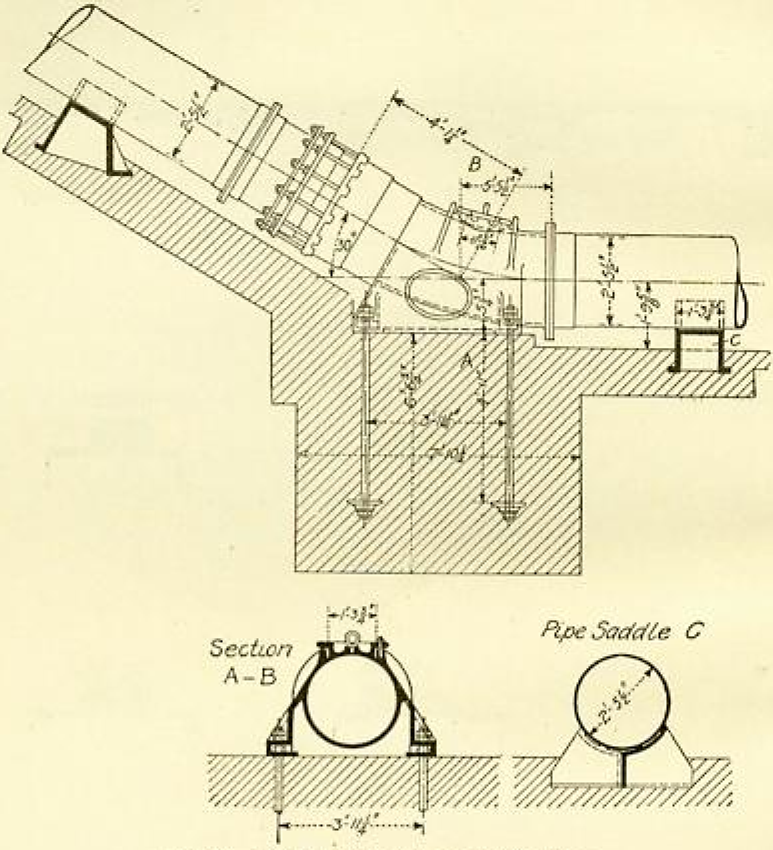


FIG. 3.—Penstock Anchorage and Saddles.

however, has the decided disadvantage of more rapid corrosion, and steel pipe should always be designed with an additional thickness above that theoretically required for strength in order to provide against loss from corrosion. In riveted pipe the stress in the plates adjacent to the rivet holes should not exceed 7000 pounds per square inch.

In designing both pipe and valves of large size, regard must be had for the transportation requirements of the railways over which they are to be transported and clearance diagrams should be obtained and adhered to. At times it is necessary to ship penstocks "knocked down," i.e., the plates are cut, rolled and punched in the shop and riveted wholly or in part in the field.

In order to guard against excessive stresses due to expansion and contraction where the pipe is to be subjected to material changes in temperature, it is well to place expansion joints at suitable intervals. A simple form of such a joint is shown in Fig. 2*f*. This joint possesses the necessary elasticity to take up variations in temperature, but the desired result is obtained by greatly increasing the applied stresses in the plates.

Welded pipe should be at least $\frac{5}{16}$-inch in thickness, although $\frac{1}{4}$-inch is used for 12-inch pipe. The stresses in the longitudinal seams should not exceed 7000 pounds per square inch. Not more than nine-tenths of the net cross-section of the welded joint should be used as an effective cross-section. In order to reduce to a minimum dangerous internal stresses, the steel for welded pipe should be rolled hot and welded immediately thereafter. It is well to draw attention to the advantages of weldless drawn tubing. As it has no longitudinal seams, the material in the pipe is practically homogeneous, and its working stress may therefore be assumed as high as 10,000 pounds per square inch.

Some of the best designs for flanged connections are shown in Fig. 2 (*a* to *g*). At the left-hand end of Fig. 2*a* is shown the usual means of riveting the flange to a penstock, while the right-hand end shows the flange welded direct to the pipe. The welded flange represents the best practice, especially for seamless-drawn tubing. Fig. 2*b* shows a riveted and a welded flange bolted together by means of loose rings riveted or welded around the pipe. Fig. 2*c* shows a flanged joint patented by the Ferrum Company of Kattowitz, Germany. It consists of two loose flanges which are bolted against the out-turned ends of the pipe, the gasket being inserted between the latter. The flanges are so designed that the bolts are not subjected to bending and a portion of one flange overlaps the joint, and thus prevents the gasket from being blown out. Fig. 2*e* shows a flanged joint patented by Sulzer Brothers of Winterthal, Switzerland. The ends of the pipe are flared, and are pressed against an internal ring by two loose flanges, the gaskets being placed between the internal ring and the ends of the pipes. Another flanged connection patented by Sulzer Brothers is shown in Fig. 4*b*, and was used at the hydro-electric plant at Engelberg near Luzerne, Switzerland, under a pressure of 285–380 pounds per square inch. The arrangement shown in Fig. 4*c* was designed by Thos. Bell & Co., but was used for pressure not exceeding 285 pounds. Fig. 2*d* shows an interesting combination of a poured and flanged joint. An excellent flanged joint is that shown in Fig. 2*g*, the same being adopted and patented by the German pipe manufacturers in Düsseldorf. In this joint the ends of the pipe are slightly flanged, the gasket being inserted between the ends. Against the projecting parts are pressed two loose rings, made in halves, the pressure being exerted by the bolts connecting the two other flanges, which are complete rings. The inside diameter of the outside rings is greater than the outside diameter of the flanged pipe ends, and the entire joint may thus be dismantled. At times this may prove a great advantage. One of the inner rings overlaps the joint and thus secures the gasket.

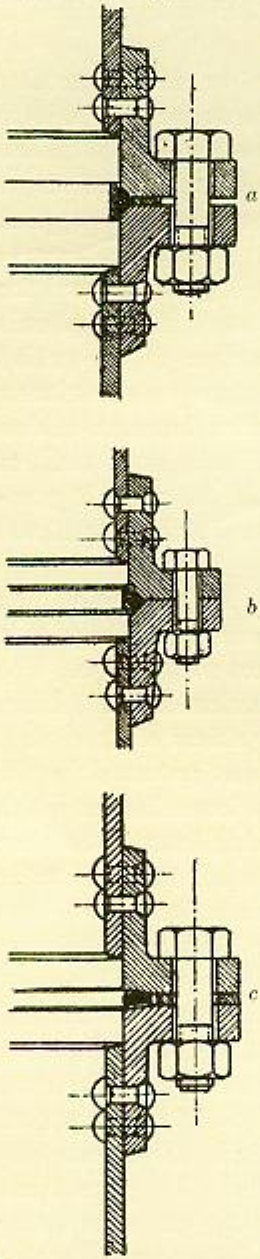


Fig. 4.—Flanged Connections with Packed Joints.

Great care should be exercised in the design of plate-steel distributing pipes leading from the main header to the wheel cases. Experience has shown that it is well to employ a greater thickness of metal than that used in the portion of the penstock or header to which such distributing pipes connect. This is due to the fact that the distributing pipes connect at an angle and must therefore be built up of a number of small sheets and consequently suffer a loss of strength on account of the seams, a loss that must be replaced in some manner. The added strength may also be supplied by saddles built of steel plates having a thickness 1½ to 2 times that of the pipe and placed under the branch pipes. In order to diminish as far as possible the loss of head caused by the diversion of water from the header pipe or penstock to the branch pipes, the latter should join the former at as acute an angle as possible—usually 45° or 60°. (See Figs. 6 and 6*a*). Penstocks should be designed in such a way that the flanges are accessible in order that the bolts may be tightened and gaskets renewed when necessary.

Anchors and Foundations for Penstocks. The simplest method of anchoring a penstock or draft tube is to cast or weld rings around it and to build such rings into the foundation walls. (See Fig. 1.) This method has the disadvantage of making it impossible to remove the section of pipe on which the rings are cast without cutting out the masonry. Where castings are employed as parts of the penstock the best anchorage is obtained by casting lugs on such sections and then securing these lugs to the masonry foundations by anchor bolts. Those sections of the penstcok which are anchored must be carefully designed so as to be capable of withstanding the stresses due both to the water pressure and the weight of water and penstock, and to transmit said stresses to the foundations. The latter must, of course, be so designed as to safely receive the stresses transmitted through the anchors. In this connection special care should be exercised in planning the foundations and anchors at angles in the penstock at its point of entry to the power house.

It is well to consider every bend in a penstock as a fixed point, while between such fixed points the straight penstock should be so erected as to allow for longitudinal expansion and contraction. Cast-iron saddles anchored to masonry foundations may be used as supports as shown in Fig. 3. In the case of a very steep penstock it is well, where the expense is not prohibitive, to connect the anchorages by a continuous wall whose foot is securely anchored to a cross-wall as shown in Fig. 3. In designing anchors for large penstocks attention must be given to the tendency of the pipe to collapse. In such cases the saddles should encircle at least one-third, and preferably one-half, of the circumference of the pipe, and should be strong enough to prevent the collapse. It is not necessary that the penstock should rest on the entire perimeter of the saddle, but it is preferable to arrange it so it may rest on several equally spaced lugs which may then be filed or chipped to equalize any irregularities in the circumference of the penstock. The same result may be accomplished by using properly fitting wooden bearing blocks secured to the saddle. It is desirable that the saddles be so designed as to reduce the friction on them of the penstock to a minimum so that it may be free to expand and contract. Rollers are sometimes used to advantage, and in an installation directed by one of the authors a pipe line nine miles in length was supported at the joints on longitudinal steel rails, anchors and expansion joints being placed alternately every one-quarter mile.

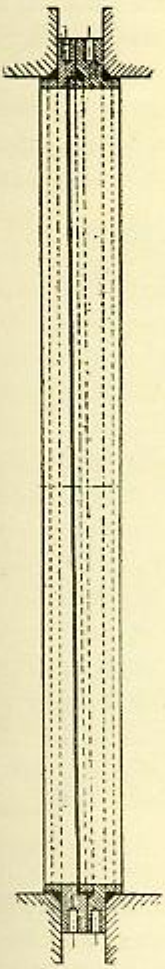


Fig. 5.—Double-Wedge Rings for Angles in Penstocks.

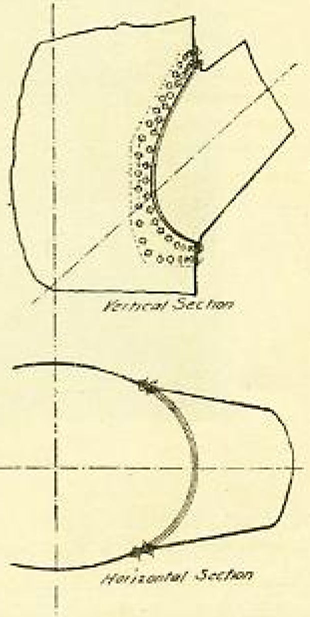


Fig. 6.—Connection of Main and Branch Penstocks.

To provide for slight angles in the penstock the use of double wedge-shaped rings is recommended. This arrangement, shown in Fig. 5, was first introduced by Sulzer Brothers, and was employed on the penstock of the hydro-electric installation at Engelberg, near Luzerne, Switzerland.

Expansion Due to Changes in Temperature. So long as a penstock is filled with water it will have practically the temperature of the latter and will consequently be subject to only slight variations. This is particularly true when the penstock is covered with earth, thus protecting it from the heat of the sun. In such cases the expansion and contraction of the penstock is negligible, and it can best be laid as a continuous, riveted length with possibly the use, at considerable intervals, of the simple form of expansion joints shown in Fig. 2*f*. Where the empty penstock may be exposed to the heat of the sun, or for other reasons a considerable expansion and contraction will take place, it becomes necessary to install well-designed expansion joints which will allow free longitudinal movement in the penstock, and thus relieve the shell of the penstock of excessive temperature stresses. Such expansion joints

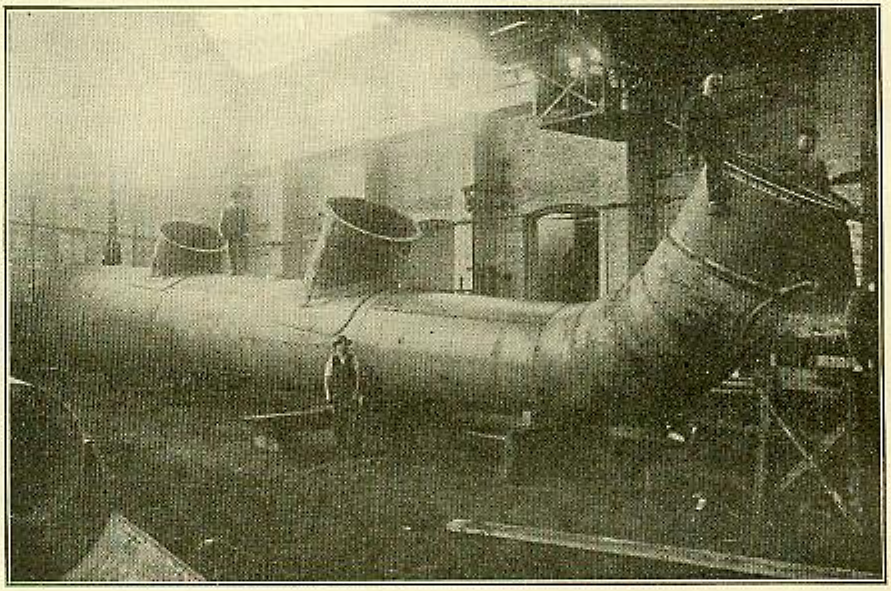


FIG. 6*a*.—Photograph of Header and its Connections.

may be placed adjacent to the elbows or other fixed points on the penstock, and may be combined with the castings which have been selected as anchor points. One form of such expansion joint is shown in Figs. 2*a* and Fig. 3, the design being such that the casting used for the elbow need not be turned. The moving piece is a separate casting held in the penstock elbow by a poured lead joint, and secured in place by bolts. The moving piece is turned so as to slide freely in the fixed piece, and is provided with a stuffing box to prevent leakage. It must be remembered that the effect of an expansion joint is entirely vitiated if the penstock cannot move on its supports between the fixed points.

Method of Venting, Filling, and Emptying a Penstock.—It is important to install air pipes at the high points of a penstock, both to admit air when the penstock

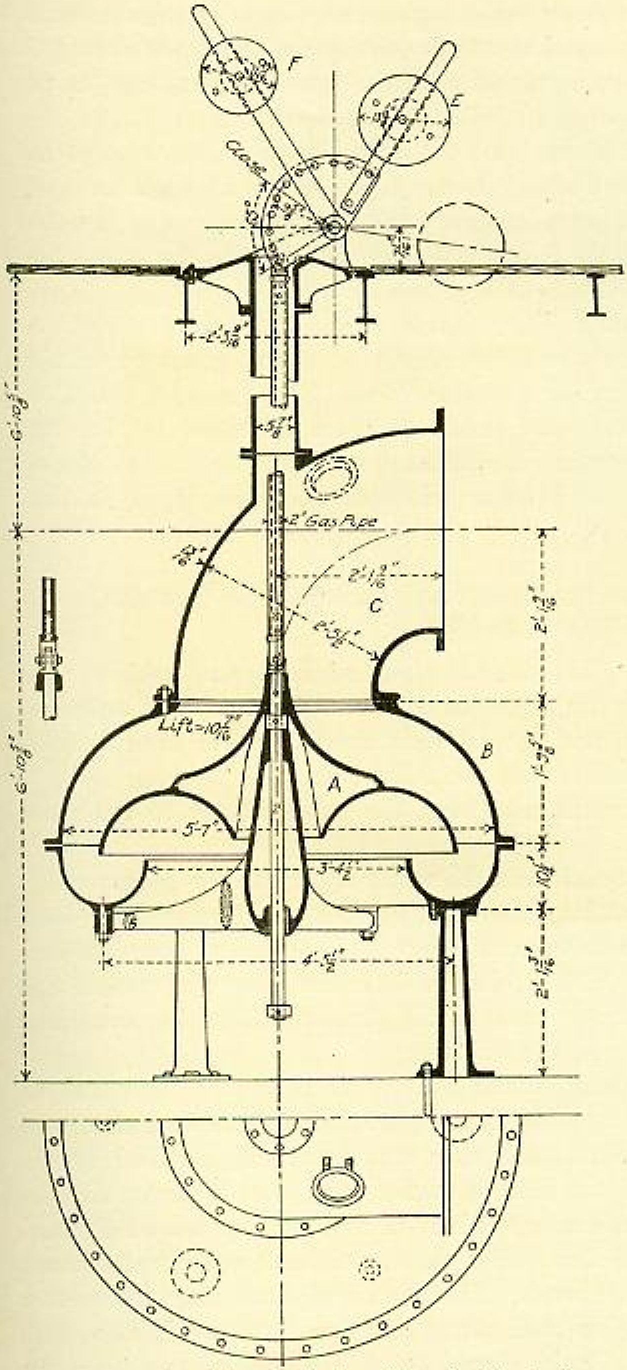


Fig. 7.—Details of Automatic Cup Valve for Regulating Flow into Penstock. Section and Plan.

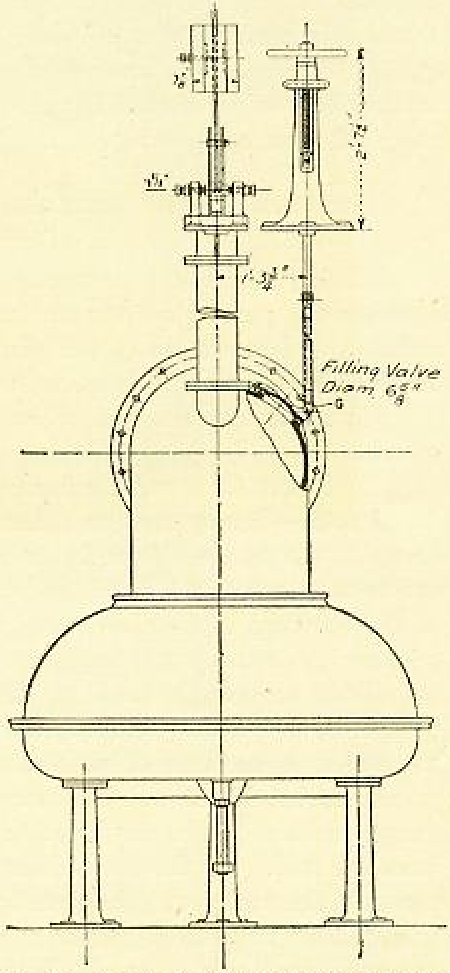


Fig. 8.—Details of Automatic Cup Valve for Regulating Flow into Penstock Elevation.

is emptied and also to prevent the accumulation of air at such points during operation. The upper end of such pipes must project above the surface of the head water. It is evident that where such an air pipe is placed near the forebay it must be on the power house side of the head gate. (See Figs. 1, 7, 8, and 13.)

In designing the head gates it must be borne in mind that a penstock must not be filled too quickly. A wicket gate should, therefore, be placed in the head gate, a by-pass pipe and valve provided, or a special form of entrance valve installed (a design of the latter is shown in Fig. 7, and will later be described). These precautions are especially necessary in the case of high heads.

In order to empty the penstock a valve which is easily accessible should be placed at its lowest point. Manholes should be placed on the penstock at suitable intervals in order to enter it for cleaning and inspection. The penstock should be thoroughly cleaned from all dirt, rust, and scale, and then protected by the best quality of paint. The best practice requires the plates of steel penstocks to be cleaned in the shop before assembling, the sand blast or acid bath being employed, immediately followed by painting.

C. THE REMOVAL FROM THE WATER SUPPLY OF INJURIOUS FLOATING AND SUSPENDED MATTER

Settling basins for preventing solid material, such as sand and pebbles, from entering the penstocks form an important part of a power plant, as they materially reduce the depreciation of the plant, and such devices should receive careful study. In this chapter we will, however, consider only the following:

(1) Racks for removal of coarse material from the water supply to the water wheels.

(2) Filters for removing fine material from the water supply to the governors.

Racks.—Racks may be either of large or small spacing, the former being those in which the bars are from $1\frac{1}{2}$ inches to 4 inches apart, and the latter where the bars are from $\frac{5}{8}$ inch to $1\frac{3}{8}$ inches apart. The spacing is determined by the cross-section of the openings in distributor and runner vanes of the turbine. It is also sometimes necessary to consider the requirements of the fish laws.

Racks are usually made of flat bars having sections from $\frac{5}{32}$ inch by 1 inch to $\frac{3}{8}$ inch by $3\frac{1}{2}$ inches held together by bolts and separators. These bars should be assembled in sections of such a weight that they can be readily handled by the raising devices installed. In place of the flats ordinarily used, bars having a torpedo-shaped section can be employed to advantage if bars of such section can be obtained from the mill. By the use of bars of this design the loss of head as the water passes through the racks may be greatly reduced. This increased head may represent a considerable amount of power in a low-head plant, and be worth much more than the additional cost of special bars. Racks should be so designed as to provide for

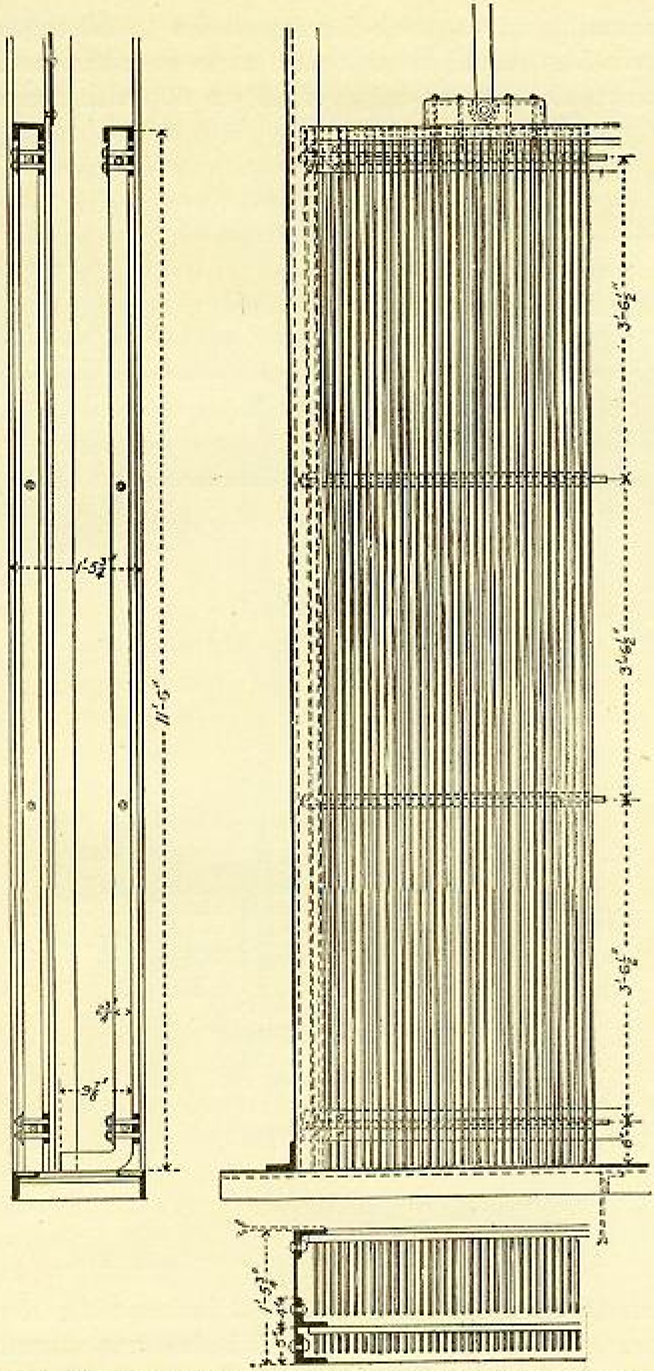


FIG. 9.—Double Vertical Rack for Protecting Entrance to Penstock.

the convenient operation of the rack-cleaning device to be employed. The bolts holding together the bars should be so located as to clear the teeth of the cleaning device, the rack should be at an angle of 45° to 60° with the vertical, and the

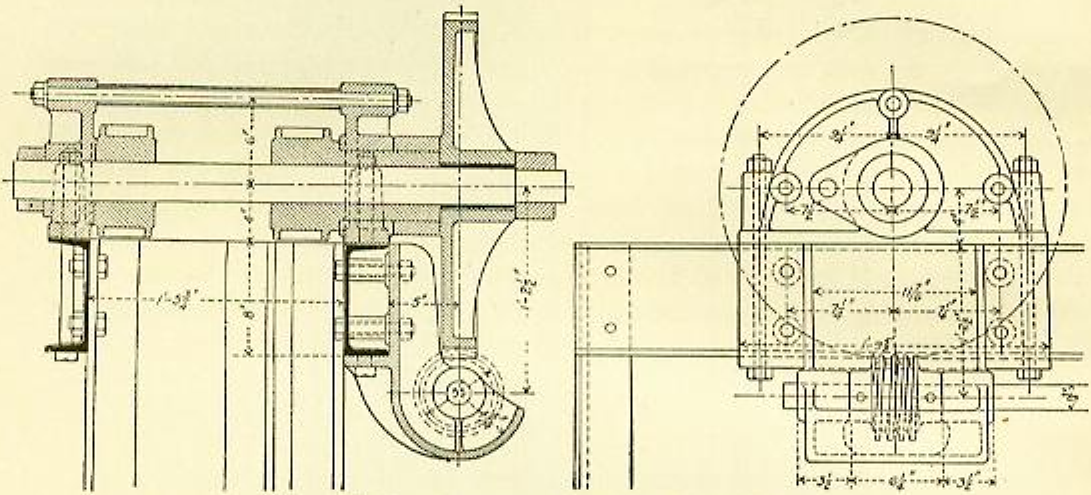


Fig. 10.—Details of Raising Device for Double Rack.

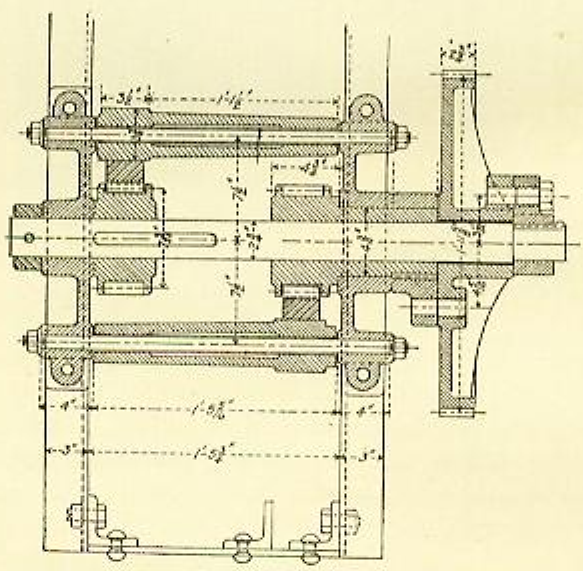


Fig. 11.—Details for Raising Device for Double Rack.

operating platform should be at least 2 feet 8 inches wide, so placed that the attendants can operate to the best advantage. It is, however, allowable, and at times even desirable, to install horizontal or even vertical racks, but in such cases a proper cleaning device is a necessity. An example of a vertical rack installed in a forebay in front of a penstock is given in Figs. 9 to 13. It consists of a double rack with the necessary lifting device, the arrangement being such that one of the racks may be raised and cleaned while the other remains in place. A working platform along the forebay wall affords an excellent opportunity for cleaning and washing the racks. Fig. 11 shows in detail an operating device consisting of a worm gear and rack and pinion which may be thrown in on either side of the racks.

An arrangement recommended by one of the authors, whereby the rack may be cleaned either by hand or by the current of the water in the forebay, is shown in Fig. 14. As shown by the drawing, a flushing valve is operated by the same mechanism (g), which operates the head gate (b). Assuming the forebay to be full of water, the head gate closed, and the flushing valve open, the water will rush backward through the racks, cleaning them of grass, scum, etc., and carry all such material through the channel (d) to the tailrace. During this operation the turbines must

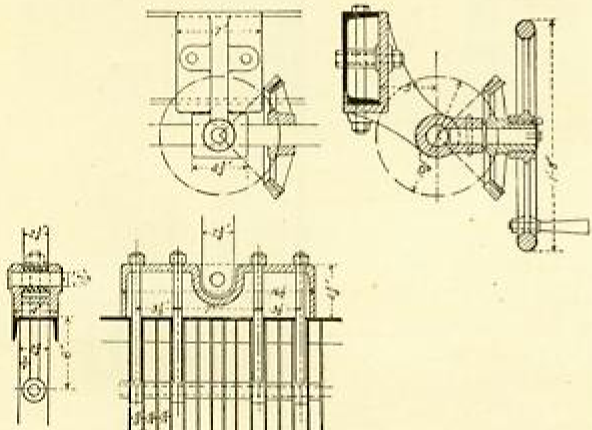


FIG. 12.—Details of Raising Device for Double Rack.

necessarily be stopped. At the same time, by means of a hand rake, the solid material adhering to the racks may be easily knocked off and dropped into the channel (d). By installing two head gates separated by a wall (c) at right angles, together with two flushing valves (a) and (a_1) with a common channel to the tailrace, it becomes possible to clean the racks when the turbines are in operation by opening (a) and (b) and closing (a_1) and (b_1). The velocity of the water in such a case depends upon the available head. The flushing valves may also act as drains for the wheel cases when the head gates (b) and (b_1) are closed.

In designing a rack it is safest to assume that it is a solid dam and that all the water behind it may be drawn off. In such a case the entire water pressure acts upon the rack. The working stress in the steel may be assumed as 17,000 pounds per

16 000

square inch. It is the usual practice to support the rack bars where the rack sections are long. Two forms of support are shown in Figs. 15 and 16.

The velocity of the water in the forebay toward the rack should be assumed as $.1\sqrt{2gh}$, which corresponds to 1 per cent of the available head. Assuming that the rack bars occupy one-quarter of the total cross-section of the rack (which is practically the case with vertical racks) the velocity through the rack will be $\frac{4}{3}\times.1\sqrt{2gh}=.13\sqrt{2gh}$, which corresponds to 1.7 per cent of the total head. To this must be added the

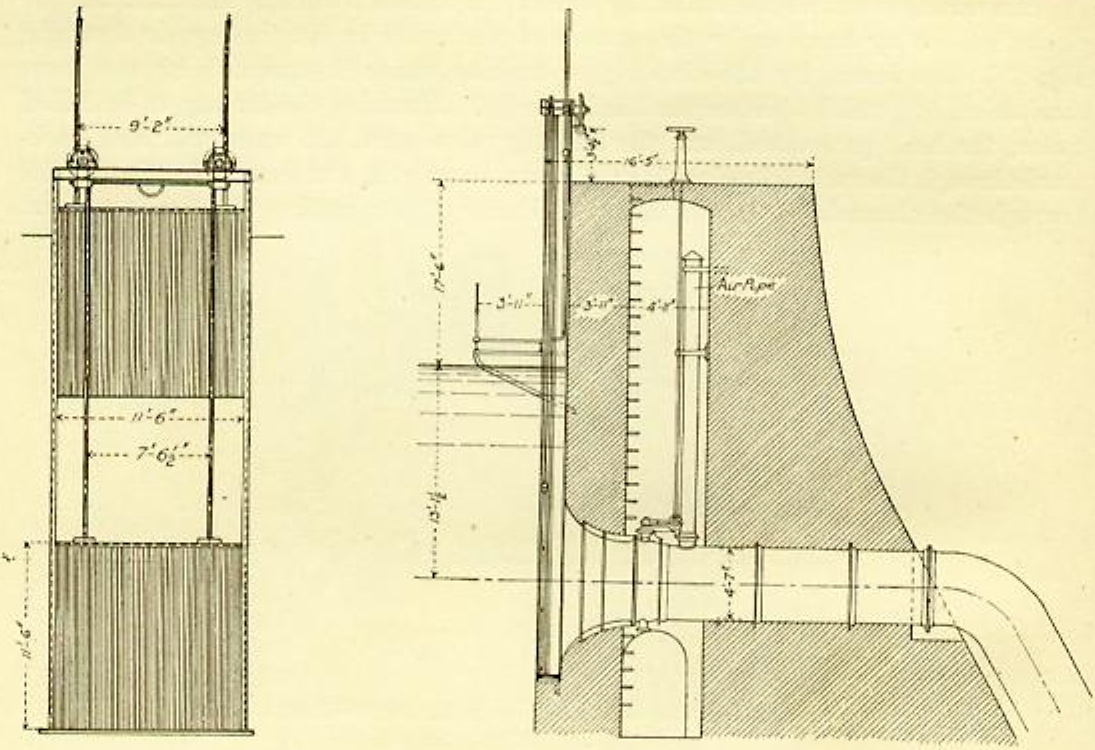


FIG. 13.—Penstock Mouthpiece Protected by Double Movable Rack.

increase in velocity due to the contraction of the water as it enters the rack between the sharp corners of the rack bars. Assuming a contraction of .7 we have a resulting velocity of $\frac{.13}{.7}\sqrt{2gh}=.19\sqrt{2gh}$, which corresponds to 3.5 per cent of the total head. If we assume that the velocity of $.1\sqrt{2gh}$ be not lost (which, for example, may be accomplished by properly guiding the water from the wheel case to the entrance to the guide vanes), we would still have a loss in head of $3.5\%-1\%=2.5\%$ of the total head, which occurs in passing the rack bars and which cannot be reclaimed.

Consequently it is advisable to use rack bars as narrow and deep as possible in order to make the spaces between the racks as large as possible. This, in turn, requires that the openings in the distributor guides of the turbine be equally large, and hence the number of buckets as small as possible. Finally, as stated above, to diminish the loss of head due to contraction in passing the rack bars the latter should be torpedo shaped and not rectangular.

Furthermore, in order to lower as much as possible the velocity of the water passing through the rack, the following features of the design should be carefully considered: (*a*) A widening of the forebay is, as a rule, only possible when the plant contains more than two units; (*b*) a deepening of the forebay below the level of the canal floor is open to the objection that the racks are thereby made more expensive and more difficult to clean and the cost of the head gates is increased; (*c*) where there are a number of inlets whose width should be restricted as much as possible from motives of economy the racks should be placed in front of and not between the inlet

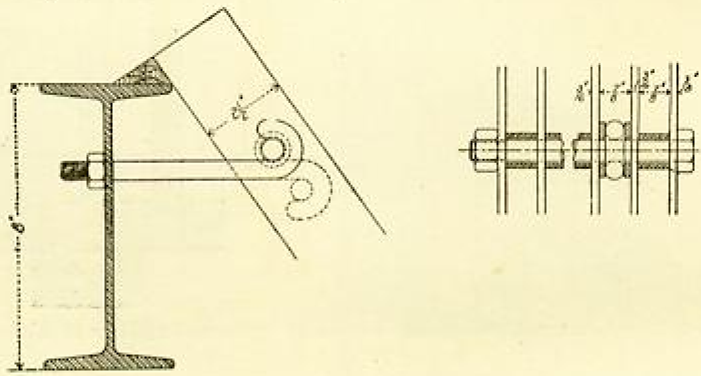


FIG. 16.—Details showing Inclined Racks and Supports.

walls; (*d*) the walls separating the inlets should have a cutwater in front so that the water may flow without eddies to the various inlets.

Filters for Removing Fine Material from the Water Supply to Governors. Under those conditions where automatic hydraulic governors operated by water are employed, it is essential for their safe and proper operation that they be supplied with clear water, as the various parts are very delicate. It is, therefore, important that the water should be filtered before using. Such filtering may take place directly by placing the filters in the penstocks or the penstock may be tapped at a point about one-third the distance from the bottom, and the water for use in the governor led into a standpipe from the top of which it flows to the governor. Alum may be added to the comparatively quiet water in the standpipe in order to clear it, but this method does not remove such coarse floating material as may have passed the racks.

By far the best method of filtering the water consists in forcing it through filters provided with fine wire gauze. In any case the filter or settling basin must be so arranged that all the material shall be removed from the water while the plant continues in operation. In the case of settling basins this may be accomplished by

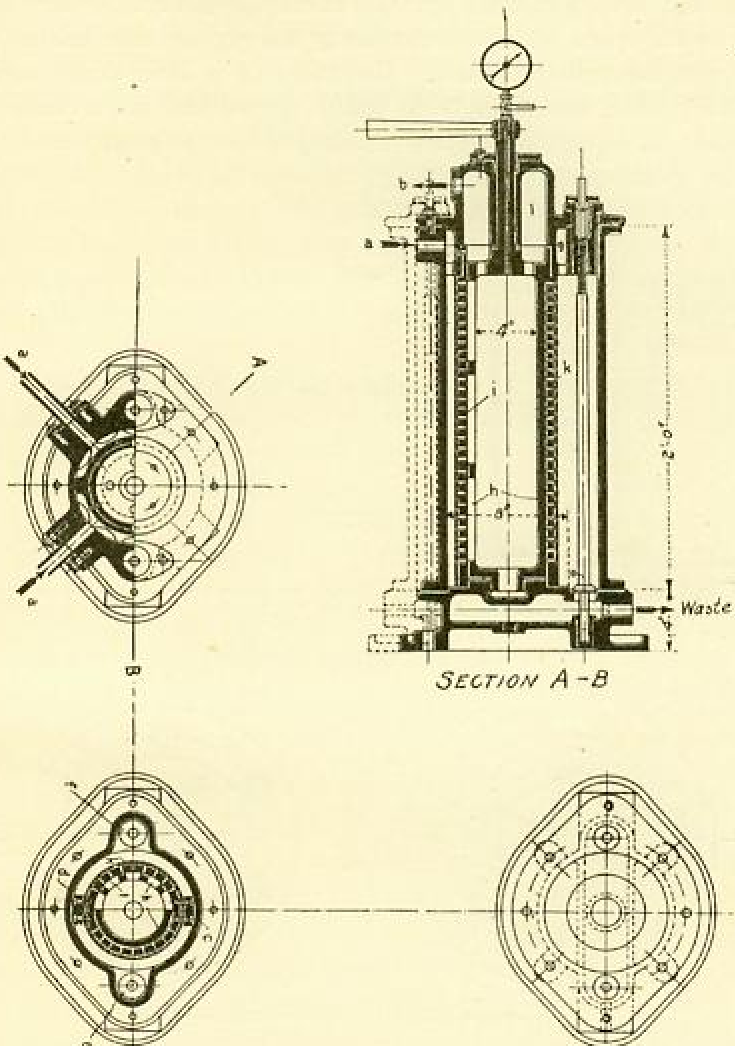
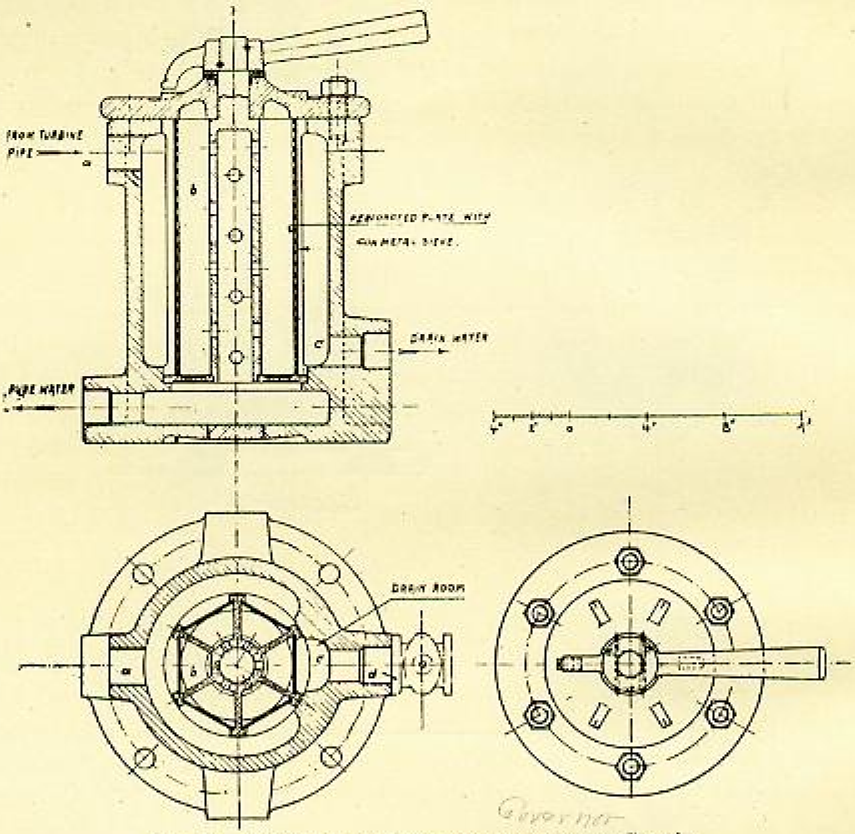


FIG. 17.—Cylindrical Filter for Governor Supply.

placing a flushing valve at the bottom of the basin, while with the use of filters the desired end may be attained in various ways.

An arrangement for cleaning filters recommended by the authors is shown in Fig. 17. The unfiltered water enters at (*a*) and passes through the sieve (*k*) which is stretched over the perforated drum (*i*). It then flows through the revolving cylinder (*h*) which is perforated for only 180°. From the space (1) the water escapes as filtered water at (*b*). The partitions (*c*) and (*d*) separate the operating portion of the filter from that used for cleaning. To clean the filter the valve (*e*) is opened, and the water in the space (*g*) then runs along the exposed one-half of the sieve and washes off the dirt through the outlet. The valve (*e*) is then closed and the cylinder (*h*) revolved 180°, when the valve (*e*) is again opened and the remaining portion of the sieve cleaned. A comparison of the reading of the gauge at the top of the filter



Governor

FIG. 18.—Compartment Filter for Government Supply.

with that of a gauge on the inlet water indicates, by the increasing difference in pressure, the necessity for cleaning.

Another filter which is much used and which may contain four, six, or more compartments, is shown in Fig. 18. One of these compartments is used for flushing only and acts through a cock simultaneously with the discharge. The water under pressure enters at (*a*) and passes through a perforated metal sheet under which is a sieve of finer mesh into the inner space (*b*) and is led from there to the governor. In order to clean the sieve the cock (*d*) is opened, when immediately a portion of the filtered water under pressure flows in a counter stream against the sieve and carries the dirt into the cleaning compartment (*c*) which is emptied by means of cock (*d*). By turning the basket each sieve may be thoroughly washed.

D. THE MEANS OF CONTROLLING THE WATER SUPPLY TO THE PENSTOCK OR OTHER FORM OF INTAKE

In the case of the turbine installation it is best to provide gates at two points, viz.: one near the mouth of the canal or penstock and one in or near the wheel case. Both should be so arranged that they may be easily operated from a point near the unit. To accomplish this in the case of the entrance gate it is necessary to install some mechanical means of transmitting the motive power or to employ an electric motor.

The head gate, as well as its opening mechanism, must be constructed in the most substantial manner, and the gears must be so proportioned as to insure the quick and complete closing of the gate under all emergencies, such as the bursting of the penstock, frozen penstock, crippled governing apparatus, short circuit in the generator, etc. (See description of automatic cup valve, page 40). The operating device must be so simple that it may be operated, if necessary, by unskilled labor, and the direction of opening the valve should be clearly indicated. Where the operator cannot see the position of the gate, an indicator showing the extent of the opening must be installed. All bearings should be kept thoroughly lubricated. The design of controlling gates may be considered under the following heads:

1. In Open Canals—
    (*a*) Head gates.
    (*b*) Revolving gates.
2. For Penstocks—
    (*a*) Gate valves.
    (*b*) Butterfly valves.
    (*c*) Cup valves.

The closing devices may be operated by the following methods:

1. Mechanically (by gearing or screws)—
    (*a*) By hand.
    (*b*) By a mechanism operated by a turbine supplied with water through some other gate.
    (*c*) By the turbine whose gate we are considering, said turbine having been brought to a slow speed by water admitted through a by-pass or by the partial opening of the gate by hand.
2. Hydraulically (by means of a hydraulic pressure cylinder)—
    (*a*) By the water pressure available from the turbine supply.

(*b*) By hydraulic pressure furnished by some means independent of the water supply to the turbine.

3. Electrically—
   (*a*) By storage batteries.
   (*b*) By current from another generator of the installation.
   (*c*) By current from a motor generator set operated from an independent plant.

Head Gates. Head gates are the simplest and cheapest means of closing off the water from canals and open penstocks and are, therefore, the most commonly used.

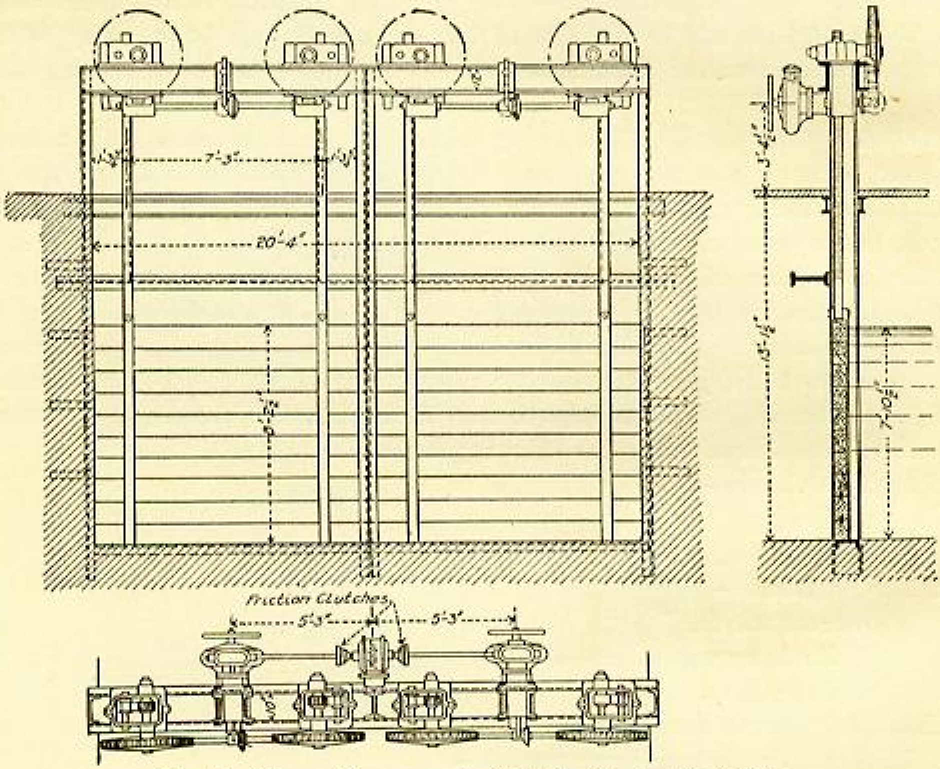


Fig. 19.—General Arrangement of Double Wooden Head Gates.

The reader's attention is directed to the several forms of head gates shown on Figs. 19, 21 and 22. A gate made of squared timbers guided by cast-iron grooves may be used where the width of the canal or inlet and the depth of the water is not too great for safe stresses in the timbers. The same design may be adopted for waterways of greater width by installing two or more gates side by side and operating them from the same shaft so that their motion will be equal. With the latter arrangement an electric motor should be used to drive the shaft, as hand operat on would be too slow. The supports for the intermediate guides must be securely anchored. They are a disadvantage, as they reduce the section of the waterway and afford a lodgment place for grass and other suspended matter. Because of the friction loss and difficulties in lubrication the use of lifting screws to replace a rack and pinion is not to be recommended.

An illustration of a double gate is shown in Fig. 19, while the details may be

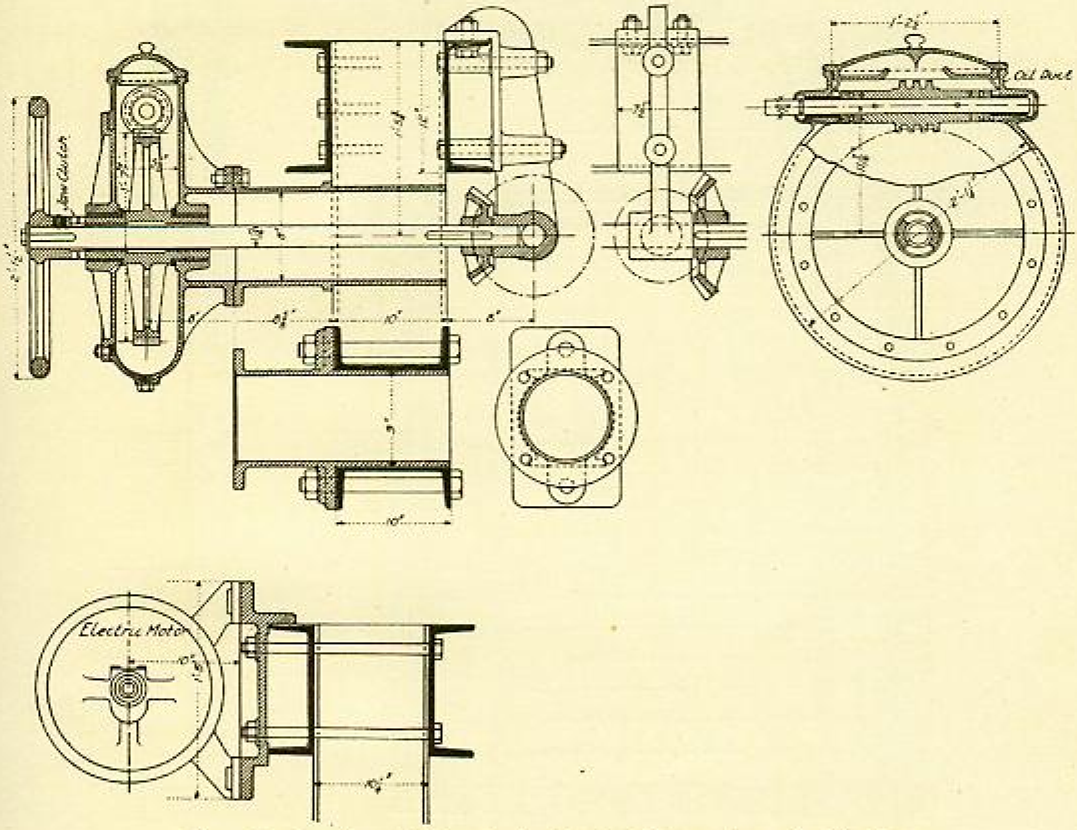


Fig. 20.—Details of Raising Device for Head Gates, shown in Fig. 19.

seen in Fig. 20. In this case the middle guides consist of two channels having their ends imbedded in the concrete and secured to another channel laid across the inlet. A truss would have effected the same result. The operating motor, placed at the center of the inlet, is connected with the transverse shaft of the worm gear by means of friction clutches, which have the advantage of avoiding an overload on the motor when starting the gate. The hand wheel is loosely keyed to the transverse shaft and may be thrown into gear by a jaw clutch, one-half of which is formed by the wheel hub. The worm on the motor shaft can also be thrown loose from the shaft. When operating by motor, the hand wheel is thrown out of gear, but when the motor is not operating the worm on its shaft does not turn with its shaft, and when the hand wheel is thrown in gear the gate may be adjusted to any position by hand.

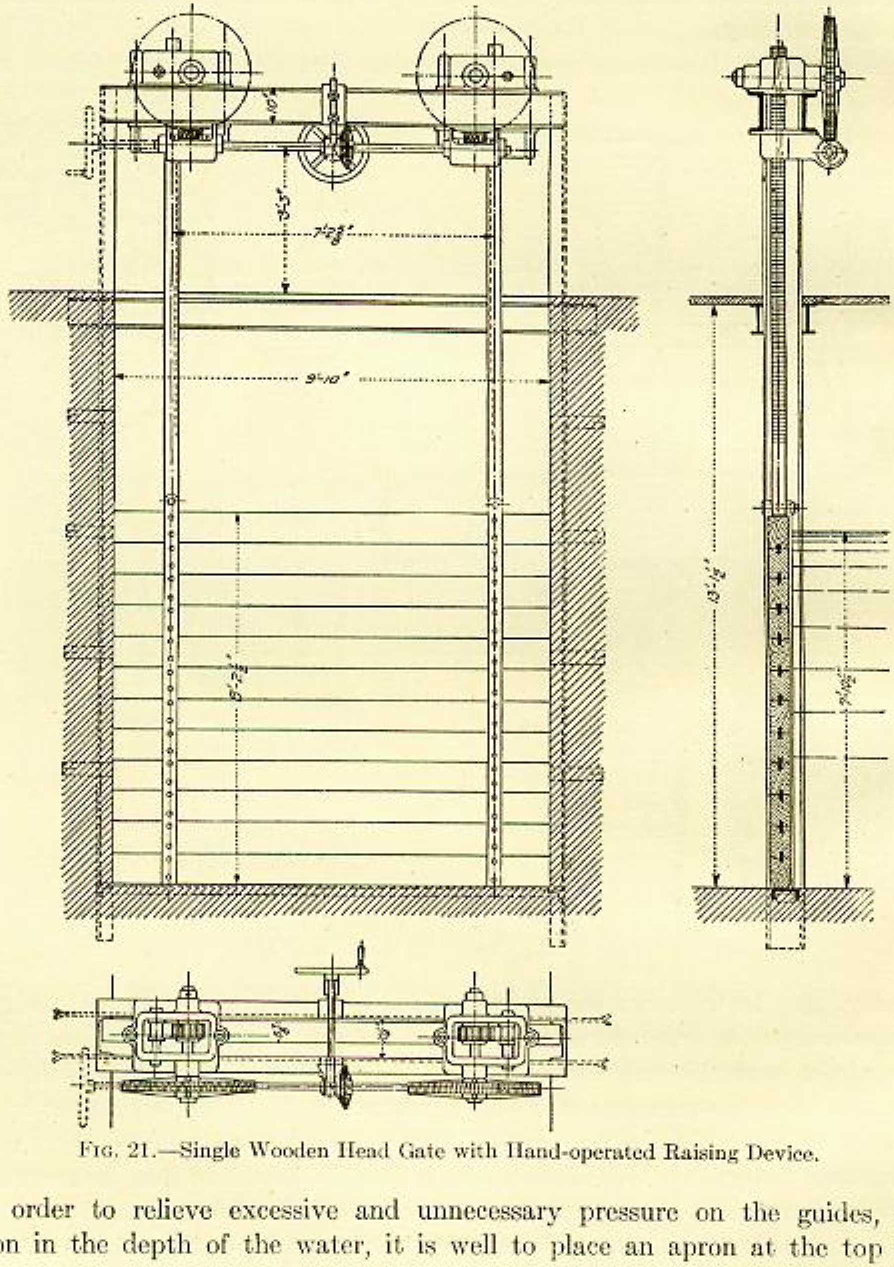


FIG. 21.—Single Wooden Head Gate with Hand-operated Raising Device.

In order to relieve excessive and unnecessary pressure on the guides, due to variation in the depth of the water, it is well to place an apron at the top of the gate ana thus keep the moving surface of uniform area. The height of the gate and gate-raising mechanism is reduced to a minimum by the use of an apron. The depth of the apron should be such that the velocity in the water channel at high water will not exceed $.1\sqrt{2gh}$ to $.12\sqrt{2gh}$.

In inlets of great depth it may be advisable, in order to reduce the pressure on any one section, to install a gate consisting of two or more horizontal sections, together with an apron. Each section may be equipped with a separate raising device or the bottom section only may be so equipped. In the latter case, when the

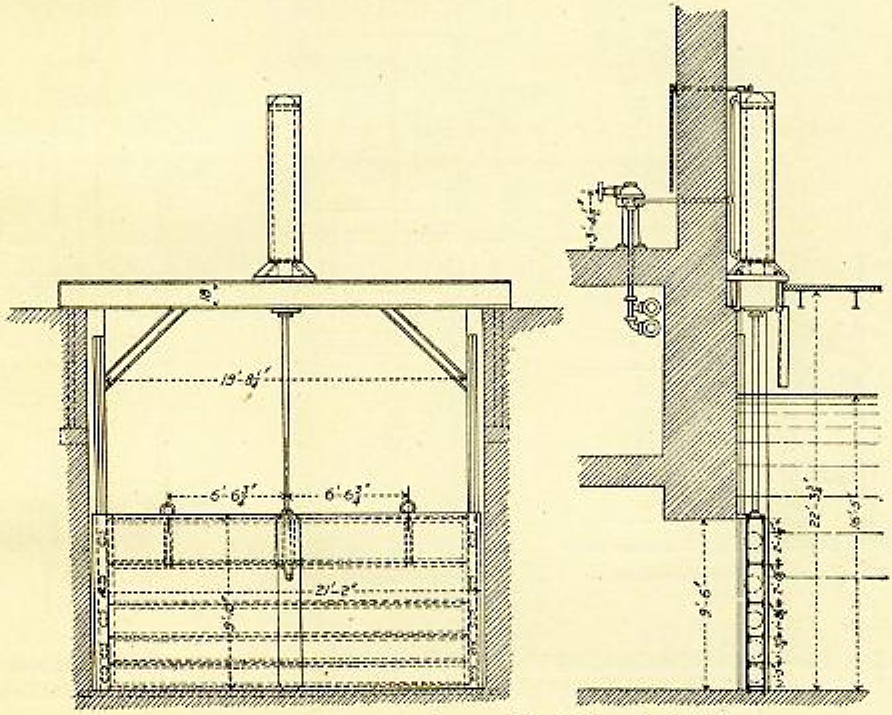


FIG. 22.—Steel Head Gate Operated by Hydraulic Cylinder.

lowest section has been raised to the lével of the next section above, it engages with and raises that, and so on, the reverse operation taking place on closing the gate. The trouble with a gate made in section lies in the difficulty of making the joints water tight between the sections. When required by the dimensions of the waterway, or when a more durable gate is desired, steel gates should be employed although they are more expensive than wooden gates. Fig. 22 shows a design of such a gate for a canal having a depth of 9 feet 10 inches, and a width of 19 feet $8\frac{1}{4}$ inches. The ribs of the gate consist of beams and channels of proper size and spacing to withstand the pressure to which they are subjected. They are coveedr on the upstream side with sheet steel and are held in place on the downstream side by steel flats.

To overcome the greater difficulty of obtaining a tight joint between a steel gate and its guides as compared with a wooden gate, it is necessary to use specially designed springs for pressing the sliding parts together, or the sliding surfaces may be carefully planed. In some of the best designs the gate is faced with a copper strip which slides against a planed casting. A wooden bearing strip secured by springs is shown in Fig. 23. Where a steel gate operates under a considerable head

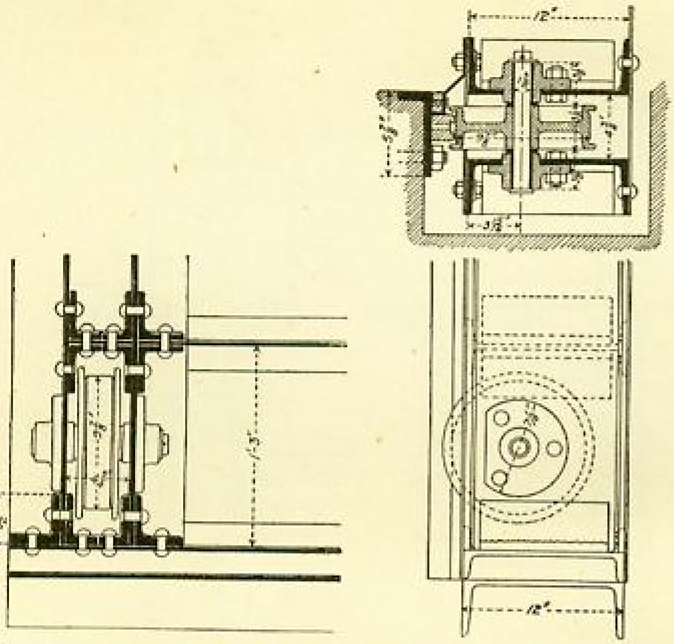


FIG. 23.—Details of Rollers and Wooden Bearing Strip for Steel Head Gate.

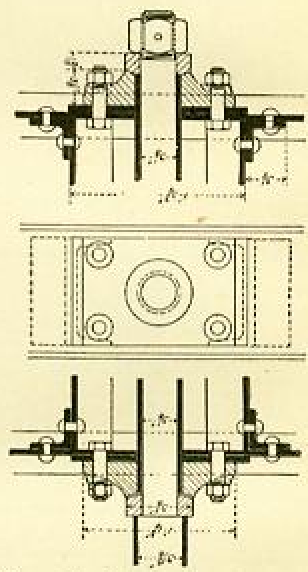


FIG. 24.—Connection between Steel Head Gate and Piston Rod from Hydraulic Cylinder.

of water, wheels or rollers are often employed, as in the design shown in Fig. 23. Rollers and wheels, however, have the disadvantage of increasing the number of moving parts liable to get out of repair, and in one of the largest installations, i.e., power house No. 1 of the Niagara Falls Power Company, where gate rollers were originally installed, they were removed and lignum vitæ strips substituted. It was found more satisfactory to provide increased motor capacity than to keep the rollers in repair.

The power required to raise a steel gate is relatively greater than for a wooden gate, and yet the gearing must be so reduced as to enable the gate to be opened by hand, even though the speed is thereby decreased and an additional amount of power is lost in such gearing. It is, therefore, desirable to employ electric or hydraulic power for raising a steel gate. If the former is employed, it should be furnished from some source other than the generator whose turbine is controlled by

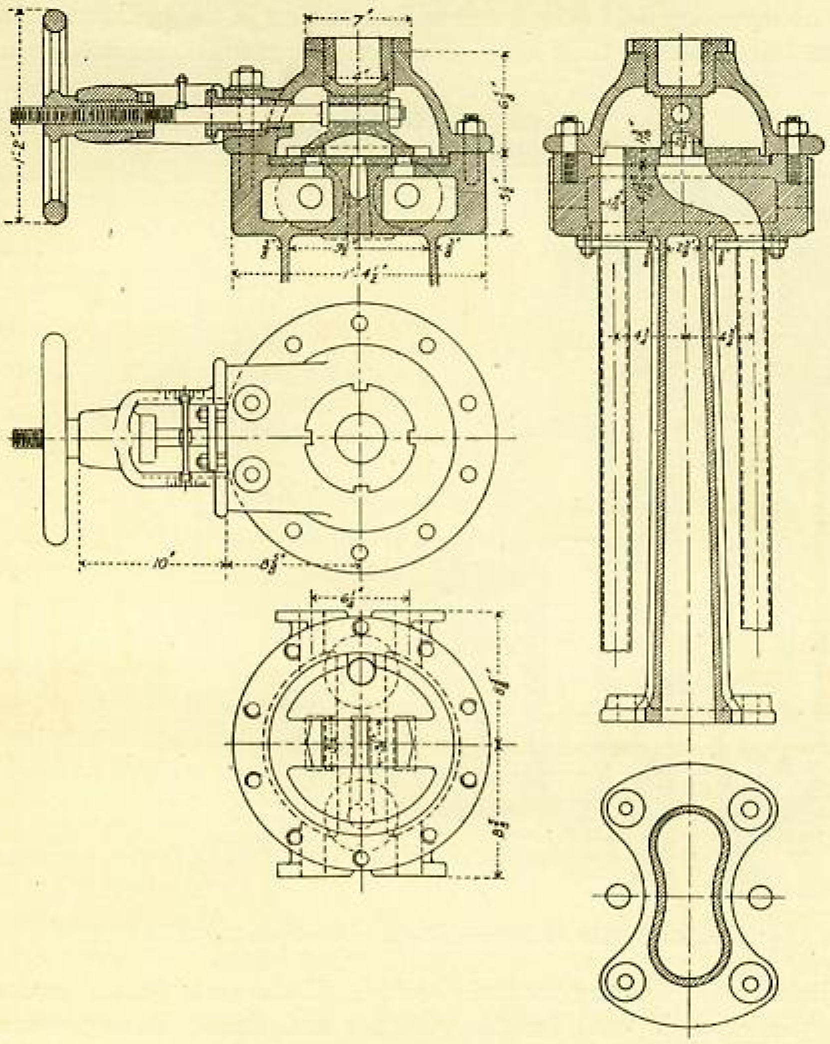


FIG. 25.—Regulating Valve for Hydraulic Cylinder on Head Gate.

the gate in question. In a large power plant—where alone steel gates are usually employed—the electric power for operating the gates may usually be obtained from the exciter plant, pumping plant, or lighting plant, as such plants are usually supplied from a separate penstock. Where gates are used to close the entire canal supplying the water to the plant, the electric power for operating them should, of course, be obtained, if possible, from storage batteries or from an independent plant.

An hydraulic lifting device has the advantage of being operated by a small hand pump at a minimum loss of power. On the other hand, a disadvantage lies in the fact that the operating fluid may freeze while standing in the cylinder, as it is usually exposed to the weather. This may, however, be guarded against by a properly

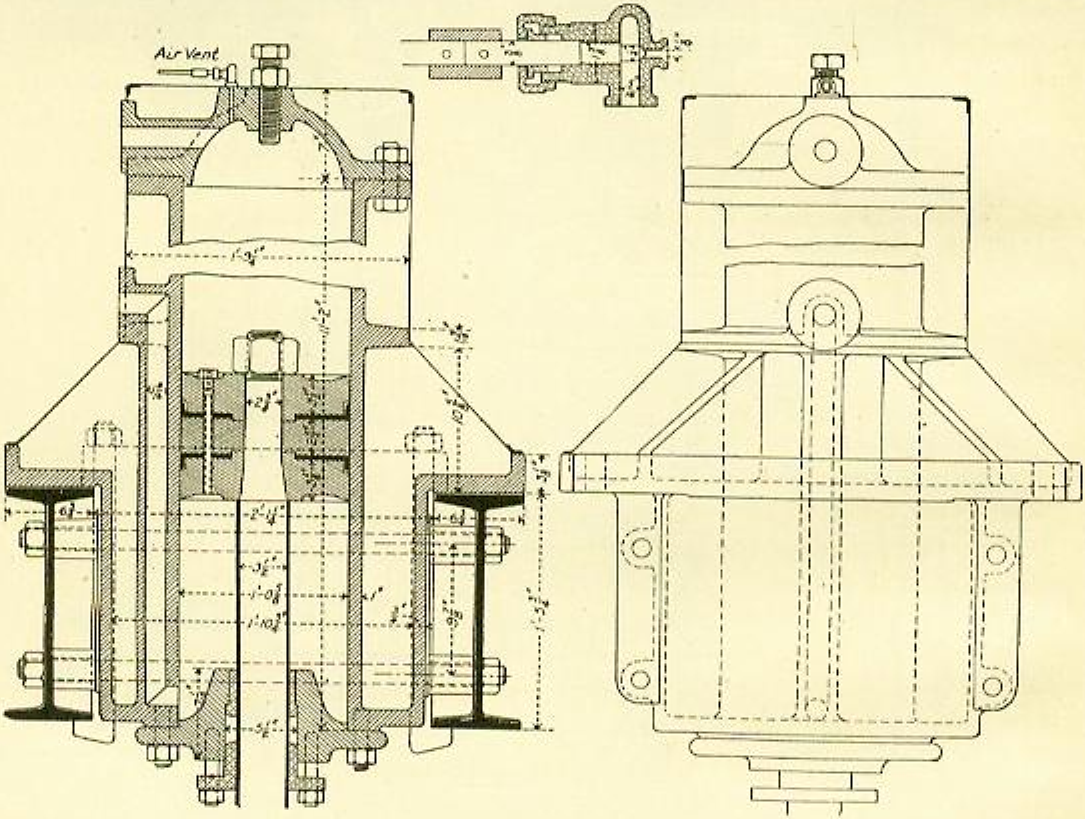


Fig. 26.—Details of Hydraulic Cylinder for Operating Head Gate.

designed jacket or by keeping the fluid heated. If the same fluid is used to operate the gate cylinder as is used for the governor and thrust bearing, we obtain the decided advantage of having the gate close and thus shut down the unit when, for any reason, the pressure in the fluid drops. Furthermore, under such a condition the cylinder of the gate-operating mechanism will for a time, at least, act as an accumulator as the gate lowers and thus temporarily protect the discs of the thrust bearing.

Fig. 26 shows further details of an hydraulic operating device. The cylinder is insulated against cold and is provided with lugs for its support. An accurately turned piston is provided with leather packing and has a variable opening for regulating the oil supply. The piston rod is not connected rigidly with the gate, but is arranged as shown in Fig. 24, to minimize the dangerous results which might arise from irregularities in construction or erection. Fig. 25 shows in detail the regulating valve through which oil under pressure is admitted to either end of the cylinder. The pet-cock (Fig. 26) for releasing the air is located at the top of the cylinder, and may be operated from the hand wheel (Fig. 22). The pressure against the gate, as designed, amounts to 165,000 pounds. The power required to lift the gate includes its dead weight, the pressure of the water on top of the gate, the sliding friction and friction on the packings, the aggregate amount being 36,300 pounds, made up as follows, viz.:

Weight of gate under water..........................	13,200 pounds.
Water pressure on top of gate........................	12,100 "
Friction under pressure 165,000 pounds	11,000 "
Total	36,300 "

An oil pressure of 290 pounds per square inch will, therefore, be required to raise the gate with a cylinder $12\frac{5}{8}$ inches diameter and each roller will be pressed against the track with a pressure of 16,500 pounds.

Revolving Gates. Revolving gates should be used only in inlets of considerable width and only in connection with large installations on account of their large cost due to their great weight. Gates of this character were used at the power plant at Rheinfelden and the electrical works at Hagneck. The bearings of the gate pivots must be designed with great care on account of the high pressures to which they will be subjected. The dimensions of the pins may be determined from the well-known formula

$$\text{Length of pin} \div \text{diameter of pin} = 1.4,$$

the allowable surface pressure being 3000 pounds per square inch, and the stress on the extreme fiber due to bending being 13,000 pounds per square inch. All of the operating parts of a revolving gate should be placed above the water and should be protected from snow and ice so as to guarantee their operation, and great care should be given to the thorough lubrication of the pins. The gate should be operated by either an electric motor or an hydraulic cylinder independently of the hand-operating device.

One advantage that attaches to a revolving gate is the ease with which it may be operated and the other is its freedom from accident due to the lodging of solid material between the gate and the inlet walls, a difficulty that sometimes occurs with sliding gates. On the other hand, it is difficult to make a swinging gate tight when closed and it reduces the section of the waterway when open.

Gate Valves. When it is desirable to insure an absolutely tight gate under a considerable water pressure, a gate valve may be placed in the penstock. It should be remembered, however, that gate valves are expensive and that they are not operated as easily or quickly as pivoted valves. Hence they are not as satisfactory for shutting off the water quickly from a penstock in the case of a break. It is, therefore, well in a high-head installation to provide a gate valve in the branch penstock to each turbine, and in addition to place a butterfly or pivoted valve in the main penstock.

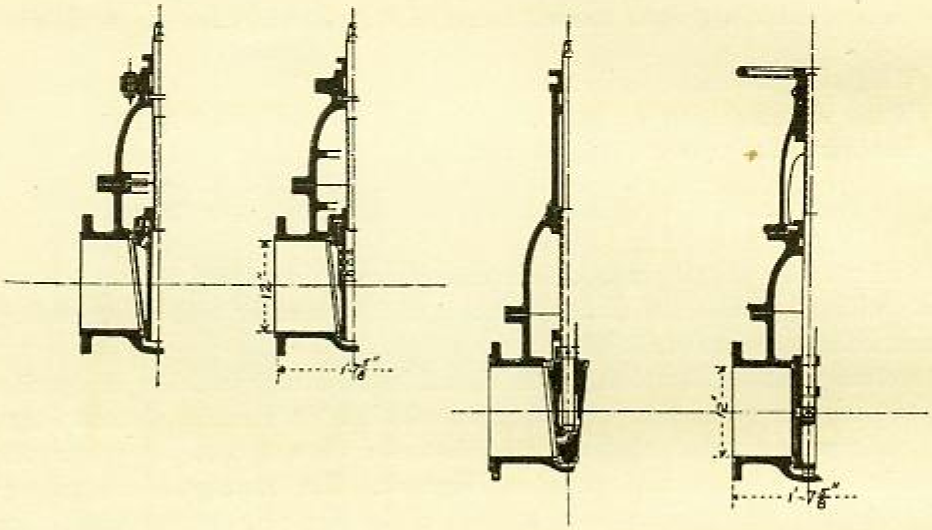


FIG. 27, 28, 29.—Gate Valves with Wedge-shaped Discs. FIG. 30.—Gate Valve with Parallel Faced Disc.

An hydraulically operated gate valve, however, is almost as quick in its action as a butterfly valve.

The tightness of a gate valve is obtained in various ways. In one form of design the disc is wedge shaped in section parallel to the axis of the penstock and has guides of the same form, the disc being forced into place by means of a screw, as shown in Figs. 27, 28, 29. In another form of valve the discs are loosely hung on the stem of the valve, and are so arranged that they are pressed against their seats by wedges which come into operation with the last few turns of the stem. The last portion of the movement of the discs is, therefore, horizontal. In a form of valve used abroad, but almost unknown in this country, the disc simply slides in its guides and depends upon the water pressure to insure tightness. Such a valve is shown in Fig. 30, and Figs. 33 and 34. In any form of gate valve it is essential that the sliding parts and their seats should be faced with bronze or babbitt, shrunk or screwed on. It is desirable that the bearing rings on the discs be renewable and that the valve be so designed that the stuffing box around the stem may be repacked when the penstock is under pressure. Great accuracy of workmanship is required in making both the discs and the guides, especially in the case of solid wedge valves. Where the disc is to move horizontally especial care must be exercised in preparing the guides so that the discs may not leave their seats or bend the screw. Some manufacturers use rollers on all horizontal gates of large size. The stem and screw should always be made of bronze having a high tensile strength and all stuffing-box bolts should be of bronze.

Whether the valve is to be operated by hand or by an electric motor, a screw is usually employed as the means of operating the valves. If a large reduction in speed

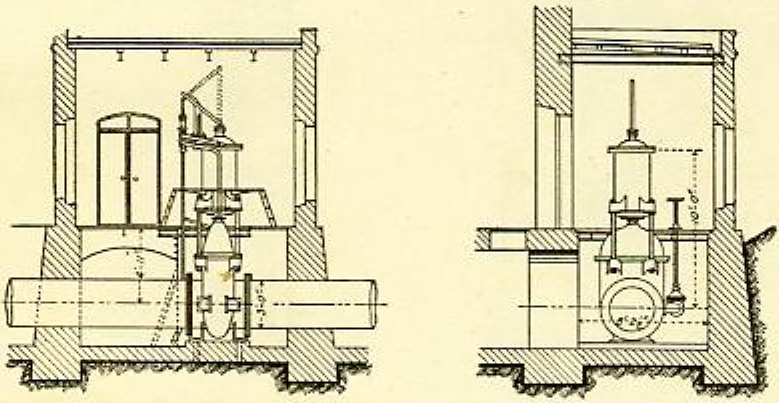


FIG. 31.—Gate Valve Operated by Hydraulic Cylinder.

is necessary, say 600 to 1, it is usual to employ spur or beveled gears, bearing in mind that the train of gears must stand at any point when stopped.

A gate valve may be operated either by a stationary inside screw as in Figs. 27, 28 and 29, or an outside rising screw as in Fig. 30. In the latter case it is customary to have the screw exposed and use a yoke. An inside screw requires less space and the screw is not so easily injured as when exposed, but an indicator should be used therewith to show the position of the disc. The outside screw has the advantage of being more easily lubricated and of indicating by its position the approximate degree of opening of the valve. The screw can also be more easily inspected but, on the other hand, when exposed in cold climates may be covered with ice, and is otherwise more liable to injury.

Gate valves of large size are sometimes operated by a hydraulic cylinder and piston as in the design shown in Figs. 31 to 34. The necessary water pressure may be obtained from the penstock or from a pump. The stem of the valve is attached to a piston moving in a cylinder, the piston being provided with rings or leather packing or ground to a fit.

Fig. 32 shows a design of a valve for applying the water pressure to either the upper or the lower side of the piston, while a slide valve for the same purpose is shown in Fig. 25. The latter is more durable but requires a larger amount of power to operate. In the former design the piston rod is connected through a lever with the valve stem. By this arrangement the speed of the piston is absolutely controlled by the hand wheel on the valve, which must be turned many times to move the piston through its

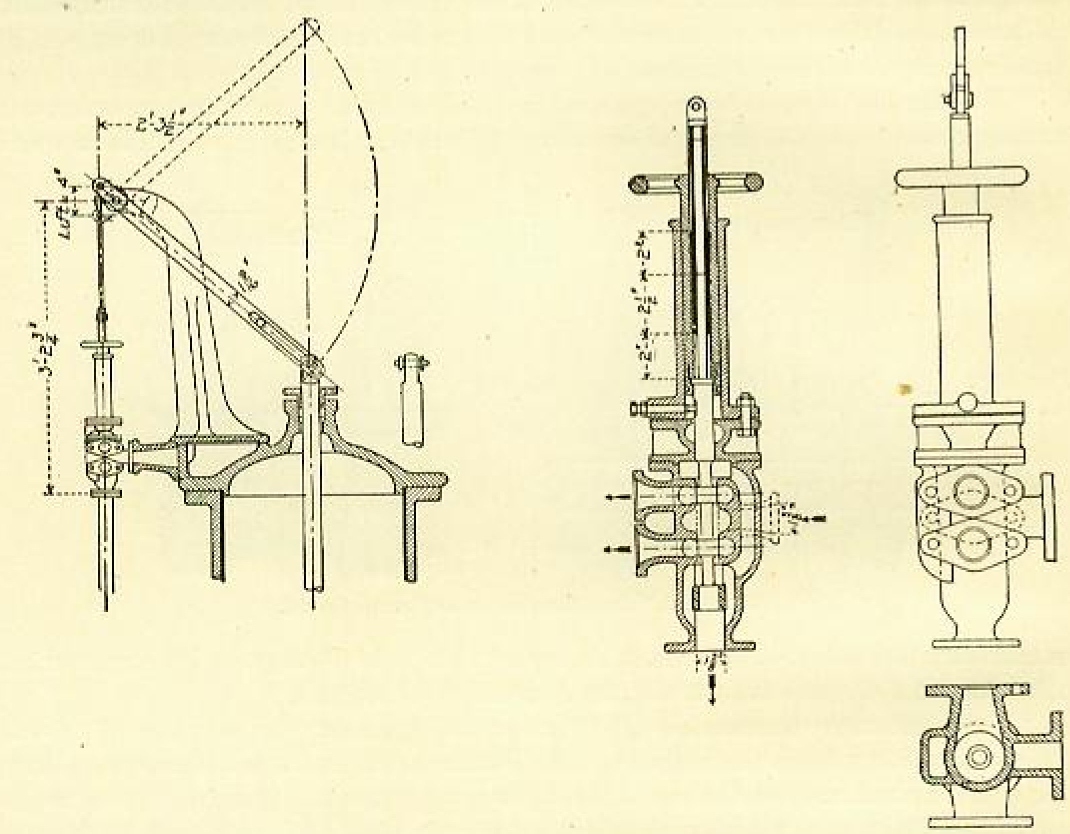


Fig. 32.—Piston Regulating Valve for Hydraulic Cylinder Operating Gate Valve.

stroke, and the latter can be locked at any desired point. This design is an excellent one but is not always necessary. Fig. 31 shows the installation of such a valve and gate valve on a penstock 3 feet in diameter under a head of 328 feet.

In order to relieve the pressure against a valve and thus open it more easily, it is customary to install a by-pass where the line is 12 inches or more in diameter and the pressure exceeds 100 pounds per square inch. A valve should, of course, be placed on the by-pass and its use presupposes a closing device at the bottom of the penstock, such as a ring valve on the turbine. A by-pass with gate valve is shown in Fig. 33, while other arrangements for equalizing the pressure before starting the main valve disc are shown on Figs. 27, 28, 29, and 30. In all of these the first few turns of the valve stem uncovers a small opening in the gate and the main disc remains closed until the penstock has filled. In Fig. 27 a conical plug is lifted from its bearing; in Fig. 28 a perforated wooden cylinder is so turned as to match with holes in the main disc; in Fig. 29 the lower end of the main stem forms a globe valve; in Fig. 30 a collar on the stem opens a small gate in the disc. None of the last-described arrangements are in ordinary use in this country, where all high-pressure valves of large size are equipped with by-passes opened by gate valves.

Gate valves frequently leak, and attention is drawn to the excellent device described in the chapter relating to the plant of the Niagara Falls Hydraulic Power & Manufacturing Company at Niagara Falls, N. Y. Back of each valve used to close the 9-foot penstock is placed a depression provided with a drain. The leakage is collected therein and the men working in the wheel case are protected from the discomfort caused by even the spray produced by water falling down the penstock. Repairs and cleaning may, therefore, be completed more rapidly and the unit put into service.

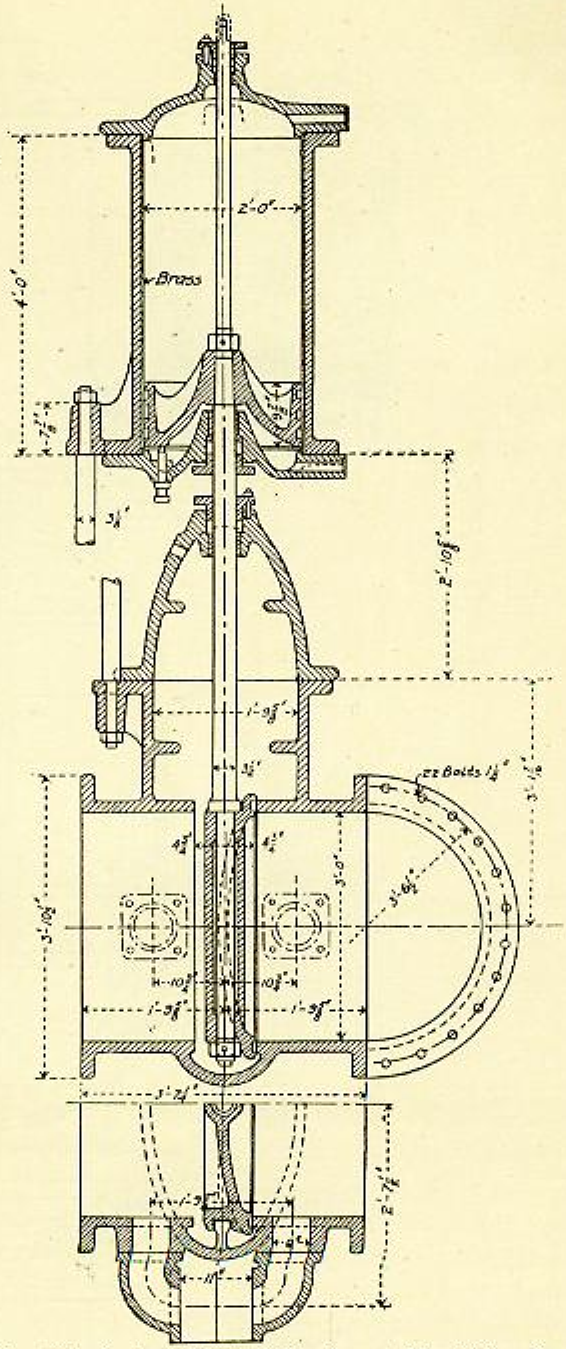


Fig. 33.—Section of Gate Valve Operated by Hydraulic Cylinder.

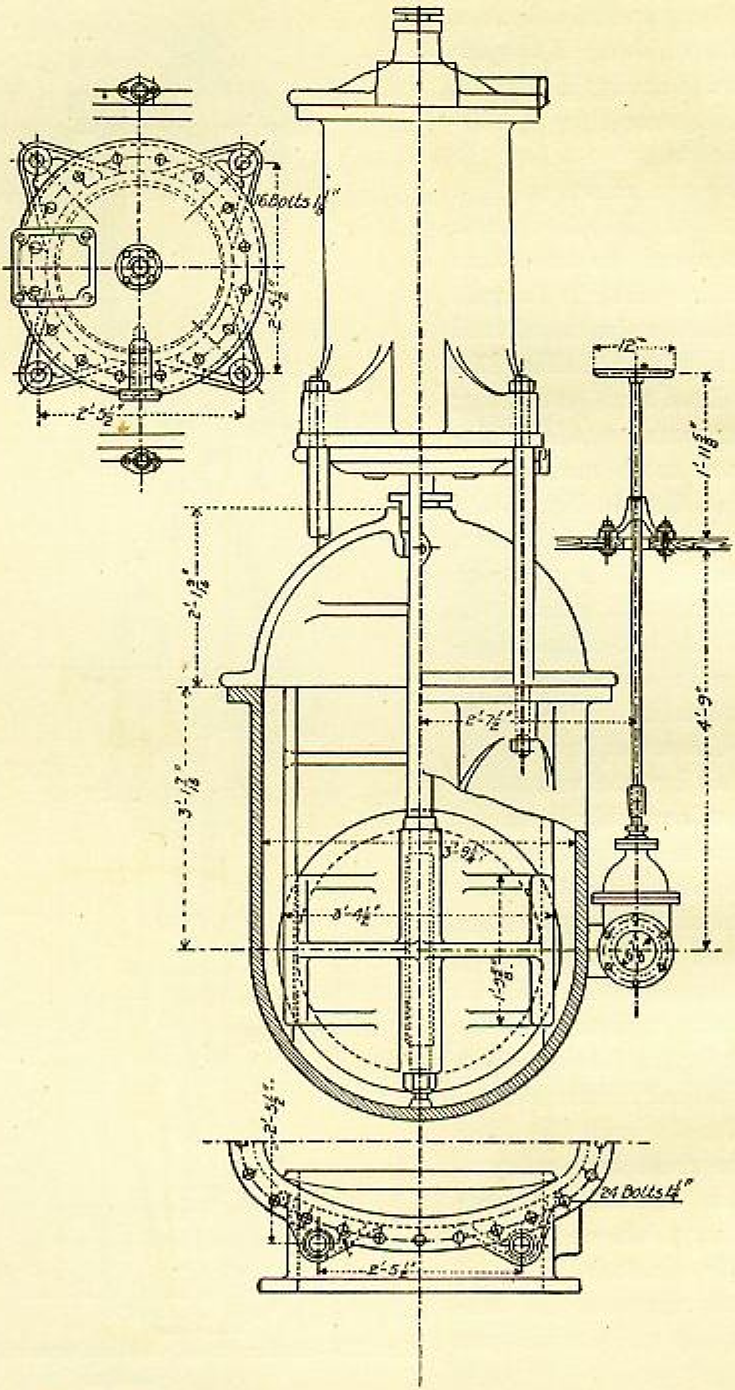


Fig. 34.—Elevation of Gate Valve Operated by Hydraulic Cylinder.

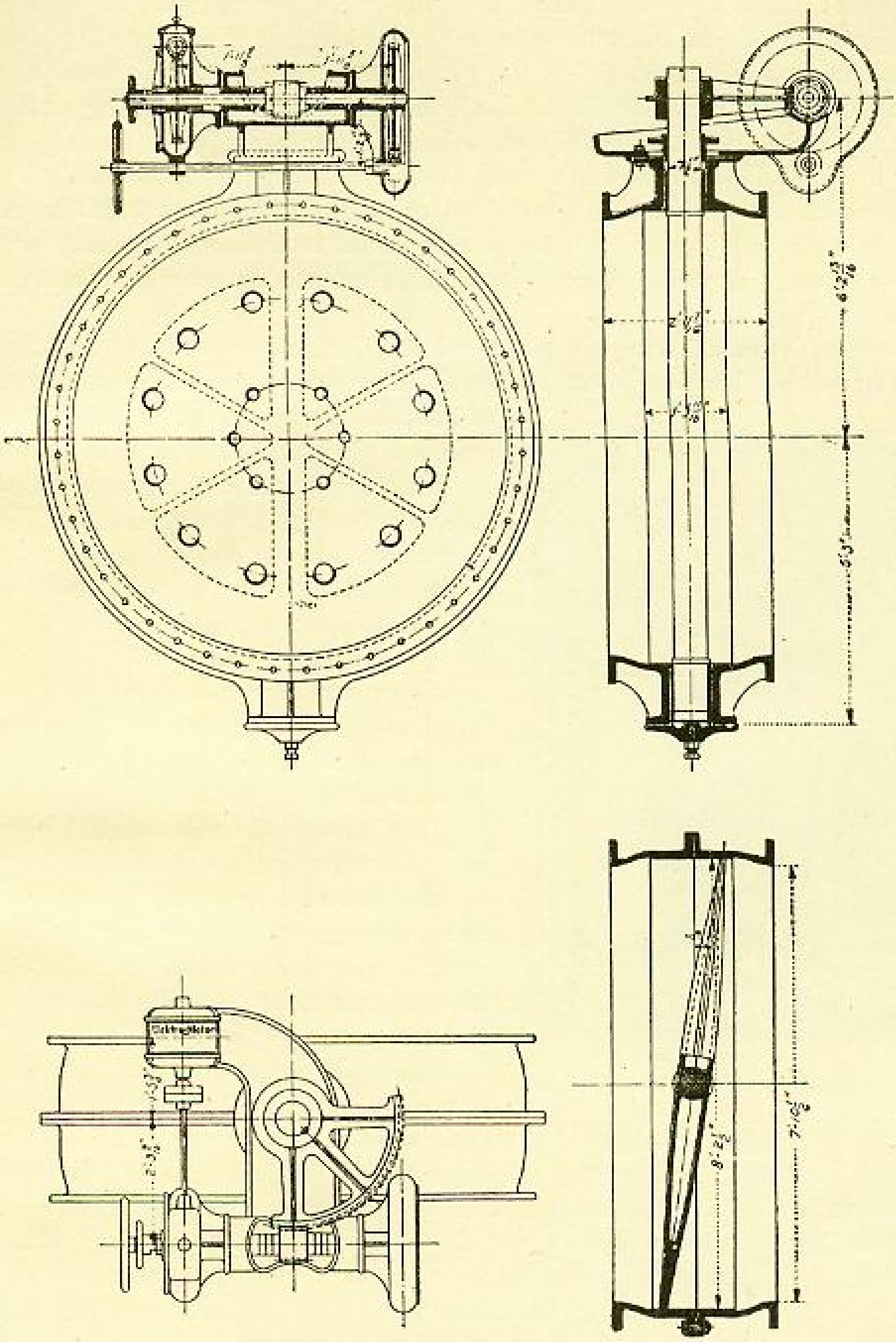


FIG. 35.—Details of Large Butterfly Valve.

In cases where by-passes cannot be installed as, for example, for overflow and ice-run gates, great care must be exercised in the design of the operating mechanism and abundant power provided. If the gates are to be operated by hand the speed must be sufficiently reduced and the coefficient of friction of the sliding surface must be liberally estimated. A butterfly valve is often preferable for use on a pipe which cannot be closed beyond the valve, even though some water may be lost.

Where the temperature is below freezing point, gate valves must be carefully packed in some heat-insulating material and should be inclosed, especially when the penstock is out of service. The cast-iron bodies are readily broken if the water in a the valve freezes.

Butterfly Valves. When the pressure head does not exceed 300 feet and it is desired to operate the valve quickly and with the minimum amount of power, butterfly valves are sometimes employed. The disadvantage of this form of valve is the tendency to leak when closed, but this may be reduced by careful design and workmanship. In order to prevent the sticking of the gate it should make when closed an angle of not less than 10° with a plane at right angles to the axis of the pipe. The circumference of the valve should be turned to a circle whose diameter is equal to the inside diameter of the penstock, and the edges when closed should be parallel to the axis of the penstock.

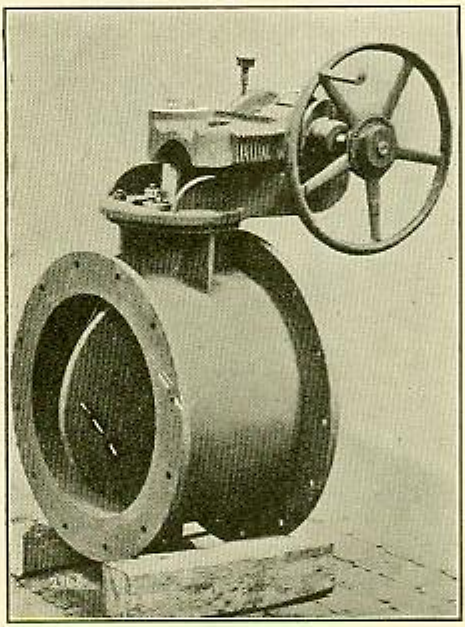


Fig. 36.—Photograph of Large Butterfly Valve.

The axis of the valve should be vertical and an adjusting screw should be placed at the bottom to prevent the moving parts from resting on the housing. This difficulty is apt to occur with a horizontal axis when the pivot and bearings are worn down and the power required for operation is thus increased. If a valve with a horizontal axis is a necessity it must be remembered that the lower half of the gate is subjected to greater stresses than the upper half, a consideration more important in the case of large valves operating under low heads. To prevent a reduction of the cross-section of the waterway at the open valve—as compared with the cross-section of the penstock—it is well to design the body of the valve in the form of a zone of a sphere, made in halves so that the gate may be inserted. Where, however, the valve is to be subjected to high pressures or where for other reasons a cylindrical body must be used, it should be made from cast steel with the pins cast integrally with the main casting. Ordinarily, an iron casting bored for a wrought iron or steel spindle will be satisfactory.

In order to reduce the power required to operate a butterfly valve under high pressure and to decrease the wear on the bearings a by-pass may be used.

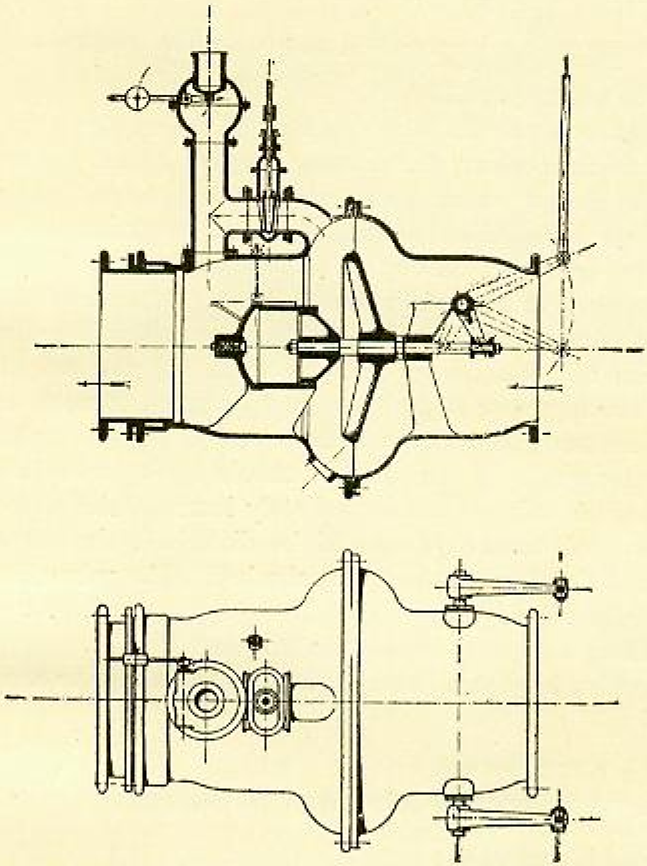


FIG. 37.—Horizontal Cup Valve for Controlling Water Entering Penstock.

The pins and bearings should be designed in the same manner as similar parts of revolving gates already described.

For operating the valve it is best to use a worm gear for the main speed reduction, as this arrangement has the advantage of locking the valve in any position. A butterfly valve used in a penstock 7 feet 10½ inches diameter is shown in Figs. 35 and 36. This valve may be operated either by hand or by an electric motor. In the former case the worm on the motor shaft is thrown out of gear by shifting the wheel with which it engages and the valve operated through the spur gears, main worm, and quadrant. The valve shown operates under a headof 60 feet 8 inches, and is subjected to a total pressure of 198,000 pounds. Assuming the coefficient of friction between the pivots and the bearings as 0.10, the total moment will be $.1 \times 198{,}000 \times .2952 = 5865$ ft.lbs. On the other hand the moment of the operating force is $44 \times .82 = 36.08$ ft.lbs. With an efficiency of .4 for the worm gear and .93 for the spur gear we have the relation $\frac{5865}{36.08 \times .40 \times .93} = 440$ (approx.). The reduction in the spur gear is 4 to 1 and in the worm gear 110 to 1.

Cup Valves. A cup valve is often installed at the upper end of a penstock conveying large quantities of water, the primary purpose of such a valve being to automatically check the flow of water in case the penstock bursts. A cup valve in connection with turbine installations should, therefore, be considered as an automatic safety device on the penstock. A cup valve is shown in detail in Figs. 7 and 8, and the general arrangement of its installation in Fig. 1. It consists of the valve A, the valve body B, with a seat for the valve on the upper elbow C and a balancing lever D. The latter has two functions: first, by means of the weight E to act as a counterweight for moving parts of the valve, and secondly, through the proper location of the weight F, to exactly hold the valve in its desired position under a given upward pressure of the water on it. If the penstock should burst, the velocity and hence the upward pressure on the valve would immediately increase, the valve would be raised, the balancing effect of the weights would be disturbed, the valve carried upward against its seat and the flow of the water checked. The valve illustrated has been designed for a normal velocity of flow of 6.28 feet per second. Attention is called to the valve G for filling the penstock and the air vent H.

A horizontal valve may sometimes be used to advantage to accomplish the same function. (See Fig. 37.)

E. THE DESIGN OF MACHINERY IN THE POWER STATION

Under this head is comprised:

(1) The prime movers, or driving machinery; i.e., the machines operated by the water, together with the devices for regulating their speed and the pressure under which they operate. The function of the prime mover is to transform the potential energy of water under head to tortional energy on a shaft.

(2) The secondary movers, or driven machinery, whose function is to receive the tortional energy from the shaft of the prime mover and to transform it into the form of energy required for local use or for transmission.

Only the first of the above classes of machines can be considered here and that only in so far as it relates to turbines and impulse wheels, together with their regulating devices. We would not be justified in devoting space to the older forms of prime movers, such as undershot, overshot and breast wheels, as they are of little consequence in the present state of the art.

We will, therefore, consider:

(1) Turbines.

(2) Speed regulators.

(3) Pressure regulators.

Turbines—General. Turbines (including water wheels) are machines which absorb the potential energy from the water and convert it into tortional energy on a shaft. This is accomplished by guiding the water through properly designed stationary guides and through buckets secured to and revolving with the shaft.

The function of the stationary guides, i.e., the parts between the wheel case and the movable buckets, is to establish the velocity and direction of the flow of the water so that it will properly enter the buckets.

The function of the revolving buckets is to abstract from the water its energy and to discharge the water without pressure and with the smallest practicable velocity. An ideal turbine receives the water without impact and discharges it without velocity.

The best method of obtaining from the several parts of the turbine their most efficient results is set forth in detail in Part II, "Turbine Design."

Turbine Parts and Their Relation. Turbines may be classified in various ways, depending on the point of view, as follows:

(*a*) In relation to the hydraulic conditions.

(*b*) As regards the relation of the guides to the buckets.

(*c*) From the relative position of the guides and buckets in respect to the shaft.

(*d*) As regards the direction of the shaft.

(*e*) In relation to the design of the wheel case or other receiver in which the turbine is placed.

(*f*) As regards the number of prime movers which operate a single shaft.

(*a*) If only a portion of the energy of the water is converted into velocity in the distributor we have what is known as a "reaction" or "pressure" turbine. If all of the potential energy of the water is converted into velocity we have what is known as an "impulse" or "limit" wheel.

(*b*) If the complete distributor is fully concentric with the movable buckets, the machine is called a "complete" turbine. If, on the other hand, only a part of the space adjacent to the buckets is occupied by guides, arranged either symmetrically or unsymmetrically, the machine is called a "partial" turbine. It may be mentioned here that "complete" turbines are usually built as pressure turbines, seldom as "limit" turbines, and still more seldom as "impulse" turbines; while "partial" turbines are almost always built as impulse wheels. (Girard & Pelton).

(*c*) If the distributor is placed outside of and concentric with the runner we have a full, inward-flow turbine, frequently known as the Francis turbine (Fig. 38), while if the concentric distributor is placed inside of the runner we have an outward flow, or Fourneyron turbine (Fig. 39). If the distributor and runner are placed one above the other about the same axis, the machine is known as an "axial" or Jonval turbine (Fig. 40). If the relative position of the distributor and runner is midway between the last two positions above described we have the historic but now seldom used

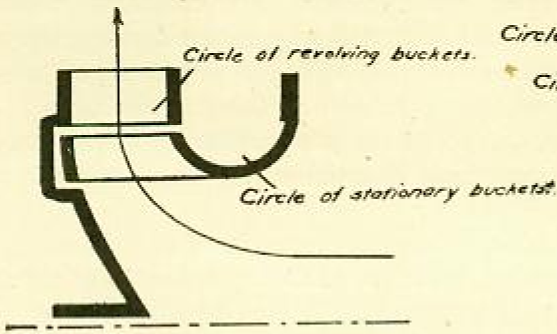


Fig. 38.—Typical Arrangement of Francis Turbine.

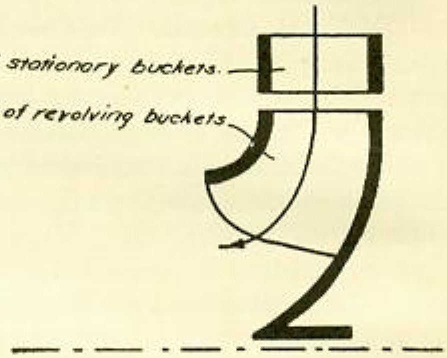


Fig. 39.—Typical Arrangement of Fourneyron Turbine.

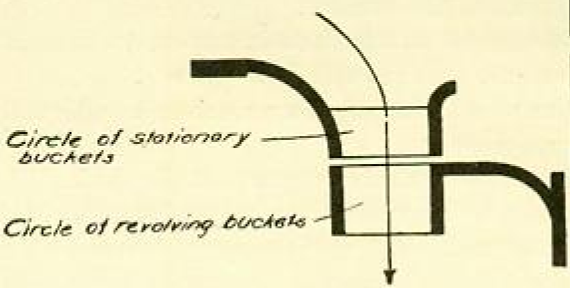


Fig. 40.—Typical Arrangement of Axial Turbine.

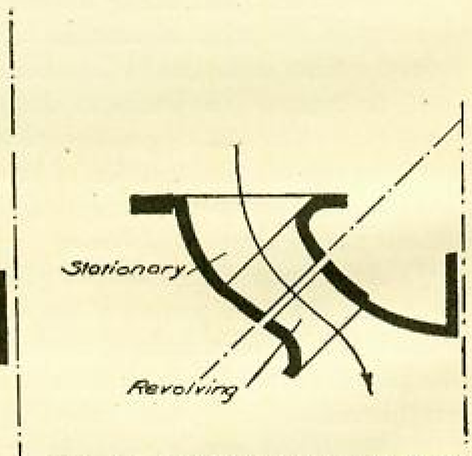


Fig. 41.—Typical Arrangement of Konus Turbine.

Konus turbine, in which the joint between the runner and distributor is a conical surface (Fig. 41).

(*d*) The turbine shaft may have any relation to the vertical, but is almost always either horizontal or vertical.

(*e*) Under this method of classification there are only two conceivable cases: first, where the turbine is set in an open flume; second, where it is set in an inclosed

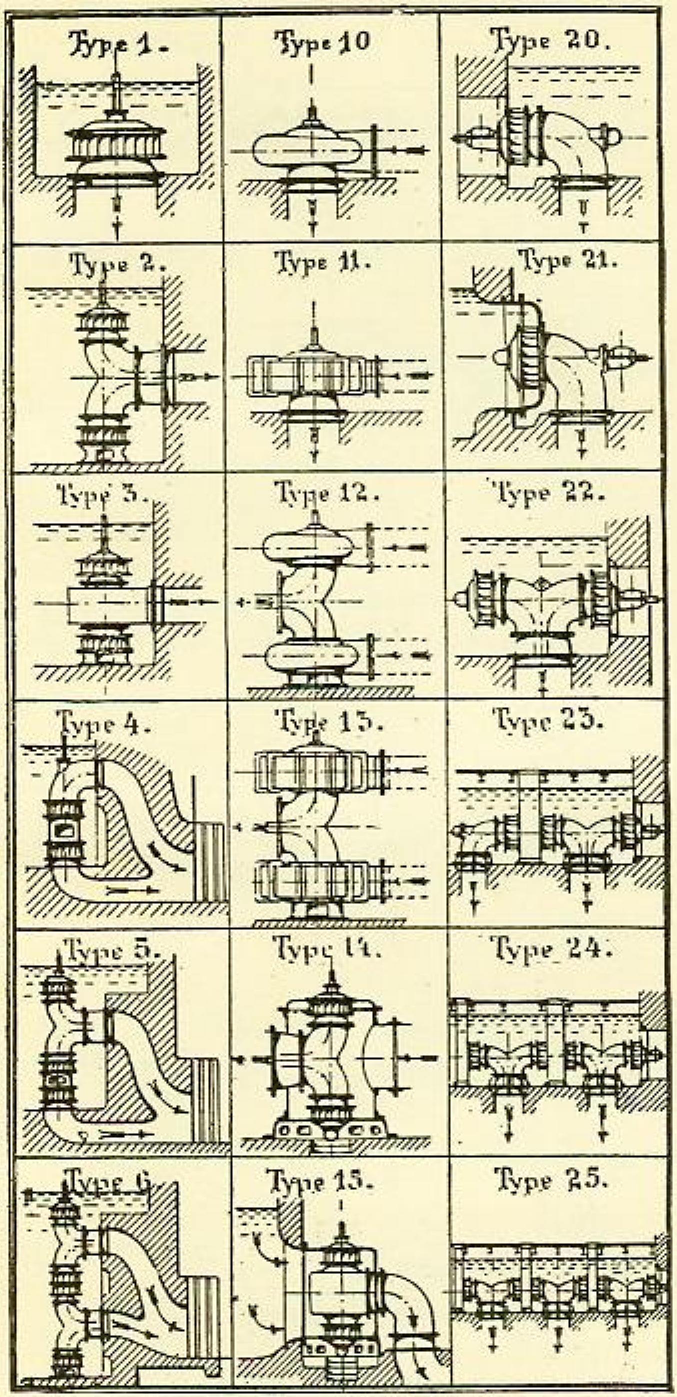


FIG. 42.—Types of Turbine Settings.

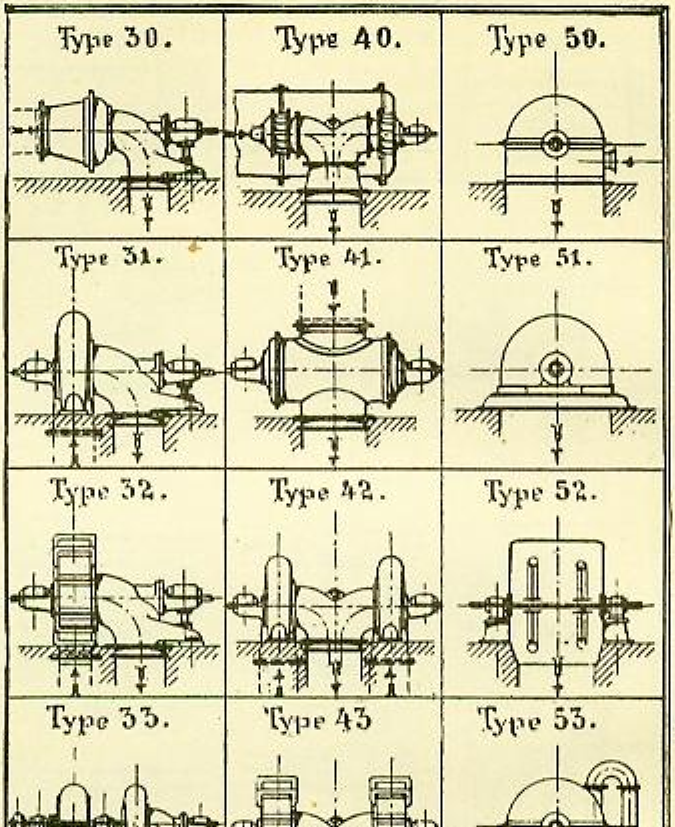


FIG. 43.—Types of Turbine Settings.

case subjected to internal pressure on all sides. In the first case the water enters the turbine through a basin in which the surface of the water stands at practically the same height as in the headrace and can be exposed to view. Such a basin may be built of masonry, wood, or steel, or a combination of these materials. The water enters the basin from an open canal and is discharged through the turbine into a masonry or steel draft tube. This is known as the "open-flume" method of development. (See Fig. 14). In the second case the wheel case is usually built of cast iron or steel, and is supplied through a penstock under pressure. If the water leaves the turbine through a closed waterway we have a "dry" turbine or turbine with restricted discharge, while if the water is discharged freely from the runner it is a "wet" turbine or turbine with free discharge.

(*f*) Almost any number of wheels may be placed on one shaft, whether it is vertical or horizontal, and is installed in an open flume or in a wheel case, but in general a larger number of wheels are placed on a horizontal shaft than on a vertical shaft. It is not unusual to install four pairs of wheels in an open flume on a horizontal shaft, while it is seldom that more than three wheels are placed on a vertical shaft.

The attention of the reader is particularly called to Figs. 42 and 43, taken from the catalogue of Amme, Giesecke & Konegan, which shows 36 different arrangements of turbine setting. A study of this illustration may often suggest a form of setting which is particularly applicable to the development which is being considered.

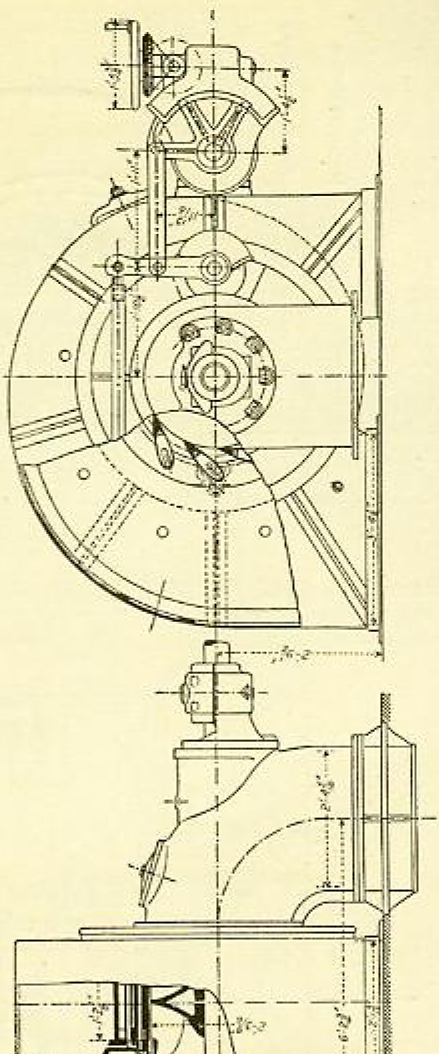


FIG. 44.—375 H.P. Horizontal Turbine with Spiral Wheel Case.

Head = 47.9 ft. Output = 375 H.P.
Quantity of water = 91.8 sec.-ft. Speed = 300 r.p.m.

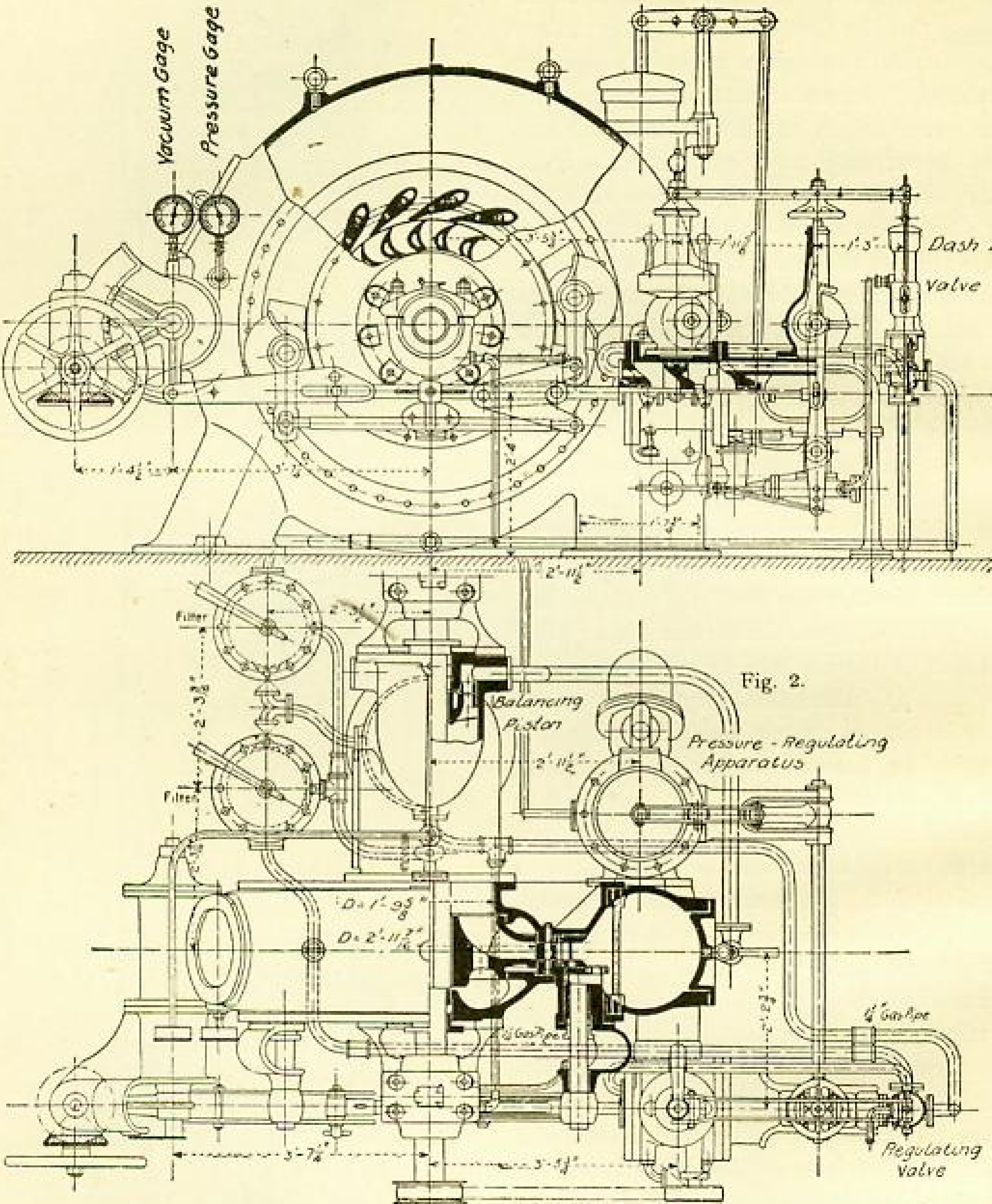


FIG. 45.—1050 H.P. Turbine with Automatic Governor.

Head = 321.5 ft.	Output = 1050 H.P.
Quantity of water = 37.78 sec.-ft.	Speed = 500 r.p.m.

Details of the Design—Form of Open Flumes and Wheel Cases. In general the cross-section of an open flume or a forebay should be reduced, as water is drawn from it for each turbine. The cost will thereby be decreased, but the velocity of approach must be kept low and the reduction in section so made as to avoid loss of head in eddies. A spiral, closed wheel case is designed on the same principle, as illustrated in Figs. 44 and 45. In the former illustration it will be observed that the penstock is tapered

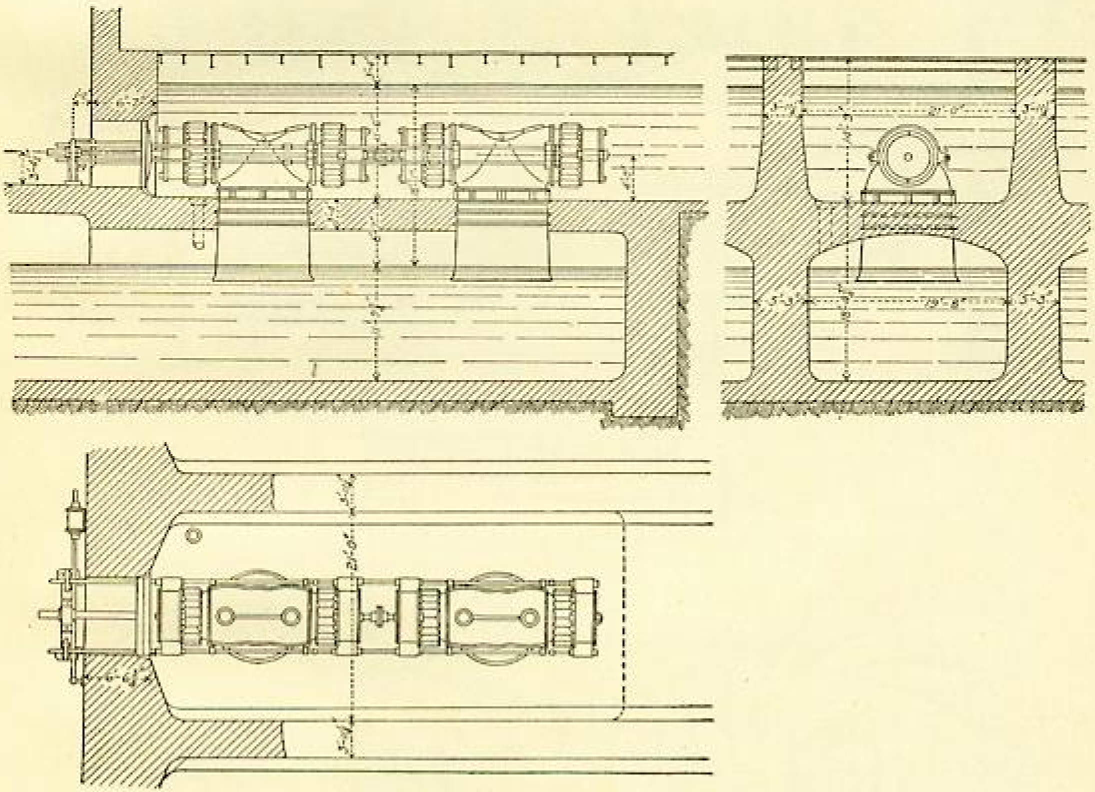


FIG. 46.—Quadruple Turbine with Cylinder Gates.

beyond the branch to the first turbine. Even in open flumes the masonry may be built in a spiral form, especially when vertical-shaft turbines are used.

The velocity of the flow in a spiral wheel case or an open flume of spiral form should be between $.2\sqrt{2gh}$ and $.24\sqrt{2gh}$ (when h is the effective head under which the turbine operates), the larger velocity being selected for turbines with low heads and large volumes. In the case of heads of 300 feet or more the velocity should be reduced to about $.15\sqrt{2gh}$, so that the material in the wheel case may not be worn away too rapidly.

When the velocity does not exceed $.08\sqrt{2gh}$ to $.12\sqrt{2gh}$ a rectangular or circular enclosure may be used for the turbine. A rectanglar open flume is shown in Fig. 46, and a circular masonry wheel case in Fig. 48, and the reader is referred to the technical press for many illustrations of this form of construction.

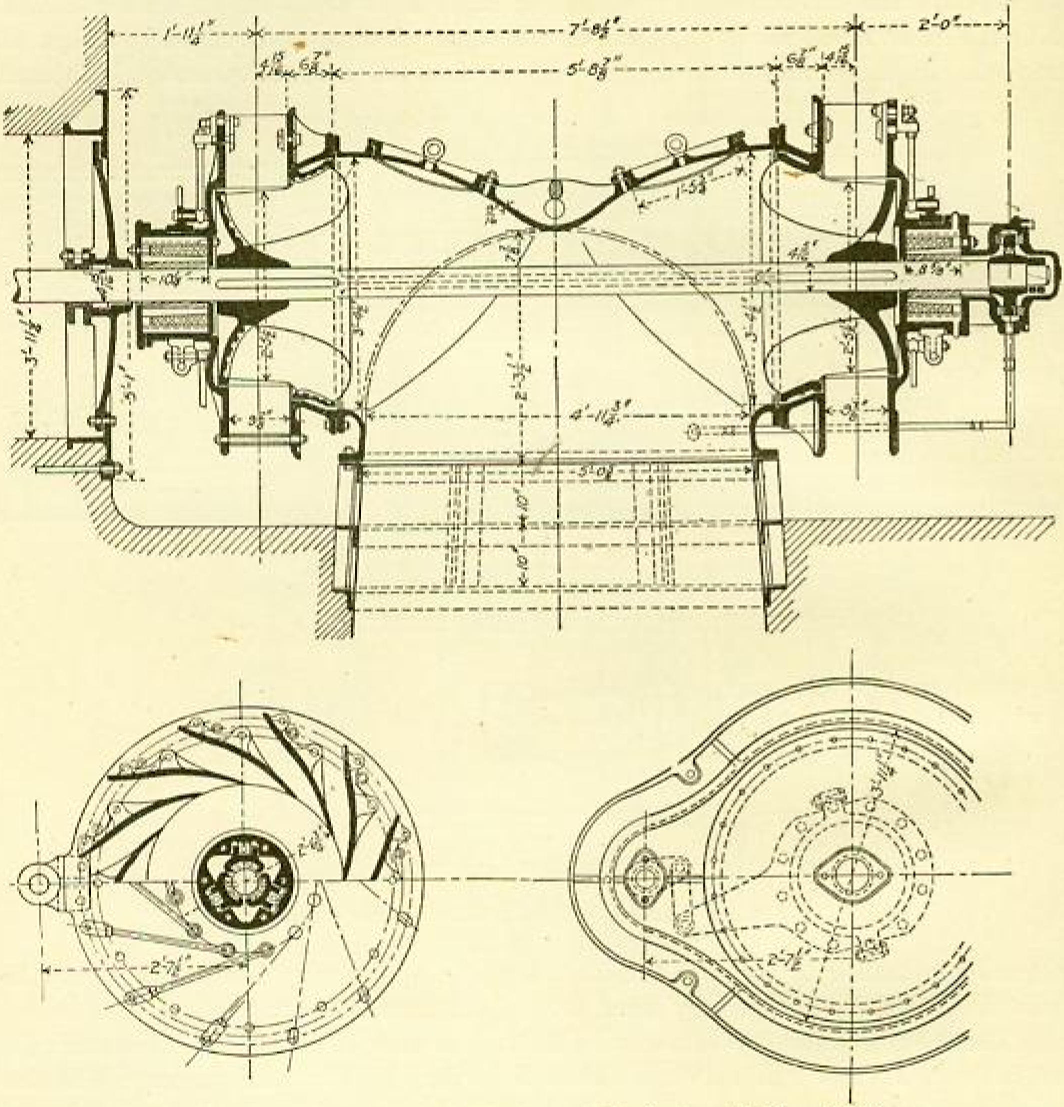


FIG. 47.—Double Francis Turbine having Distributor with Movable Vanes.

If the turbine is placed in an eccentric position in a circular wheel case so that the effect of a spiral is approximated, Figs. 49, 50, and 51, the entrance velocity of the water may be $.15\sqrt{2gh}$. The same velocity is allowable if the circular wheel case has

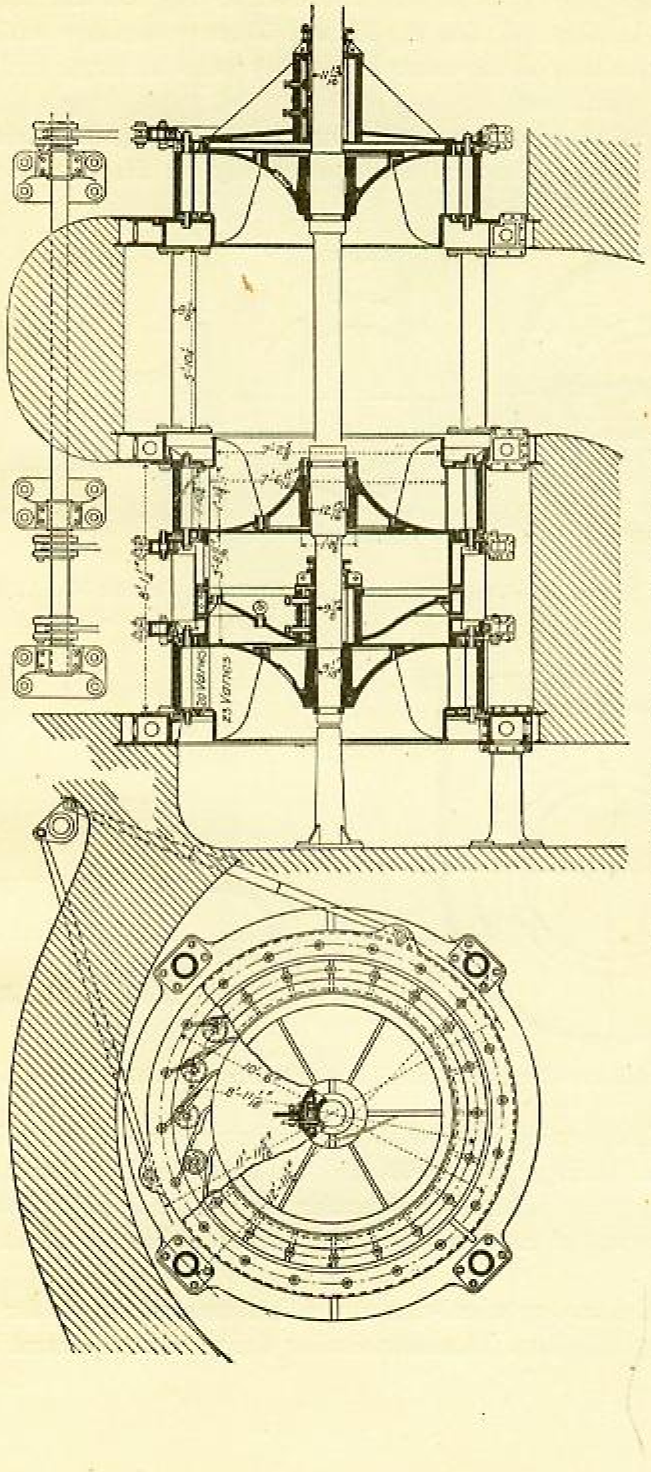
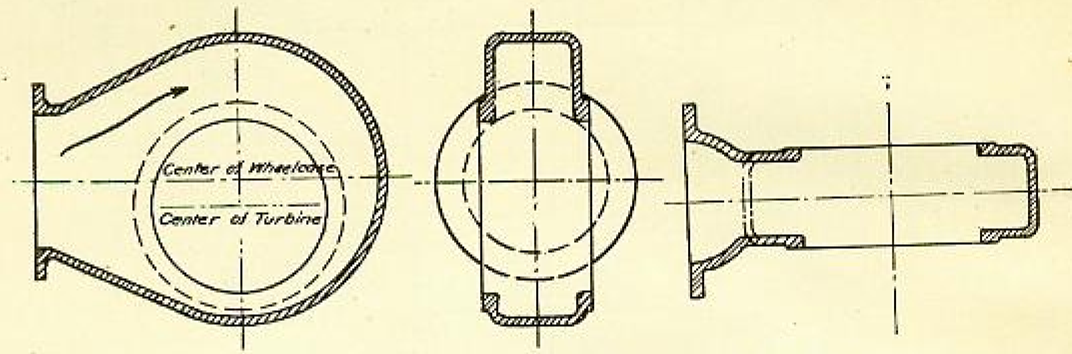


Fig. 48.—Triple Francis Turbine with Vertical Shaft.

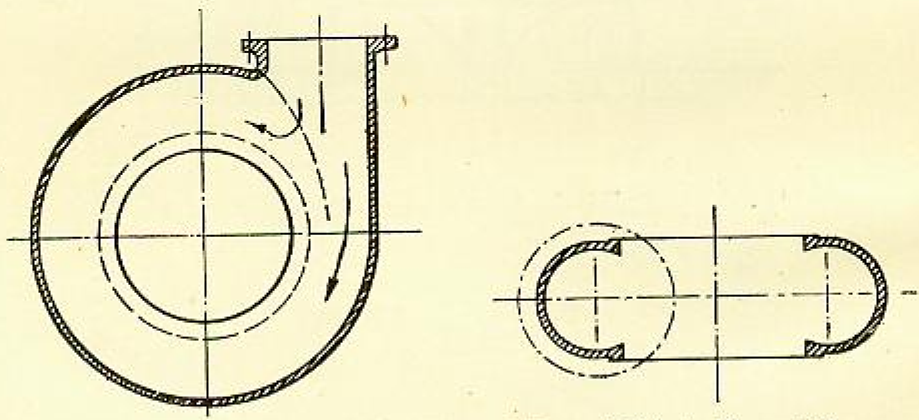
Mean head—14.43 ft.
Quantity of water—794.5 sec.-ft.
Output—1000H.P.
Speed—67 r.p.m.

the inlet penstock cast to it on a tangent as shown in Figs. 52 and 53. Wheel cases of the form shown in Figs. 49, 50, 51, 52 and 53 may be more easily constructed than a true spiral case but, on the other hand, they must be larger and hence require more metal than a spiral case.

Where any form of wheel case operates under considerable pressure the case should be strengthened by ribs and fillets, especially on the entrance side, so that



FIGS. 49, 50, and 51.—Turbine Eccentric in Relation to Circular Wheel Case.



FIGS. 52 and 53.—Circular Wheel Case with Penstock Entering Tangentially.

it may have abundant strength to resist the water pressure. Drum-shaped distance pieces are illustrated in Figs. 66 and 67, and the same result is partially obtained in the Fink regulator by the bolts of the annular moving buckets, but the bolts are so small that they are not of much use for the purpose in hand without increasing the number.

A well-designed dome, or wheel-case cover, is shown in Fig. 54 for use with axial or outward-discharge turbines. Another similar design with a central hub and cover

RADIAL OUTWARD FLOW TURBINE.

HEAD	164 FT.
VOLUME OF WATER	56,5 CUFT.
CAPACITY	800 HP.
REVOLUTIONS	360 PR. MIN.

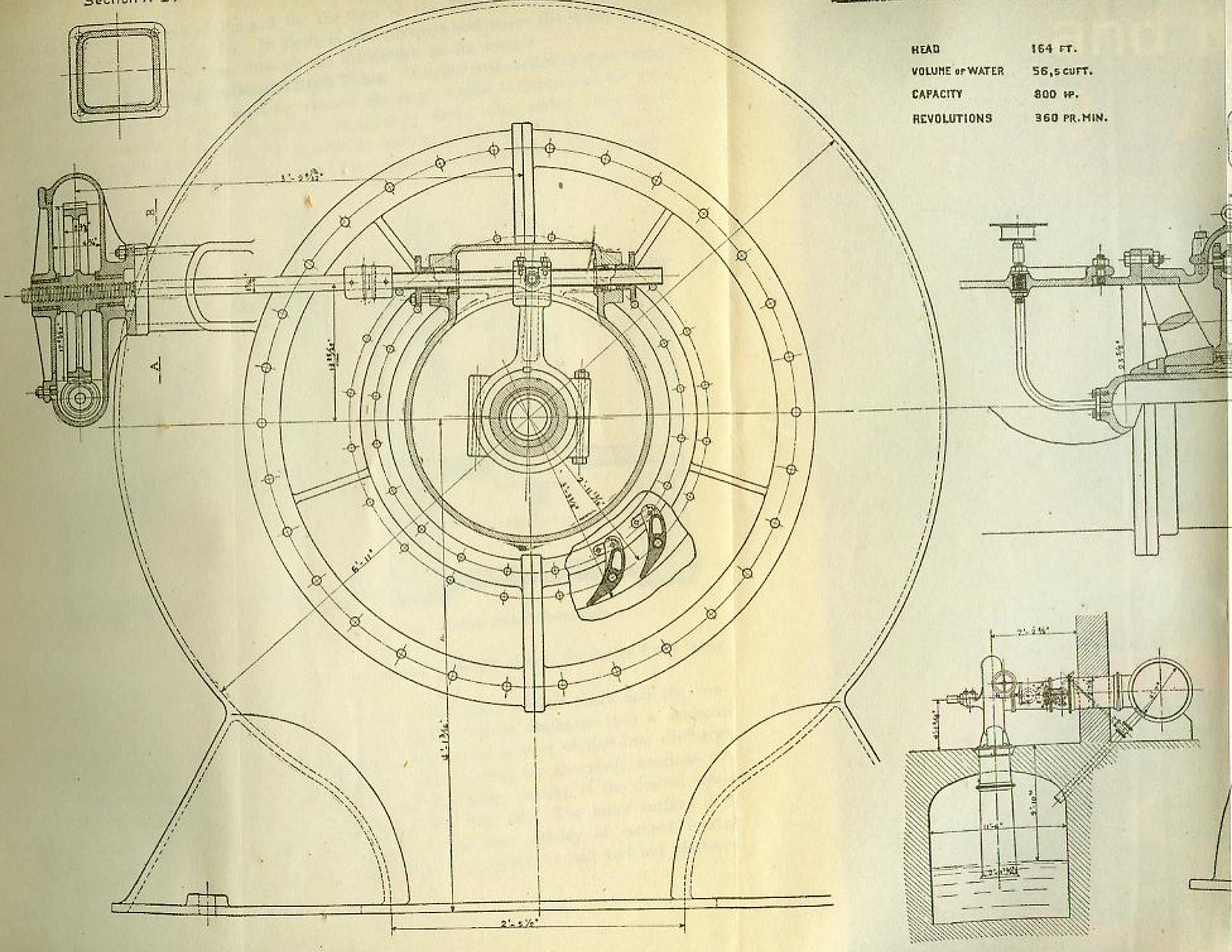
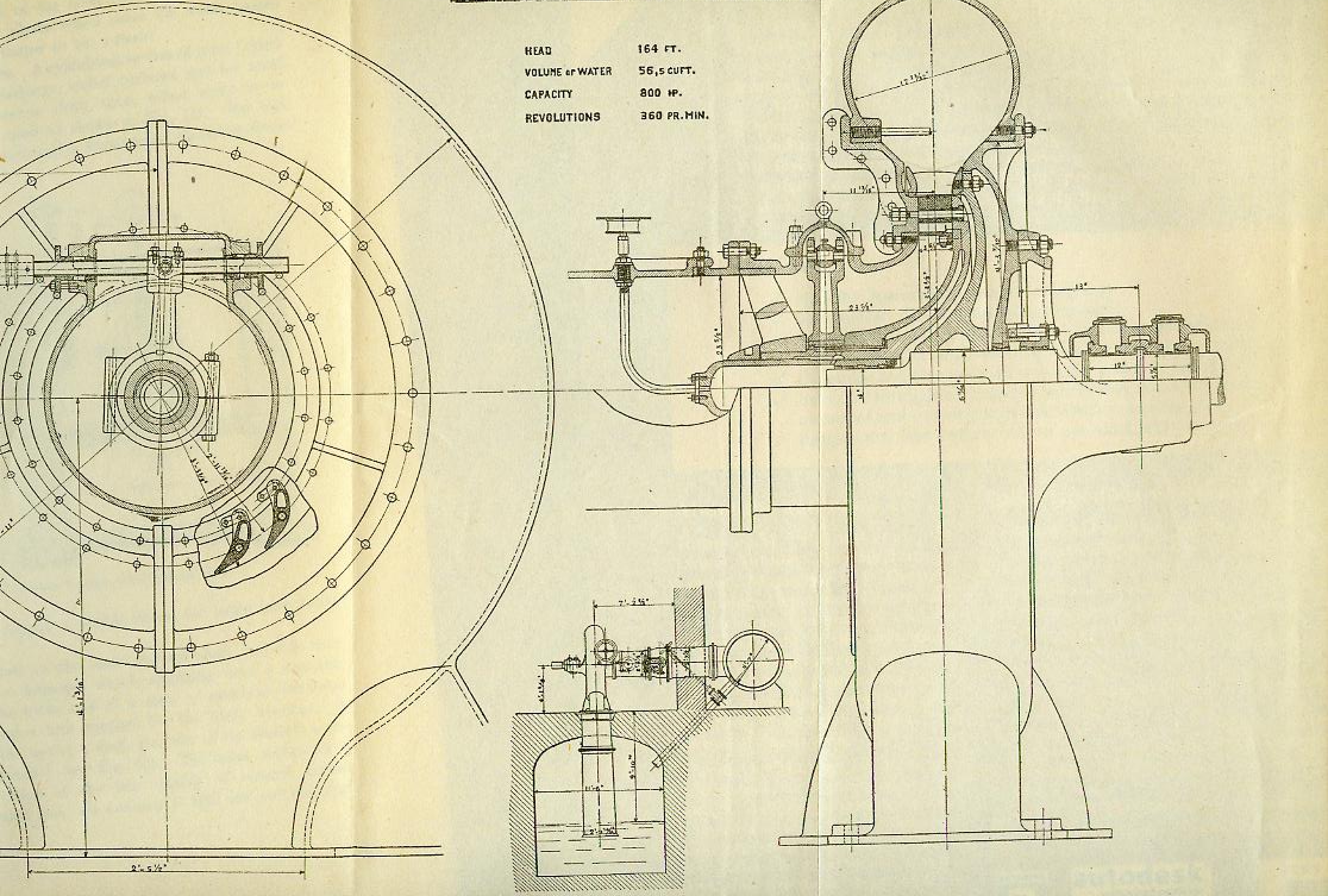


Fig. 54.

RADIAL OUTWARD FLOW TURBINE.

HEAD	164 FT.
VOLUME OF WATER	56,5 CUFT.
CAPACITY	800 HP.
REVOLUTIONS	360 PR. MIN.



is that made by Faesch & Piccard for the first of the 5500 H.P. turbines installed at Niagara, shown in Fig. 55. In both of the above wheel cases the water passes without eddies from the case through the distributor to the runner.

Form of Wheel Case Outlet and Draft Tubes. A cylindrical section of pipe, bolted to the wheel case, may be used for inward discharge, radial turbines and for axial turbines as shown in Figs. 56 to 59. A concrete draft tube, either alone or in connection with a metal elbow, is sometimes used as shown in Fig. 111. For two turbines on a common shaft there may be employed a semicircular discharge dome

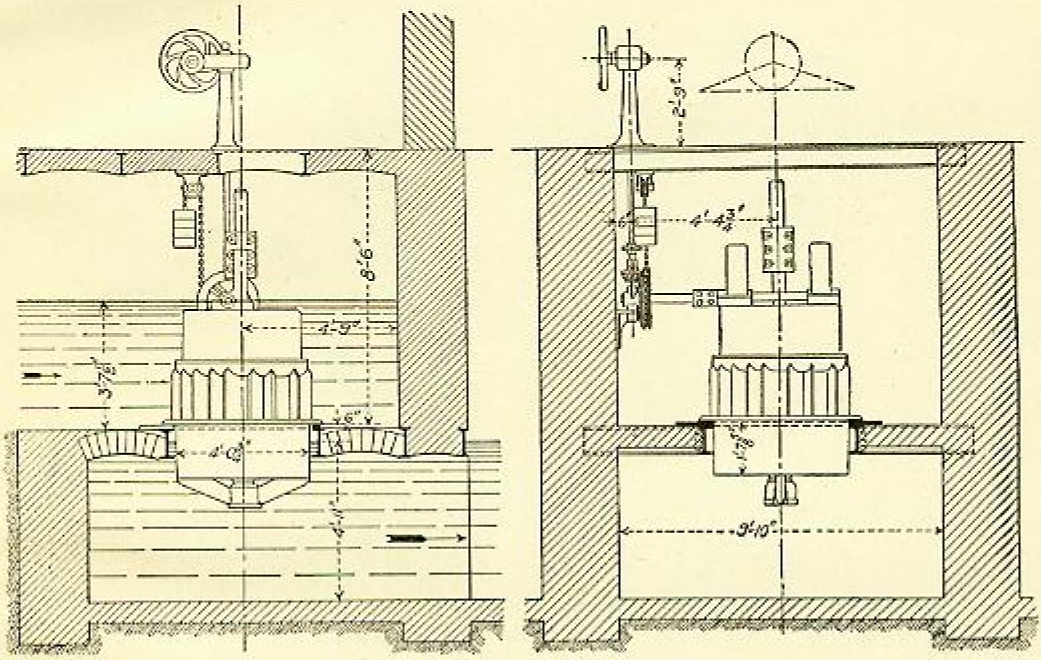


FIG. 56.—General Arrangement of Vertical Turbine with Cylinder Gate.

with a rectangular entrance or more commonly a double elbow, the latter form being shown in Figs. 47, 69 and 72.

Where several turbines are attached to the same vertical shaft and the conditions permit, it is well to have two adjacent wheels discharge into a common concrete draft tube, thereby saving the extra cost of a steel or cast-iron discharge dome, and at the same time affording a firm support for the shaft bearings. A concrete draft tube may be built by the use of a single pattern of the desired cross-section if its shape is not too complicated (see Fig. 60). The inner surface of a concrete draft tube should be composed of the best quality of cement mortar deposited against the forms at the same time the concrete is laid and not plastered

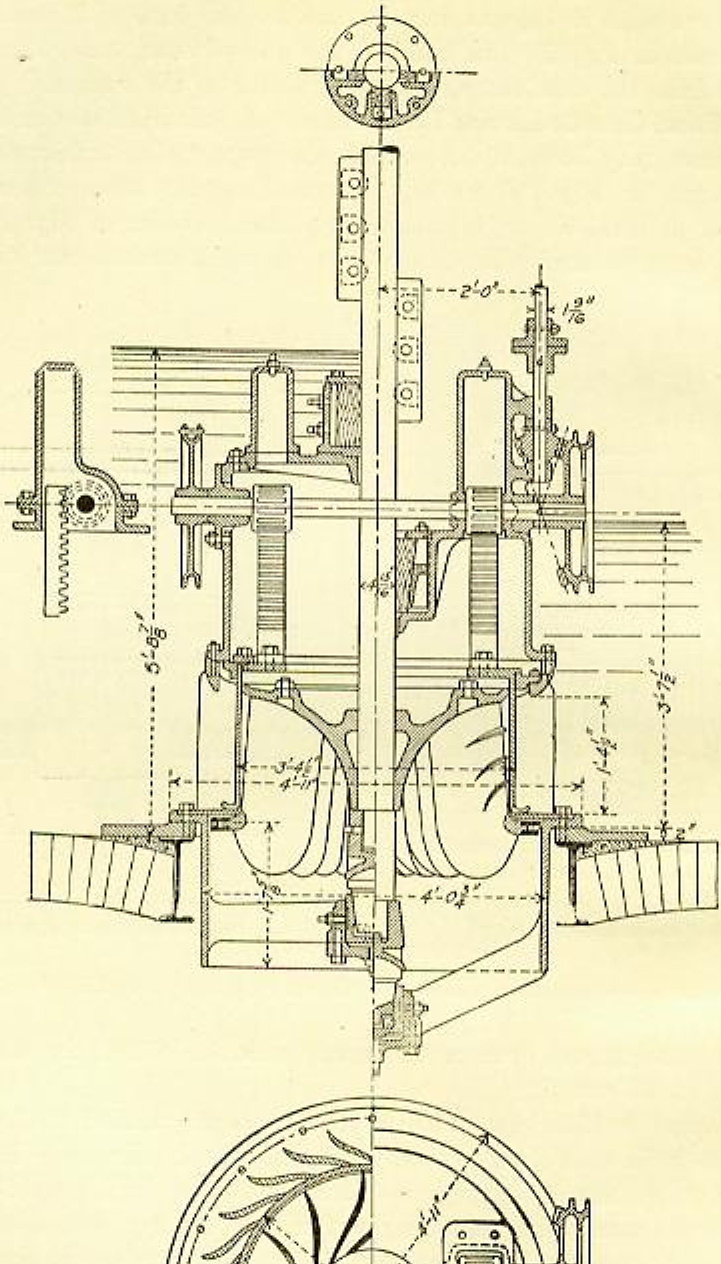


FIG. 57.—Details of Vertical Turbine with Cylinder Gate.

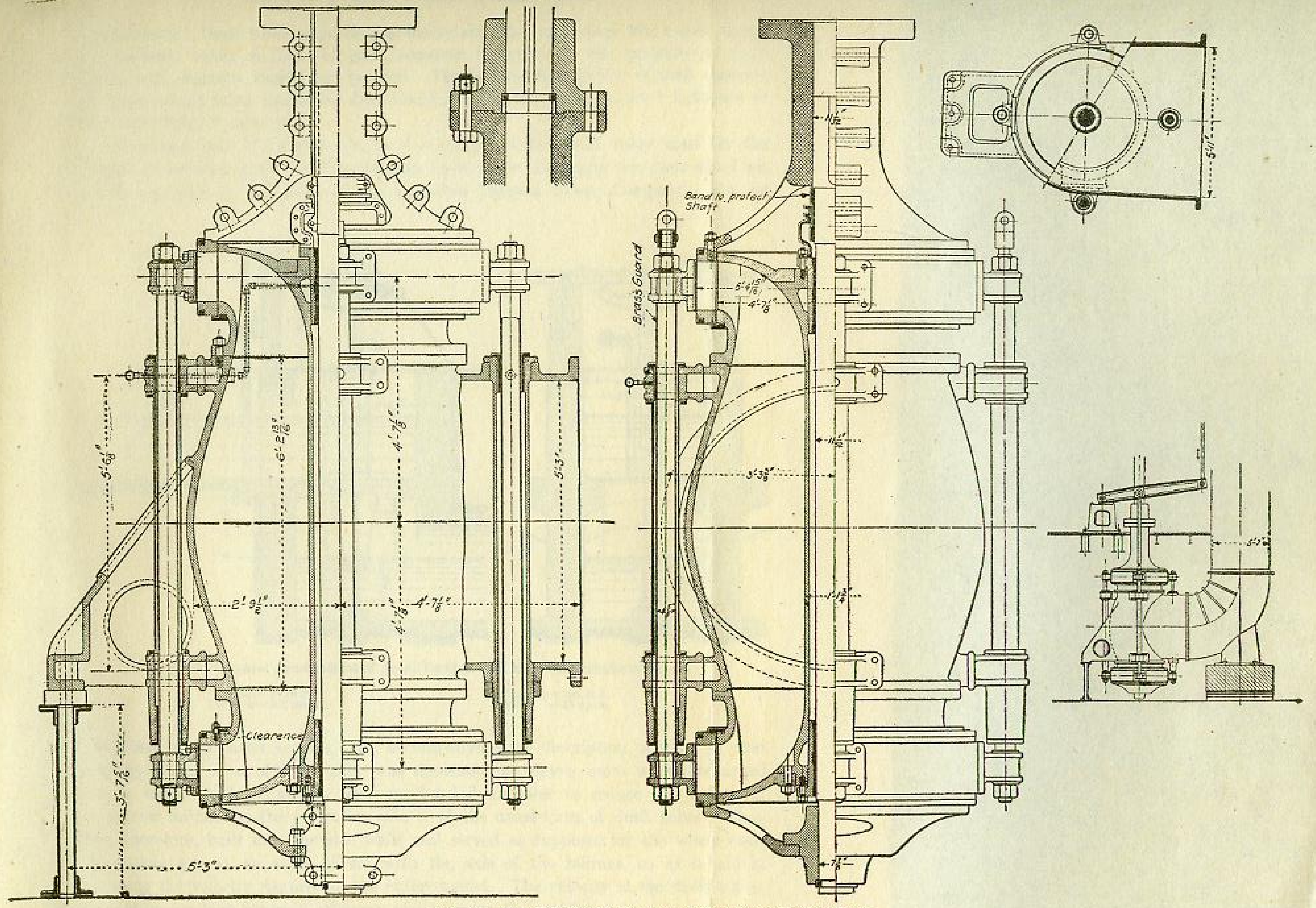


Fig. 55.—5000 H.P. Double Fourneyron Turbine on Vertical Shaft.

Head = 131.2 ft. Output = 5000 H.P.
Quantity of water = 438.5 sec.-ft. Speed = 250 r.p.m.

[To face page 59]

on afterward. Draft tubes must be practically air tight, and where this result cannot be obtained, owing to lack of good concrete materials or the presence of poor foundations, concrete should not be used. The proper cross-section of both concrete and metal draft tubes should be determined by the rules given under "Influence of the Draft Tubes," page 142.

A unique form of construction is that of the forked draft tubes used for the 5500 H.P. turbines made for The Niagara Falls Power Company for their wheel pit No. 2, and also in the plant of the Canadian Niagara Power Company. For an

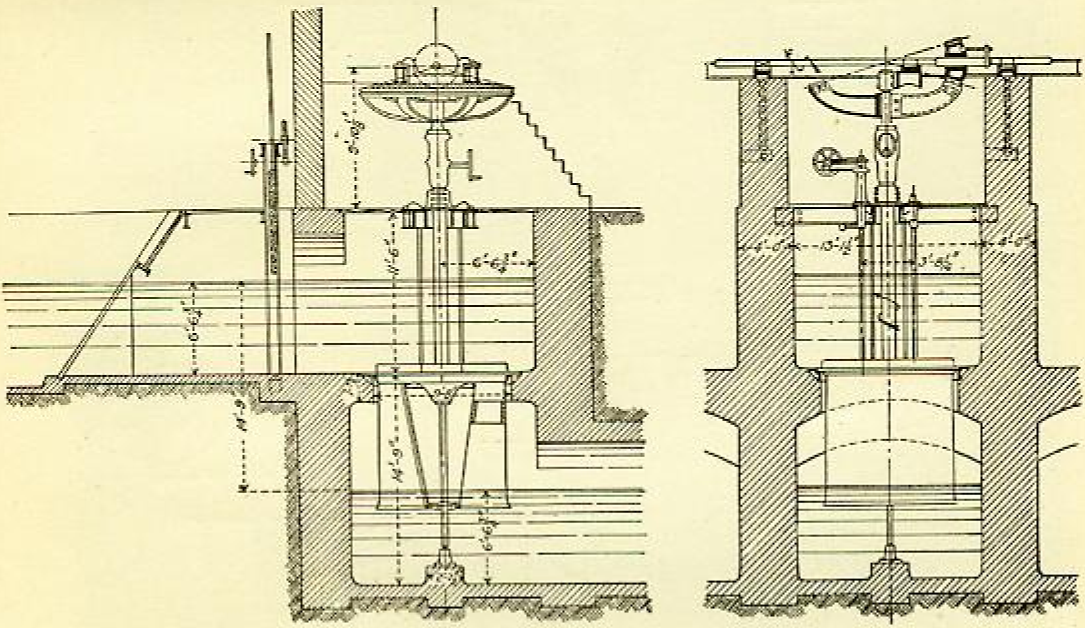


Fig. 58.—General Arrangement of Axial Turbine with Movable Distributor Vanes.

Head = 14.76 ft.
Volume = 307 sec.-ft.

Capacity = 390 H.P.
Speed = 55 r.p.m.

illustration of the latter see Fig. 144, accompanying the description of plant of that company. This form of draft tube was necessary, as eleven units were discharged into the same wheel pit, and it was considered inadvisable to reduce the section of the narrow tailrace by the projection into it of the usual form of draft tubes. They were, therefore, built into the side walls and served as supports for the wheel case. The outlets are at an angle of 45° with the axis of the tailrace, so as to aid in producing the velocity required in the outlet tunnel. The velocity in the draft tubes, however, is too high for good efficiency of the hydraulic installation.

For outward discharge, radial turbines the form of discharge dome shown in

Fig. 54 may be used, as it provides a smooth flow for the water. For impulse wheels the form of housing shown in Figs. 130, 132, 137 and 138 are the best.

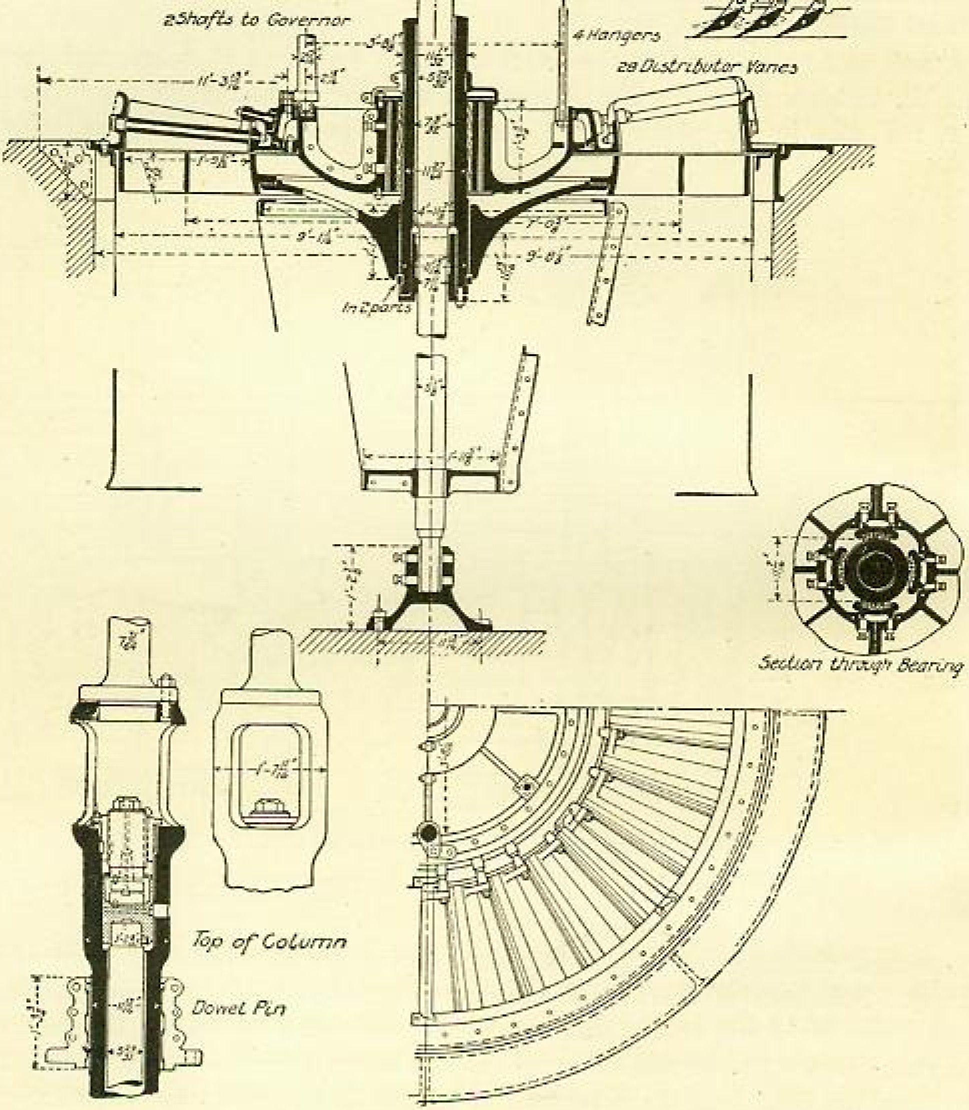


FIG. 59.—Details of Axial Turbine shown on Fig. 58.

The velocity of the discharge water at the point of its entry into the draft tube, assuming its diameter to be that of the outside of the runner, should be from

$.22\sqrt{2gh}$ to $.30\sqrt{2gh}$. As the water leaves the double elbow of twin turbines, if it has no eddies, its velocity may be $.19\sqrt{2gh}$ to $.26\sqrt{2gh}$. On the other hand, in semicircular metal discharge domes and in concrete domes or chambers with rectangular outlets the velocity should not exceed $.15\sqrt{2gh}$. The velocity at the outlet

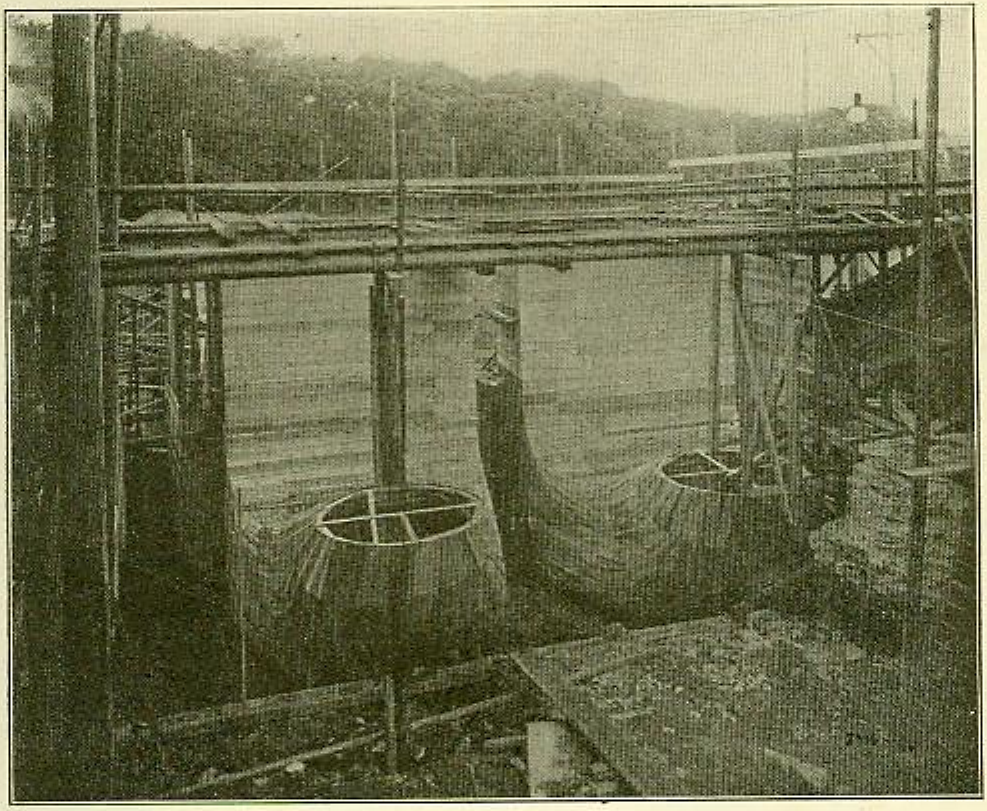


Fig. 60.—Wooden Form for Concrete Draft Tubes.

of a draft tube may be $.1\sqrt{2gh}$, especially if the discharge is in the direction of the flow.

Regulation of the Flow. In former years the turbine most commonly used, especially abroad, was an axial or a Girard radiating turbine, which, as a rule, were regulated by hand and were so arranged that one guide after another was closed to the passage of the water so as to obtain the highest efficiency. This arrangement has been almost entirely superseded by one in which all of the guides are open or closed at the same time. The increase in the number of radial turbines brought about this change, as the modern form of regulating device could be more simply and easily used with this class of turbines. In order to effect quick regulation the designer was also forced to make the movement of the controlling valves from " open " to " closed " as short as possible and this, in turn, caused a change in the cross-section of the guides.

It is evident that movable guides may be operated in a number of different ways, and it is not necessary that they should always maintain the same relation to each other

In the present state of the art, a well-designed regulating device should fulfill the following requirements:

(1) Short throw.

(2) The maintenance of a high, average efficiency for both partial and full gate openings.

(3) A uniform sensitiveness at all times to changes in load whether the regulator acts on the entering or discharge water.

(4) Ease of movement of the operating device under pressure in any position between " open " and " closed."

(5) Freedom from injury by dirty water or by matter carried by the water.

(6) Simplicity of construction, especially in respect to ease of inspection, the replacement of broken parts, and the durability of the parts most subject to wear.

Keeping in mind the object to be attained in a regulating device we will mention a few of those best known, some of which have long been in use.

Regulation by Means of a Butterfly Valve on the Penstock

This means of regulating the water supply to the turbine is by far the simplest form of regulating device, as the wheel and distributor may each be then made in one piece. Requirements (4), (5), and (6), above set forth, are completely fulfilled. The valve may be turned by means of worm gears or racks and pinions operated either by hand or by some mechanical or electrical device. The movement may also be effected either by levers, winches, or hydraulic apparatus acting directly on the shaft of the valve. Requirement (1) is, therefore, complied with.

It is so difficult, however, to attain objects (2) and (3) that regulation by butterfly valves, though frequently used in the past, has now become obsolete, and can be recommended only in extreme cases.

Regulation by Means of Movable Vanes

When the Distributor has Movable Vanes only (*Fink Regulator*). Illustrations of devices coming under this head are shown in Figs. 125, 126 and 127; in Fig. 44; Fig. 45, and in Figs. 66 and 67. The first two of these devices is successfully built by

Escher-Wyss & Company of Zurich, Switzerland, and has proven very effective in practice. The arrangement shown in Figs. 126 and 127 affords an easy means of interchanging the several parts of the vanes, an excellent point in its favor, as with

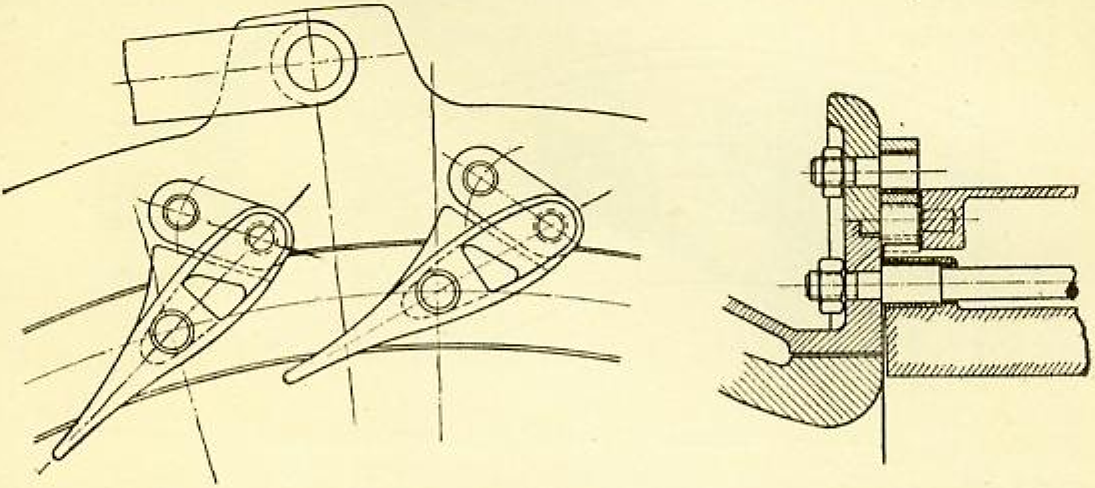


FIG. 61.—Movable Vanes for Distributors as built by Amme, Giesecke & Konegan Co.

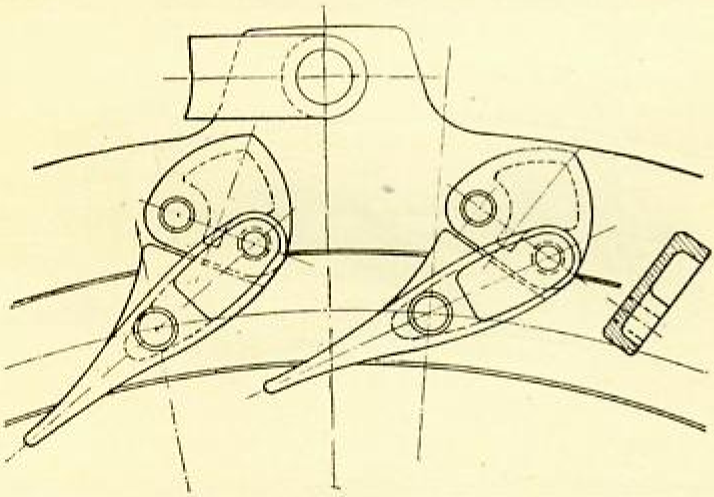


FIG. 62.—Movable Distributor Vanes with Arrangement for Guiding the Water into the Distributor.

this style of turbine interchangeability of parts is an important consideration. It is also important to guard against the wear of adjacent portions of moving parts by inserting washers between them.

A regulator with movable vanes shown in Fig. 61, built by The Amme, Giesecke &

Konegan Company of Brunswick, Germany, has been widely adopted by engineers. This regulator possesses the following characteristics, viz.: the buckets are interchangeable; are symmetrical about their axis, and, therefore, require only one pattern

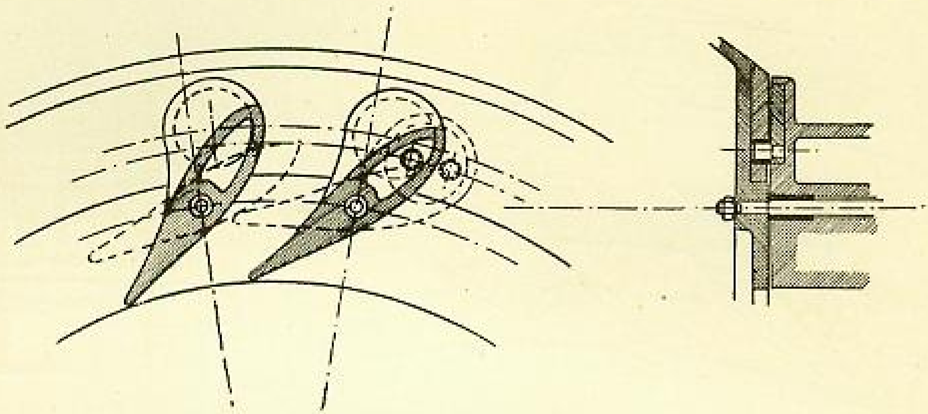


FIG. 63.—Arrangement of Movable Distributor Vanes invented by Storrer.

for either a right-handed or left-handed turbine; the pivots are attached to the ends of the vanes; to prevent eddies in the water a deflector is attached to the guides which performs the office of filling up the dead space behind the guides and giving the

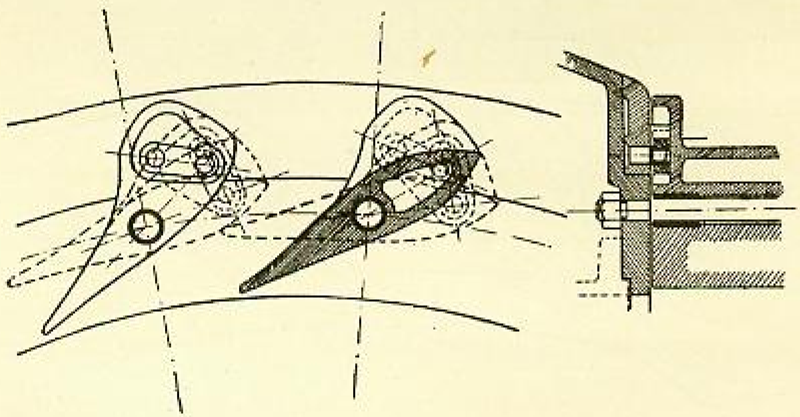


FIG. 64.—Arrangement of Movable Distributor Vanes originally adopted by J. M. Voith.

proper direction to the water at these points; all bearing parts of the vanes and guides are protected by metal bushings. Fig. 62 shows an arrangement by which the flow of the water is properly directed before it enters the guides, an arrangement which is desirable in narrow distributors.

The construction adopted by J. M. Voith, of Heidenheim, Germany, and shown in Fig. 64, has become well known. It involves the principle of placing the pivot mechanism in a small compartment so as to protect it against the admission of dirt. Practical experience, however, demonstrates that just the opposite result obtains, the injurious effect of dirt being augmented. The pockets become filled with fine particles of it which harden on the pivots and finally completely block the openings and

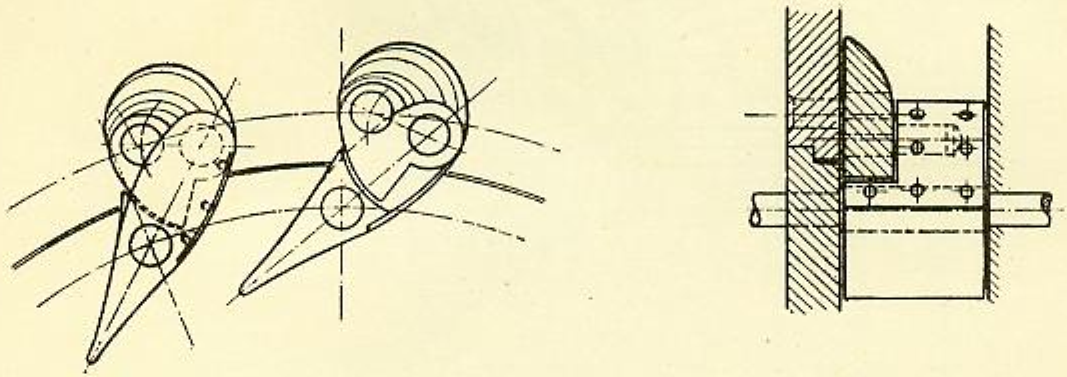


Fig. 65.—Form of Movable Vanes for Distributors used by J. M. Voith.

necessitate frequent cleaning of the pivots. Warned by this difficulty, Voith has abandoned the construction above referred to and has recently adopted the arrangement with exposed pivots shown in Fig. 65.

A similar construction, invented by Storrer and also patented, is shown in Fig. 63. This arrangement is simpler than that of Voith, and also aims to protect the bearing parts of the apparatus from foreign material, but it has not proven a success

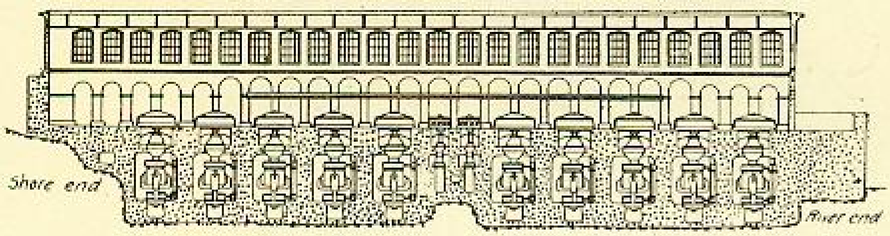


Fig. 66a.—Longitudinal Section of Power House, McCall Ferry Power Co.

in practice. In addition to the losses in friction which the arrangement entails, and which in themselves retard the proper movement of the parts, dirt is deposited on the surface of the eccentric pin which finally hardens and creates still more friction. Both the earlier Voith and the Storrer design have the further disadvantage that the pins of the movable guides increase their section and thus decrease the cross-section of the water passages, a serious matter when the distributor is small.

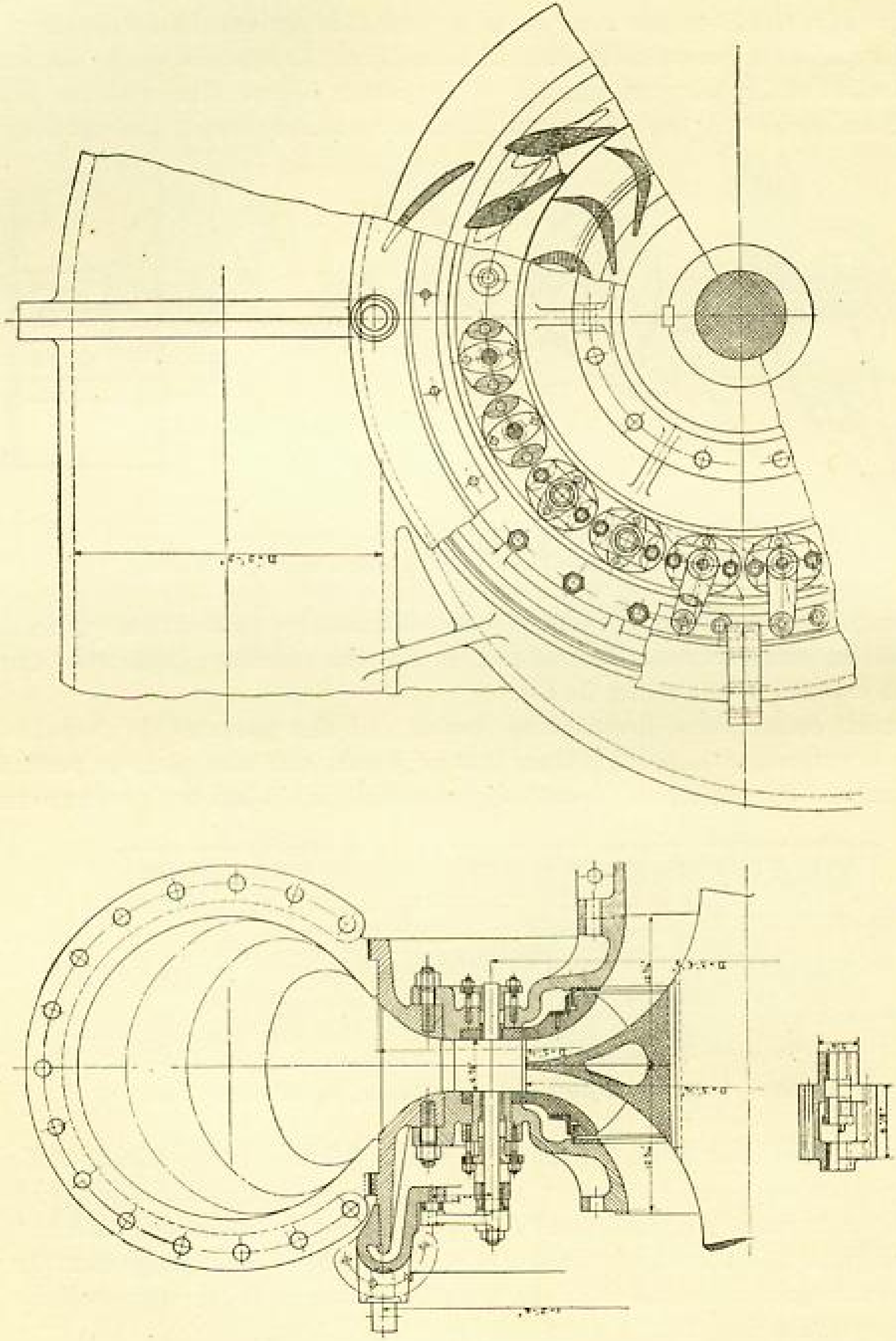


FIG. 66.—Details of 4000 H.P. Turbine built for Anglo-Newfoundland Development Co.

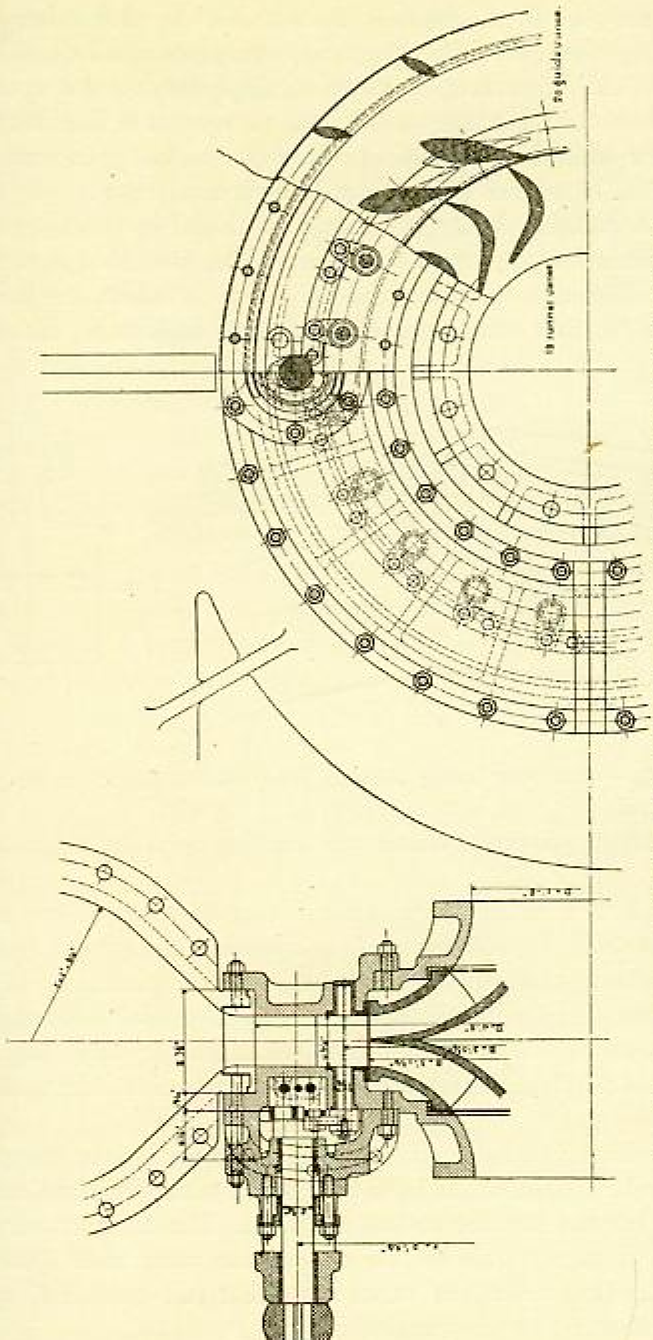


FIG. 67.—Arrangement with Regulating Ring Attached to Distributor having Movable Vanes.

A device patented by Foresti is shown in Fig. 68. This arrangement does away with a distributor ring, as all the buckets are attached to each other by a system of levers and move together, the operating rods being attached at two diametrically opposite points. With this arrangement the guides nearest the operating rods are acted upon first when the latter are moved; the movement is then transmitted to the other guides in their order in diminishing intensity, the last guide receiving the least, as it must be affected by all the lost motion in each connection.

A well-known regulating device frequently employed in this country is that used on the Victor turbines and shown in Fig. 54. Here, too, the pivot arrangement is adopted, with the difference, however, that in order to reduce the friction the cross-section of the guide is made small and the pin long. In Europe, Hausen & Company

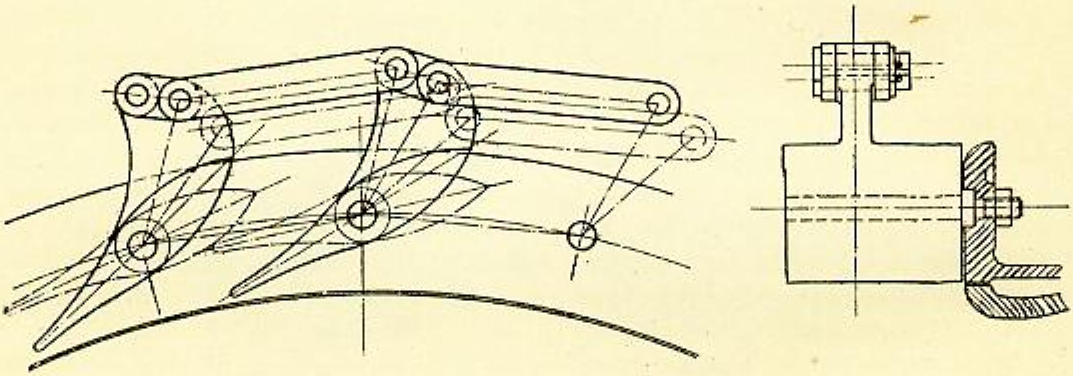


FIG. 68.—Distributor having Movable Vanes without Regulating Ring.

have adopted the above principle, which was the first used for regulation by movable distributor vanes.

Fig. 54 shows how a similar arrangement may be adapted for an outward discharge radial turbine and Fig. 59 shows its adaptation to an axial turbine, the latter form of application being protected by a German patent granted to W. Suchowiah.

All of the above designs for distributors with movable vanes satisfy conditions (1), (2), (3), (5) and (6) of a proper form of regulator and may, therefore, be regarded from the theoretical standpoint as the most perfect forms of apparatus yet suggested.

Regarding condition (4), however, it may be stated that in one position only of the vanes may they be considered as balanced; in all other positions between "open" and "closed" the balance is more or less unstable. The designer may, nevertheless, make the distance from the axis to the end of the vane such that the balancing point may be at the time when the vanes are either full open, half open, or closed. This will also be influenced by the number of guides selected.

Where automatic regulation is not provided, as in the case illustrated in Fig. 44, care should be exercised to provide, by speed reduction or otherwise, for the certain and positive closing of the vanes against the water pressure. A hand wheel larger than shown would be a simple means of obtaining increased power, but would reduce the speed of closing.

From a construction standpoint and in view of requirement (6) it should be remembered that for a turbine with movable vanes a firm support is required for the distributor rings in which the vanes are pivoted and have their bearings. For turbines set in closed chambers this point is to be considered under the head of "Form of Wheel Case." For turbines set in an open flume the necessary supports may be obtained by the use of separate bolts with spacers as in Fig. 47, by columns, or as shown in Fig. 59. In each of these arrangements the pivots of the vanes are relieved of their load and the distributor held firmly together as a whole. In a vertical-shaft turbine such a provision is not necessary if the turbine is single and the head low. It is, nevertheless, a good plan to take up the pressure on the covers of the distributor by bolts and thus relieve the pins of the vanes from unnecessary stresses. Such an arrangement for an axial turbine is shown in Figs. 58 and 59.

The bearings of the pins of the movable vanes and of the shafts of the operating mechanism should be designed with great care, as the parts are almost constantly in motion, and it is evident that the friction should be reduced to a minimum. This result may be attained in various ways. One method of accomplishing it is by the use of a small distributor ring as shown in the illustration in Fig. 47.

With a vertical-shaft turbine the distributor ring may rest on ball bearings which are so designed as to prevent the entrance of dirt (Fig. 48). An unusual foreign practice consists in placing curved pieces of round iron in the bearing grooves between the balls, the diameter of the iron ring being slightly smaller than that of the balls.

Where turbines are installed in wheel cases, especially when double turbines are used, it is well to use a double distributor ring as shown in Fig. 44 and Fig. 45, and also in Fig. 125. As shown on the latter plate, special transverse separators may be used between the rings in addition to the pins of the vanes, and the rings may thus be firmly secured together. By this arrangement the rings are held firmly and the vanes are properly acted upon by the water.

For use in connection with a turbine inclosed in a wheel case a method of arranging the mechanism for operating the movable vanes of the distributor, differing from those above described, has been developed in recent years, and has been widely adopted, especially in the United States. It has been used by the Platt Iron Works Company and by the Allis-Chalmers Company, and was used by J. M. Voith in the design of the 11,340 H.P. turbines for the Ontario Power Company at Niagara Falls, Ontario. This construction is fully illustrated in *Engineering News*, March 29, 1906, in connection with a description of the 10,000 H.P. turbine manufactured by the Platt

Iron Works Company for the power development at Snoqualmie Falls, and the attention of the reader is called to Fig. 166 in the chapter of this book describing the plant of The Hydraulic Power & Manufacturing Company at Niagara Falls. It will be seen that the arrangement consists essentially in bringing the pins of the movable vanes to the outside of the casing through stuffing boxes. The operating mechanism is exposed to view and the parts are interchangeable. The motion of the governor is variously communicated to the pins. In the case of the Snoqualmie turbine, lugs on the vane pins have square ends fitting into brass discs set in the circumference of the movable ring, while for the exciter turbine shown in Fig. 166 the shaft of each pin is connected by a double link with the movable ring which, in turn, is operated by two rods connected with a quadrant driven by the governor.

An entirely new development of the idea is shown in Fig. 185, which illustrates the 4000 H.P. turbine built by Amme, Giesecke & Konegan Company for the Anglo-Newfoundland Development Company. The construction adopted, which was suggested by Mr. Gelpke, is shown in detail in Fig. 66. In the design it was sought to fill the following requirements, viz.: the turbine was to be as small as possible; the pins of the movable distributor vanes were to be firmly supported; the stresses in the pivots were to be equally distributed; the distributor was to be capable of easy and accurate assembling, similar parts being interchangeable. This design of distributor vanes provides abundant and properly formed passageways for the water, and in this respect may be considered ideal. Yet on account of the large number of stuffing boxes, the increased size of the turbine and the increased cost, the design may be considered an extreme one to accomplish the desired result. It can be used for turbines in wheel cases only, while other arrangements of distributor vanes and their operating mechanism may be used both for wheel-case and open-flume installations.

Fig. 67 shows a less expensive arrangement in which the regulating ring and its operating mechanism is attached to the distributor. The pivots of the distributor vanes are secured to them in holes bored or cast in the vanes. The distributor vanes may be made of wrought iron with the pins welded to them, but where the increased cost is justified the vanes should be of cast steel with the pins cast as an integral part, this being done recently for the larger installations, like that at McCall Ferry.

A split-spiral wheel case is expensive and not always necessary.

The moving distributor ring should be attached to its operating mechanism at two diametrically opposite points so as to avoid eccentric stresses. This is best accomplished by the use of two opposite shafts with levers attached to the distributor ring by plates. The connections between the shafts and the ring must be so designed that equal forces are applied to the points of connection with the ring, suitable compensating or adjustable devices being employed.

In the case of turbines installed in open flumes the motion of the operating shaft can best be transmitted to the distributor ring by means of a crank and levers, or a series of levers, where the distance between the operating shaft and the main turbine shaft will permit. Such an arrangement is shown in Fig. 69. It will be seen that the points of attachment of the rods to the ring are not diametrically opposite and hence the pull on the ring is one-sided, but the difficulty from this cause decreases as the distance between the operating and main shaft increases.

The use of a toothed segment and rack for transmitting the operating shaft motion to the distributor ring cannot be recommended, but if used for an open-flume installation the teeth must be thoroughly protected from dirt in the water and proper provision must be made for lubrication. The teeth should be machine cut when an automatic governor is used for operation as lost motion must be eliminated. Distributor rings operated from only one point should never be used except for small turbines and then only when provision has been made for relieving the excessive friction.

Where the effective head exceeds 50 feet, cast steel should be used for the distributor vanes, not only to safely withstand the water pressure, but also the stresses caused by an attempt to move the vanes after any solid material has become wedged between them. To overcome the breakage of vanes through the lodgment of solid matter in them, Piccard & Pictet have designed an arrangement whereby the vanes are provided

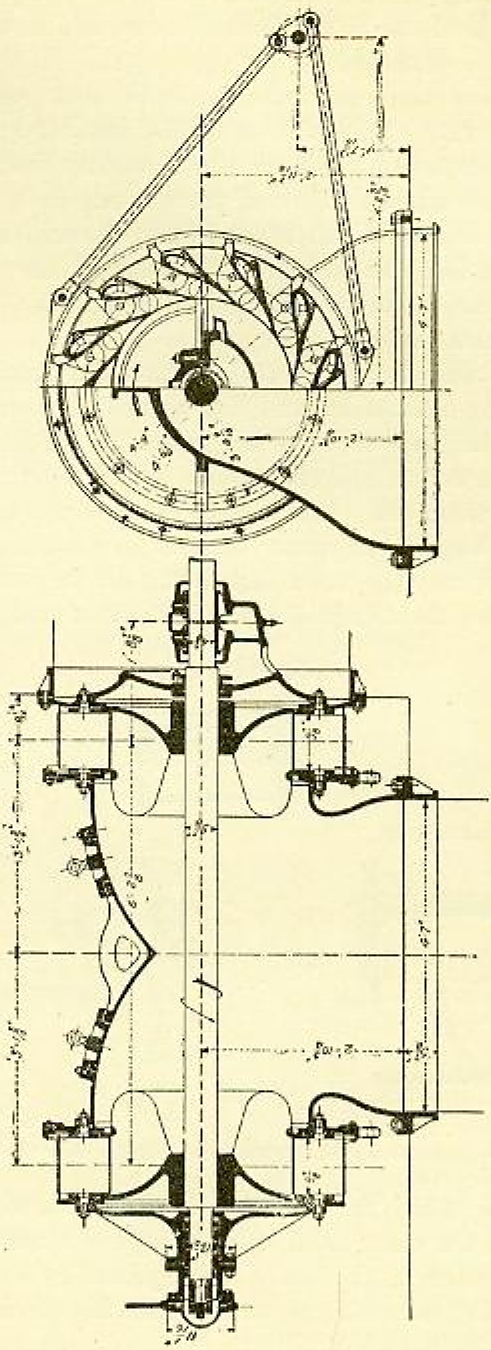


FIG. 69.—Francis-Turbine with Horizontal Axis.

Head = 24.28 ft. Output = 240 H.P.
Quantity of water = 107.3 sec.-ft. Speed = 240 r.p.m.

with springs so that the vanes will slightly separate when pressed against the obstacle lodged in them. This form of construction may be criticised for the complications which it introduces.

Where the conditions are such that vanes are frequently broken, the interchangeability of parts is of special importance in order that repairs can be quickly made, for the entire distributor is usually made useless if one vane is out of commission. Interchangeability of parts is not so important, however, from the standpoint of replacing worn parts, for it may be anticipated with proper design and uniform material that similar parts will wear at the same rate. This is a more theoretical than practical consideration and in no way detracts from the advantage of interchangeable parts.

In order to reduce the wear on the moving parts of the distributor vanes and operating mechanism, and thus reduce the repairs to a minimum, the bearing surfaces should be made as large as possible, a consideration especially important where the parts are submerged and cannot well be lubricated. It is desirable, but not essential, that it should be possible to remove each vane separately, as it is convenient to be able to inspect the runner buckets by removing a distributor vane. In many plants, however, only the entire distributor can be removed.

When the Distributor has both Fixed and Movable Vanes. Among the patented arrangements involving this principle of construction are those of Thomas Bell & Company, shown in Figs. 48 and 69, and that of Victor Gelpke, shown in Fig. 70. With the use of both fixed and movable vanes requirements (1), (3), and (6) are satisfied, especially (6). Regarding requirement (2), however, it will be noted that in partial-pressure turbines the dead space between the fixed and the movable vanes produces eddies which doubtless reduce the efficiency of the prime mover and also cause erosion of the distributor, especially where the operating head exceeds 66 feet or where the water contains sand.

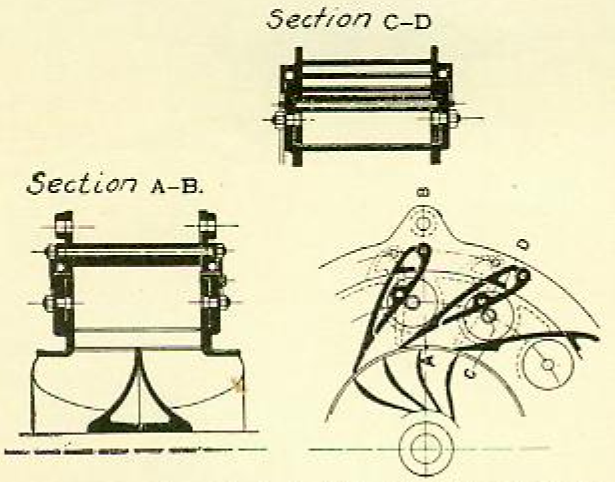


FIG. 70.—Distributor having both Fixed and Movable Vanes.

On the other hand, the arrangement now being considered conduces to the ease of operation of the vanes, as required by (4), to a greater extent than in designs containing movable vanes only, and this consideration has led to the adoption of the design shown in Fig. 70.

Referring to Fig. 48, it may be said that the lever arrangement is much to be preferred to one with a pinion and rack, for the latter is opposed to requirement (5), as solid material may become wedged between the teeth and thus increase the power required for operation and perhaps break the teeth.

It may be stated that where both fixed and movable vanes are used in the distributor the entrance angle becomes greater when the vanes are partially closed and reaches its maximum when they are fully closed, which, moreover, agrees with a condition of an entrance free from impact up to a degree of reaction equal to one-half (i.e., $\phi = .63$). On the contrary, where the vanes are all movable the entrance angle becomes smaller at partial vane openings and is zero when fully closed.

Regulation by Means of Circular Valves or Ring Gates

This form of regulation of the water entering the distributor has been much used. It is shown in Figs. 46, 56, 57, 71, 72, 72*a* and 72*b*. Among the larger installations for which this arrangement has been employed are the following, designed by Escher, Wyss & Company:

5500 H.P. single turbines in wheel pit No. 2 of The Niagara Falls Power Company.
10,000 H.P. double turbines for Canadian Niagara Power Company.
3000 H.P. turbine at Glommen.

Other installations are as follows:

600 H.P. Fourneyron turbine built by Thomas Bell & Company.
1100 H.P. Fourneyron turbine built by J. J. Rieter & Company, Winterthur, for the hydro-electric works at Montboven, toegther with many installations in this country.

The circular valve, or ring gate, consists of a smooth cylinder sliding in a space between the distributor and the wheel and can therefore be used for radial turbines only. It satisfies requirements (4), (5) and (6) very effectively, meets requirements (1) and (3) reasonably well, but does not fulfill condition (2).

Referring to requirement (1) it may be said that in general the motion of the governor rod is equal to or slightly greater than the height of the gate, but this movement may be decreased by the use of a properly designed mechanism such as a series of levers or the rack and pinions shown in Figs. 72, 72*a* and 72*b*. In the case of the Niagara turbines designed by Faesch & Piccard and Escher, Wyss & Company, the rack actuated by the governors has a movement materially in excess of the height of the gate, but nevertheless the gates can be closed in two seconds, if desired, owing to the use of counterbalances.

Where ring gates are used the efficiency is high, as a rule, only when the gate is wide open. To overcome the relatively low efficiency at partial gate openings and

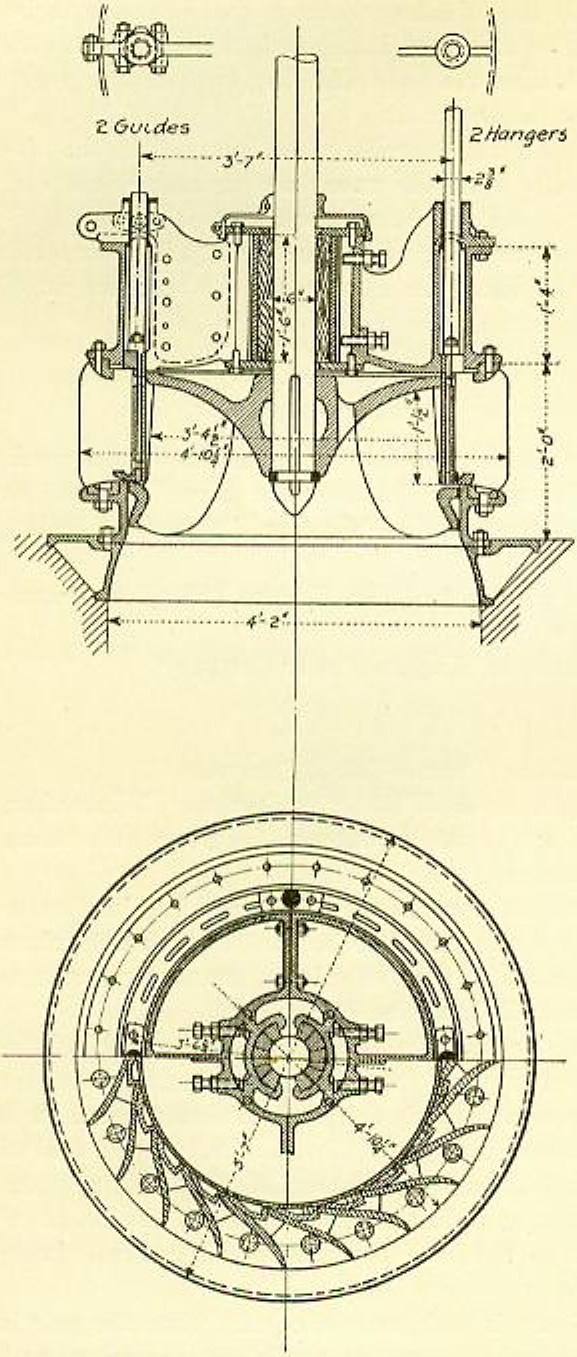


FIG. 71.—Francis Turbine with Cylinder Gate and Vertical Shaft.

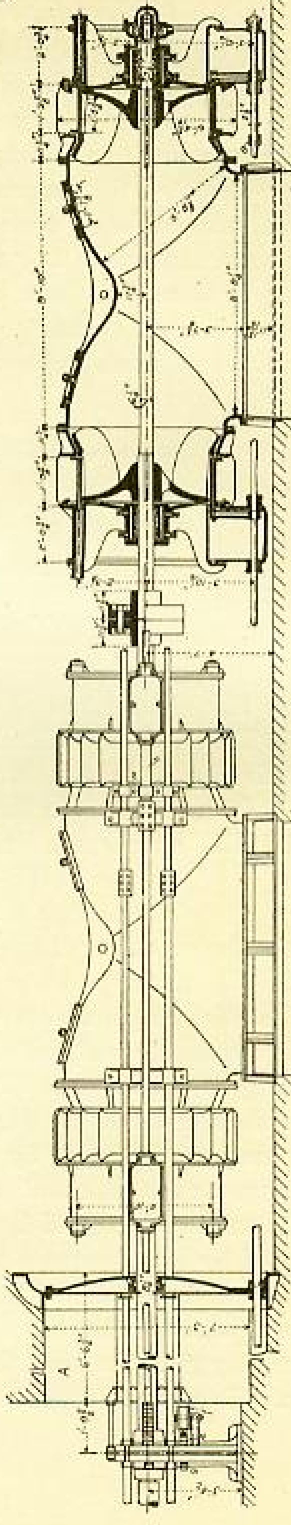


FIG. 72.—General Arrangement of Quadruple Turbine Regulated by Cylinder Gates.

at the same time permit the use of the simple ring gate, this form of construction has recently been modified as shown in Fig. 71 and Figs. 72, 72*a* and 72*b*. In this design special guiding lugs projecting as far as possible into the distributor are attached to the bottom of the circular valve and thus at partial gate openings afford a smooth passage for the water with less contraction, thereby increasing the efficiency.

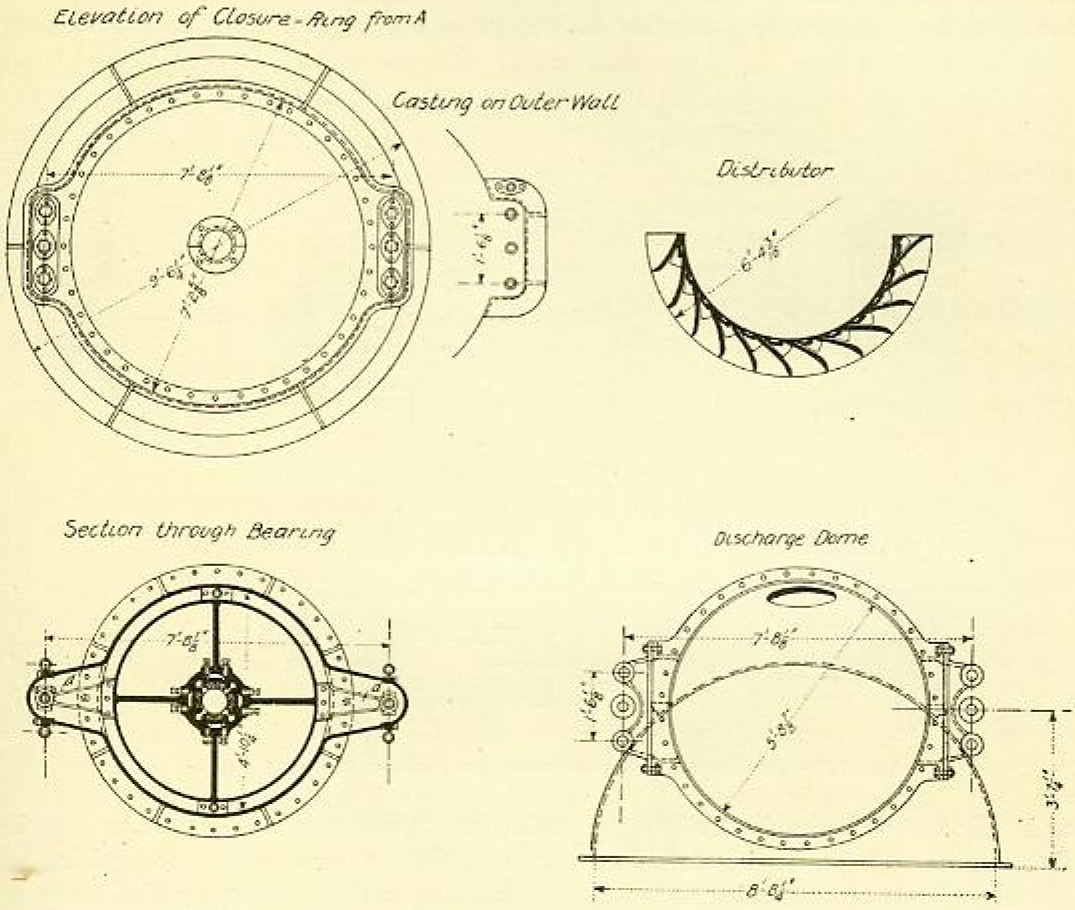


FIG. 72*a*.—Details of Regulating Device for Turbine shown in Fig. 72.

Referring further to the conditions of proper regulation it will be noted that with a ring gate the discharge is not proportional to the gate opening. As, for example, when the gate is half open the discharge is nearly two-thirds as great as when the gate is wide open. This condition is taken into account in the form of construction patented by Zodel by making the entrance angle and consequently the width of the opening relatively smaller as the gate closes. In the use of hydraulic or mechanical occupy nearly one-half of the cross-section of the distributor, "dead" spaces occur both in the runner and in the valve, especially at partial gate openings. These produce eddies and thus reduce the hydraulic efficiency and cause unnecessary wear.

It may be added in respect to condition (4) that the moving valve may be properly guided and supported on the outside of the cover only in the case of vertical turbines. To accomplish this result in the case of turbines with horizontal shafts it is necessary to have an additional support for the valve, as otherwise the gate would tilt from its own weight when the bearings became worn and this would result in the gate wedging fast. The necessary support may be obtained by the use of a ball bearing on the stationary distributor. This method was formerly frequently employed. In the case of both horizontal and vertical turbines it should be remembered that the movable ring receives a strong axial thrust in a direction opposite to the discharge, this being due to the suction. This must be provided for in some manner or a considerable additional turning movement will result and the movement of the valve will be made much more difficult by overcoming it.

It may be said in regard to condition (6) that it is difficult to cast the distributor and ring vanes in parts accurately enough so that the openings will fit closely over each other without considerable additional fitting. It should also be borne in mind that on account of the eddies above referred to erosion will take place in the valve and on the edges of the runner under high heads. Also that the increased thickness of the vanes of the distributor and movable valve, as compared with a normal cylinder gate, results in a greater diameter of distributor and runner and hence increases the space occupied by the turbine machinery.

Finally, it may be stated that the runner vanes as they revolve are subjected to water hammer and vibration due to the constant change from eddies to discharge water. It is, therefore, best in this form of construction to cast the entire runner, including its vanes, in one piece and not to use vanes made of sheet steel. The simple, solid, and homogeneous construction of the distributor constitutes one of the advantages of this form of construction.

Regulation by Means of a Cylinder Gate Outside of the Distributor Moving Parallel to the Shaft

This method of regulation has been successfully employed by Thomas Bell & Company, having been used by that firm on a large reaction wheel for Alten-Aarburg. It was also used by Escher, Wyss & Company in connection with the radial turbines installed at Rheinfelden. This design, however, has the disadvantage that even when the valve is wide open the entrance cross-section at the outer diameter of the distributor is so contracted because of the flange around the upper portion of the ring valve that the velocity of the water is nearly as great there as within the vanes.

The water in passing from the wheel case to the distributor is subjected to a sudden increase in velocity which results in eddies and reduces the hydraulic efficiency.

In reference to the conditions requisite for a satisfactory regulating device it may be stated that conditions (4), (5), and (6) are properly met by the construction above considered. Regarding condition (1) the same remarks apply as to Regulation by Means of Circular Valves. It should be noted in respect to condition (2) that with the form of construction under consideration the maximum efficiency is obtained when the gate is wide open and that it decreases rapidly as the gate closes. It is therefore desirable that the valves be equipped with lugs to properly guide the water into the distributor—an arrangement practical only in a radial turbine. Regarding condition (5) it appears that at partial gate openings a cylinder valve on the outside of the distributor is not as satisfactory as a regular ring valve between the distributor and runner unless it is equipped with the lugs above described.

In respect to condition (6) it may be said that the form of construction under consideration surpasses the usual ring valve in simplicity, strength, accessibility, and interchangeability, especially in the case of a number of turbines, where it is the simplest arrangement conceivable. Attention is called to the opportunity for firm connections between the valve rods and the gates and for rigid supports for the bearings. This method of regulation is recommended for a turbine installation of a large number of units where a high efficiency is desired at full gate opening only, but for the usual case where good efficiency at partial gate openings is a necessity it should not be used.

Regulation by Means of a Turning Ring Gate Placed at the Entrance to the Distributor (Register Gate)

This method of regulation has been used in a number of installations, especially for horizontal, axial turbines.

Conditions (1), (4), and (5) are as satisfactorily met by this arrangement as by that described under the 4th method of regulation, and it better fulfills condition (6). Respecting condition (2), the efficiency at full gate openings will be high, but at partial gate openings the efficiency will decrease more rapidly than with the usual form of movable vanes. The diameter of the distributor is not unduly increased, as is the case with the fourth method.

The arrangement satisfactorily meets requirement (3).

Regulation by Means of a Cylinder Gate Moving Parallel to the Shaft on the Outside of the Runner

As an example of this arrangement see Fig. 55, which illustrates the 5000 H.P. Fourneyron turbines installed in wheel pit No. 1 of The Niagara Falls Power Company and also Fig. 75, which shows the patented construction used by Escher, Wyss & Company at Chevres.

In regard to the conditions of proper regulation the same remarks apply as already made in respect to the fifth method of regulation.

Where this form of construction is employed care must be taken that the velocity of the water continually increases from the point where it leaves the penstock until it reaches that in the distributor; also that the draft chest shall so inclose the runner that no sudden change of cross-section occurs in passing from the latter to the former. The height of the draft chest must also be made equal to the width of the double runner in a design like that shown in Fig. 75.

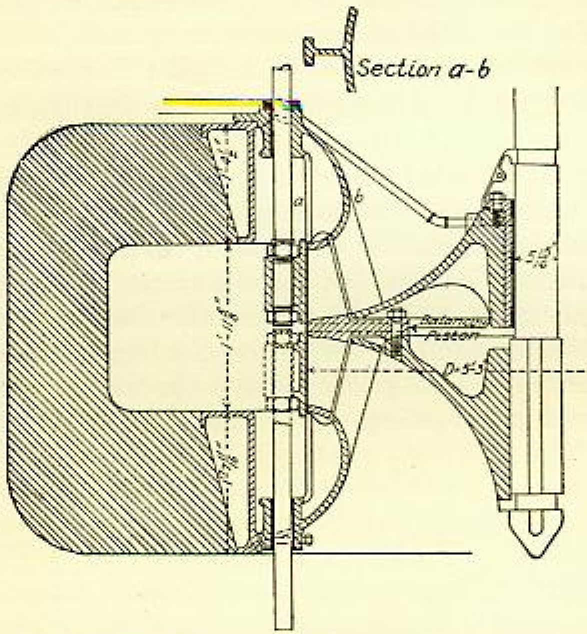


Fig. 75.—Double Vertical Shaft Turbine Regulated by Cylinder Gate on Outside of Runner.

Hand Regulation.—In order to insure the desired regulation at all times it is necessary in every case to install a hand-regulating device, even when the turbine is equipped with an automatic regulator. In the latter case the hand regulator is used in starting up and also when the automatic regulator is out of order. The conditions which determine a well-designed hand-operated governor are the following:

(*a*) Certainty of action and a condition of constant readiness for service.

(*b*) The property of self locking.

(*c*) Easy operation and quick closing.

(*d*) A hand wheel placed about three feet above the floor, an arrow on the same indicating the direction of opening and closing, the latter preferably clockwise and the former counter-clockwise.

(*e*) Reliability in the change from hand to automatic regulation and vice versa.

The safest and most reliable method of hand regulation is by means of a worm or screw or a combination of both. (See Figs. 45 and 54.) Such a mechanism has the advantage of locking itself in any position, and wherever possible should be independent of the automatic apparatus.

Hydraulic hand regulation is used only in connection with an automatic governor having the necessary cylinder and piston. Frequently the necessary pressure is obtained by means of a separate hand pump and is conducted to the cylinder. Such a regulating apparatus is not absolutely sure and quick in its action, as it depends absolutely upon the tightness with which the pistons of the hand pump and governor fit the cylinder. It also depends on the rapid filling of the cylinder, and a clear liquid must be used. For example, if the piston leaks or if the valve connecting the two ends of the cylinder is not absolutely tight, the device is not self-locking, for the unbalanced distributor system produces a pressure on the piston and consequently the piston will move if it does not fit tightly. If the liquid leaks past the piston it will also require a much larger and unnecessary amount of pumping. Should the cylinder be partially filled with air a "dead center" will be produced and the requirements for good regulation will not be complied with. To insure the tightness of the piston it should be ground or provided with leather-cup packings, the latter being better than piston rings. Ground globe valves or gate valves should be used for controlling the flow of the liquid and the hand pump should be equal in workmanship to that used for a diving apparatus. All auxiliary devices, such as cocks and valves, must be absolutely tight.

Hand regulation is frequently employed in connection with the piston and hydraulic pressure with the automatic governor. A three-way cock is sometimes used in connection with the piping from the hand-regulating device, the fluid under pressure being connected through it at a given time with one side of the piston while the same position of the cock allows the liquid to escape from the opposite side of the piston. With a differential piston the same result is obtained by operating on the high-pressure side only. In this case the hand-regulating device becomes simply a means of checking the movement of the piston at the desired point.

In many of the best hydraulic governors the hand mechanism operates the pilot valve and through it causes the main piston to move, the automatic device being thrown out of gear when the hand regulation is thrown in.

To fulfill requirement (*c*) it is necessary to limit as much as possible the space between the hand wheel and the apparatus which moves the distributor or, in other words, to have as direct a transmission as possible to the operating device of the power applied to the hand wheel. This condition (*c*) also requires the reduction of the friction losses to a minimum, the attainment of a high combined efficiency, and a small reduction in gearing so that the valve may be closed quickly. Worm gears have been found by experience to be superior to screws for doing the same work. Condition (*c*) may be met by the use of a well-made hand pump. The hand wheel will operate more easily if pins are attached to the circumference. The proper lubrication of all moving parts must be provided for.

In regard to condition (*e*) it may be said that the transition from automatic to hand regulation may be effected in various ways by hand-operated devices, among which may be mentioned a claw coupling as shown in Fig. 20, a plate with a hole for the connection of the hand control or a slot for the automatic control, as shown in Fig. 45, the shifting of the worm from its wheel or the disengaging of the nut from a screw drive, etc.

In an installation with an hydraulic hand control the transition must be effected by means of cocks operated simultaneously.

The hand wheel and the mechanism for shifting from hand to automatic control should be as close as possible to one another so that the change may be made quickly and certainly. The best arrangement is that in which the cutting out of the hand control throws in the automatic mechanism and vice versa.

The Control of Forces Acting Parallel to the Shaft.—On account of the differences in the static pressure of the water operating the runner, of the changes in the velocity of the water as it passes through the moving vanes and also because of the dead weight of the revolving parts in the case of vertical turbines, forces are exerted which tend to shift the revolving parts from their proper position, and these forces must be controlled.

The exact determination of the forces to be resisted resolves itself into the following four problems:

(*a*) The calculation of the weight of the revolving parts of a vertical turbine. In determining the weight of the runner the actual dead weight must be reduced by the weight of the water displaced by it, provided that the wheel is entirely submerged, which, however, is not the case with a wheel having a free discharge or with a Pelton wheel.

(*b*) The determination of the pressure p on both the inner and outer faces of the runner and the resolution of the same into its radial and axial compounds, p_r and p_a. For example, let the pressure at the point P distant r from axis equal p, a pressure determined from the velocity of the water at this point and the useful head on the turbine. Then the upward pressure on a circular strip of the area $2\pi rdr = 2\pi rdrp_a = 2\pi rdrp \cos \phi$, where ϕ represents the angle which the surface of the runner at the point P makes with the horizontal. The summation $\Sigma 2\pi rdrp \cos \phi$ is the axial pressure which is exerted upward on the inner face of the runner vane. In a similar manner there can be obtained the pressure on the outer face of this vane as well as on the other vanes and finally by the difference between the upward and downward axial forces we can obtain the total axial pressure due to the static pressure.

(*c*) To determine the weight of the water contained between the runner vanes in the case of a vertical turbine.

(*d*) A determination of the reaction pressures due to the changes in the velocity of the water in its passage through the moving channels, that is, the dynamic action. The necessary calculation is approximately as follows, viz.:

In the sketch of the turbine vane shown in Fig. 80 which uses dG pounds of water having a mass $\frac{dG}{g}$, let the velocity at entrance be $c_p = c_e \sin \alpha = w_e \sin \beta$, and the velocity of the discharge be $c_a'' = c_a \sin \delta = w_a \sin \gamma$, both velocities being in a radial plane, i.e., the plane of the paper.

Now let us denote by μ and r the angles which the entrance and exit edges of the vane make with the direction of the axis. Then the components of the axial pressure are respectively $c_p \sin \mu$ and $c_a'' \sin r$. Then we have on account of the changes of velocity a total downward axial pressure of the amount

$$=\frac{dG}{g}(c_p \sin \mu - c_a'' \sin r);$$

and of the entire turbine a sum total of

$$\Sigma\frac{dG}{g}(c_p \sin \mu - c_a'' \sin r);$$

or

$$=\Sigma\frac{dG}{g}(c_e \sin \alpha \sin \mu - c_a'' \sin \delta \sin r);$$

or

$$=\Sigma\frac{dG}{g}(w_e \sin \beta \sin \mu - w_a \sin \gamma \sin r).$$

The following are the means of controlling, resisting, or supporting the axial pressure:

(*a*) A symmetrical arrangement of the turbine units in the form of double turbines, quadruple turbines, etc.

(*b*) An arrangement for reducing the axial pressure by relieving the water pressure acting on one side of the runner vanes.

(*c*) An arrangement for the production of counter forces generated by the pressure of water against a piston keyed to the shaft and revolving with it, that is, a balance piston.

(*d*) A mechanical support in the form of a pressure bearing, step bearing, or thrust bearing.

(*a*) Although with a symmetrical arrangement of turbines it might be expected that the axial forces would be balanced, yet it has been found in practice that even with such a design it is necessary to provide a step bearing to receive a certain amount of unbalanced pressure. (For an example see Fig. 69.) This is due to the fact that while theoretically the balancing is perfect, yet practically it is easily disturbed; for instance, in a double turbine at partial gate opening an unequal vacuum may occur when two turbines equal in all respects use an unequal amount of water. Inequalities may occur in the cross-section of the runner or distributor vanes; draft tubes or stuffing boxes may be unequally tight, etc. All of these inequalities cause unequal axial pressures.

(*b*) This arrangement involves the connection of the space between the runner and the top of the wheel case with the draft or tailrace so that pressure cannot accumulate there from the water which leaks past the wheel. This result may be accomplished by means of a pipe containing a valve for regulating the pressure, or holes may be provided in the runner hub so that the space above the runner is connected with the draft tube. In the latter case the axis of the holes should be in the relative direction of the water as it leaves the runner (w_a) in order to make the vacuum over the runner as large as possible, but should never be at right angles to (w_a), as the connection between the draft tube and the space over the runner would be practically cut off. If a separate pipe is installed to carry away the leakage from the space above the runner, its area should be at least one-third of the minimum cross-section of the clearance space. It must also be remembered that as the gate opening of the turbine changes so the discharge will also vary. For this reason it is necessary to change the valve opening on the discharge pipe so as to preserve the balancing effect at the desired amount.

(*c*) The installation of a balancing piston operating under the full pressure of the water from the headrace is to be considered, preferably for vertical turbines, as it then serves two purposes: first, to balance the axial forces due to the water and, secondly, to carry the entire weight of the revolving parts of the turbine and alternator. It will thus reduce to a minimum the downward pressure to be carried by the step bearing.

In some cases two balancing pistons are used on the same turbine, the lower face of one being subjected to the pressure while the upper face of the other is exposed to a vacuum from the draft tube, the opposite faces of each piston being subject to reverse pressures. A simpler form is shown in Fig. 48 in connection with the middle wheel. In the turbine shown in Fig. 75 the space between the upper distributor and the runner hub is connected with the discharge dome, while the lower runner hub is acted upon by the full pressure of the water used for operating the turbine. In the design of the 10,000 H.P. turbines for the Canadian Niagara Power Company, a combination of the two forms of balancing piston was employed, one of the runners and a separate piston being used to balance the downward thrust and dead weight. For an illustration of this arrangement see Fig. 144 in the chapter describing the plant of the above-named company.

In most cases it is best to provide means for regulating the pressure under the balancing piston. This may be done by a valve or automatic pressure regulator in the supply or discharge pipe.

All forms of balancing pistons may be considered as hydraulic pistons whose operation involves the use of a corresponding amount of motor water as well as certain mechanical losses. They exhibit, therefore, no advantage in respect to the losses in the turbine operation over a purely mechanical step bearing. However, their use increases the certainty of operation of the whole hydraulic installation as the pressure on the step bearing, and hence the friction and consequent heating, are reduced by their use.

(*d*) As the design of a step bearing for a horizontal shaft must naturally be very different from that for a vertical shaft, each will be considered separately.

Pressure Bearings for Horizontal Shafts. In most cases a pressure bearing is employed which, in its simplest form, consists of a collar on the revolving shaft bearing against a fixed surface. The bearing surface should be made as large as possible and must be well lubricated. Pressure or step bearings, so called, are often installed in connection with the bearings carrying the weight of the rotor and are lubricated in connection therewith.

The thrust bearing shown in Fig. 54 in connection with the double-babbitted bearing is not satisfactorily lubricated. The oil ring hangs on the bearing at its center and the passage of the oil to the vertical surfaces of the step bearing is interfered with by the movement of the shaft in its bearing. This difficulty could be corrected by the use of a sufficient number of oil rings or by properly grooving the bearing.

A better design than that above referred to is shown in Fig. 78, two oil rings and three collars being used.

In order to prevent as far as possible the heating of the oil, and consequently to improve the reliability of the operation, the oil chamber about the bearing inside the frame of the machine should be made as large as possible.

If, on account of high speed or great pressure, or both, the ordinary self-lubricating, ring-oiled bearing heats the oil to an undesirable temperature, special cooling devices must be installed. The bearing may be water jacketed, a cooling coil containing circulating water may be inserted in the oil chamber, or a separate circulating system may be installed, the oil after passing through the bearing passing to a tank containing a cooling coil and being pumped from thence to the bearings. Oil rings are, of course, unnecessary with such an arrangement.

The cast-iron bearing boxes should be lined with the best quality of white metal rather than with bronze bushings.

A bearing secured to the wheel case is usually more satisfactory and rigid than one having a separate support, as the latter is more apt to get out of alignment and thus cause heating. Where the bearing is secured to the wheel case it is desirable

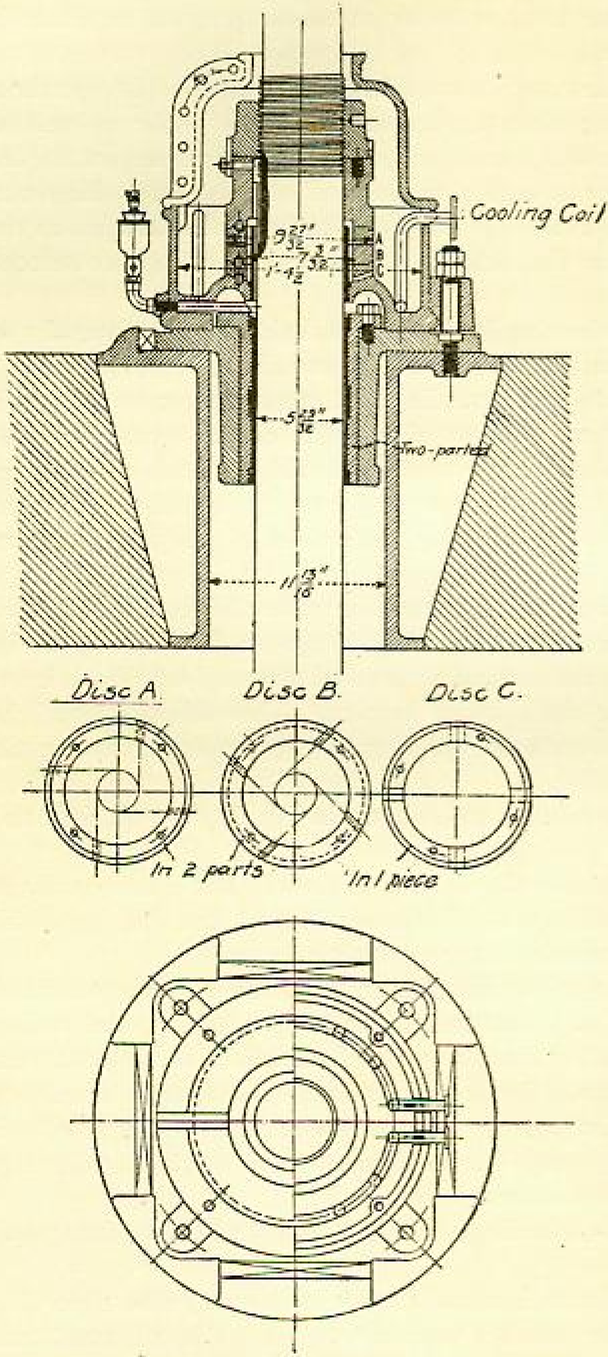


Fig. 76.—Submerged Oil Step-bearing for Vertical Shaft.

to prevent distortion of the latter by means of a strut placed against the bearing and acting in the line of the applied thrust.

In this country bearings are often lined with lignum vitæ, this form of construction being satisfactory where the bearing is submerged in water and therefore lubricated thereby.

Pressure or Step Bearings for Vertical Shafts. Step bearings for turbines with vertical shafts are nowadays almost always supported above the water. The following are the most usual forms of construction employed for such bearings:

The usual form of step bearing used in connection with a hollow shaft and a vertical supporting column is shown in Fig. 59, and is used by J. M. Voith.

An annular bearing revolving in a bath of oil with or without a cooling coil is a second form and is illustrated in Fig. 76, and is one of the most frequently employed and successful designs of step bearings. The bearing discs are made in two parts so as to be easily removed for repairs. Where conditions permit, however, it would be better to make each as a single casting. The best grade of charcoal iron should be used in their construction, and if made in parts the halves of each disc should be cast from the same ladle to insure equal hardness. The casting below the

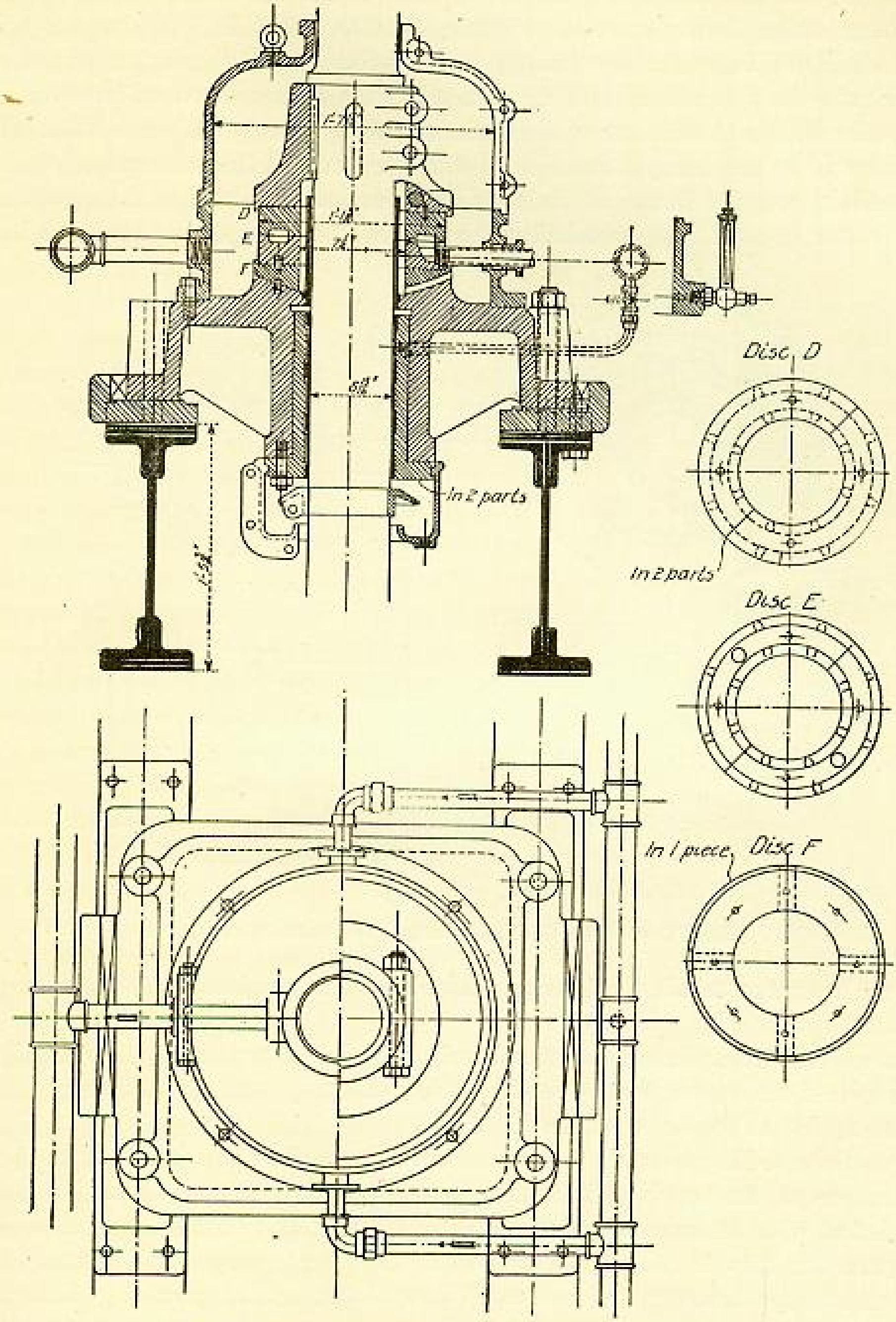


FIG. 77.—Oil Pressure Step-bearing for Vertical Shaft.

lower bearing disc is made in a spherical form to provide for inaccuracies in alignment of the same either during erection or subsequent thereto. In the illustration it also serves to hold the halves of the bearing disc together, an office also performed by the casting above the upper bearing disc. A better form of construction, however, is that in which the halves of the bearing discs are united by horizontal ream bolts. The oil grooves should be so arranged that the oil will flow outward from the space between the metal bushing and the inside of the bearing discs and then return through the large groove in the base of the spherical casting. Jack screws passing into the bedplate

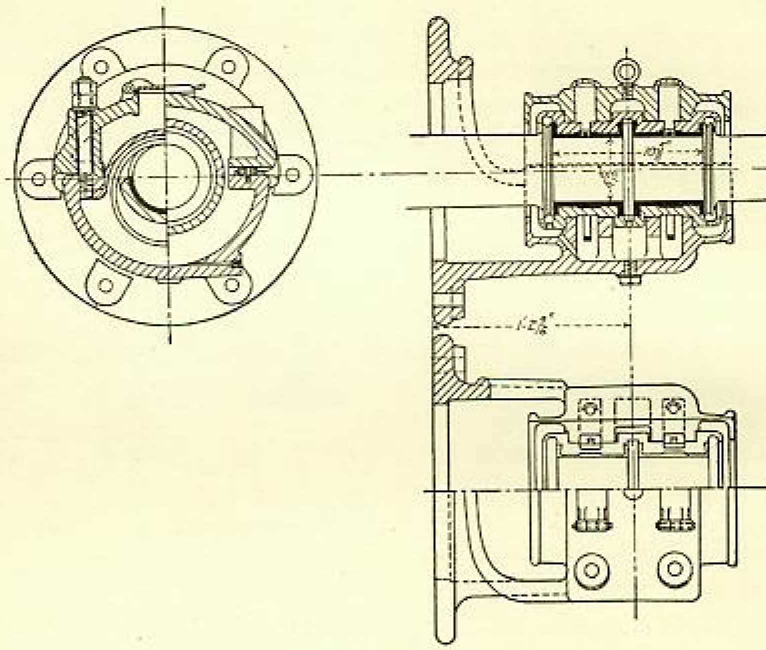


Fig. 78.—Double Oil Ring-bearing for Horizontal Shafts.

enable the operator to lift the bearing with the entire weight of the rotor suspended upon it.

The guide bearing below the step bearing should be separately lubricated, either by a drop-feed oil cup or from the circulating oiling system, the discharge oil being reclaimed or not as the designer elects.

If p = the specific pressure per square inch and v = the velocity in feet at the outer circumference of the bearing discs, then $pv \sim 127$ if no artificial oil-cooling device is provided, but with the use of cooling coils this figure may be materially increased.

An annular step bearing lubricated with oil under pressure is illustrated in Fig. 77, which shows a bearing designed by Victor Gelpke. The revolving discs contain one or more circular grooves into which the oil under pressure is conducted through grooves in the stationary discs, the connection to these grooves being provided at two opposite points where possible. The oil acting between the discs raises the moving parts until its upward pressure exactly balances the downward forces. In the bearing illustrated on Fig. 77 the discs are intended to separate .007 inch. No grooves are necessary in addition to the annular ones, but if used should not be carried too close to the outside edge of the discs or the oil consumption will be excessive. If properly arranged oil grooves are provided, a high-pressure oil bearing will operate as an ordinary step bearing if the oil supply is shut off, but, of course, with a diminished carrying capacity.

In order to supply oil to the guide bearing under the step bearing it is well to tap the high-pressure main, placing a pressure reducer on the supply pipe. Of course if a low-pressure circulating system is employed for all guide bearings in the plant this is not necessary. The oil used in the guide bearing is thrown by the shedder into the oil catcher and thence piped to filters to be used again.

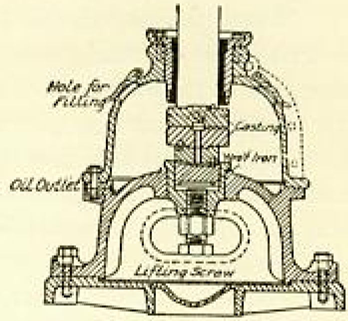


FIG. 79.—Submerged Oil Step-bearing at Lower End of Vertical Shaft.

Even though motives of economy may prevent the use of the spherical casting under the stationary disc, nevertheless, some means, such as jack screws, wedges, or shims, must be provided so that the bearing may be erected with great accuracy. The success of such a bearing depends largely upon proper erection and upon the finish of the bearing faces of the discs. Only the best workmanship can be used on this class of bearings.

The construction just discussed suggests an equally simple and practical method of supporting the turbine shaft, viz.: the connection of a disc with the lower end of the turbine shaft. Such an arrangement is as efficient, as accessible, as interchangeable and as firm as any of those previously mentioned, and should therefore be considered. Such a step bearing is shown in Fig. 79, the design having been made by Victor Gelpke.

Lignum vitæ step bearings have been used to a much greater extent in America than in Europe, and should receive attention on account of their cheapness. The lubrication of these bearings need not be discussed except to state that it is effected by water and that only clean water can be successfully employed. Their use is limited by the latter condition, but where the water is reasonably clean they wear well. An example of this form of bearing is shown in Fig. 57.

Guide Bearings—Tightness.—Various forms of guide bearings are shown by the several illustrations, but whether they consist of cast-iron boxes lined with white metal or of castings lined with adjustable wooden strips, whether they are used alone or in conjunction with step bearings, the designer must consider each case separately. In all cases special attention should be given to the lubrication, and where a gravity system is used the supply pipe to the bearing and especially the discharge pipe should be made of ample size. All piping and fittings should be of brass.

Hemp packing in stuffing boxes is very largely used abroad, and was originally employed in the Niagara turbines. Its use was, however, abandoned, as the shafst revolving with a speed of 250 rev. per min. were badly scored when the packing was drawn up sufficiently tight. Leather packings were substituted with good results. If hemp packing is used the usual clearance is 0.02 inch. Means should be provided for conducting away the water which squirts or oozes through the bushings. The same form of stuffing boxes are necessary for sliding rods wherever it is necessary to have them pass between points where different pressures exist. A groove or small chamber placed between the two halves of the stuffing box and fed with water from the penstock may be used to insure tightness against vacuum.

Water packing should be considered when it is necessary to prevent water from entering the operating room around the revolving shaft which passes from a wheel case free from pressure, but filled with spray—as in the case of Pelton wheels.

Leather packings, piston rings, or long, accurately turned pistons and cylinders have been found the best means of insuring tightness in hydraulic cylinders used in connection with the governors. When oil is used as the fluid the parts do not, of course, require to be as carefully protected against corrosion as when water is employed.

General.—In order to simplify the erection and dismantling of the individual parts of the machine, and to insure their accessibility there are a large number of construction details to be considered, many of which are often overlooked.

Among the most important are the installation of a sufficient number of lifting screws, jack bolts, and separating screws to force apart such parts as fit closely into each other; eye bolts for securing the crane hook to the heavier parts; handholes and manholes for inspection, etc.

Furthermore, every wheel case should have a cock at its highest point and a flushing valve at its lowest point, and a plane surface or instrument board should be provided for the speed indicator, pressure gauge, and vacuum gauge. All operating devices, whether for the turbine, governor, or head gate, should be readily accessible, and all indicating instruments should be located where they are within the convenient observation of the operator. In preparing the design care should be taken to so locate the several operating wheels that the entire operation of the turbine may be carried on from as compact a space as possible, or, in other words, that where the general design will permit a separate operating station be provided for each turbine.

PART II

TURBINE DESIGN

PART II

TURBINE DESIGN

GENERAL NOTATION

Q = volume of water in cubic feet per second.

H = effective head in feet, being the sum of the pressure and suction heads measured on the center line of the turbine outlet.

η = revolutions per minute.

$\omega = \frac{\pi\eta}{30}$ = angular velocity in feet per second.

ε = efficiency in percentage.

W = weight of one cubic foot of water.

$P_t = \frac{QWH}{550}$ = theoretical horse power.

$P = \varepsilon P_t$ = hydraulic horse power delivered to the turbine shaft.

$P_t - P = (1-\varepsilon)P_t$ = hydraulic operating losses in distributor, clearance, runner, and draft tube.

$P_e = \eta R_t$ = brake horse power.

$P - P_e = (\varepsilon - \eta)P_t$ = mechanical losses in bearings together with hydraulic losses due to use of balancing piston, cooling devices, etc.

h_E = head in feet from surface of head water to center line of entrance to distributor.

h_A = head in feet from surface of head water to center line of discharge from distributor.

$h_A - h_E$ = head in feet consumed in distributor.

h_o = head in feet from surface of head water to the center line of the outside of the runner at the entrance side.

h_e = head in feet from surface of head water to the center line of the cross-section of the entrance to the runner.

$h_e - h_A$ = head in feet consumed between distributor and runner.

h_a = head in feet from the surface of the head water to the center line of the cross-section of the runner discharge.

$h_a - h_e$ = head in feet consumed in runner.

$H - h_a$ = suction head in feet.

y = distance in feet from the surface of the tail water to any element of the true draft tube discharge section.

R_E, R_A, R_o, R_e, R_a = gauge pressures expressed as head in feet at the several cross-sections.

D_E, D_A, D_o, D_e, D_a = diameter in feet through the middle points of the several cross-sections.

r_E, r_A, r_o, r_e, r_a = radii in feet of same.

$\Delta_E, \Delta_A, \Delta_o, \Delta_e, \Delta_a$ = the minimum width in feet of the same.

l_E, l_A, l_o, etc. = the measured length of the cross-sections of a bucket.

b_E, b_A, b_o, etc. = the effective breadth of the cross-sections of a bucket.

$b_E\Delta_E, b_A\Delta_A, b_o\Delta_o$, etc. = the effective cross-section of a bucket.

$\Sigma b_E\Delta_E, \Sigma b_A\Delta_A$, etc. = the total effective cross-section of one runner.

c_O = velocity in headrace.

c_U = velocity in tailrace.

c_E, c_A, c_o, c_e, c_a = the absolute velocities in feet at the several points on the distributor and runner above referred to.

v_o, v_e, v_a = respectively the circumferential velocity in feet of the runner periphery, the middle point of the entrance cross-section, and the middle point of the discharge cross-section.

w_o, w_e, w_a = respectively the relative velocity in feet of a particle of water passing the runner rim, the middle of the cross-section at the entrance to the runner and the middle of the cross-section of the outlet from the runner.

$c_d'', c_d''', c_d'''' \ldots c_d^n$ = the velocity in feet in a draft tube in the direction of a normal thread of water.

α = the terminal angle of the distributor bucket = the angle formed by c_o with v_o and lying between 0° and 45°.

β = the entrance angle to the runner bucket = the angle formed by w_o and v_o when the entrance is free from impulse, and is greater than 90° for high-speed turbines and less than 90° for slow-speed turbines and action turbines.

γ = the terminal angle of the runner buckets = the angle made by w_a with v_a, and is always less than 90°.

δ = the actual angle of a jet of water at the point of discharge from the runner bucket = the angle between c_a and v_a. This angle is always varying about 90°, being less than 90° when the projections of c_a and v_a have the same direction as v_a.

m = number of turbines.

Z = number of vanes in the distributor.

z = number of vanes in the runner.

T = pitch of distributor vanes.
t = pitch of runner vanes.
s = thickness of distributor vanes.
s' = thickness of runner vanes.
r = distributor radii.
r' = runner radii.

DETERMINATION OF TURBINE CROSS-SECTIONS

Planes of Rotation and Normal Planes of Intersection

The lines 00, 11, 22, etc., shown in Fig. 82, are assumed to be the lines of inter-

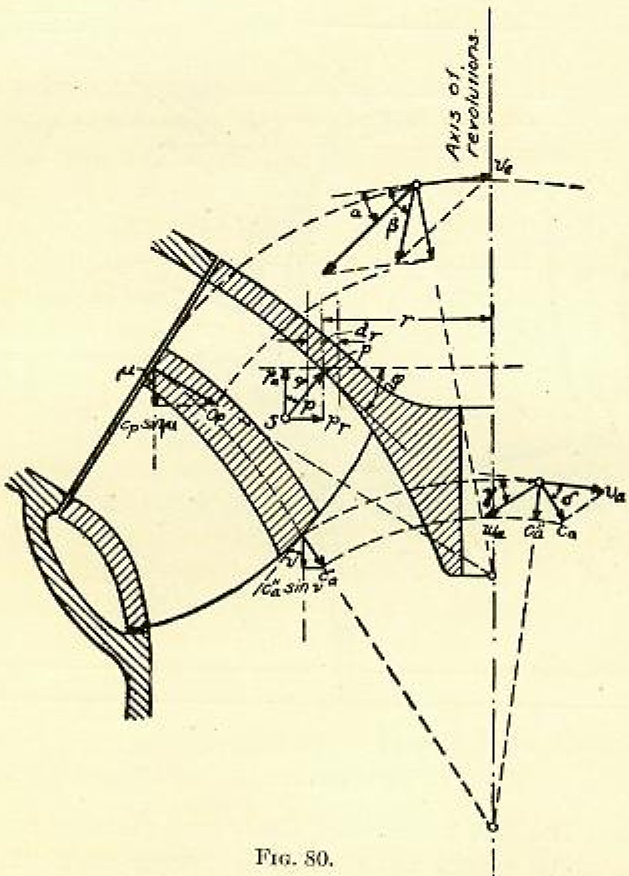


FIG. 80.

section of planes of rotation with the plane of the paper. The axis of rotation in this case, as in future studies, is understood to be the axis of the turbine. The limit 00, the outer plane of rotation and 44, the inner plane of rotation, are assumed for the present. Let us consider that a fixed volume of water, Q, passes between these two planes of rotation in such a manner that all at times each particle of water moves in

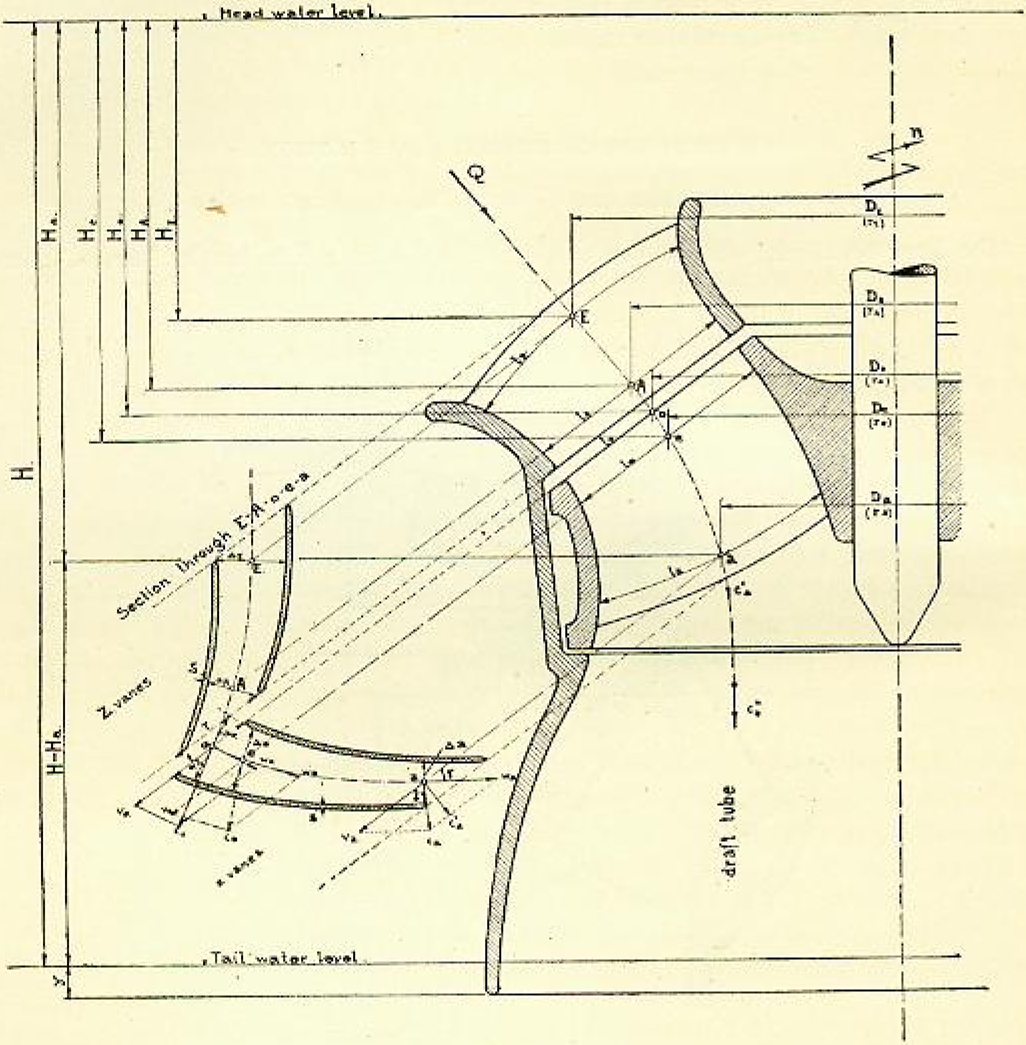


FIG. 81.

a revolving surface. The lines representing the several planes of rotation, 11, 22, etc., are so fixed that an equal volume of water passes between each. As shown in Fig. 82, such a volume $=\frac{Q}{4}$, and in general $=\frac{Q}{n}$. The intermediate planes of rotation, 01–01, 12–12, etc., are, in turn, so chosen that the volume of water flowing between 01–01 and 1–1 is equal to the volume between 1–1 and 12–12, etc. In the case shown in Fig. 82 such volume $=\frac{Q}{8}$, and in general $=\frac{Q}{2n}$. Now let the lines 0–0, 01–01, etc., represent at the same time the paths in which the particles of water travel, assuming that there are no buckets between the limits 0–0 and 4–4; i.e., that the turbine has only an inner and an outer crown.

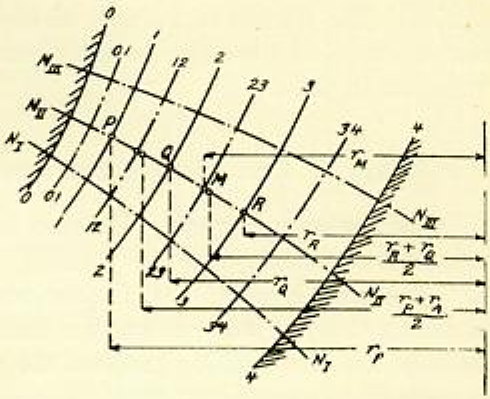


FIG. 82.

This form of a path will therefore be designated as a normal thread of water and must be assumed to be moving in a plane passing through the axis. The surface formed by the rotation of a normal thread of water around the axis of the turbine we will call a normal surface of revolution of the water thread, or in short a rotation surface.

The lines N_1N_1, N_2N_2, N_3N_3, etc., are lines of intersection with the plane of the paper of surfaces which are at right angles to the normal water threads. We therefore designate the said lines as normal lines of intersection, and their respective surfaces as normal surfaces of intersection.

The following calculation is based on the possible assumption that all particles of water which are in one and the same normal plane intersection have an equal velocity in the direction of the normal water threads, that is, at right angles to the surface of intersection. This assumption is an arbitrary one in some cases. From the assumption in connection with the above-mentioned condition that equal quantities of water pass between two adjacent planes of rotation we have the following from Fig. 82 for rotation planes 1–1, 2–2, and 3–3, which cut the normal line of intersection N_{11}–N_{11} at the points P, Q, and R.

$$RQ\cdot 2\pi\frac{r_R+r_Q}{2}=PQ\cdot 2\pi\frac{r_P+r_Q}{2},$$

or

$$RQ(r_R+r_Q)=PQ(r_P+r_Q).$$

If further we consider 23–23, the middle plane of rotation between 2–3 and 3–3, which cuts the normal line of intersection N_{11}–N_{11} at M, we have

$$MR=2\pi\frac{r_M+r_R}{2}=MQ2\pi\frac{r_M+r_Q}{2}=RQ\cdot 2\pi\frac{r_R+r_Q}{4},$$

or

$$MR(r_M+r_R)=MQ(r_M+r_Q)=RQ\frac{r_R+r_Q}{2}, \text{ etc.}$$

By introducing vanes between the limiting rims 0–0 and 4–4 certain strips will be cut from the normal intersection surface which will form a greater or less percentage of the normal surface, depending on the thickness of the vanes, the number of the vanes, and the angle of the vanes. The summation of those strips subtracted from the normal surface of intersection will give the effective normal surface of intersection. The volume of water Q divided by this effective normal surface will give the component of the velocity in the direction of the normal threads of water when the vanes are in place.

The Bucket, the True Plane of Intersection and the Effective Cross-Section of the Bucket

The water passageway inclosed between two adjacent vanes and the circular crowns is commonly called a "bucket"; in the case of the stationary circle of vanes, a "distributor bucket," and for the revolving parts a "runner bucket." The course of the particles of water through a bucket of a distributor or runner can be indicated by lines, and the resulting paths of the water should be shown as actual water threads in contrast to the normal ones. The series of planes at right angles to the true water threads and also at right angles to the bounding walls of the bucket (the vanes and crowns) will be called the true surface of intersection.

Taking a point in the true plane of intersection, an indefinite number of lines of intersection can be drawn through it in the true surface of intersection, from vane to vane; of all these lines one is the shortest and it has the property of being at right angles to the bounding vanes at its point of intersection with them. We call the length of this shortest line of intersection the minimum width of the bucket, or the bucket clearance, for a given point inside the bucket. We may then consider the shortest lines of intersection drawn throughout the whole surface of a given true surface of intersection inside a bucket and the middle of the resulting clearance lines connected by a line which extends from one rotation plane to another and in all from crown to crown. The length of such a median line is commonly called the breadth of the bucket.

The superficial area of a true plane of intersection measured between two vanes and the crowns is considered the effective cross-section of the bucket.

By dividing the cross-section of the bucket by the breadth of the bucket we obtain the average clearance in the bucket cross-section. If on a given true plane of intersection we select the shortest intersection lines having equal lengths and divide each into two equal parts at a common point, we obtain a point which may be designated as the center point of the bucket cross-section. If we do the same with all the remaining cross-sections of the same bucket we obtain a row of center points which may be connected by a line representing the average thread of the water flowing through that bucket.

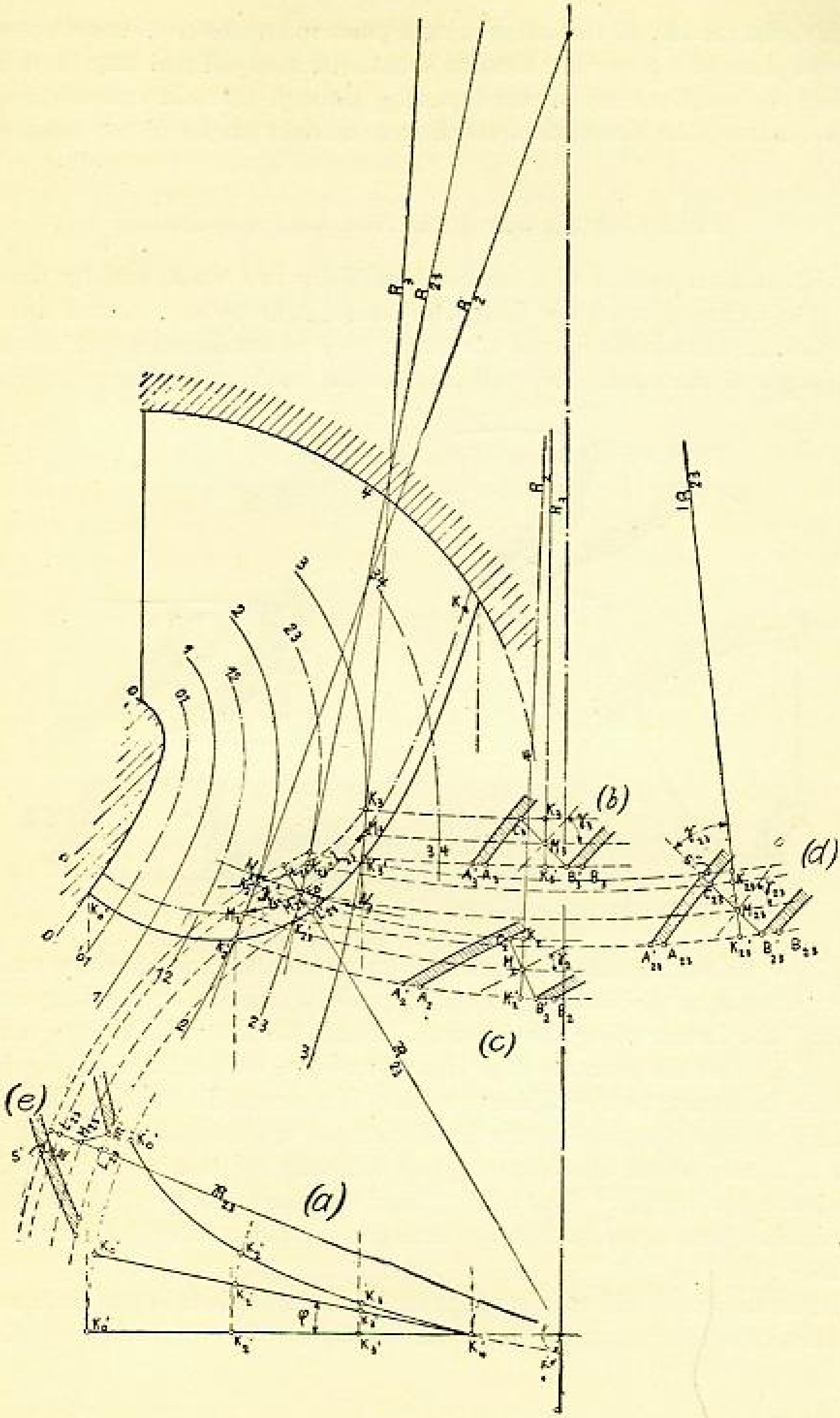
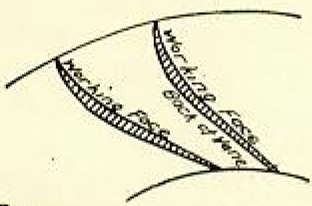


FIG. 83.

Finally, the velocity of the water at each point in an effective bucket cross-section or in a true plane of intersection is to be considered as equal and may be determined by dividing the total volume of water passing through the said cross-section by the area of the same. The direction of the flow is at right angles to the effective cross-section.

Fixed Cross-Sections of the Distributor and Runner

Fig. 83 shows a section of a bucket limited by two vanes and by the circular crowns. The inclosing vanes are limited in their extent by two curved lines, drawn at will, which, as regards the flow of the water, may be considered as the entrance and discharge edges of the vanes. We will consider the vanes as having a material thick-



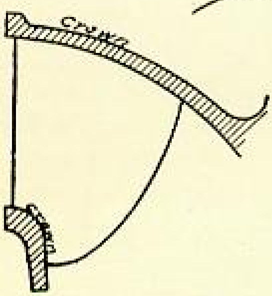


Fig. 84.

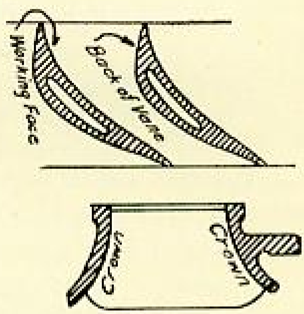


Fig. 85.

ness and have shown the same as measured on a clearance line on every vane of the distributor as well as the runner. We distinguish between the "working face" and the "back," the former being the concave face and the latter the convex face.

From points on the entrance edge of a given vane, let the shortest effective intersection lines be drawn to the adjacent vane. The summation of these lines forms a plane which we will designate as the fixed entrance cross-section of the bucket. By a similar operation at the discharge edge of the vane we obtain the fixed discharge cross-section of the bucket. (The word "fixed" is employed for the reason that in a finished turbine these cross-sections are available for measurement and hence may be determined or fixed.)

By a summation of said cross-sections for all of the buckets in the circle we obtain, in the case of the distributor, the fixed cross-section at the entrance to the distributor and the fixed cross-section at the distributor discharge; and as regards the runner, the fixed cross-section at the runner entrance and the fixed cross-section of the discharge from the runner.

In general the plane of the fixed cross-section does not coincide with an effective surface of intersection or with an effective cross-section of the bucket. In all these cases it is necessary to calculate the effective cross-section of the bucket from the fixed cross-section and from the remaining dimensions of the bucket.

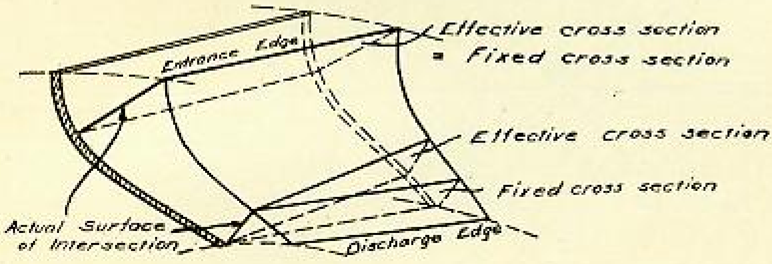


Fig. 86.

It is only in special cases that the true water-thread falls at right angles to the fixed cross-section, in which case the effective cross-section coincides with the fixed cross-section and the calculation then becomes particularly simple. This occurs, for example, at the entrance to a Francis turbine. (See Fig. 86.)

The Determination of the Fixed Cross-Section of a Bucket, the Discharge Edge Being Radial

In order to represent a bucket by a drawing it is necessary to make use of spherical projection; that is, in making the drawing, we consider that every point lying outside of the plane of the paper is projected on it by revolving the same about the axis of the turbine. (For the following see Fig. 83.)

We have given the limits of the bucket, that is, the inner plane of rotation 4–4 and the outer plane of rotation 0–0. Between these let the planes of rotation 1–1, 2–2, and 3–3 be drawn in the manner hereinbefore described, as well as the planes of rotation of the average normal water threads 01–01, 12–12, 23–23, and 34–34. We unite and project together between the vane limits 0–0 and 4–4 the two vanes which together with the lines 0–0 and 4–4 bound the bucket. On account of the spherical projection employed the ends of the vanes appear in the drawing as a curved line, K_0'–K_4'. This line, which is called the discharge edge, is so designed as to provide a proper path for the water from the planes of rotation 0–0 to 4–4, and in plan may be either a radial line, a straight line not radial, or a curve. (See Fig. 83.) For the present we will consider that the discharge edge is a radial line in plan.

Further, let us select the line 23–23 as the middle line of the part of the bucket

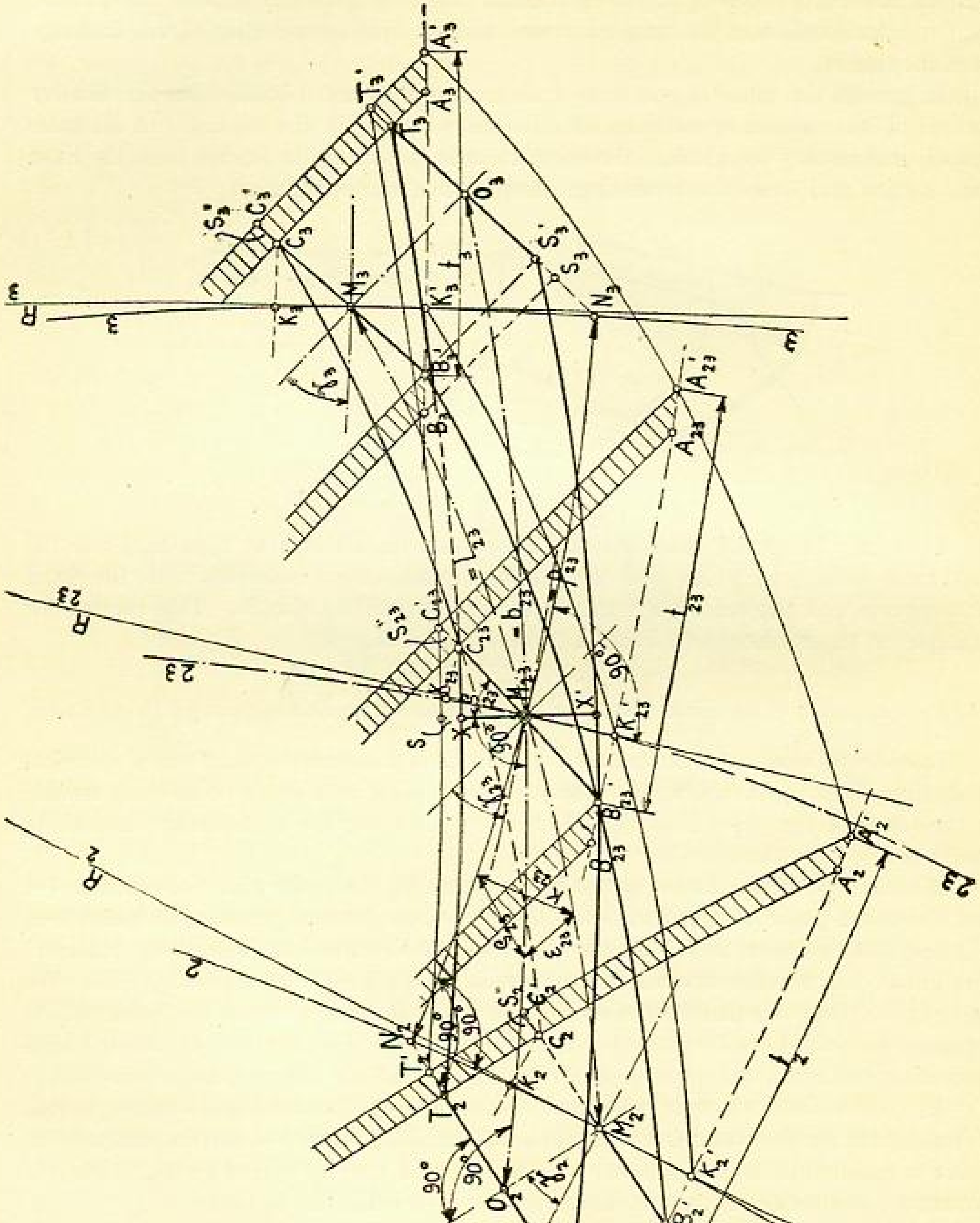


FIG. 87.

lying between 2–2 and 3–3, and concern ourselves only with the determination of the cross-section at the point of discharge, because this gives the general solution and can, without difficulty, be hereafter used for the determination of the cross-section at the entrance to the runner. The portion of the bucket intersected by the plane of rotation 23–23 is shown in section and projection in Fig. 83. It will be observed that the point K'_{23} represents the projection of the four points, A'_{23}, A_{23}, B'_{23}, and B_{23}, in which the rotating plane 23–23 cuts the two vanes at the discharge edge. It will be noted that from the point B'_{23} in the working plane in this section a line of intersection in the cross-section of the bucket may be drawn to the back of the adjacent vane which it hits at the point C_{23}. By bisecting the line B'_{23}, C_{23} we obtain the point M_{23}, which may be considered as the middle point of the cross-section to be determined.

The tangent to the path of an element of water passing through M_{23} makes with the circumferential tangent the discharge angle γ_{23}, whose calculation is fundamental, and which appears in the discharge diagram for the point M_{23} as the angle of v_a with w_a. On the other hand the thickness of the vane s'' is not shown in this projection in its true size, at least so long as the line of rotation is at right angles to the discharge edge. If in the sketch the point M_{23} be projected back to Fig. 83, we may lay out at its intersection with the plane of rotation 23–23 the tangential conical surface whose generating line is R_{23}. We may then with sufficient accuracy consider the plane of rotation and the conical plane identical for points which are not far distant, especially in Fig. 83, and such points may be regarded as lying on the conical plane.

By repeating this process in the case of the planes of rotation 2–2 and 3–3, we obtain the middle points M_2 and M_3. The connecting line $M_2M_{23}M_3$ can now, as a close approximation, be considered as lying in a diametrical plane, so long as the discharge edge, as already presupposed, lies in such a plane. The conical surfaces tangent at the points M_2 and M_3 have the generating lines R_2 and R_3, and are shown in projection in Fig. 83. If we connect the points C_2 and C_3, which have been determined in the same manner as C_{23}, by a curve, we obtain (see Fig. 87) the section $B'_2C_2C_3B'_3$, having the center line $M_2M_{23}M_3$, which is the fixed cross-section of the partial bucket, i.e., the cross-section as it is found in the finished wheel. It is important to know the width of this cross-section, that is, the length of the line $MM_{23}M'$ (see Fig. 83*e*), which lies in the plane $B'_2C_2C_3B'_3$, and is at right angles to $M_2M_{23}M_3$. In order to determine this width we lay out through M_{23} a normal surface of intersection which cuts the planes of rotation 2–2 and 3–3 at the points N_2 and N_3, respectively.

For convenience we designate the line $M_2M_{23}M_3$ as l_{23}, the line $N_2M_{23}N_3$ as p_{23}, and the angle which l_{23} makes with p_{23} as k_{23}. It will be noted that these three values may be obtained from the drawing and that further by reference to a sufficient number of partial turbines we have

$$p_{23} = l_{23} \cos k_{23}. \quad \quad (1)$$

Now let a conical surface be passed through M_{23}, the projection of which is R_{23}, at right angles to l_{23}. In this conical section, whose projection is shown in Fig. 83*e*, the desired line $MM_{23}M'$ must appear as the clearance line of the bucket, the thickness of the vane in this section being S'.

The relation between the dimensions is now readily obtained from Fig. 83*b* and 83*e* if one compares the circular projection of $B'_{23}C_{23}$, (i.e., $K'_{23}K_{23}$) with the circular projection of $M'M$ (i.e., $L'_{23}L_{23}$). We then have

$$K'_{23}K_{23} \cos k_{23} = L'_{23}L_{23}. \qquad (2)$$

It is clear that we must also have

$$C_{23}K_{23} + B'_{23}K'_{23} = ML_{23} + M'L'_{23}. \qquad (3)$$

From which we obtain

$$M'M = \sqrt{\overline{L'_{23}L_{23}}^2 + (ML_{23} + M'L'_{23})^2}. \qquad (4)$$

With the number of vanes z and the distance of the point M_{23} from the axis $= r_{23}$ we have:

$$K'_{23}K_{23} = \left(\frac{2\pi r_{23}}{z} \sin \gamma_{23} - S''\right) \cos \gamma_{23}. \qquad (5)$$

and

$$C_{23}K_{23} + B'_{23}K'_{23} = \left(\frac{2\pi r_{23}}{z} \sin \gamma_{23} - S''\right) \sin \gamma_{23}. \qquad (6)$$

and these values, in connection with formulas (2) and (3), substituted in (4) give

$$M'M = \left(\frac{2\pi r_{23}}{z} \sin \gamma_{23} - S''\right) \sqrt{\cos^2 \gamma_{23} \cos^2 k_{23} + \sin^2 \gamma_{23}}, \qquad (7)$$

also, the thickness of the vane S', measured in the direction of $M'M$, can be determined as follows from the thickness S'', measured in the direction $B'_{23}C_{23}$:

$$S' = S'' \sqrt{\cos^2 \gamma_{23} \cos^2 k_{23} + \sin^2 \gamma_{23}}. \qquad (8)$$

It is also possible, as we shall see further on, to express the value of S'' in terms of s, the true thickness of the vane measured at right angles to the face of the vane, as follows:

$$S'' = s \frac{\sqrt{1 - \sin^2 k_{23} \sin^2 \gamma_{23}}}{\cos k_{23}}. \qquad (9)$$

Eq. (7) may therefore be written

$$M'M=\left(\frac{2\pi r_{23}}{z}\sin\gamma_{23}-s\frac{\sqrt{1-\sin^2 k_{23}\sin^2\gamma_{23}}}{\cos k_{23}}\right)\sqrt{\cos^2\gamma_{23}\cos^2 k_{23}+\sin^2\gamma_{23}}. \quad . \quad (10)$$

In this equation all of the values are known, and if no further condition is imposed it may be at once used for each remaining plane of rotation by substituting for r_{23}, k_{23}, and γ_{23} their corresponding values.

The area of the fixed cross-section of the partial bucket may be obtained by multiplying $M'M$ by l_{23}, or in general, the fixed cross-section =

$$l\left(\frac{2\pi r}{z}\sin\gamma-s\frac{\sqrt{1-\sin^2 k\sin^2\gamma}}{\cos k}\right)\sqrt{\cos^2\gamma\cos^2 k+\sin^2\gamma}. \quad . \quad . \quad . \quad . \quad (11)$$

Determination of the Effective Cross-section, the Discharge Edge Being Radial
(See Fig. 87)

We again assume that the discharge edge of the vanes and, approximately, the line $M_2M_{23}M_3$, lies in a radial plane. A plane is now passed through the normal intersection line $N_2M_{23}N_3$, which is at right angles to the surface of the vanes. This intersects the vanes on the plane of rotation 2–2 in the points T'_2, T_2, S'_2, and S_2, and on the plane of rotation 3–3 in the points T'_3, T_3, S'_3, and S_3. Then the surface $T_2S'_2S'_3T_3$ represents the effective cross-section of the partial bucket. In this lies the line $O_2M_{23}O_3$, the center line of the effective cross-section. At right angles to this is the clearance line $X'X$, which at the same time is in general the shortest distance, vane surface to vane surface, at the point M_{23}. The line $O_2M_{23}O_3$ may be designated for convenience by b_{23} and $X'X$ by Δ_{23}, so that the effective cross-section will be $b_{23}\Delta_{23}$.

The lines $M_2M_{23}M_3$, $N_2M_{23}N_3$, and $O_2M_{23}O_3$ form with each other at the point M_{23} the angles k_{23}, c_{23}, and e_{23}. Of these three angles only k_{23} is given and may be taken from the drawing. We may, however, deduce the following from the right-angled triangles $M_2N_2M_{23}$, $N_2O_2M_{23}$, and $M_2O_2M_{23}$:

$$\sin e_{23}=\sin k_{23}\sin\gamma_{23}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

and

$$\cos c_{23}=\frac{\cos k_{23}}{\sqrt{1-\sin^2 k_{23}\sin^2\gamma_{23}}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

In these equations γ_{23} represents, as before, the angle which the direction of the flowing water makes with the circumferential tangent at the point M_{23}, that is, the angle which w_a makes with v_a at the point M_{23}. Then

$$b_{23}=M_2M_{23}M_3\cos e_{23}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

the length of the center line, and

$$b_{23} = l_{23}\sqrt{1-\sin^2 k_{23}\sin^2 \gamma_{23}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

from which $l_{23} = M_2M_{23}M_3$, as we know.

Finally, taking into consideration the given thickness of the vane s we have determined $\varDelta_{23}$, which is measured in the direction of the shortest distance from vane surface to vane surface.

$$\varDelta_{23} = \frac{2\pi r_{23}}{z}\sin \gamma_{23}\cos c_{23} - s, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

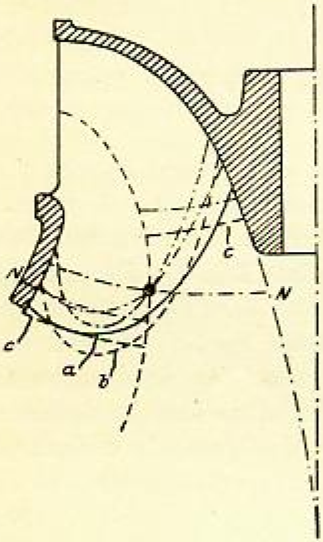


Fig. 88.

in which as above r_{23} represents the distance from the point M_{23} to the axis of revolution and z = the number of the buckets. With the above value of $\cos c_{23}$ (see 13) we have finally

$$\varDelta_{23} = \frac{2\pi r_{23}}{z}\sin \gamma_{23}\frac{\cos k_{23}}{\sqrt{1-\sin^2 k_{23}\sin^2 \gamma_{23}}} - s, \quad . \quad . \quad (17)$$

and the cross-section of any portion of the turbine (omitting the indices)

$$b\varDelta = l\frac{2\pi r}{z}\sin \gamma \cos k - ls\sqrt{1-\sin^2 k \sin^2 \gamma}.^* \quad . \quad (18)$$

Now observing that $l\cos k$ represents the length of the normal line of intersection which may be measured in the plane of the drawing ($N_2M_{23}N_2$, Fig. 83), and omitting from consideration the thickness of the vanes, we have the following simple definition:

The effective cross-section of any portion of a bucket is equal to the length of the normal line of intersection, drawn through the center point of the cross-section, multiplied by the width of the cross-section, which lies on the plane of rotation of the mean water thread.†

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* In order to find the fixed cross-section from the effective cross-section it is necessary to first multiply by the factor $\sqrt{\cos^2 \gamma + \frac{\sin^2 \gamma}{\cos^2 k}}$. This factor = 1 when $k = 0$; that is, when the edge of the vane is at right angles to the water thread.

† In the drawing (Fig. 83) this width appears on the conical section which is tangent to the plane of rotation at M_{23}, and whose generating line has the length R_{23}.

Therefore it is practically immaterial in the determination of the size of the cross-section and hence of the quantity of water flowing through it whether the limiting edges of the bucket vanes are chosen as the full line (a), the dotted line (b) or the zigzag line (c) Fig. 88, provided only that the mean normal line of intersection NN of the total discharge cross-section is the same in all cases.

The Determination of the Effective Cross-Section When the Discharge Edge is Any Convenient Curve

We will now abandon the assumption that the discharge edges of the vanes are radial in plan and will suppose that the discharge edge as well as the line passing through the middle point of the discharge cross-section is shown in plan as a convenient curve whose several elements 00, 11, 22, etc., we consider as revolved by circular projection into the reference plane of the elevation until the center points of said elements, i.e., M_{01}, M_{12}, M_{23}, etc., lie in said reference plane, a method easy of execution. In this manner we obtain the drawing shown in Fig. 89, in which the reference plane of the elevation is indicated as a radial line. The angle which each element of a line passing through a middle point makes with said radial line may in general be designated as K_g' in contrast to the projection in elevation where it may be called K_a. The length of the elements of the lines passing through the middle points may by analogy be represented by l_g and l_a. The bucket being subdivided into a sufficient number of parts, we will consider that the end points of the line l_{23} lie respectively in plan at a distance $\frac{l_{g23}}{2}\sin k_{g23}$ over and under the reference plane, such distance being actually measured on the conical surfaces whose generating lines have respectively the lengths R_2 and R_3. For example, if measured on the conical surface whose generating line is R_2 the distance $\frac{l_{g23}}{2}\sin k_{g23}$ subtends an angle T_2, we may with sufficient accuracy write

$$\sin T \sim \tan T_2 \sim \text{arc } T_2 = \frac{\frac{l_{g23}}{2}\sin k_{g23}}{R_2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

a value which is practically zero where $\frac{l_{g23}}{2}\sin k_{g23}$ is sufficiently small and R_2 very large. The corresponding angle for the conical surface which is tangent to the plane of rotation 33 at M_3, the middle point of the cross-section, may be called T_3.

If we again select the partial bucket 22–33 having the center line 23–23 and draw the bucket cross-section lying between the planes of rotation 2–2 and 3–3 with the discharge angle γ we obtain the perspective sketch shown in Fig. 90.

A plane passing through the normal line of intersection $N_2M_{23}N_{23}$ (whose length is $p\cos k_{a23}$) may again be laid out at right angles to the center line of the bucket cross-section; this is possible because this normal line of intersection $N_2M_{23}N_{23}$ lies at right angles to the intersection lines of the several tangential conical planes on which the bucket cross-section is drawn and with which we may assume the path of flowing water will probably coincide. This plane also intersects the middle lines of the bucket

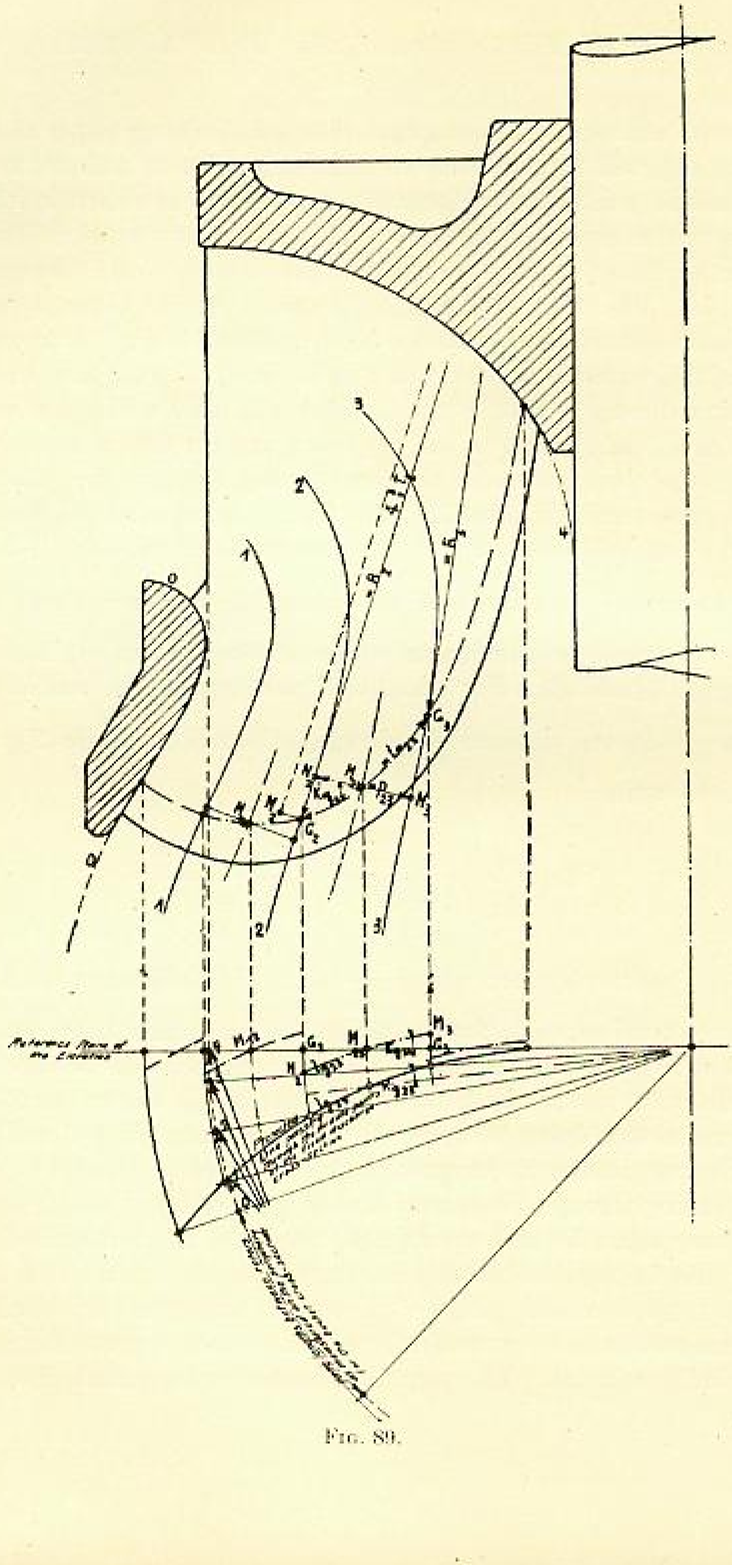

FIG. 89.

cross-section lying between the planes of a rotation 2–2 and 3–3 at the points O_2 and O_3 on the assumption that the line $O_2M_{23}O_3$ is likewise at right angles to said middle lines. We have again designated the length of the effective cross-section as $O_2M_{23}O_3 = b_{23}$, and its width as $\varDelta_{23}$. Indicating as C_{23} the angle with which $O_2M_3O_2$ intersects

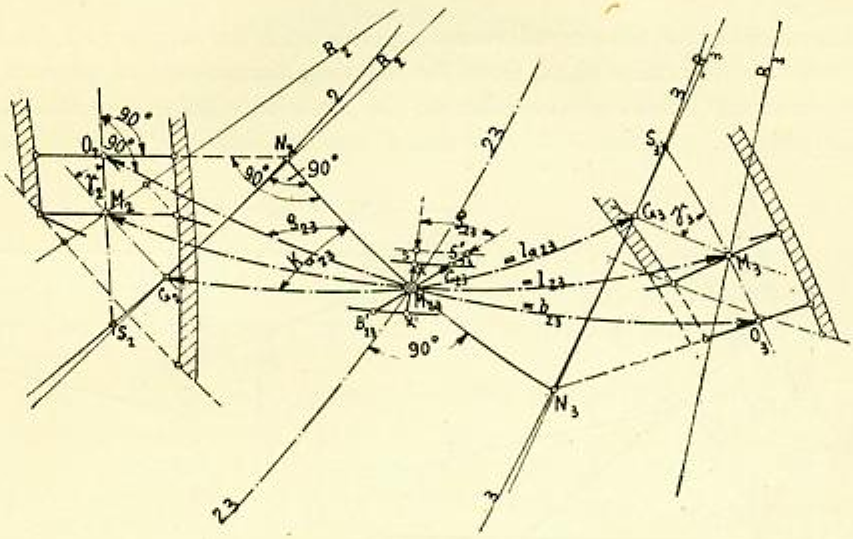

FIG. 90.

the line $N_2M_{23}N_3$ at the point M_{23} we may then obtain in conformity with the previous calculations and designations

$$b_{23} = l_{a23} \cos k_{a23} \times \frac{1}{\cos c_{23}}, \qquad (20)$$

and the value of the minimum clearance line is then

$$\varDelta_{23} = \frac{2\pi r_{23}}{z} \sin \gamma_{23} \cos c_{23} - s, \qquad (21)$$

in which s denotes the vane thickness measured in the direction of $\varDelta_{23}$.

The area of the effective cross-section is then

$$b_{23}\varDelta_{23} = \frac{2\pi r_{23}}{z} l_{a23} \sin \gamma_{23} \cos k_{a23} - s\frac{l_{a23} \cos k_{a23}}{\cos c_{23}}. \qquad (22)$$

As the thickness of the vanes decreases the area approaches the product of the normal line of intersection, $l \cos k$, multiplied by the width $\frac{2\pi r}{z} \sin \gamma$, measured on the planes of rotation, or in the planes of the flow of the water. Further, we have the following relation between the vanes thickness S'' measured on the plane of rotation and the true thickness of the vane measured in the direction Δ.

$$S''=\frac{s}{\cos c}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

The determination of the several values depends upon the angle c and the function $\cos c$. In order to determine those from the drawing dimensions we show in Fig. 91 a development of the bucket cross-section on the plane of rotation 2–2 and in Fig. 92 that on the plane of rotation 3–3. We again consider that the plane of rotation is

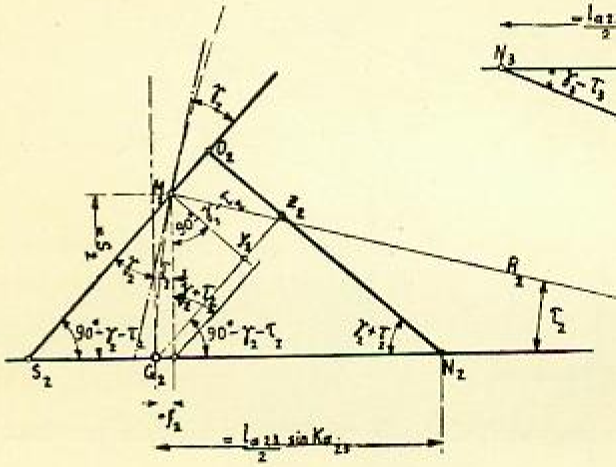

Fig. 91.

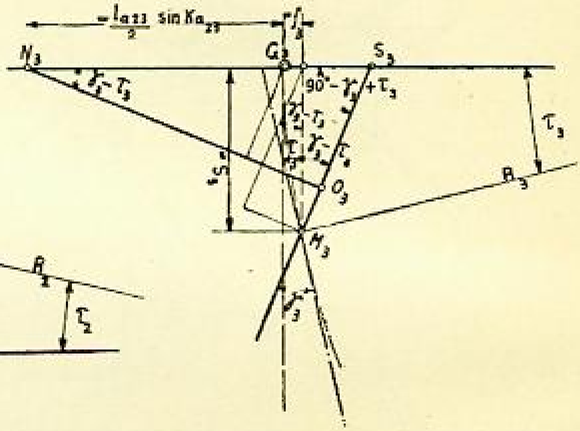

Fig. 92.

identical with the tangential conical surface in the case of the section which we are considering.

Denoting by S_2 the half-chord corresponding to the half-arc M_2G_2 we have

$$S_2=R_2 \sin T_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

and if f_2 represents the corresponding versine

$$f_2=R_2-R_2 \cos T_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

Through M_2 let a line be drawn parallel to O_2N_2; through G_2 a line be drawn at right angles to the last-named line, intersecting it at y_2 and intersecting O_2N_2 at Z_2. Then the angle $O_2N_2G_2=\gamma_2+T_2$ and we have (see Fig. 91)

$$N_2O_2=\frac{l_{a23}}{2} \sin k_{a23} \cos (\gamma_2+T_2)+S_2 \sin (\gamma_2+T_2)-f_2 \cos (\gamma_2+T_2), \quad . \quad (26)$$

and, by analogy, Fig. 92

$$N_3O_3 = \frac{l_{a23}}{2} \sin k_{a23} \cos (\gamma_3 - T_3) + S_3 \sin (\gamma_3 - T_3) + f_3 \cos (\gamma_3 - T_3). \quad (27)$$

The length of $N_2M_{23}N_3$ is given as before by the equation

$$N_2M_{23}N_3 = p_{23} = l_{a23} \cos k_{a23}. \quad (28)$$

Finally the angle c is determined by the equation

$$\tan c_{23} = \frac{N_2O_2 + N_3O_3}{N_2M_{23}N_3}. \quad (29)$$

The length of the lines N_2O_2, N_3O_3, and $N_2M_{23}N_3$ may be calculated by the use of Eqs. (19), (25), (26), (27), and (28), or may be determined graphically. In this manner the problem is readily solved.

We may now introduce certain permissible simplifications; in particular that in which the generating lines R_2 and R_3 are infinite, or that in which the subdivision of the bucket into parts is carried so far that T_2 and T_3 are very small, the versed sines f_2 and f_3 are zero and the sum of the sines S_2 and S_3 equals $l_{g23} \sin k_{g23}$, or

$$S_2 + S_3 = l_{g23} \sin k_{g23}. \quad (30)$$

With these assumptions we have

$$N_2O_2 + N_3O_3 = l_{a23} \sin k_{a23} \frac{\cos \gamma_2 + \cos \gamma_3}{2} + l_{g23} \sin k_{g23} \frac{\sin \gamma_2 + \sin \gamma_3}{2}, \quad (31)$$

substituting

$$\frac{\cos \gamma_2 + \cos \gamma_3}{2} = \cos \gamma_{23}, \quad (32)$$

$$\frac{\sin \gamma_2 + \sin \gamma_3}{2} = \sin \gamma_{23}, \quad (33)$$

and introducing the value of $N_2M_{23}N_3$ in Eq. (29) we have

$$\tan c_{23} = \frac{l_{a23} \sin k_{a23} \cos \gamma_{23} + l_{g23} \sin k_{g23} \sin \gamma_{23}}{l_{a23} \cos k_{a23}}, \quad (34)$$

omitting the indices we have in general

$$\tan c = \frac{l_a \sin k_a \cos \gamma + l_g \sin k_g \sin \gamma}{l_a \cos k_a}, \quad (35)$$

or
$$\tan c = \tan k_a \cos \gamma + \frac{l_g \sin k_g}{l_a \cos k_a} \sin \gamma. \quad \ldots \quad (36)$$

Moreover we have

$$\cos c = \frac{1}{\sqrt{1+\tan^2 c}} \quad \text{and} \quad \frac{1}{\cos c} = \sqrt{1+\tan^2 c}. \quad \ldots \quad (37)$$

Special Case I.—A case met with in design, but which is of little practical importance, is that in which k_g in plan has such a value that the center line of the fixed cross-section (shown as $M_2M_{23}M_3$ on Fig. 90) coincides with the center line of

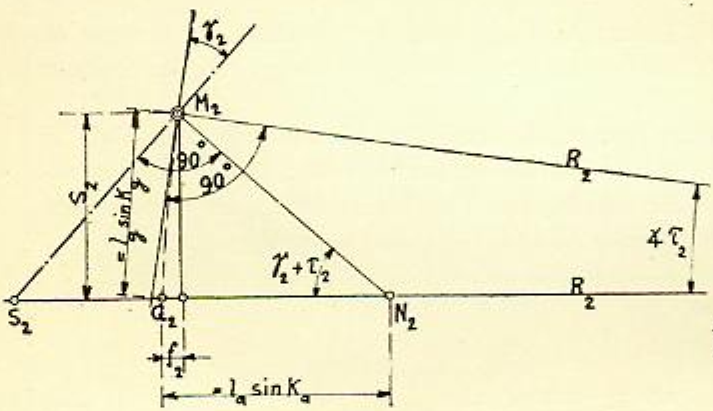

Fig. 93.

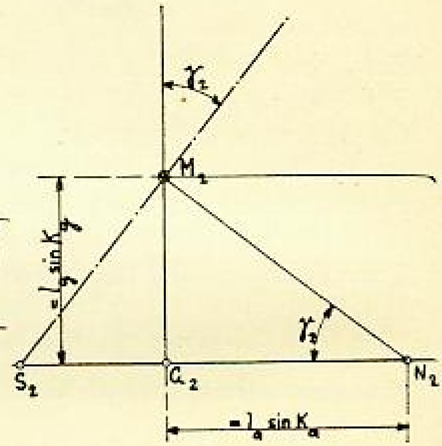

Fig. 94.

the effective cross-section, $O_2M_{23}O_3$. In this case, T_2 and T_3 being assumed as zero, the values which determine the projection in plan must satisfy the condition that

$$l_g \sin k_g = l_a \sin k_a \tan \gamma. \quad \ldots \quad (38)$$

and the value of $\tan c$ then takes the form

$$\tan c = \frac{\tan k_a}{\cos \gamma}, \quad \ldots \quad (39)$$

and we have

$$\cos c = \frac{\cos k_a}{\sqrt{1+\sin^2 k_a \tan^2 \gamma}}, \quad \ldots \quad (40)$$

and

$$M_2M_{23}M_3 = O_2M_{23}O_3 = l = b = l_a\sqrt{1+\sin^2 k_a \tan^2 \gamma} = \sqrt{l_a^2 + l_g^2 \sin^2 k_g} \quad . \quad (41)$$

and the minimum clearance line in this special case becomes

$$\Delta=\frac{2\pi r}{z}\frac{\cos k_a \sin\gamma}{\sqrt{1+\sin^2 k_a \tan^2\gamma}}-s. \quad . \quad . \quad . \quad . \quad (42)$$

Special Case II.—A further special case arises in which the discharge edge of the bucket as well as the center line of the discharge cross-section lies in a plane which does not pass through the axis of the turbine, but is tangent to a cylinder around the same which has the radius $r_n \sin\phi$. In the discussion of this case we will call the angle made by this discharge plane with the reference plane of the elevation ϕ and will let r_n = the radius of cylinder, also of the innermost bucket cross-section. With this designation we find that for a given distance r_{23} the value $l_{g23}\sin k_{g23}$ cannot be chosen at will, but must have the value

$$\frac{r_n \sin\phi}{r_{23}}\frac{l_{g23}}{2},$$

or

$$\sin k_{g23}=\frac{r_n \sin\phi}{r_{23}}. \quad . \quad . \quad (43)$$

But l_{g23} may no longer be chosen arbitrarily, but as l_{a23} makes the angle ϕ_{23} with a horizontal line drawn through M_{23}, we have

$$l_{g23}=\frac{l_{a23}\cos\phi_{23}}{\cos k_{g23}}. \quad . \quad . \quad (44)$$

Therefore, in the special case we have in general

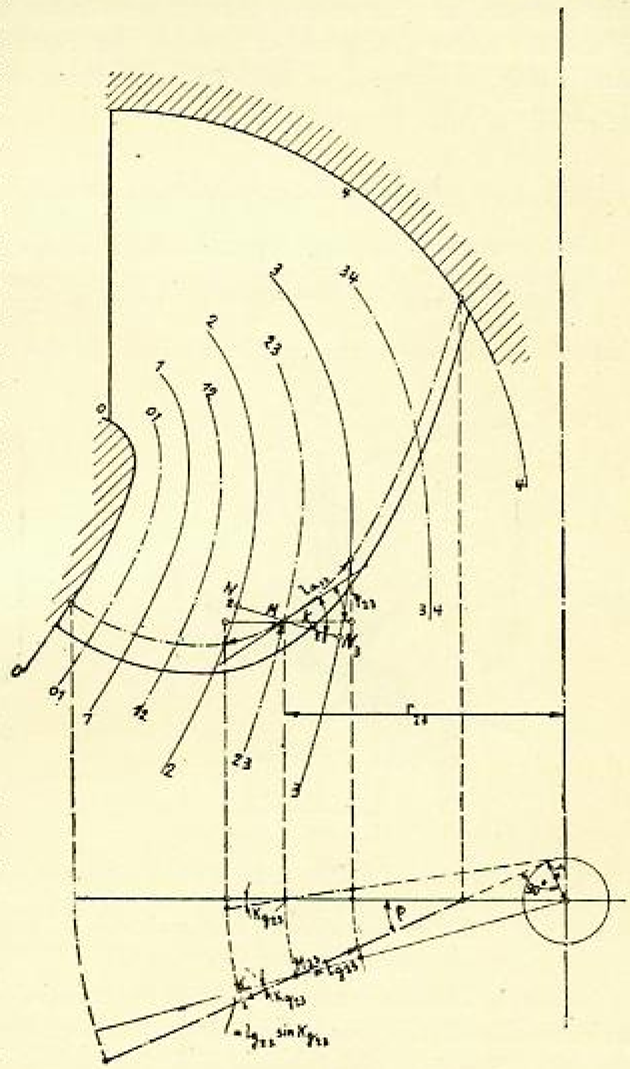

Fig. 95.

$$\tan c=\tan k_a\cos\gamma+\frac{\cos\phi}{\cos k_a}\tan k_g\sin\gamma. \quad . \quad . \quad . \quad . \quad (45)$$

In which, as stated above, k_g may not be chosen arbitrarily but has its value fixed by the equation

$$\sin k_g \frac{r_n \sin \phi}{r}.$$

It will be observed that in the expression for $\tan c$ the second term of the right-hand side decreases as r increases; that is, the further we go from the axis the smaller is ϕ, and the nearer ϕ approaches to zero. As soon as the second term becomes so small that it may be dropped we have the same value of $\tan c$ as in the general case where the discharge edge is in a radial plane.

Graphical Representation of the Effective Cross-Section

There are two ways to represent by a drawing the effective cross-section.

First. We may lay out the half value of $\frac{2\pi r}{z} \sin \gamma$ on both sides of the normal lines of intersection which are drawn through the center of the cross-section. In this

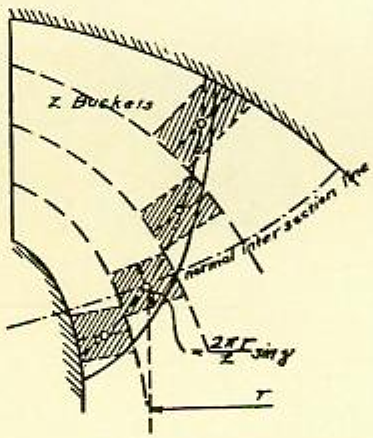

Fig. 96.

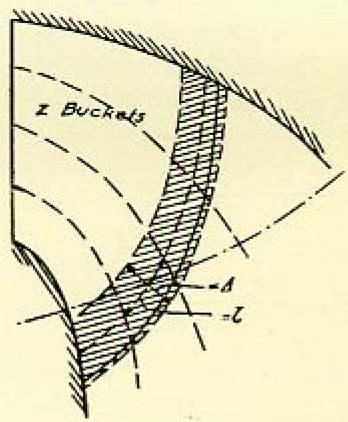

Fig. 97.

manner we obtain the shaded areas shown in Fig. 96, and the summation of these results in the effective cross-section.

Second. The true value of $\varDelta$ may be calculated and one-half of the same plotted on each side of the center line of the cross-section as shown in the shaded area on Fig. 97. The area thus plotted will then represent $\Sigma l \cdot \varDelta$. In order to obtain from its value that of the effective cross-section $= \Sigma l \cos e \cdot \varDelta$, it is necessary to multiply every fixed partial value of l by its corresponding $\cos e$, determined according to (14), and to make a summation of such products, or instead of plotting the values of $\varDelta$ we may lay out directly the values of $\Delta \cos e$ in the manner above described. Then the shaded area will represent at once the effective cross-section.

The variation between the values $\Sigma l \cos e \cdot \Delta$ and $\Sigma l \cdot \Delta$ becomes larger with an increase in the angle between the discharge edge and the normal line of intersection, and of the discharge angle γ.

In practice the mistake is often made of considering $\Sigma l \cdot \Delta$ (i.e., the shaded area in Fig. 97), as the true effective cross-section of the bucket, although, as a matter of fact, the effective cross-section may be 20 per cent smaller than $\Sigma l \cdot \Delta$, particularly with the high-speed type of turbines.

The Determination of the Clearance and Vane Thickness in a Selected Cross-Section of a Bucket

In Fig. 98 Δ and s are shown respectively as the clearance and vane thickness

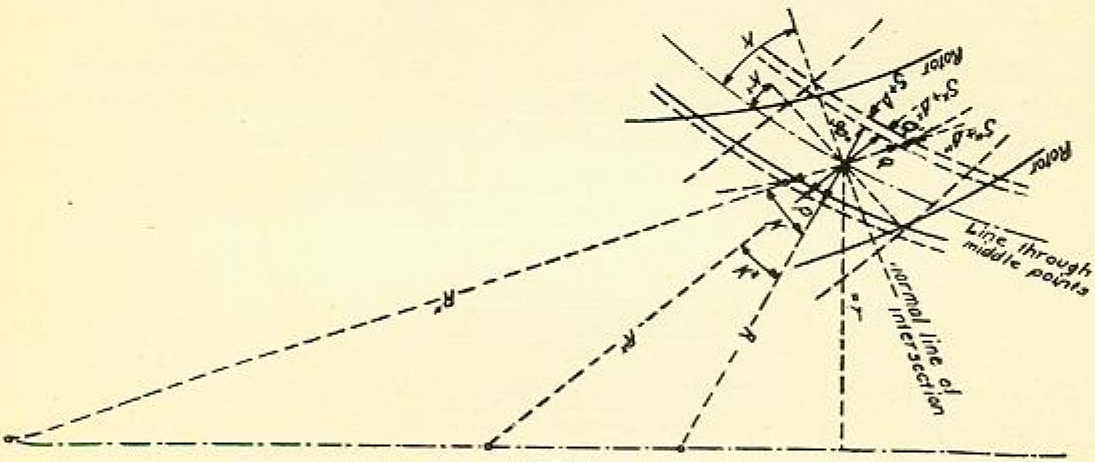

FIG. 98.

in an effective cross-section, the same being measured on a conical surface having the generating line R which is at right angles to a line through the center point of the cross section in the plane of the drawing.

From Eq. (17) we have

$$\Delta + s = \frac{2\pi r}{z} \sin \gamma \frac{\cos k}{\sqrt{1 - \sin^2 k \sin^2 \gamma}}.$$

By further reference to Fig. 98 it will be seen that Δ'' and s'' represent the clearance and vane thickness of a cross-section lying in the middle water thread, or what is the same thing, on a conical surface having the generating line R'', in which case

$$\Delta''+s''=\frac{2\pi r}{z}\sin\gamma.$$

We may now consider the last value of $\Delta''+s''$ to have been calculated by means of Eqs. (17) and (21), and we will have

$$\Delta''+s''=\frac{2\pi r}{z}\sin\gamma=\frac{(\Delta+s)\sqrt{1-\sin^2 k\sin^2\gamma}}{\cos k}, \quad \ldots \ldots \quad (46)$$

or

$$\Delta''=\Delta\frac{\sqrt{1+\sin^2 k\sin^2\gamma}}{\cos k}=\frac{\Delta}{\cos c} \quad \ldots \ldots \quad (47)$$

and

$$s''=s\frac{\sqrt{1+\sin^2 k\sin^2\gamma}}{\cos k}=\frac{s}{\cos c}. \quad \ldots \ldots \quad (48)$$

Let us now draw at will a bucket cross-section on a conical surface whose generating line is R^x, the angle between R^x and R being k^x. Then by a calculation similar to the above, using the proper symbols for the new section, we have

$$\Delta^x+s^x=\frac{(\Delta+s)\sqrt{1-\sin^2 k^x\sin^2\gamma}}{\cos k^x}, \quad \ldots \ldots \quad (49)$$

or

$$\Delta^x=\frac{\Delta\sqrt{1-\sin^2 k^x\sin^2\gamma}}{\cos k^x}, \quad \ldots \ldots \quad (50)$$

and

$$s^x=\frac{s\sqrt{1-\sin^2 k^x\sin^2\gamma}}{\cos k^x}. \quad \ldots \ldots \quad (51)$$

These equations enable us to find the clearance and vane thickness for any selected bucket cross-section, the effective width and vane thickness being given or calculated and k^x being found by scale from the drawing.

It is very simple to determine Δ^x and s^x when k^x is small; that is, when the selected bucket section is but slightly inclined to the section which contains the effective width Δ and the vane thickness s. In such a case $\sqrt{1-\sin^2 k^x\sin^2\gamma}$ may be considered to be equivalent to 1, and we obtain the approximate values

$$\Delta^x=\frac{\Delta}{\cos k^x}, \quad \ldots \ldots \quad (52)$$

and
$$s^x = \frac{s}{\cos k^x}. \quad \dots \dots \dots \dots \quad (53)$$

These values may be obtained directly from the drawing from the lengths of the lines PQ and QQ' respectively.

In connection with drawings showing the vane surfaces we will use with advantage this approximate method, which is accurate enough for all practical purposes.

Numerical Examples

Problem I. Given:

$$r = 1.3583 \text{ ft.};$$
$$z = 15;$$
$$\gamma = 31^\circ\ 00';$$
$$s = \tfrac{5}{16} \text{ in.} = 0.02604 \text{ ft. (sheet-steel vanes)};$$
$$t = \frac{2\pi r}{z} = 0.5690 \text{ ft.};$$
$$(a)\ k = 58^\circ; \quad (b)\ k = 30^\circ; \quad (c)\ k = 0.$$

To determine:

(1) The clearance in the fixed cross-section of the partial bucket $= M'M$.

(2) The clearance in the effective cross-section of the partial bucket $= X'X$.

1*a* and 2*a*. Using Eqs. (8), (9), and (10) with $k = 58^\circ$, we have

$$M'M = 0.5690 \times 0.515\sqrt{0.2063 + 0.2652} - 0.02604 \times \frac{\sqrt{1 - 0.7191 \times 0.2652}}{0.53}$$
$$\sqrt{0.2063 + 0.2652} = 0.2007 - 0.0304 = 0.1703 \text{ ft.} = 2\tfrac{1}{16} \text{ in.}$$

In the above 0.0304 is the vane thickness measured in the direction $M'M$; whereas by Eq. (17)

$$\Delta = X'X = 0.5690 \times 0.515 \frac{0.530}{\sqrt{1 - 0.7191 \times 0.2652}} - 0.02604$$
$$= 0.1726 - 0.02604 = 0.1466 \text{ ft.} = 1\tfrac{13}{16} \text{ in.}$$

In this case 0.02604 ft. or $\tfrac{5}{16}$ in. is the true thickness of the vane measured in the direction $X'X$.

It can be seen by this numerical example that with large values of k the clearance and vane thickness in the fixed cross-section differ materially from those of the effective cross-section, as was stated above.

1*b* and 2*b*. Using the same equations as before and making $k = 30^\circ$

$$M'M = 0.5690 \times 0.515\sqrt{0.5508+0.2652} - 0.02604 \times \frac{\sqrt{1-0.25\times 0.2652}}{0.866}\sqrt{0.5508+0.2652}$$

$$= 0.2646 - 0.02613 = 0.2385 \text{ ft.} = 2\tfrac{7}{8} \text{ in.};$$

$$X'X = 0.5690 \times 0.515 \frac{0.866}{\sqrt{1-0.25\times 0.2652}} - 0.02604;$$

$$= 0.2626 - 0.02604 = 0.2365 \text{ ft.} = 2\tfrac{27}{32} \text{ in.}$$

It is apparent from this example that with an angle of 30° between the discharge edge and the normal intersection line the clearance in the measurable and in the effective cross-section is practically the same.

1*c* and 2*c*. In this case where the edge of the vane coincides with the normal intersection line, that is when $k=0$, we have

$$M'M = 0.5690 \times 0.515 - 0.02604 \frac{\sqrt{1-0}}{1}\sqrt{l}$$

$$= 0.2930 - 0.02604 = 0.2670 \text{ ft.} = 3\tfrac{3}{16} \text{ in.};$$

$$X'X = 0.5690 \times 0.515 \frac{1}{\sqrt{1-0}} - 0.02604$$

$$= 0.2930 - 0.02604 = 0.2670 \text{ ft.} = 3\tfrac{3}{16} \text{ in.}$$

In this case, then, the fixed and effective cross-sections are identical.

From a consideration of the above examples it will be seen that if the angle γ remains the same the effective clearance decreases about as rapidly as the obliquity of the normal water threads to the discharge edge increases.

Problem II. In the effective cross-section of a partial bucket there is given

Clearance $\Delta = 0.1837$ ft. $= 2\tfrac{3}{16}$ in.

Thickness of vane $= 0.02604$ ft. $= \tfrac{5}{16}$ in.

Discharge angle γ, measured on the middle plane of rotation $= 31°$.

(*a*) If a section is now drawn through the middle point, making an angle of 15° with the normal intersection surface, what will be the clearance Δ^x and the vane thickness s^x in the new section?

From Eqs. (50) and (51) we have

$$\Delta^x = \frac{0.1837\sqrt{1-\sin^2 15° \sin^2 31°}}{\cos 15°} = 0.991\,\frac{0.1837}{\cos 15°} = 0.1885 \text{ ft.} = 2\tfrac{1}{4} \text{ in.};$$

$$s^x = \frac{0.02604\sqrt{1-\sin^2 15° \sin^2 31°}}{\cos 15°} = 0.991\,\frac{0.02604}{\cos 15°} = 0.02672 \text{ ft.} = \tfrac{21}{64} \text{ in.}$$

By the approximate Eqs. (52) and (53) we obtain

$$\Delta^x = \frac{0.1837}{\cos 15^\circ} = 1.000\,\frac{0.1837}{\cos 15^\circ} = 0.1902 \text{ ft.} = 2\tfrac{9}{32} \text{ in.};$$

$$s^x = \frac{0.02604}{\cos 15^\circ} = 0.02696 \text{ ft.}$$

In this example the error due to the use of the approximate instead of the correct formula would be less than 1 per cent, an error permissible in practice.

(*b*) The same problem when $k=30^\circ$. Substituting the new value we have as before

$$\Delta^x = \frac{0.1837\sqrt{1-\sin^2 30^\circ \sin^2 31^\circ}}{\cos 30^\circ} = 0.2048 \text{ ft.} = 2\tfrac{29}{64} \text{ in.};$$

$$s^x = \frac{0.02604\sqrt{1-\sin^2 30^\circ \sin^2 31^\circ}}{\cos 30^\circ} = 0.0290 \text{ ft};$$

and the approximate values

$$\Delta^x = \frac{0.1837}{\cos 30^\circ} = 0.2121 \text{ ft.} = 2\tfrac{35}{64} \text{ in.};$$

$$s^x = \frac{0.02604}{\cos 30^\circ} = 0.03007 \text{ ft.}$$

The error accompanying the use of the approximate equations for this example is therefore about 3 per cent, and they cannot be recommended for such oblique sections.

Problem III. There being given

$$r = 1.3583 \text{ ft.};\quad z = 15;\quad s = 0.02604;\quad k = 58^\circ;$$

and the length of the projected line drawn between the adjacent planes of rotation through the center point of the cross-section

$$= l = M_2M_{23}M_3 = 0.7150 \text{ ft.}$$

to determine the effective and the fixed cross-section of the partial bucket.

(*a*) *To Determine the Area of the Fixed Cross-Section.*

The length of the center line of this cross-section as drawn in the plane of the paper is 0.7150 feet, and the clearance in the fixed cross-section has been found in

Problem I to be $M'M=0.1703$ feet. Therefore the area of the fixed cross-section $=0.7150\times0.1703=0.1218$ sq.ft.

(*b*) *To Determine the Area of the Effective Cross-Section.*

The length of the center line is obtained by Eq. (15) $b=0.7150\sqrt{1-0.7191\times0.2652}$ $=0.7150\times0.900=0.6435$ feet, and the clearance we have found by Problem I to be 0.1466 feet. Accordingly the value of the area of the effective cross-section becomes

$$b\Delta=0.0435\times0.1466=0.0943 \text{ sq.ft.}$$

The same result can be obtained by first determining the length of the normal intersection line $N_2M_{23}N_3$ as $l\cos k=0.3789$, and then calculating the clearance of the bucket cross-section which lies on the rotation plane of the middle water thread. Using our previous designations this is obtained by the equation

$$B'_{23}C_{23}=\frac{2\pi r}{z}\sin\gamma-s''=0.5690\times0.515-0.0443=0.2487.$$

In which s'' the vane thickness, measured in the direction of $B'_{23}C_{23}$ is determined by Eq. (48) as follows:

$$s''=s\frac{\sqrt{1-\sin^2\gamma\sin^2 k}}{\cos k}=0.02604\frac{\sqrt{1-0.1907}}{0.53}=0.0443.$$

The desired area of the effective cross-section of the partial bucket may then be obtained by the product of the last-determined quantities.

$$N_2M_{23}N_3\times B'_{23}C_{23}=0.3789\times0.2487=0.0942 \text{ sq.ft.}$$

Note.—Often in the finished turbine there is measured the length of the center line $M_2M_{23}M_3$ of the fixed cross-section as well as the shortest distance $X'X$ which belongs to the effective cross-section, and the product of these two dimensions is considered the area of the partial bucket. Assuming the conditions of the previous case a cross-section so measured would have an area $=0.7150\times0.1466=0.1048$ square feet, an area intermediate between that of the fixed and of the effective cross-section, differing by 13 per cent from the former and 11 per cent from the latter. Such a method is evidently erroneous.

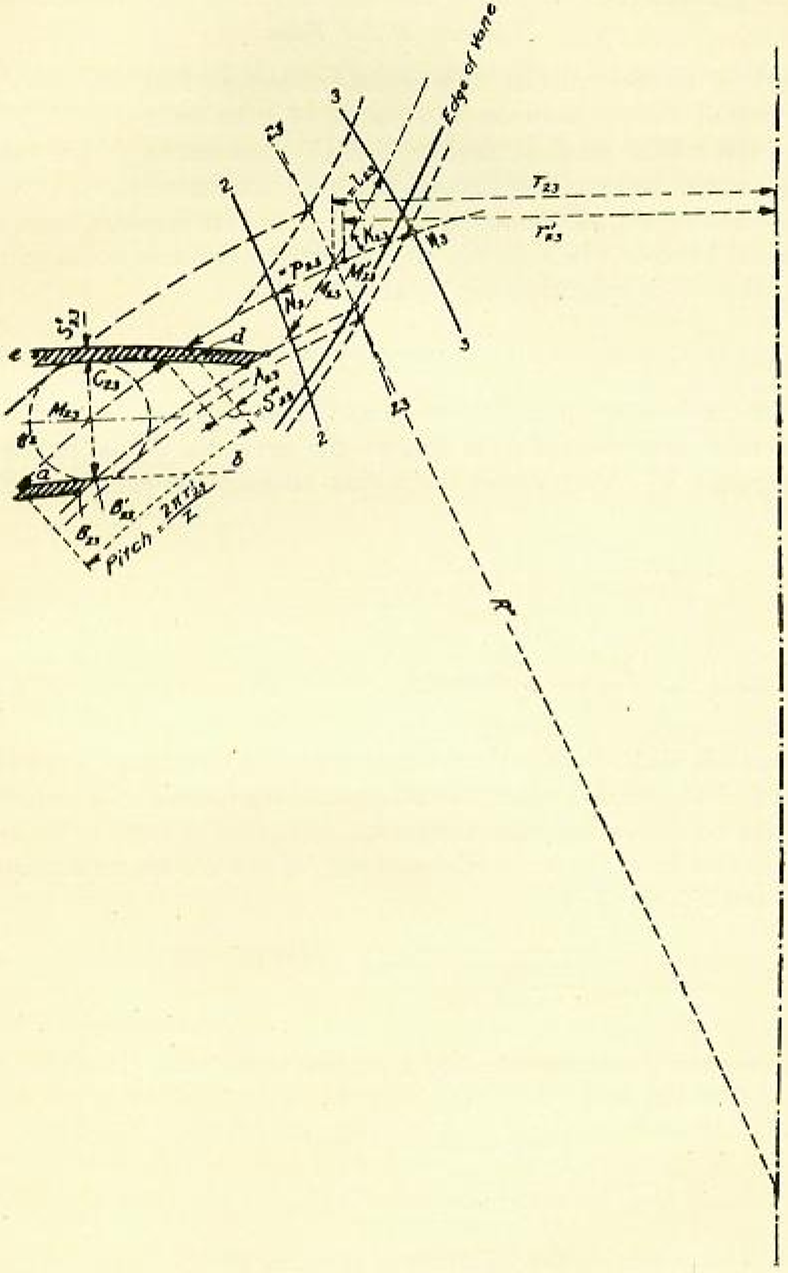

FIG. 99.

The Area of the Normal Surface of Intersection Taking into Consideration the Thickness of the Vane

The length of the normal line of intersection through M_{23} will be calculated as p_{23} for the partial bucket between the planes of rotation 2–2 and 3–3, that is, $N_2M_3N_3=p_{23}$ the middle point of the line $N_2M_3N_3$ does not in general coincide with the point M_{23}, whose distance from the axis we have designated as r_{23}. We will then designate the middle point of the line p_{23} as M'_{23}, and its distance from the axis as r'_{23}. For partial buckets which do not lie too near to the axis we may in practical cases consider that r'_{23} is approximately equal to r_{23}, or

$$r'_{23} \sim r_{23}. \qquad (54)$$

With these designations and those before employed and without regard to the vane thickness, we may then determine as follows the area N_T of the normal plane of intersection through M_{23} lying within the bucket between the planes of rotation 2–2 and 3–3.

$$N_{23T}=\frac{2\pi r'_{23}}{z}p_{23}=\frac{2\pi r'_{23}}{z}l_{23}\cos k_{23}. \qquad (55)$$

or in general

$$\text{Area } N_T=\frac{2\pi r'}{z}p=\frac{2\pi r'}{z}l\cos k. \qquad (56)$$

The section thus calculated does not accurately represent the area of the water flowing in the direction of the normal water thread, especially because of the reduction in the latter due to the corresponding vane thickness.

By Eq. (43) we have the vane thickness s_{23}'' of the bucket section lying on the planes of rotation 2–2 and 3–3.

$$s_{23}''=s_{23}\frac{\sqrt{1-\sin^2 k_{23}\sin^2\gamma_{23}}}{\cos k_{23}}.$$

The resulting vane thickness measured on the normal intersection plane passing through M_{23} is then (See Fig. 99.)

$$s_{23}'''=s_{23}\frac{\sqrt{1-\sin^2 k_{23}\sin^2\gamma_{23}}}{\sin\gamma_{23}\cos k_{23}}. \qquad (57)$$

Taking into consideration the thickness of the vane we may then obtain as follows the true area N_e of the normal intersection plane of the partial bucket above discussed.

$$N_{23e}=\left(\frac{2\pi r'_{23}}{z}-s_{23}'''\right)p_{23}=\left(\frac{2\pi r'_{23}}{z}-s_{23}'''\right)l_{23}\cos k_{23}. \quad . \quad . \quad . \quad . \qquad (58)$$

By substituting the value of s''' as obtained from Eq. (57) we then have in general for a partial bucket

$$N_e=\left(\frac{2\pi r'}{z}-s\frac{\sqrt{1-\sin^2 k \sin^2 \gamma}}{\sin \gamma \cos k}\right)p, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (59)$$

$$N_e=\left(\frac{2\pi r'}{z}-s\frac{\sqrt{1-\sin^2 k \sin^2 \gamma}}{\sin \gamma \cos k}\right)l \cos k. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (60)$$

Having determined the area of the normal intersection plane of the partial bucket diminished by the vane thickness we can now, from the volume of the water ΔQ, obtain the velocity c_a'' with which the water flows in the direction of the normal water thread from the equation

$$c_a''=\frac{\Delta Q}{N_e}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (61)$$

This value is necessary for drawing the discharge diagrams.

Numerical Examples

Problem I. The bucket being definitely determined by the drawing, to obtain the area of the normal surface of intersection when

(*a*) The thickness of the vanes is neglected.

(*b*) Sheet steel vanes $\frac{5}{16}$ inch thick are used.

(*c*) Cast-steel vanes $\frac{5}{8}$ inch thick are used.

We obtain from the drawing

$$r'=0.3517 \text{ ft.}; \quad \gamma=31°; \quad k=58°; \quad z=15; \quad l=0.7150 \text{ ft.}$$

Then $p=l\cos k=0.3789$ ft.

$$\frac{2\pi r}{z}=0.5663 \text{ ft.}$$

For (*a*) $\qquad N_{\gamma'}=0.3789\times 0.5663=0.2146$ sq.ft.

For (*b*)

$$N_e=0.2146-0.02604\frac{\sqrt{1-0.7191\times 0.2652}}{0.515\times 0.53}\cdot 0.3789$$

$$=0.2146-0.03256=0.1820 \text{ sq.ft.}$$

For (*c*) $\qquad N_e=0.2146-0.06512=0.1495$ sq.ft.

It therefore appears that sheet steel vanes of usual thickness decrease the area of the normal intersection plane by 15.2 per cent, and cast vanes having double the thickness of the sheet-steel vanes decrease it by 30.4 per cent, so that in case (*b*) only 84.8 per cent, and in case (*c*) only 69.6 per cent of the theoretical area remains.

If we assume, for example, that the discharge velocity in the direction of the normal water threads is equal and that the losses are equal, then in case (*a*) there will be discharged Q cubic feet, in case (*b*) 0.847 Q cubic feet, and in case (*c*) 0.694 Q cubic feet.

This example teaches that in wheels using a large quantity of water sheet steel vanes should certainly be used and that they should be made as thin as the requirements for strength will permit.

It will also be seen that a small wheel built like a large wheel will have a lower hydraulic efficiency, because in the former case the thickness of the vanes will be larger relatively to the area of the bucket cross-section.

Problem II. This problem is similar to Problem I except that $k=0$ instead of 58°, i.e., the discharge edge of the bucket lies in a normal surface of intersection.

For case (*a*)	$N_T=0.2146$ sq.ft.
For case (*b*)	$N_e=0.2146-0.0192=0.1954$ sq.ft.
For case (*c*)	$N_e=0.2146-0.0384=0.1762$ sq.ft.

From this example in connection with that preceding it, it will be observed that the harmful effect of the vane thickness on the effective area of the normal plane of intersection decreases with a decrease in the angle k; i.e., as the edge of the vane approaches coincidence with the normal intersection line. In the extreme case where $k=0$ the said area will be decreased by the vane thickness only about 9 per cent for case (*b*), and about 18 per cent for case (*c*). From this point of view the discharge edge of the vanes, which may be chosen at will, should be so designed that the normal water threads will make therewith an angle which shall be as near 90° as possible. This consideration, however, must not prejudice a proper form of the vane.

THE RELATIONS OF PRESSURE AND VELOCITY IN TURBINES

Impact of Water Against a Movable Plane.

Assume that in Fig. 100 a particle of water strikes a moving plane at the point A in such a manner that its velocity may be represented in direction and amount by c, while the direction and amount of the velocity of the moving plane is indicated by v, the angle between v and c being denoted by α and the angle between v and the direction of the plane by β'. It then follows: As the particle of water has the velocity c and must acquire the velocity v, it then has a tendency to flow in the direction and with the amount of velocity represented by w_0. But the plane will not allow the particle of water to move forward in the direction w_0, and hence its velocity must be divided into two components, one parallel to the plane and one at right angles thereto represented respectively by w_0' and c_n. The velocity c_n is called the impact component and is considered as lost. The corresponding loss in pressure head from eddies is $\frac{c_n^2}{2g}$. The remaining velocities which may be used are, therefore w_0', and v. Without taking into consideration the friction along the plane, another loss occurs when the particle of water flows freely from the plane at B, namely, the velocity head which corresponds

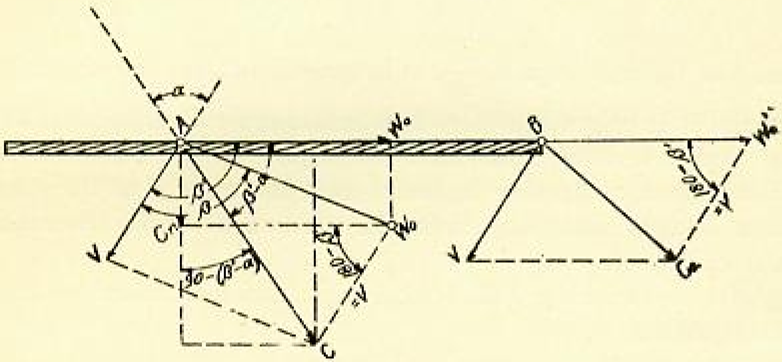

FIG. 100.

to the resultant of the velocities w_0' and v. If this resultant be designated by c_a the corresponding velocity head will be $\frac{c_a^2}{2g}$.

From Fig. 100 we have

$$c_n = c \sin(\beta' - \alpha) - v \sin \beta', \quad \ldots\ldots \quad (67)$$

$$w_0' = c \cos(\beta' - \alpha) - v \cos \beta', \quad \ldots\ldots \quad (68)$$

and finally

$$c_a = \sqrt{v^2 + w_0'^2 + 2vw_0' \cos \beta'}. \quad \ldots\ldots \quad (69)$$

The value of w_0' in (69) may be obtained by (68).

The effective head acting on the plane, therefore, is

$$A = \frac{c^2}{2g} - \frac{c_a^2}{2g} - \frac{c_n^2}{2g} = \frac{2vc_n \sin \beta'}{2g} = \frac{2v \sin \beta' [c \sin(\beta' - \alpha) - v \sin \beta']}{2g}. \quad . \quad (70)$$

This value is a maximum when $(\beta' - \alpha) = 90°$, i.e., when the particle of water impinges on the plate at right angles thereto, and when the velocity of the plane is such that $v=\frac{c}{2\sin\beta'}$. Then

$$A_{\max}=\frac{1}{2}\frac{c^2}{2g}, \qquad (71)$$

and then

$$\frac{c_n^2}{2g}=\frac{1}{4}\frac{c^2}{2g}, \qquad (72)$$

$$\frac{c_a^2}{2g}=\frac{1}{4}\frac{c^2}{2g}. \qquad (73)$$

The head lost through impact, $\frac{c_n^2}{2g}$, is in general not so large as might appear at a first glance, and it is only when there is a considerable divergence in the directions of w_0 and w_0' that it becomes noticeable. As applied to the turbine it may be said that the efficiency is not noticeably decreased until the angle β' at the entrance to the runner buckets departs materially from the theoretical angle determined for an entrance free from impact.

If in Fig. 100 we denote by β the bucket angle thus determined we may obtain its value from the equation

$$\sin(\beta-\alpha)=\frac{v\sin\alpha}{\sqrt{v^2+c^2-2vc\cos\alpha}}. \qquad (74)$$

We will now investigate the values of the lost pressure head by means of the following practical examples:

To determine the conditions at the entrance to a runner bucket there are given

$$\alpha=30°;\quad v=0.65\sqrt{2gh};\quad c=0.71\sqrt{2gh}.$$

Required:

(1) The entrance angle for an entrance free from impact.

(2) The loss in pressure head if instead of the correct angle we choose one of 120°.

(3) The same if one of 110° is chosen.

Required (1) Eq. (74) gives

$$\sin(\beta-30°)=\frac{0.65\sqrt{2gh}\times 0.500}{2gh\sqrt{0.65^2+0.71^2-2\times 0.65\times 0.71\times 0.866}},$$

$$\beta-30°=65°\ 30';$$

$$\beta=95°\ 30'.$$

The bucket angle is therefore 95° 30′ for an entrance free from impact.

Required (2) The choice of $\beta=120°$ instead of $=95° 30'$ results in a deflection of $24° 30'$. The impact velocity c_n may then be determined by Eq. (67),

$$c_n = 0.71\sqrt{2gh}\sin(120° - 30°) - 0.65\sqrt{2gh}\sin 120°$$

$$= 0.15\sqrt{2gh},$$

or

$$\frac{c_n^2}{2g} = 0.0225h.$$

That is, there is the loss of pressure head $=2.25$ per cent.

Required (3) The choice of $\beta=110°$ results in a deflection of $14° 30'$. The corresponding impact component then is

$$c_n = (0.71\times 0.985 - 0.65\times 0.940)\sqrt{2gh}$$

$$= 0.0883\sqrt{2gh}.$$

Therefore,

$$\frac{c_n^2}{2g} = 0.0078h.$$

The loss from impact is therefore only 0.78 per cent.

Pressure Relations in a Turbine—Derivation of the Fundamental Equation

In its passage through a turbine a particle of water traverses the following distinct and several paths or lines:

Line I. From the entrance place immediately in advance of the turbine (whether a ring gate, a butterfly valve, or the flume), to the first fixed cross-section of the distributor, where the pressure is p_E and the velocity is c_E.

Line II. From the entrance cross-section of the distributor to its last fixed discharge cross-section, where the pressure is p_A and the velocity is c_A.

Line III. From the discharge cross-section of the distributor through the clearance space to the edge of the vanes at the entrance to the runner where the pressure is p_O and the velocity is c_O.

Line IV. From the entrance edge of the runner vanes to the first fixed entrance cross-section of the runner bucket, the pressure at that section being p_e and the velocity w_e.

Line V. From the entrance cross-section of the distributor to the last given discharge cross-section of the runner buckets, where the pressure is p_a and the velocity w_a, and from which the water enters the draft tube with an absolute velocity c_a.

Line VI. From the discharge cross-section of the runner buckets to the point of discharge into the tailrace, a line which in most cases coincides with the average line of the draft tube. The particle of water leaves the draft tube at a depth of y feet below the surface of the tail water and has a velocity c_y at right angles to the true discharge cross-section of the draft tube.

A portion of the available head is lost in each line through friction, resistance due to angles and irregularities in the flow of the water. These losses of pressure head we will designate in the above order by t_1H, t_2H, etc.

By the consideration of those losses we obtain the following fundamental relations between the absolute pressures, the pressure heads corresponding to the velocities of the water, and the pressures which in the form of centrifugal forces result from the rotation of the runner.

Line I.
$$p_E+\frac{c_E^2}{2g}=h_E-t_1H. \qquad (75)$$

Line II.
$$p_A+\frac{c_A^2}{2g}=p_E+\frac{c^2_E}{2g}+h_A-h_E-t_2H. \qquad (76)$$

Line III.
$$p_O+\frac{c_0^2}{2g}=p_A+\frac{c_A^2}{2g}+h_O-h_A-t_3H. \qquad (77)$$

Line IV.
$$p_e+\frac{w_e^2}{2g}=p_O+\frac{w_0^2}{2g}-\frac{v_0^2-v_e^2}{2g}+h_e-h_0-t_4H. \qquad (78)$$

Line V.
$$p_a+\frac{w_a^2}{2g}=p_e+\frac{w_e^2}{2g}-\frac{v_e^2-v_a^2}{2g}+h_a-h_e-t_5H. \qquad (79)$$

Line VI.
$$y+\frac{c_y^2}{2g}=p_a+\frac{c_a^2}{2g}+(H-h_a)+y-t_6H. \qquad (80)$$

By the addition of (75), (76), and (77) we obtain

$$p_0+\frac{c_0^2}{2g}=h_0-(t_1+t_2+t_3)H. \qquad (81)$$

From this equation we may calculate the pressure at the circumference of the runner.

From (78) and (77) we have

$$p_a+\frac{w_a^2}{2g}=p_0+\frac{w_0^2}{2g}+h_a-h_0-\frac{v_0^2-v_a^2}{2g}-(t_4+t_5)H. \qquad (82)$$

This equation may be used to calculate the pressure in the discharge cross-section of the runner bucket.

The term $\frac{c_v^2}{2g}$ in Eq. (80) represents lost pressure head—at least when the axis of the draft tube is at right angles to the surface of the tail water—and may be expressed as t_7H. Then

$$1-(t_1+t_2+t_3+\ .\ .\ .\ t_7)=\varepsilon. \qquad (83)$$

In which ε represents the hydraulic efficiency of the turbine from the entrance of the water to its discharge. By the addition of Eqs. (80), (81), and (82), and by using the value of ε given in (83) we obtain the fundamental equation expressing the relation of pressures within a turbine:

$$\frac{c_0^2-c_a^2}{2g}+\frac{w_a^2-w_0^2}{2g}+\frac{v_0^2-v_a^2}{2g}=\varepsilon H. \qquad (84)$$

Pressure Losses in the Turbine and Their Reduction

It is the problem of the designer to avoid, where possible, each partial loss of efficiency and where they are unavoidable to keep them as small as possible. This may be accomplished in the following manner:

I. *Partial Loss* t_1H.

(*a*) By a reduction in the velocity of the water in cases where its course is not uniform, its direction and the cross-sections being subject to sudden changes.

Let c, the velocity of the flow of the water at the entrance to the distributor, be so chosen that

$c \leqq 0.10\sqrt{2gH}$ in an open flume with a rectangular cross-section built of concrete or steel for turbines with horizontal or vertical shafts.

$c \leqq 0.14\sqrt{2gH}$ in an open flume having the form of a half-circle in plan with a vertical arrangement of the turbine or in the case of a circular sheet steel or cast-iron wheel case in connection with a horizontal or vertical arrangement of the turbine, the turbine in all cases being installed in the center of the flume or case.

$c \leqq 0.17\sqrt{2gH}$ in a half-round concrete chamber or for a sheet-steel or cast-iron housing in which the turbine is placed excentrically so that the cross-sections through which the water flows decrease in area in proportion to the water passing through them.

(*b*) By carefully regulating the velocity of the water from the headrace or penstock to the first measurable cross-section of the distributor by conducting it through a concrete chamber having a spiral form or through spiral-formed wheel cases made of sheet steel or cast iron and having circular or rectangular sections. In these cases the velocity may be so chosen that

$$c \leqq 0.20\sqrt{2gH} \text{ up to a maximum of } c \leqq 0.25\sqrt{2gH}.$$

The value of t_1H should not exceed 3 per cent of the total pressure head.

II. *Partial Loss* t_2H.

(*a*) By placing the vanes of the distributor in such a relation to the entrance chamber that the inflowing water may enter the distributor buckets without impact or abrupt changes of velocity. Above all the vanes of the distributor must be so formed that no contraction of the water can take place at its entrance and so that no "dead spaces" may be produced outside of the distributor and thus cause eddies.

(*b*) By sharpening the vanes on the entrance edges.

(*c*) By using vanes which are as thin and smooth as possible as well as those having curves which are as easy as possible so as to prevent the loss of head which is caused by abrupt bends.

(*d*) By reducing the number of vanes. This cannot be carried too far, for the control of the water must be maintained between the fixed entrance and the fixed discharge cross-section. The latter is the case when the length of a middle water thread between the above-named sections is not less than 1.2 times the clearance line in the discharge cross-section.

t_2H should not exceed 2.5 per cent of the total pressure head.

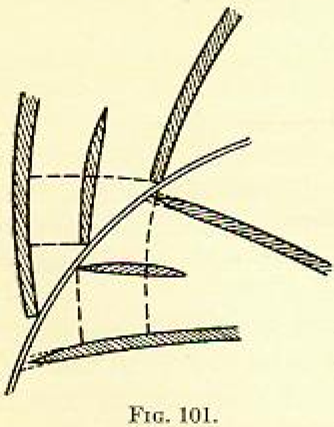

Fig. 101.

III. *Partial Loss* t_3H.

(*a*) By selecting the forms of the vanes in such a manner that the actual water threads are parallel to each other in the discharge cross-section.

(*b*) By constructing the ends of the vanes in such a manner that the backs as well as the working surfaces are placed at the same angle to the surface which includes the discharge ends of the vanes. Requirements (*a*) and (*b*), for example, may be combined in the case of a radial turbine by curving the ends of the vanes in the form of an evolute.

(*c*) By sharpening and rounding off the ends of the vanes at the discharge edge.

(*d*) By making the flow of the water as uniform as possible from the fixed cross-section of the distributor discharge to the entrance of the runner. This requirement may, in some cases, be combined with that of II (*d*) by the introduction of short, thin intermediate vanes of fish form placed at the angle α between the other vanes, as shown in Fig. 101.

(e) By reducing the clearance losses, that is, the resistance to the water flowing between the rims of the distributor and runner. This may be accomplished:

(1) Through the greatest possible reduction of the clearance space by means of excellent balancing and exact turning of the runner as well as by accurate boring of the distributor.

(2) By turning down the outside of the runner and distributor in steps or some similar form (see Fig. 66).

Let f be the area in square feet of the cylindrical surface of the clearance space; p the head in feet of the water column; μ the coefficient of contraction, which is about 0.5. Then the loss of water in cubic feet approximates

$$q=\mu f\sqrt{2gp}, \qquad (85)^*$$

t_3H should not altogether amount to more than 2 per cent of the pressure head.

IV. *Partial Loss* t_4H.

(a) By the use of vanes as thin as possible.

(b) By sharpening the edges of the vanes at the entrance, beveling them well back.

(c) By placing the backs of the vanes as well as their working faces at the same angle.

(d) By insuring a uniform flow of the water from the entrance edge to the first fixed cross-section of the runner bucket. This may be accomplished as shown in Fig. 68 in connection with requirement V (b). If a uniform flow does not obtain it is a question whether the centrifugal force $\frac{v_0^2-v_e^2}{2g}$, as contained in the fundamental equation, really acts with its full intensity.

We may consider the loss t_4H to be 1.5 per cent of the pressure head.

V. *Partial Loss* t_5H.

(a) By using the thinnest possible vanes with easy curves.

(b) By the use of the least possible number of vanes.

(c) By so selecting the limits of the vanes and the form of the runner crowns that a uniform flow of the water, both relatively and absolutely, is obtained through all the cross-sections and so that the path of the water through the runner is a continuation of that from the distributor buckets.

(d) By causing the particles of water to flow in parallel lines through the last

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* An actual determination shows that the losses due to lack of tightness are dependent upon the length of the joint and upon μ, varying with the play between the packing rings.

fixed cross-section of the runner buckets so as to avoid a contraction of the flow beyond the runner.

(*e*) By designing the back and the working surface of the vanes parallel at their ends.

(*f*) By sharpening the ends of sheet-steel buckets and by rounding off those that are cast.

With a proper design the loss t_5H should not exceed 2.5 per cent as a maximum.

VI. *Partial Loss* t_6H.

(*a*) By so designing and connecting the draft tube that its bounding lines form a natural prolongation of the outer rim of the runner buckets or, in the case of an axial turbine, of both rims.

The velocity of the water in the direction of the draft-tube axis, that is, the velocity in the direction of the normal water threads, varies at the point of entrance from $0.10\sqrt{2gH}$ to $0.33\sqrt{2gH}$ and averages $0.22\sqrt{2gH}$. This value depends upon the length of the draft tube and above all on the characteristics of the turbine in respect to the water consumption.

(*b*) By so designing the draft tube that the passage of the water through it will have the proper relation to that through the runner; there should be no sudden changes in the velocity or direction of flow at the point where the water leaves the runner.

(*c*) By allowing a sufficient distance from the mouth of the draft tube to the bottom of the tailrace so that tail water may not be readily dammed up by obstacles in the flow of the discharge. This distance should be from 0.6 to 1.0 times the diameter of the draft tube at its lower end. The channel through which the water flows from the draft tube to the tailrace canal or river should have no sudden changes of cross-section.

The velocity of discharge from a draft tube with a vertical discharge section be about $0.1\sqrt{2gH}$ and t_6h will be from 3 per cent to 5 per cent of H, depending upon the length and form of the draft tube. As the water leaves the runner with a revolving motion (see discharge diagram, Fig. 116) t_6H is increased on account of the circular component of the absolute velocity of discharge, and this fact must be considered in calculating t_6H.

VII. *Partial Loss* t_7H.

This loss can be entirely obviated if the form of the draft tube is so chosen that the velocity and direction of the water leaving the runner are gradually changed to those of the water in the tailrace. This can be accomplished by the use of properly designed concrete, plate steel, or cast-iron draft tubes.*

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* As VI and VII deal with retardations of velocity it would be of great interest to be able to determine what is the minimum length of draft tube which is necessary to change a given velocity c_a' to c_{r_1}'' a problem which can be solved in each case only by experiment.

Total Efficiency

It is sufficient for the practical calculation of the turbine, especially those with movable vanes, to know the hydraulic efficiency as a whole. In the following calculations of the areas of the cross-sections of the turbine we will once for all deduct from the total head the losses $(t_1+t_2+t_3+t_4+t_5)H$, that is, these up to the discharge from the runner, and will then calculate the turbine as an "ideal" turbine without further losses.

The total hydraulic losses being

$$(t_1+t_2+t_3+t_4+t_5+t_7)H=\varepsilon H.$$

We will separate the value $(t_6+t_7)H$, i.e., the losses in the draft tube, and for convenience in calculation will consider that this term represents 5 per cent of the total head, that is,

$$(t_6+t_7)H=0.05H.$$

Strictly speaking a distinction should be made at this point between large and small turbine units. As the result of experience we determine ε as a function of the size of the unit, i.e., the horse power of the machine, and give it the following values, viz.:

$\varepsilon \cong 78$ per cent for a turbine unit of.......................	30 H.P.
$\varepsilon \cong 87$ " " " "	10,000 H.P.

Intermediate values may be obtained from the curve shown in Fig. 102. For example,

$\varepsilon \cong 84$ per cent for a turbine unit of.......................	1,000 H.P.
$\varepsilon \cong 81$ " " " "	100 H.P.

It is to be carefully noted that these values obtain only when the discharge of the water from the runner occurs on the normal water threads and when the runner entrance is free from impact.

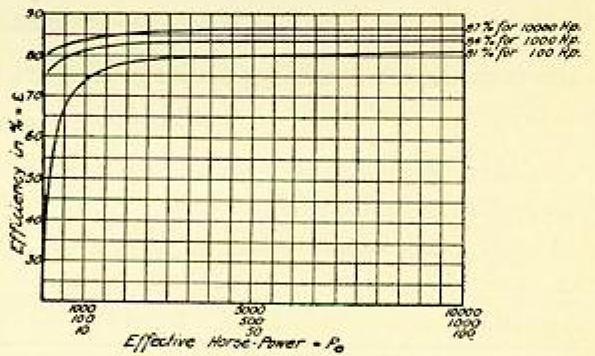

FIG. 102.

We must also make the necessary deductions for the friction losses in step and guide bearings and for hydraulic losses in balancing pistons and cooling devices. In the case of a turbine of 30 H.P. these losses will amount to 3 per cent, and for a 10,000 H.P. unit to $1\frac{1}{2}$ per cent. By plotting these values in a curve similar to that shown in Fig. 102 we thus obtain a curve showing the mechanical efficiency which may be guaranteed for a turbine regulated by hand by means of a mechanical device.

If the turbine is regulated automatically by means of the water available for operation the efficiency must be reduced by 1 per cent for a unit of 30 H.P., and by ½ per cent for a unit of 10,000 H.P. in order to allow for the power expended in regulation if we desire to obtain the mechanical efficiency which may be guaranteed for a self-regulated unit.

The Fundamental Equation for an Entrance Free from Impact

From Fig. 103 we obtain from the entrance diagram showing an entrance free from impact the equation

$$w_0^2 = c_0^2 + v_0^2 - 2v_0c_0 \cos \alpha, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (86)$$

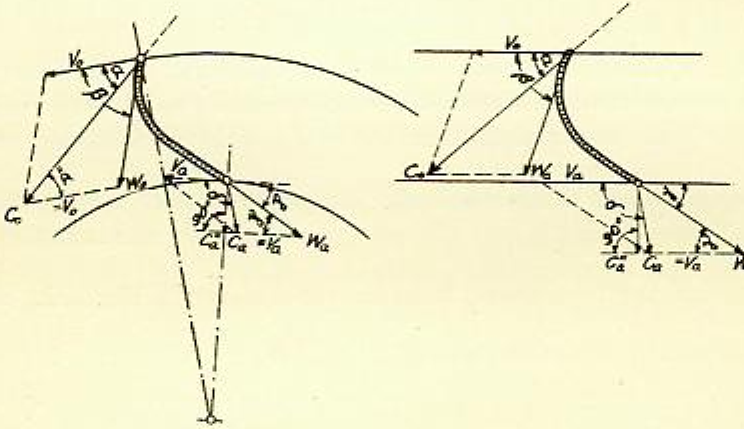

FIG. 103.

and from the corresponding discharge diagram

$$w_a^2 = c_a^2 + v_a^2 - 2c_av_a \cos \delta, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (87)$$

obtaining from the above equations the value of $w_a^2 - w_0^2$, and inserting the same in the fundamental Eq. (84) we have

$$c_0^2 - c_a^2 + c_a^2 + v_a^2 - 2v_ac_a \cos \delta - c_0^2 - v_0^2 + 2v_0c_0 \cos \alpha + v_0^2 - v_a^2 = 2g\varepsilon H,$$

or

$$v_0c_0 \cos \alpha - v_ac_a \cos \delta = g\varepsilon H,^* \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (88)$$

* In this equation, obtained by following a water thread, $c_o \cos \alpha$ and $c_a \cos \delta$ represent the components of the absolute velocity of a particle of water which are produced entirely by revolution and are correctly measured at the entrance and exit respectively of a similarly formed rotating bucket.

The points of entrance and discharge are thereby so located that at them the centrifugal forces grasp and this is the fundamental equation for a turbine having an entrance free from impulse.

In the practical application of this equation α and δ may be chosen, subject to correction later, and the velocities v_0 and v_a may be considered as known values. Finally the velocity c_a'' may be considered as obtained from Eq. (56), said velocity being

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or release, respectively, the water element, or, what is the same thing, so that at these points an element of the water, possessed with energy, begins or ceases to exert a turning effect on the wheel.

Let us represent $c_o \cos \alpha$ and $c_a \cos \delta$, the respective components of the velocity, by w_{n_o} and w_{n_a}; multiplying the entire equation by $dQdW$ and dividing the same by the constant $\omega = \frac{\pi n}{30}$, we may write Eq. (88) in the form

$$dQdW(r_o w_{n_o} - r_a w_{n_a}) = \frac{dQdW\varepsilon Hg}{\omega},$$

or

$$\frac{dQdW}{g}(r_o w_{n_o} - r_a w_{n_a}) = \varepsilon \frac{dQdWH}{\omega},$$

in which r_o and r_a, respectively, represent the distances of the entrance and discharge points of the water elements from the axis of revolution.

It is important to remember at this point the important law first discovered by Euler in 1750 and which, though for a time forgotten, is still to-day the foundation of the calculations for any turbine; that is

$$\frac{dQdW}{g} = dm,$$

where m represents the mass.

Then
$$dWdQH = dP_T,$$

dP_T being the differential of the theoretical potential force of the particle of water and

$$\frac{dP}{\omega} = dM_T,$$

where dM_T represents the theoretical turning forces, i.e., those turning forces which would act if there were no losses inside the turbine. We may now write the fundamental equation in the form

$$dm(r_o w_{n_o} - r_a w_{n_a})\omega = \varepsilon dP_T,$$

or

$$dm(r_o w_{n_o} - r_a w_{n_a}) = \varepsilon dM_T.$$

In this form these equations clearly express the relations existing inside the turbine between the action of the turning forces and the forces tending to velocity.

These equations apply without further change to centrifugal pumps having a definite control of the water providing that for ε is substituted its reciprocal value $\frac{1}{\varepsilon}$.

Considering the left side of the above equations we find that the first term gives the value of the turning forces which, with a given number of revolutions n, act directly upon the wheel at the point of entrance to drive it forward. The second term of the equation gives the value of the forces which act upon the wheel at the point of discharge indirectly, i.e., by back pressure.

In case the direction of c_a and c_o falls within the same quadrant these two turning forces act against each other, which is clearly shown by the minus sign.

Under the assumption that every element of water $dQdW$ delivers toward the axis an equal amount in the direction of a normal water thread at the point where the water leaves the runner and enters the draft tube. c_a may be determined from c_a'' by the equation

$$c_a = \frac{c_a''}{\sin \delta}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (89)$$

By substituting this value of c_a in (88) we have

$$v_0 c_0 \cos \alpha - v_a c_a'' \cot \delta = g \varepsilon H. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (90)$$

In order to express this equation in a form independent of the amount of head we will make the following substitutions:

$$v_0 = \phi \sqrt{2gH}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (91)$$

$$c_0 = k_{c_0} \sqrt{2gH}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

$$v_a = k_{v_a} \sqrt{2gH}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (93)$$

$$c_a'' = k_{c_a''} \sqrt{2gH}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (94)$$

and thus obtain

$$\phi k_{c_0} \cos \alpha = k_{v_a} k_{c_a''} \cot \delta = \frac{\varepsilon}{2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (95)$$

This equation enables us to determine k_{c_0} and thus c_0, the velocity of discharge

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$r_0 w_{n_0} - r_a w_{n_a}$, we may obtain at once a summation for the whole turbine and may write the fundamental equation in the form

$$\frac{QW}{g}(r_0 w_{n_0} - r_a w_{n_a}) = \varepsilon M_T.$$

The assumption on which this is based holds good in the case of a turbine which receives the water on a cylindrical surface whose axis is the axis of the turbine and discharges the water parallel to said axis, as in a normal Francis turbine; for in such a case $w_{n_a} = 0$ and r_0 and w_{n_0} are constants.

Lorenz, in his book "Theorie und Berechnung der Kreiselräder," has drawn attention to the limits of accuracy of this equation and has shown a way in which an equality of the values $r_0 w_{n_0} - r_a w_{n_a}$ may be effected by a suitable design of the entrance and discharge edges. The suggested treatment is, however, not practicable in connection with the type of regulating apparatus now most frequently employed (i.e., that with movable distributor vanes), for in this case for constructive reasons the distributor must be a cylinder concentric with the runner. Moreover it is clearly shown in the footnote to example 3, page 211, that the inequality between the values of $r_0 w_{n_0} - r_a w_{n_a}$ as calculated for a partial turbine of modern design is extremely small,—even with an unfavorable discharge diagram,—amounting to scarcely 1%. The example referred to also emphasizes the agreement between theory and practice and, at least in this direction, makes it easier to believe in the high efficiencies obtained in some finished wheels.

Unfortunately, Lorenz in his investigations omitted a consideration of the friction on the walls of the buckets and draft tube and assumed an endless number of buckets. He was therefore not able to harmonize his theory with the results of practice, particularly in the case of high-speed turbines.

Friction has an important influence on the efficiency of a turbine and must not be omitted from consideration, especially in the case of turbines operating at high relative velocities.

from the distributor and the angle of the reaction (see numerical example, -8-, page 98).

Strictly speaking the value of ε to be used in (95) should be somewhat smaller than that shown by the curve, because if the water is not discharged parallel to the axis a further loss of pressure head occurs under the runner, the amount of which is $\frac{c_a^2 \cos^2 \delta}{2g}$. This should be expressed in percentage and deducted from ε.

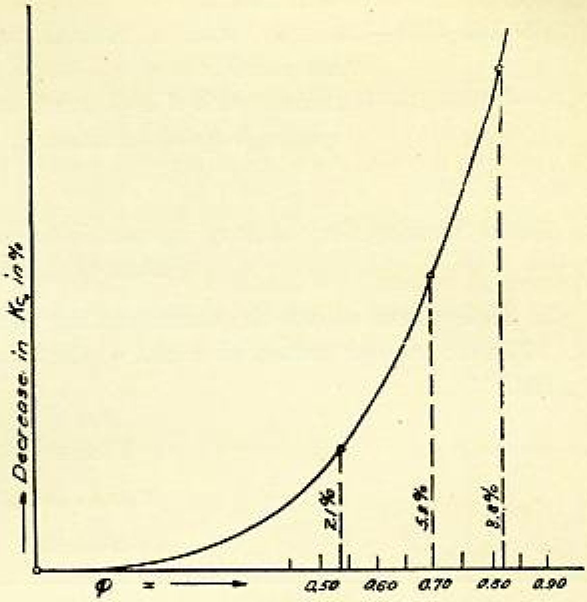

Fig. 104.

Fundamental Equation under the Conditions of an Entrance Free from Impulse and a Discharge Parallel to the Axis.

A case of special importance is that where c_a'' and c_a have the same direction, or in other words where the absolute velocity has the direction of a normal water thread. In this case $\delta = 90°$ and $c_a'' = c_a$, and the fundamental equation may be written

$$v_0 c_0 \cos \alpha = g \varepsilon H, \qquad (96)$$

or in a form for practical use

$$\phi k_{c_0} \cos \alpha = \frac{\varepsilon}{2}. \qquad (97)$$

In order to make it possible to use this equation in practical work a curve showing the values of ε for different cases has been shown in Fig. 102. From this table we may select the proper value of ε for a unit having a given horse power and number of revolutions, and after choosing the entrance angle α may then calculate k_{c_0} and c_0 (see numerical examples).

In the present state of turbine construction the values of ϕ vary all the way from 0.48 to 0.95, and dependent upon these values we may make the following classification of turbines, viz.:

$\phi = 0.48$–0.51—Freely radiating turbines (i.e., action or Girard turbines) with the smallest number of revolutions.

$\phi = 0.51$–0.56—Limit, or Haenal, turbines, that is, turbines with a small amount of reaction.

$\phi = 0.56$–0.68—Turbines with a normal reaction, the vane angle β varying about 90°.

$\phi = 0.68$–0.95—Turbines with a high reaction and a large number of revolutions, i.e., high-speed turbines.

Tables Showing the Values of k_{c_0} and ϕ Obtained by Employing the Most Usual Values of α and ε

In Table I are shown the corresponding values of k_{c_0} and ϕ as calculated from Eq. (97) with various values of α and ε, the latter being selected from the curves in Fig. 102.

TABLE I

VALUES OF k_{c_0}

$\varepsilon = 0.78$

$\phi =$	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.85	0.90	$\delta =$
$\alpha = 16°$	0.811	0.738	0.676	0.624	0.580	0.541	0.507	0.478	0.451	90°
	0.789		0.654		0.558		0.485		0.429	100°
$\alpha = 20°$	0.831	0.755	0.692	0.639	0.594	0.553	0.519	0.488	0.462	90°
	0.803		0.664		0.566		0.491		0.434	100°
$\alpha = 24°$	0.854	0.766	0.711	0.656	0.610	0.570	0.534	0.502	0.474	90°
	0.819		0.676		0.575		0.499		0.439	100°
$\alpha = 30°$	0.901	0.819	0.750	0.694	0.643	0.600	0.563	0.530	0.500	90°
	0.854		0.703		0.596		0.516		0.453	100°
$\alpha = 38°$		0.900	0.825	0.761	0.707	0.659	0.618	0.582	0.550	90°
	0.920	0.830	0.755		0.637		0.548		0.480	100°

$\varepsilon = 0.800$

$\phi =$	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.85	0.90	$\delta =$
$\alpha = 16°$	0.833	0.756	0.695	0.641	0.595	0.555	0.520	0.490	0.463	90°
	.811		0.673		0.573		0.498		0.441	100°
$\alpha = 20°$	0.850	0.773	0.708	0.654	0.608	0.568	0.532	0.501	0.472	90°
	0.822		0.680		0.580		0.504		0.444	100°
$\alpha = 24°$	0.878	0.798	0.732	0.675	0.626	0.585	0.548	0.517	0.487	90°
	0.843		0.697		0.591		0.513		0.452	100°
$\alpha = 30°$	0.925	0.840	0.770	0.710	0.660	0.617	0.578	0.544	0.514	90°
	0.878		0.723		0.613		0.531		0.467	100°
$\alpha = 38°$		0.921	0.846	0.779	0.725	0.676	0.635	0.596	0.564	90°
		0.851	0.776		0.655		0.565		0.494	100°

TABLE I—*Continued*

$\varepsilon = 0.82$

$\phi =$	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.85	0.90	$\delta =$
$\alpha = 16°$	0.854	0.776	0.710	0.657	0.610	0.569	0.533	0.502	0.474	90°
	0.832		0.688		0.588		0.511		0.452	100°
$\alpha = 20°$	0.874	0.794	0.728	0.672	0.622	0.582	0.545	0.514	0.485	90°
	0.846		0.700		0.594		0.517		0.457	100°
$\alpha = 24°$	0.900	0.818	0.750	0.692	0.642	0.600	0.562	0.529	0.500	90°
	0.865		0.715		0.607		0.527		0.465	100°
$\alpha = 30°$	0.948	0.861	0.790	0.728	0.676	0.631	0.592	0.558	0.527	90°
	0.901		0.743		0.629		0.545		0.480	100°
$\alpha = 38°$		0.945	0.867	0.800	0.743	0.694	0.650	0.612	0.578	90°
		0.875	0.797		0.673		0.580		0.508	100°

$\varepsilon = 0.84$

$\phi =$	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.85	0.90	$\delta =$
$\alpha = 16°$	0.875	0.795	0.729	0.672	0.625	0.583	0.546	0.514	0.485	90°
	0.853		0.707		0.603		0.524		0.463	100°
$\alpha = 20°$	0.895	0.814	0.745	0.688	0.638	0.597	0.558	0.527	0.497	90°
	0.867		0.717		0.610		0.530		0.469	100°
$\alpha = 24°$	0.920	0.836	0.767	0.708	0.657	0.612	0.675	0.541	0.512	90°
	0.885		0.732		0.622		0.540		0.477	100°
$\alpha = 30°$		0.880	0.808	0.746	0.692	0.647	0.606	0.571	0.539	90°
		0.833	0.761		0.645		0.559		0.492	100°
$\alpha = 38°$		0.970	0.889	0.820	0.762	0.710	0.666	0.628	0.592	90°
		0.900	0.819		0.692		0.596		0.522	100°

The formula upon which the above table is based and the significance of the several terms employed are as follows, viz.:

Formula 1. $k_{c_a} = \dfrac{\varepsilon}{2 \cos \alpha \cdot \phi}$ for discharge parallel to axis.

Formula 2. $k_{c_a} = \dfrac{\varepsilon}{2 \cos \alpha \cdot \phi} + \dfrac{k_{v_a} k_{c''_a} \cot \delta}{\phi \cdot \cos \alpha}$ for δ in excess of 90°.

k^{c_a} = ratio of the absolute velocity of discharge from the distributor to the spouting velocity of water, $\sqrt{2gh}$;

ϕ = ratio of the peripheral velocity of the runner to $\sqrt{2gh}$;

$k_{c''_a}$ = ratio to $\sqrt{2gh}$ of the absolute discharge velocity, measured at right angles to the circumferential velocity of the runner at point of discharge;

k_{v_a} = ratio of the circumferential velocity of the runner at the point of discharge to $\sqrt{2gh}$.

As in many cases it is specified that the turbine shall give its best efficiency at three-fourths gate opening, which is equivalent to a value of δ of about 100°, we have also calculated the corresponding values of k_{c_0} and ϕ for this special case of a Francis turbine, and have inserted them in the table. These values are also obtained by Eq. (95), δ being made equal to 100°. The relations of k_{v_a} to ϕ are then those which exist in a normal and practically designed Francis turbine. The difference between the values of k_{c_0} when $\delta=90°$ and $\delta=100°$, as given in the second term of the right-hand side of Formula 2 are given in the following table:

$\delta=90°+10°$, i.e., $\cot\delta=\tan 10°=-.07633$

$\alpha=16°$	$\alpha=20°$	$\alpha=24°$	$\alpha=30°$	$\alpha=38°$
$k_{c_a}=0.655\phi$	0.690ϕ	0.725ϕ	0.780ϕ	0.890ϕ
$k_{c''_a}=0.18$	0.215	0.25	0.295	0.35
Correction $=0.022$	0.028	0.035	0.047	0.070

Numerical Example

(*a*) For a slow-speed Francis turbine of 500 H. P. there is given:

$$\varepsilon=0.83;\ \alpha=14°;\ \phi=0.54;\ k_{v_a}=0.346=0.64\phi;\ k_{c''_a}=0.15$$

To solve for

(1) k_{c_0} with the discharge angle $\delta=90°$.

(2) k_{c_0} with the discharge angle $\delta=100°$.

Equation (97) gives for (1)

$$k_{c_0}=\frac{0.83}{2\times0.970\times0.54}=0.792.$$

Considering (2), we may disregard the reduction in ε due to a discharge not parallel to the shaft, for with a 10° leading discharge diagram, the loss in efficiency is negligible and in the example before us is only $0.0007H$. We then have by Eq. (95)

$$k_{c_0}=\frac{\frac{0.83}{2}-0.346\times0.15\times0.176}{0.54\times0.970}=0.775.$$

The value of k_{c_0} in (2) is therefore 2.1 per cent smaller than in (1).

(*b*) For a normal-speed Francis turbine of 500 H. P. there is given:

$$\varepsilon=0.83;\quad \alpha=24°;\quad \phi=0.70;\quad k_{v_a}=0.508=0.725\phi;\quad k_{c''_a}=0.25.$$

To solve for

(1) k_{c_0} with the discharge angle $\delta=90°$.

(2) k_{c_0} with the discharge angle $\delta=100°$.

Eq. (97) gives for (1)

$$k_{c_0}=\frac{0.83}{2\times0.914\times0.70}=0.648.$$

In regard to (2), the loss in eddies beyond the wheel, corresponding to a 10° leading discharge diagram may be regulated, as it is only $0.0019H$ in this example. ε therefore remains unchanged at 0.83. Eq. (95) then gives

$$k_{c_0}=\frac{\frac{0.83}{2}-0.508\times0.25\times0.176}{0.70\times0.914}=0.614,$$

or 5.2 per cent smaller than for a discharge parallel to the shaft.

(*c*) For a high-speed Francis turbine of 500 H.P. there is given:

$$\varepsilon=0.83;\quad \alpha=33°;\quad \phi=0.82;\quad k_{v_a}=0.664=0.81\ \phi;\quad k_{c_{a''}}=0.315.$$

To solve for

(1) k_{c_0} with the discharge angle $\delta=\ 90°$.

(2) k_{c_0} with the discharge angle $\delta=100°$.

Eq. (97) gives for (1)

$$k_{c_0}=\frac{0.83}{2\times0.839\times0.82}=0.603.$$

The loss in eddies beyond the wheel for (2) amounts to only $0.003H$ and may again be neglected.

We then have from (95)

$$k_{c}=\frac{\frac{0.83}{2}-0.664\times0.315\times0.176}{0.82\times0.839}=0.550,$$

or 8.8 per cent less than for a discharge parallel to the shaft.

These three examples teach especially that in order to obtain an angle of 100° for the absolute discharge from the runner the velocity of discharge from the distributor must be decreased, in comparison with the conditions existing when $\delta=90°$, by the following amounts:

In a slow-speed turbine by 2.1 per cent.
In a normal-speed turbine by 5.2 per cent.
In a high-speed turbine by 8.8 per cent.

In Fig. 104 these values are shown graphically in connection with the increase in peripheral velocity.

Determination of the Distributor Cross-Section

If $c_0 = k_{c_0}\sqrt{2gh}$ has been determined from the fundamental equation for a discharge parallel to the shaft, or for the case when δ exceeds 90°, we may then obtain the area in square feet of the cross-section of one of the Z distributor buckets of a radial turbine at that point where the middle water thread intersects the periphery of the runner, for its value is

$$\frac{Q}{Z}\frac{1}{c_0}.$$

In preparing a diagram the middle water thread should be constructed as equally distant from the backs of the distributor vanes, which are curved in the form of an involute, and the angle α thus determined. (See Fig. 125 for the cross-section of the distributor.)

Assuming that $c_A = c_0$, an assumption that may be approximated by a correct bending of the ends of the vanes of the distributor buckets (see Fig. 125, Diagram of the absolute velocities), it then follows that the cross-section area thus found is equal to that of the last fixed cross-section of the distributor buckets.

In the case of axial turbines it is necessary to divide each individual distributor bucket into m partial buckets, through each of which flows $\frac{Q}{mZ}$ cubic feet of water. If b_A is the effective breadth of any partial bucket at the last discharge cross-section and Δ_A the clearance, then the area in square feet of the effective cross-section of the partial bucket is as before

$$b_A \Delta_A = \frac{Q}{mZ} \cdot \frac{1}{c_0}. \qquad (98)$$

As the angle α has been assumed, thus fixing Δ_A, we may use this formula to determine the breadth of the partial bucket.

It is advisable to increase by about 10 per cent the cross-section of the distributor buckets as obtained by calculation, especially in the case of turbines regulated by movable distributor vanes or cylinder gates. On the other hand the runner bucket cross-sections should be designed exactly as calculated.

The additional area of the distributor buckets serves the following purposes:

(1) To insure the full capacity of the distributor at full gate opening even though errors in construction might occur.

(2) To meet the condition that in reality c_A is somewhat smaller than c_0.

(3) To provide for losses in the clearance space between the distributor and runner.

(4) To compensate for the difference in pressure $h_0 - h_A$ in the case of an axial turbine, this difference having been neglected.

In general no detrimental effect on the efficiency results from the increase in the cross-section of the distributor buckets, at least with regulation by movable vanes.

The Pressure Relations in the Spaces Adjacent to the Runner

In the case of a normal-speed turbine we may obtain an expression for the difference in pressure which exists in the water inside of a turbine next to the runner crown and that which obtains outside of the turbine, both pressures being measured in terms of the head acting on the space adjacent to the runner. The pressure outside the wheel may have either a negative or a positive value as compared with atmospheric pressure, depending on whether the turbine is placed above or below the surface of the water in the tailrace. The discharge channel should of course in the last case be called a discharge pipe rather than a draft tube.

As the pressure on the several points of the entrance edge of the runner vanes does not have a constant value (at least in an axial turbine), as shown by the fundamental equation, therefore the pressure at both runner crowns will in general have differing values. This pressure will be a minimum at the inner crown and a maximum at the outer one.

By subdividing the turbine into a sufficiently large number of parts we may determine the absolute pressure of the water in the neighborhood of the wheel crowns and at the entrance to the runner.

If c_0, co-ordinate to v_0, is the entrance velocity as fixed by the fundamental equation, then the corresponding value of the absolute pressure p_0 at those points on the vane edges where the circumferential velocity v_0 obtains may be obtained as follows:

(*a*) For a reaction turbine with a draft tube

$$p_0 = \left[(\varepsilon + t_4 + t_5 + t_6 + t_7)H - \frac{c_0^2}{2g}\right]0.434 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (99)$$

pressure in pounds per square inch.

In the equation $(t_4 + t_5)\ H$ are the losses in head which take place in the runner and $(t_6 + t_7)\ H$ are those in the draft tube.

(*b*) For a turbine operating below the surface of the tailrace without draft tube or discharge pipe

$$p_0=\left[(\varepsilon+t_4+t_5)H+\frac{c_a^2}{2g}-\frac{c_0^2}{2g}\right]0.434. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (100)$$

In this equation $\frac{c_a^2}{2g}$ represents the lost pressure head corresponding to c_a. This lost head is, as a rule, greater than $(t_6+t_7)H$.

Reaction turbines in which a part of the available head is lost because the turbine is placed above the surface of the tail water without a draft tube are no longer built and need not be discussed.

(*c*) For a freely radiating turbine with a draft tube or a limit turbine with a draft tube

$$p_0=0 \text{ because } \frac{c_0^2}{2g}=(\varepsilon+t_4+t_5+t_6+t_7)H. \quad . \quad . \quad . \quad . \quad . \quad . \quad (101)$$

(*d*) For freely radiating turbines without draft tubes (Pelton, and Girard turbines)

$$\text{As in } (c)\ p_0=0 \text{ whereas } \frac{c_0^2}{2g}=\varepsilon H_0, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (102)$$

in which H_0 denotes the pressure head on the turbine. This head is less than H on account of the pressure head ζH lost in friction in the penstock and by the distance X at which the turbine is set above the surface of the tail water, that is

$$H_0=H-\zeta H-X \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (103)$$

Eq. (102) may be written in the form

$$c^0=\sqrt{\varepsilon}\,\sqrt{2gH_0}.$$

Dependent upon the form and smoothness of the distributor buckets, the discharge coefficient $\sqrt{\varepsilon}$ varies from 0.94 to 0.98. If the pressure head is measured by a pressure gauge we must add to its reading the head corresponding to the velocity in the fixed cross-section.

Determination of the Vane Angle β, The Relative Velocity w_0 and the First Fixed Cross-Section of the Runner Buckets

In order that the runner vane may receive no impulse either forward or backward, it is placed at the angle β with the direction of the peripheral velocity v_0. As will be seen by reference to Fig. 105, this angle is determined for a point on the entrance edge of the vane by the equation

$$\sin \beta=\frac{c_0 \sin \alpha}{\sqrt{v_0^2+c_0^2-2 v_0 c_0 \cos \alpha}},* \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (104)$$

in which v^0, c^0 and α are considered to be known quantities.

The relative velocity w_0 at this point is determined by the equation

$$w_0=\sqrt{v_0^2+c_0^2-2 v_0 c_0 \cos \alpha} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (105)$$

If the vane is constructed at an angle other than β, say β', then w_0 is reduced to w_0' and its value may be obtained by the equation

$$w'_0=c_0 \cos (\beta'-\alpha)-v_0 \cos \beta' \quad . \quad (106)$$

For the development of this equation see Fig. 100 and Eqs. (67) and (68).

The velocity which is lost through impact becomes

$$c_n=c_0 \sin (\beta'-\alpha)-v_0 \sin \beta'. \quad . \quad (107)$$

The loss in pressure head itself is $\frac{c_n^2}{2g}$, which in the calculation of the bucket cross-sections is to be deducted from the available head

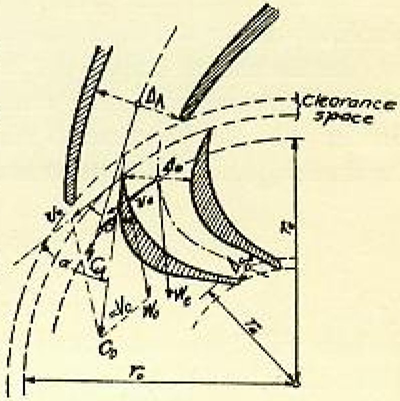

FIG. 105.

$$\varepsilon H+(t_6+t_7) H.$$

The velocity w_e in the first fixed cross-section of the runner bucket may as a first approximation be considered as equal to w_0, said section being designated in Fig. 125 as I. That is, the velocity at I may be considered equal to that at the point O, the latter being determined either graphically or by means of Eq. (105) from v_0 and c_0. In this way the cross-section and the clearance at the point I is fixed preliminarily. The direction of the flowing water is given by the entrance diagram, that is, by the triangle formed by v_0, c_0 and w_0. We may now draw out freehand the form of the bucket, as, for example, the bucket cross-section from the point O to the point VI, bearing in mind the clearance in the discharge cross-section of the bucket, which has been calculated in accordance with Section 14. In like manner one proceeds in the case of the bucket

* It is evident that in this equation the coefficient k_{c_0} and ϕ may be substituted for c_o and v_o, so that it may be written finally

$$\sin \beta=\frac{\phi \sin \alpha}{\sqrt{\phi^2+k^2_{c_0}-2 \phi k_{c_0} \cos \phi}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (104a)$$

sections *LMK* and *PQR*, for which the vane lengths are determined both from the calculated width of the discharge and from the previous section *O–VI*. After some practice it is not difficult to find a rather satisfactory shape for the bucket on the first trial. We may then complete the development on the bucket sections from the clearances at the points *I*, *II*, and as the width of the bucket at the several points on the elevation is known we may multiply these dimensions together and thus obtain the areas of the cross-sections at right angles to the flowing water. Further, by dividing the volume of water flowing through the several cross-sections by the areas thus found, we may obtain the relative velocities at each of the points *I*, *V*. The angular velocity at each point on the moving wheel may then be found and with this data we may finally obtain the absolute velocity and the absolute waterway by the following methods:

On the conical surface 0–1, whose vertex lies on the axis of revolution at the point 01, there moves a particle of water whose relative velocity is given in amount and direction by w_0. As the assumed relative path of the water has the length 0–1, therefore the time required for the water to flow through this path is equal to

$$\frac{0-1}{w_0}=t_0.$$

By multiplying this time by the peripheral velocity v_0 we obtain the length of the path through which the water flows in said time as such path is measured on the periphery. This path is to be drawn on a circle passing through 0 and having the center 01 (the vertex of the cone for both the absolute and the relative path of the water). By uniting the relative path and the peripheral path we obtain the point *I* of the absolute waterway. We may now proceed from point *I* to point *II* in a manner similar to the above except for the fact that the movement is now considered to take place on the conical surface *I–II*, whose vertex is at *I–II* on the axis of revolution. We must also consider the path *I–II* through which the water flows and the relative velocity at the point *I*. We thus find the time required for the passage of a particle of water from point *I* to point *II*, and with it a new section of the true curve through which the water flows. The union of the relative path with this section of curve gives the desired point *II* on the absolute waterway, and so on. At the same time we may obtain the absolute velocity c_e at the point *I* by combining the relative velocity w_e with the peripheral velocity V_e, etc. If we then use as abscissas the lengths of the absolute waterway as thus found, and as ordinates the absolute velocities, we obtain a curve whose form may be ascertained not only for the runner but also for the distributor and the draft tube. (See Fig. 125, "Diagram of absolute velocities.")

Assuming that the continuity of the curve holds at the points of transition from the distributor to the runner and from the runner to the draft tube, a correction is obtained in the absolute velocities, and as the peripheral velocities remain unchanged there results a corresponding correction in the relative velocities. The latter may also be assembled in a curve ("see Diagram of Relative Velocities," Fig. 125.) With such corrected values of the relative velocities we may finally obtain new values of the several clearances at points O to VI in the bucket, and with the latter may at last construct the accurate corrected form of the bucket to replace that originally assumed.

The transition from the runner to the draft tube may also be somewhat more accurately fixed by the requirement of continuity in the diagram of the relative velocities and a form thus obtained for the upper part of the draft tube by which it may be more accurately connected to the runner.

Influence of Centrifugal Force in Reaction and Girard-Type Turbines

In a reaction turbine the water is inclosed on all sides as it passes through the bucket from e, the effective entrance cross-section to a, the effective discharge cross-section (see Figs. 106 and 107) and must consequently revolve with the bucket. It therefore receives the influence of the centrifugal force by its revolution about the turbine axis.*

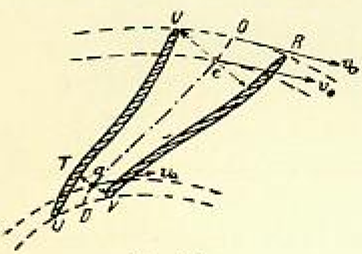

Fig. 106.

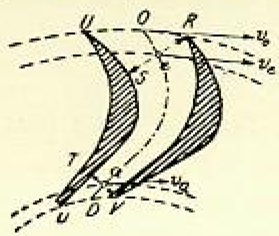

Fig. 107.

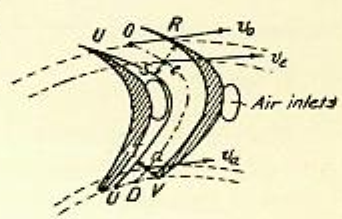

Fig. 108.

This influence is not so marked on the lines between O and e and between a and D, because the water is controlled on one side only; i.e., either by the working surface of a vane or by the back of a vane. Postponing the determination of the cross-section at a, we will consider that the centrifugal force acting from O to a has the value $\frac{v_0^2 - v_a^2}{2g}$, and that the work done in the whole bucket is completed between O and a.

In a Girard-type turbine where the jet generally exerts a pressure on the working surface only (see Fig. 108) it is questionable whether the centrifugal force exerts an influence, and it has been positively denied.

The authors are inclined to the opinion that centrifugal force exerts an influence in the case of Girard-type turbines just as long as the water jet is engaged in doing work on the vane, this opinion being based on the following considerations:

A particle of water entering the rotating bucket at the point R is compelled by the position of the vane to acquire the relative velocity w_0, and must necessarily possess the circumferential velocity v_0 both in amount and in direction, because v_0 is only the result-

* We will not consider here the centrifugal force due to the water following a curved path.

ant of the two velocities c_0 and w_0 which produce it. As the said particle of water passes through the bucket it must be subject to the same changes of velocity as the several points along the vane from R to V, for it steadily presses against the working surface of the vane, and in fact must do so in order to perform its work. At V, the point of discharge, it will, in addition to its relative velocity, have both in amount and direction the circumferential velocity which exists there. At the moment when the element of water has received its circumferential velocity and must receive it, as the fundamental equation teaches, in order to perform work at this instant, the centrifugal force begins to act. The working surface of the vane is first subjected at the point S to the influence of the centrifugal force arising from the action of the most distant particle of water which enters the bucket at U and the action of this particle ceases at T. The lines RS and VT represent true lines of intersection and are at right angles to the normal water thread ea.

A particle of water which moves on the middle water thread is subject to the action of the centrifugal force only between e and a, as shown above, and this force then has the value in head $\frac{v_e^2 - v_a^2}{2g}$. This value should be used in the calculation of Girard-type turbines in place of $\frac{v_0^2 - v_a^2}{2g}$.

Influence of the Draft Tube

Before we can proceed to fix the discharge cross-section of the runner we must determine the influence of the draft tube.

We will consider the draft tube limits as shown in Figs. 109, 110 and 111 to be the natural continuation of the runner crowns. In the manner already described, we will then lay out any desired number of rotating surfaces between the runner crowns and between the draft-tube limits in such a way that an equal amount of water flows between any two consecutive surfaces of rotation. At right angles to these planes of rotation we lay out any desired number of normal surfaces of intersections. In the draft tube these normal intersection surfaces are at the same time true surfaces of intersection. That is, if the draft tubes were cylinders they would be true circular cross-sections. We will now assume that similar particles of water in a given surface of intersection have approximately equal velocities at right angles to the said surfaces of intersection, that is, in the normal water threads, the latter at all points being in the periphery of circular planes. The value of this velocity in feet per second at any given point is then given by the quotient:

$$\frac{\text{Quantity of water in cubic feet per second}}{\text{Area of the intersection surface in square feet}}.*$$

* Professor Prazil, in his paper from which the above is quoted, in regard to the above assumption shows that in a rotating space without vanes the characteristic surfaces of intersection are, strictly speaking, only surfaces of equal potential and that the surfaces of equal velocity are somewhat different from these. Our assumption is, however, sufficiently accurate for practical calculations.

At the entrance to the draft tube this velocity is

$$c_a{}''' = \frac{Q}{\text{Area of surface } N_{111}N_{111}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (108)$$

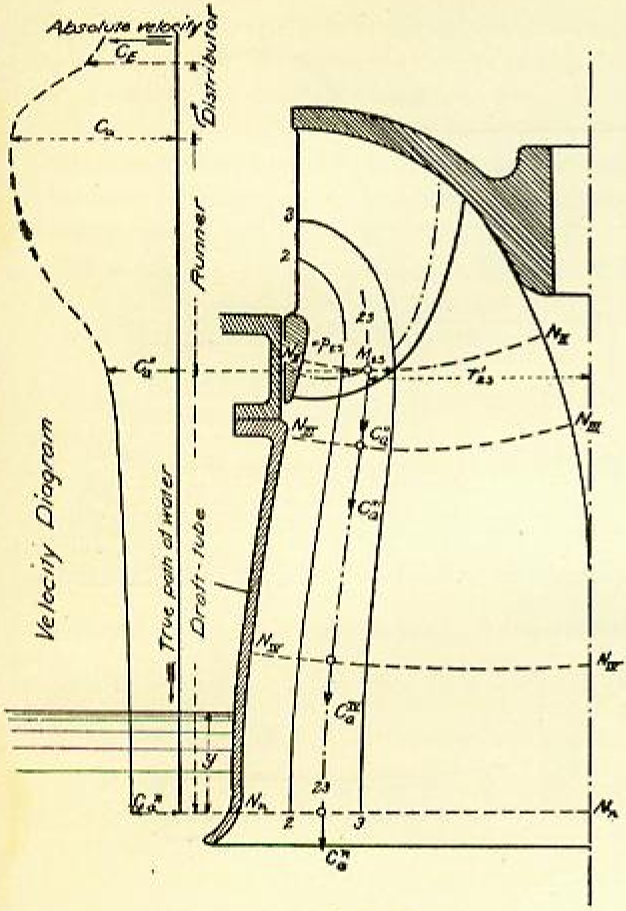

Fig. 109.

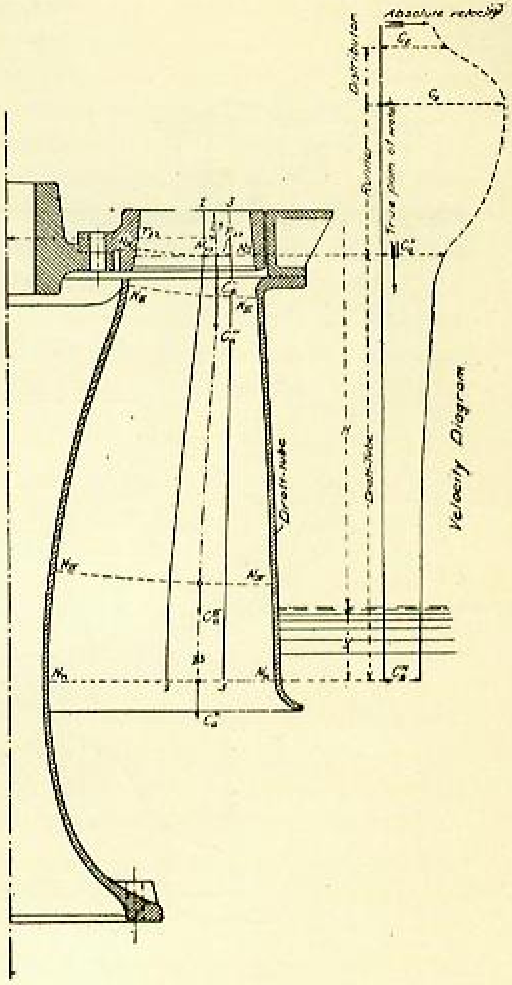

Fig. 110.

At the discharge from the draft tube it is

$$c_a{}^n = \frac{Q}{\text{Area of surface } N_nN_n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (109)$$

If we again investigate that one of the m partial turbines which is included between the rotating surfaces 2–2 and 3–3, without considering the vane thickness, it will be found by equation (50) that the velocity of a particle of water leaving the runner at the point M_{23} is

$$\frac{\frac{Q}{m}}{2\pi r'_{23}\cdot p_{23}}.$$

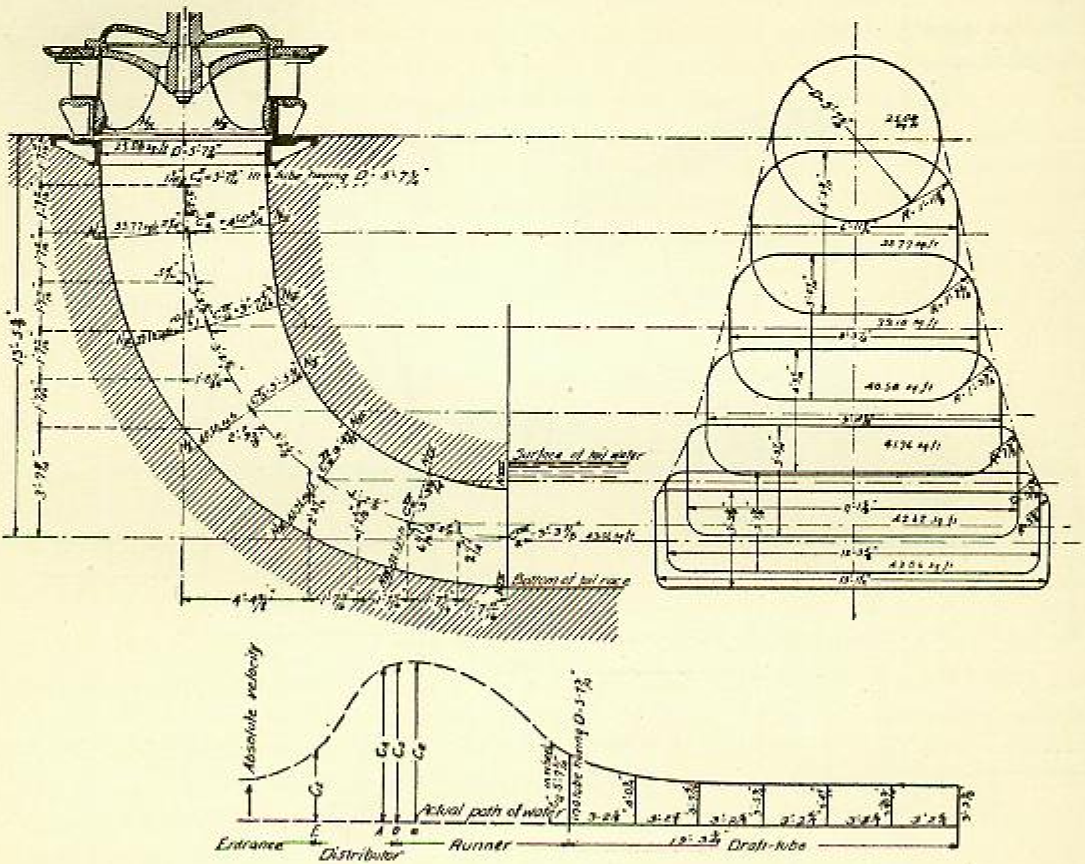

FIG. 111.

If we take into consideration the vane thickness when there are z runner buckets the velocity in the direction of the normal water thread will have the value [see Eq. (54)]

$$c_{n23}''=\frac{\frac{Q}{mz}}{\left(\frac{2\pi r'_{23}}{z}-s_{23}\frac{\sqrt{1-\sin^2 k_{23}\sin^2\gamma_{23}}}{\sin\gamma_{23}\cos k_{23}}\right)p_{23}}, \quad . \quad . \quad . \quad . \quad (110)$$

or in general, the value will be

$$c_a'' = \frac{\frac{Q}{mz}}{\left(\frac{2\pi r}{z} - s\frac{\sqrt{1-\sin^2 k \sin^2 \gamma}}{\sin \gamma \cos k}\right)p}. \quad \quad (111)$$

By properly choosing the shape of the runner crowns and that of the draft tube it makes it possible to establish a steady or smooth transition from c_a'' to c_a''' and finally to c_a^n so that we may avoid losses in pressure at the transition from the runner to the draft tube and in the draft tube itself, due to sudden changes in cross-sections. We have a wide choice of means by which to fulfill these requirements as to changes in velocity. (For illustration, compare the diagrams shown in Figs. 109, 110 and 111.) In order to have a free choice of forms, it is well to use draft tubes of concrete or cast iron which may be built in any shape. The use of concrete is advantageous, for it enables us to gradually change the direction and velocity of the water discharged from the runner to that of the tail-water canal and thus make use of the head which would otherwise be lost: i.e., $\frac{c_a^{n2}}{2g}$ (see Fig. 111). If we use draft tubes made of steel plates they must have a conical form because of the requirements of construction. If the law governing the changes in velocity is to be followed we are therefore more or less restricted in our choice of draft-tube forms.

By gradually changing the velocity c_a'' to c_a^n we tend to overcome the pressure lost in the resistance arising from the passage of the water through the draft tube, that is, the head $\frac{c_a''^2 - c_a^{n2}}{2g}$. Only a part of this lost head may be eliminated by a judicious choice of the form of the draft tube, viz.: $(t_6+t_7)H$, which is about 3 per cent – 5 per cent of the total head. The remainder is $\frac{c_a''^2 - c_a^{n2}}{2g} - (t_6+t_7)H$, which usually has a positive value and it must act to perform work. We will consider it as having the action of an injector increasing the action of the suction and will designate it as the "Sub-Vacuum." With this in mind we will consider that the suction head at M_{23}, for example, which lies at a distance of about h_{23} below the head-water surface, there exists a suction head which has a value of not only $H - h_{23}$, but

$$H - h_{23} + \frac{c_{a23}''^2 - c_a^{n2}}{2g} - (t_6+t_7)H.$$

The discharge velocity at the point M_{23} will therefore be increased by the amount due to the sub-vacuum.

This sub-vacuum is not influenced by the value of y, which represents the distance by which the draft-tube outlet is immersed below the surface of the water in the tail-race.

The Relative Discharge Velocity in the Runner Buckets. The Discharge Cross-Section

The pressure head which corresponds to the relative velocity w_a may be obtained by means of the following discussion: When the entrance to the runner is free from impulse the available pressure head is

$$\varepsilon H + (t_6 + t_7) H - \frac{c_0^2}{2g} + \frac{w_0^2}{2g}.$$

If for brevity we substitute in this equation

$$\varepsilon_0 = \varepsilon + t_6 + t_7 = 1 - (t_1 + t_2 + t_3 + t_4), \quad . \quad . \quad . \quad . \quad . \quad . \quad (112)$$

the pressure head becomes

$$\varepsilon_0 H - \frac{c_0^2}{2g} + \frac{w_0^2}{2g}.$$

As a result of the action of centrifugal force this head is reduced at the point of discharge from the runner by the amount $\frac{v_0^2 - v_a^2}{2g}$, this term being $\frac{v_e^2 - v_a^2}{2g}$ for Girard-type turbines. To this must be added the amount of the sub-vacuum, expressed in head, that is

$$\frac{c_a''^2 - c_a^{n2}}{2g} - (t_6 + t_7) H.$$

The desired pressure head for a reaction turbine with a draft tube will then have the value

$$\frac{w_a^2}{2g} = \varepsilon_0 H - \frac{c_0^2}{2g} + \frac{w_0^2}{2g} - \frac{v_0^2 - v_a^2}{2g} + \frac{c_a''^2 - c_a^{n2}}{2g} - (t_6 + t_7) H, \quad . \quad . \quad . \quad (113)$$

or from Eq. (112)

$$\frac{w_a^2}{2g} = \varepsilon H - \frac{c_0^2}{2g} + \frac{w_0^2}{2g} - \frac{v_0^2 - v_a^2}{2g} + \frac{c_a''^2 - c_a^{n2}}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (114)*$$

* If the entrance is not free from impact the head εH will be still further reduced by the amount $\frac{c_n^2}{2g}$, which corresponds to the component c_n due to impact losses.

This equation corresponds to the fundamental Eq. (84), if we substitute $\frac{c_a^2}{2g}$, the head equivalent to the discharge velocity, for the somewhat smaller value $\frac{c_a''^2 - c_a^{n2}}{2g}$.

Eq. (114) enables us to calculate the true discharge velocity w_a for any effective discharge cross-section of a partial bucket. The area of the latter when there are z vanes in the runner and m partial turbines is:

$$\text{Discharge cross-section of a partial bucket} = \frac{\frac{Q}{mz}}{w_a}. \quad . \quad . \quad . \quad . \quad (115)$$

As by the design of the discharge edge of the vane the effective breadth of the cross-section may be obtained from Eq. (35) in connection with Eqs. (62) and (64), the effective width of the bucket may now be easily calculated.

Special Cases

(*a*) Reaction turbines in which the sub-vacuum = 0. That is when

$$\frac{c_a''^2 - c_a^{n2}}{2g} = (t_6 + t_7)H.$$

For this case we then have

$$\frac{w_a^2}{2g} = \varepsilon_0 H - \frac{c_0^2}{2g} + \frac{w_0^2}{2g} - \frac{v_0^2 - v_a^2}{2g}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (116)$$

from which we may write

$$\varepsilon_0 H = \varepsilon H + (t_6 + t_7)H. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (117)$$

The value of $\varepsilon_0 H$ may in practice be assumed as 5 per cent larger than εH, as $(t_6 + t_7)H = 0.05H$ (approx.)

(*b*) Reaction turbines operating in the tail water without discharge pipe.

For this case

$$\frac{w_a^2}{2g} = \varepsilon_0 H - \frac{c_0^2}{2g} + \frac{w_0^2}{2g} - \frac{v_0^2 - v_a^2}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (118)$$

From which we have

$$\varepsilon_0 H = \varepsilon H + \frac{c_a^2}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (119)$$

(*c*) Girard-type turbines with draft tubes.

By substituting v_e for v_0 and making

$$\frac{c_0^2}{2g}=\varepsilon_0 H, \qquad (120)$$

we have

$$\frac{w_a^2}{2g}=\frac{w_0^2}{2g}-\frac{v_e^2-v_a^2}{2g}+\frac{c_a''^2-c_a^{n2}}{2g}-(t_6+t_7)H. \qquad (121)$$

In special cases where

$$\frac{c_a''^2-c_a^{n2}}{2g}=(t_6+t_7)H,$$

it follows that

$$\frac{w_a^2}{2g}=\frac{w_0^2}{2g}-\frac{v_e^2-v_a^2}{2g}. \qquad (122)$$

(*d*) Girard-type turbines without draft tubes. In this case

$$\frac{w_a^2}{2g}=\frac{w_0^2}{2g}-\frac{v_e^2-v_a^2}{2g}.$$

In all of the preceding cases we have considered the losses in the runner to have the values ε_0, $\frac{c_0^2}{2g}$ and $\frac{w_0^2}{2g}$, but in the case of axial turbines with vertical shafts we must also take into consideration the value of the height of the runner.

Determination of the Discharge Angle γ and the Discharge Velocity c_a

The velocity c_a'' in the direction of the normal water thread at the discharge from the runner is determined by means of Eq. (111) as

$$c_a''=\frac{\frac{Q}{mz}}{\left(\frac{2\pi r'}{z}s-\frac{\sqrt{1-\sin^2 k\sin^2\gamma}}{\sin\gamma\cos k}\right)p}.$$

From Fig. 103 we have:

$$\sin\gamma=\frac{c_a''}{w_a}=\frac{\frac{Q}{mz}}{w_a\left(\frac{2\pi r'}{z}-s\frac{\sqrt{1-\sin^2 k\sin^2\gamma}}{\sin\gamma\cos k}\right)p}. \qquad (123)$$

From this equation γ, the only unknown quantity may be determined by trial. All the remaining are given by the drawing or may be chosen at will, except w_a, which may be determined by Eq. (114).

As a first approximation for the value of γ, we may of course use the result of the equation

$$\text{Sin } \gamma = \frac{\frac{Q}{m}}{2\pi r' - w_a \cdot p}. \quad \ldots \ldots \ldots \quad (124)$$

As soon as γ has been calculated nearly enough, we use then the equation (see Fig. 103)

$$c_a = \sqrt{v_a^2 + w_a^2 - 2v_a w_a \cos \gamma}, \quad \ldots \ldots \ldots \quad (125)$$

and finally as the last unknown quantity

$$\cos (\delta - 90°) = \frac{c_a''}{c_a}. \quad \ldots \ldots \ldots \quad (126)$$

The value of δ here used must agree with that obtained from the fundamental equation.

The problem may be solved graphically as follows (see discharge diagram, Figs. 125, 126 and 127):

At a distance c_a'' from v_a draw a line parallel to v_a, the value of c_a'' being calculated from Eq. (111), the angle γ being assumed to have the value given by Eq. (124). From M let an arc be struck with a radius w_a, intesecting the first drawn line at B, w_a being calculated by Eq. (114). AMB is now the correct angle γ from which c_a'' may be determined afresh and correctly. c_a and δ are found by direct measurement from the corresponding velocity triangle.

Directions for Drawing the Normal Surfaces of Intersection and Rotation Surfaces

We will consider the limiting lines of the distributor and runner buckets and of the draft tube to be given, these having been so selected that at the points of transition from one to the other continuity obtains in respect to the area of the normal surfaces of intersection. (See Figs. 109, 110 and 111.)

In Fig. 112 the limits of the runner buckets are shown by lines only. Now in order to find a normal intersection line let the point $O_,$ be chosen as the center of a circle which is tangent to the bucket limits at the points $N_,$ and $N'_,$. The lines connecting $O_,$ with $N_,$ and $N'_,$ are then tangents to the desired normal line of intersection. We may draw the intersection line $N_,$ $N'_,$ freehand, subject to correction later. In a similar manner we may determine approximately the normal intersection lines $N_{,,} - N'_{,,}$ and $N_{,,,} - N'_{,,,}$. The line $N_, N'_,$ is then divided by locating the point $A_,$ so that

$$\text{Surface } N_, A_, \cdot 2\pi r_2 = N'_, A_, \cdot 2\pi r_{,,} \quad \ldots \ldots \quad (127)$$

or in simple form

$$N_{,}A_{,}r_2 = N'_{,}A_{,}r_{,,} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (128)$$

where $r_{,}$ and r_2 are the respective distances from the turbine axis to the middle points of the lines $N_{,}A$ and $N'_{,}A$. A is then a point on that surface of rotation which divides the space between the runner crowns into two parts such that through each there flows an equal quantity of water. In a similar manner we may determine the points $A_{,,}$ and $A_{,,,}$. The connecting line $A_{,}\ A_{,,}\ A_{,,,}$ is then the projection of the mean rotating surface. In accordance with the construction method above described we may now consider that the line $A_{,}\ A_{,,}\ A_{,,,}$ represents a new bucket limit to which the lines $N_{,}N'_{,,}$ $N_{,,}N'_{,,,}$ etc., are at right angles.

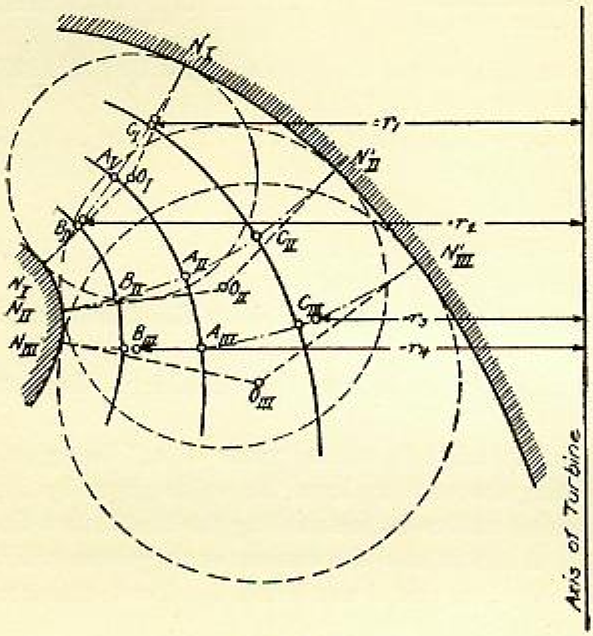

FIG. 112.

The surfaces $N_{,}A_{,}2\pi r_2$ and $N_{,}A'_{,}2\pi r_{,,}$ may in a similar manner be divided into parts and the points $B_{,}$ and $C_{,}$ thus obtained. On the surfaces of intersection $N_{,,}N'_{,,}$ and $N_{,,,}N'_{,,,}$ the points $B_{,,}$, $C_{,,}$, $B_{,,,}$ and $C_{,,,}$ may similarly be located and thus obtain the new rotating surfaces B, $B_{,,}$, $B_{,,,}$ and $C_{,}$, $C_{,,}$, $C_{,,,}$ from which we may if necessary make a still further correction of the chosen normal intersection line.

This operation may be repeated until the bucket is sufficiently subdivided. The number of partial buckets in is always a multiple of 2, if determined by the simple method just described.

By this method one is able by practice to obtain the intersection line the first time so exactly that generally no further correction is necessary.

For examples for the application of this method see Figs. 125, 126, 127 and 128. It is particularly interesting in connection with Francis turbines as shown in Figs. 126 and 127, to note the method of obtaining the smallest surfaces of intersection on which the components of the velocity obtain their greatest values in the direction of the normal water threads.

CONSTRUCTION OF TURBINE WHEELS

Full Radial Turbines with Inward Discharge—Francis Turbines

The Amount of the Water Consumption as Determined by Practice. The area of the first surface of intersection of the runner has the value $\pi D \times B$, when the diameter of the runner is D and B is the breadth (or height) of the distributor, both measured in feet (see Fig. 113). If the water were to enter at right angles to this surface, that is, if it flows radially (which would be the case if there were no vanes in the distributor or runner) then the velocity in feet c_p with a consumption of Q cubic feet of water is expressed by the equation

$$c_p = \frac{Q}{\pi D \cdot B} = k_{cp}\sqrt{2gh}. \qquad (129)$$

We will designate k_{c_p} as the "Requirement Factor" of the turbine. It forms at the same time a measure of the entrance angle α, for we have found by Eq. (92) that

$$c_0 = k_{c_0}\sqrt{2gh},$$

and from Fig. 113 we see that

$$c_p = c_0 \sin \alpha, \qquad (130)$$

$$\therefore \; k_{cp} = k_{c_0} \sin \alpha \text{ or } \sin \alpha = \frac{k_{c_p}}{k_{c_0}}. \qquad (131)^*$$

By assuming that the discharge from the wheel is parallel to the shaft there follows from Eqs. (97) and (131)

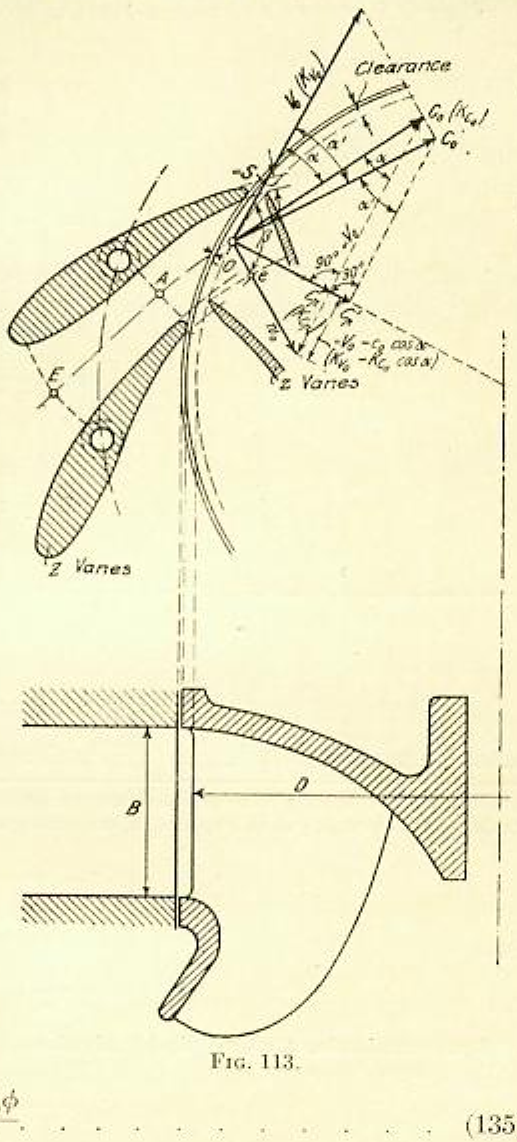

FIG. 113.

$$\tan \alpha = \frac{2k_{c_p}\phi}{\varepsilon}. \qquad (135)$$

* Strictly speaking, these relations are subject to change due to the influence of the thickness of the distributor vanes, while the influence of the runner vanes may be neglected if they are properly sharpened

If the thickness of the distributor vanes is considered we have the corresponding equation

$$\tan \alpha' = \frac{2\phi k_{c_0}}{\varepsilon}\left(\frac{k_{cp}}{k_{c_0}} + \frac{ZS}{\pi D}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (136)$$

Now it is clear that with a turbine which consumes a large amount of water this

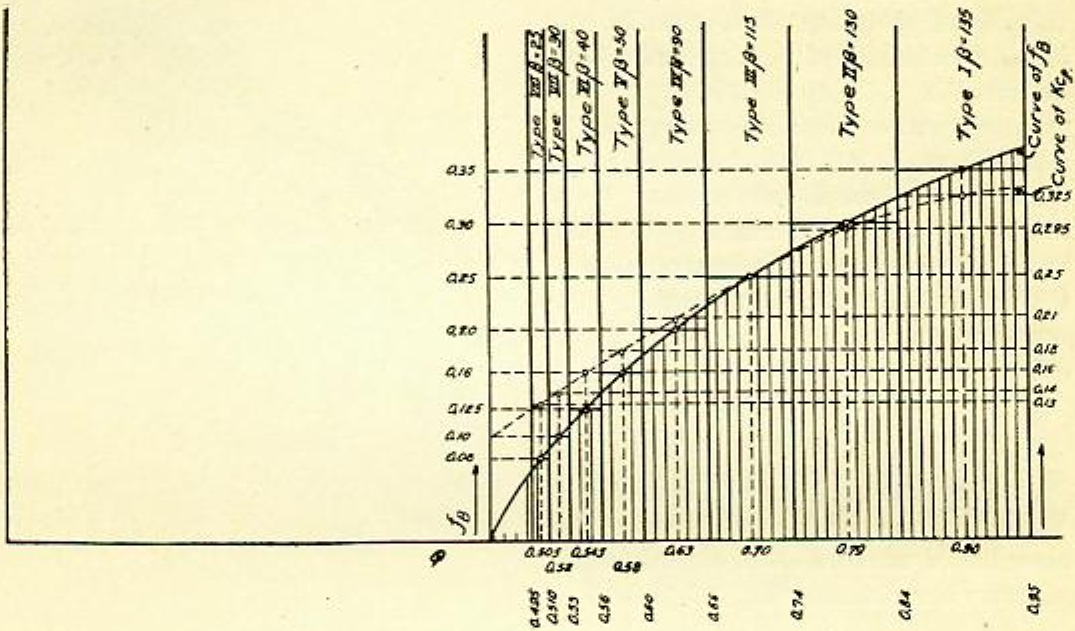

FIG. 114.

factor k_{cp} must be very large. This is the case with the so-called "high-speed" runners.

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in a long bevel. Taking into consideration the thickness S of each of the Z distributor vanes and using the designations employed in Fig. 113 we may determine the more exact entrance angle α' and velocity c'_p.

$$c'_p = \frac{Q}{B \cdot \pi D - BZ\dfrac{S}{\sin \alpha'}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (132)$$

and

$$\sin \alpha' = \frac{c'_p}{c_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (133)$$

By the elimination of c'_p and the introduction of the requirement factor k_{cp}, whose value is determined by Eq. (129) we have

$$k_{cp} = \frac{Q}{B \cdot \pi D\sqrt{2gH}},$$

and finally obtain

$$\sin \alpha' = \frac{k_{cp}}{k_{c_0}} + \frac{ZS}{\pi D}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (134)$$

Conversely, k_{c_p} will be small for slow-speed turbines, i.e., in the case of turbines for high heads and low degree of reaction.

As the result of experience we will select this factor k_{c_p} as a function of the peripheral velocity factor ϕ in accordance with the broken line curve shown on Fig. 114. In this way we may obtain the corresponding values for the eight usual types of turbines defined later.

Type.	VIII	VII	VI	V	IV	III	II	I
$\phi=$	0.505	0.520	0.545	0.580	0.630	0.700	0.790	0.900
$k_{c_p}=$	0.110	0.120	0.140	0.170	0.205	0.250	0.295	0.325

By the use of an average value of $\varepsilon=0.80$ and without consideration of the thickness of the distributor vanes, the entrance angle α may then be determined by Eq. (135) to be as follows:

Type.	VIII	VII	VI	V	IV	III	II	I
$\alpha=$	8°	8° 50′	10° 50′	13° 50′	18°	23° 40′	29° 30′	36° 20′
$\cos\alpha=$	0.990	0.988	0.982	0.971	0.951	0.916	0.870	0.806

With a runner discharge parallel to the shaft k_{c_0} then has the following values according to the fundamental Eq. (97).

Typ	VIII	VII	VI	V	IV	III	II	I
$k_{c_0}=$	0.80	0.78	0.75	0.71	0.67	0.62	0.58	0.55

Then for an entrance angle free from impulse the vane angle β will be as follows:

Type.	VIII	VII	VI	V	IV	III	II	I
$\beta=$	21°	25° 30′	36° 20′	57° 20′	88° 0′	117° 50′	134° 0′	144° 30′

Instead of using Eq. (104*a*) we may calculate β from the simpler formula

$$\operatorname{tang}(\beta-90°)=\frac{\phi-k_{c_0}\cos\alpha}{k_{c_p}}, \quad \ldots \ldots \ldots \quad (137)$$

whose correctness may be readily seen from Fig. 98.

Similarly we may determine α from the equation

$$\cot \alpha = \frac{k_{c_0} \cos \alpha}{k_{c_p}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (138)$$

In both these cases the value of $k_{c_0} \cos \alpha$ may be determined as a whole from Eq. (97). The values of α and β thus obtained are shown by the broken line curves of Fig. 115

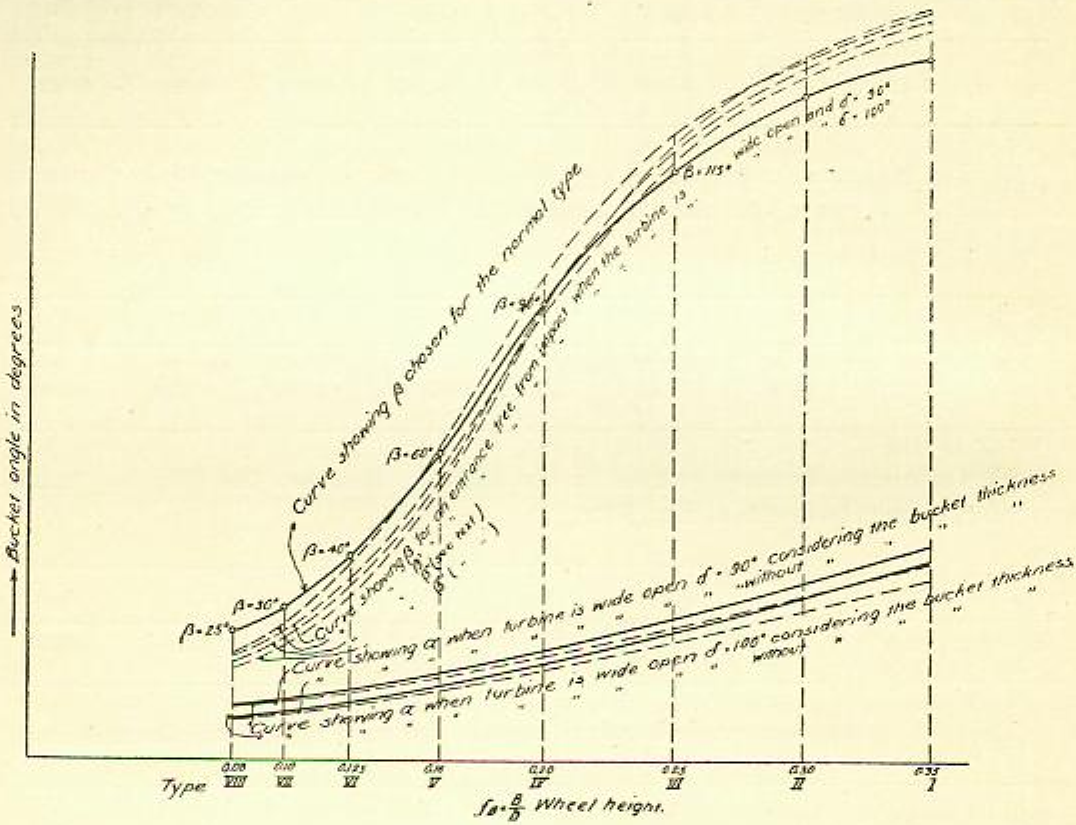


FIG. 115

as functions of the wheel breadth, or what is more commonly known as the height of the runner.

We will refer further on to the values of α, k_{c_0} and β which obtain when the runner discharge is not parallel to the shaft.

Choice of the Height of the Runner Entrance. In order to determine the diameter of the wheel it is necessary to assume the relation of the runner height B to the diameter D. This relation, for practical reasons, must be such that for a given head if the peripheral velocity is high a large quantity of water will be used, and conversely if the peripheral velocity is low a small quantity of water will be used.

If we let

$$B = f_B \cdot D \qquad (139)$$

then in accordance with the above f_B must be chosen as a function of ϕ. The interdependence of these two values is shown by the curve in Fig. 114 in which they are so selected as to fulfill the requirements of good practice.

However, it would be irrational in building turbines to fix upon a new runner diameter and to build a new model for every different peripheral velocity and every change in reaction. Furthermore it is well to choose the same runner diameter for a certain range of changes in reaction and thus to reduce to a minimum the number of wheels built with a given diameter. It is also wise to so determine the factor f_B that the heights of the runner have gradually changing values.

We will accordingly select for each runner diameter a certain corresponding height of runner for each of the eight types of wheels referred to in the following table and also indicated on Fig. 114.

Type.	$B = f_B D$	f_B.	Peripheral Velocity Factor ϕ.	Turbine Classification.
VIII	0.080 D	0.080	0.495 – 0.51	Girard-type and limit turbines
VII	0.100 D	0.100	0.51 – 0.53	Slowest-speed turbines
VI	0.125 D	0.125	0.53 – 0.56	Slow-speed turbines
V	0.160 D	0.160	0.56 – 0.60	Slow-speed normal turbines
IV	0.200 D	0.200	0.60 – 0.66	Normal turbines
III	0.250 D	0.250	0.66 – 0.74	High-speed normal turbines
II	0.300 D	0.300	0.74 – 0.84	High-speed turbines
I	0.350 D	0.350	0.84 – 0.96	Highest speed turbines

With these eight types, which may be called the normal types, we are in a position to fulfill all the practical requirements as to quantity of water, pressure head, and number of revolutions which may arise in practice. This is especially true because we may divide among two, three, or more wheels secured to the same shaft the quantity of motor water required and thus increase the means at our disposal for securing the desired result.

If, nevertheless, one desires to build abnormal wheels, the calculations and construction of the same do not present any special difficulties, but it must be remembered that abnormal models are not economical and that the construction of such turbines undoubtedly requires a larger amount of work and time than that of normal types.

Of the eight normal types, Types II, III, IV, and sometimes V, are to be preferred and are most used for low and medium heads. Type I is best fitted for high-speed wheels operating under low heads. In respect of water consumption and speed this type is not excelled even by the American high-speed wheels.

Types VI, VII and VIII are intended for high heads and small quantities of water.

As a rule, in such cases a low relative speed is desired and these types provide for accomplishing this result.

Water Consumption with the Normal Types. To reduce the water consumption of a given wheel to that which would be used under a head of one foot, we have in cubic feet

$$Q_{,}=\frac{Q}{\sqrt{H}}. \qquad (140)$$

For a wheel with a diameter D and a height at entrance $B=f_B D$, both expressed in feet, the above-mentioned quantity of water in cubic feet will then be

$$Q_{,}=\pi D^2\cdot k_{c_p}\cdot f_B\cdot\sqrt{2g}=25.196 k_{c_p} f_B D^2. \qquad (141)$$

If a normal-type wheel is not chosen, then k_{c_p} and fB should each be chosen independently so that, if possible, their values shall agree with those shown on the curve in Fig. 114.

The following table has been prepared to show the consumption of water in cubic feet for the several normal types of turbines, the corresponding values of k_{c_p} and f_B being first given:

Type.	VIII	VII	VI	V	IV	III	II	I
k_{cp} =	0.11	0.12	0.14	0.17	0.205	0.25	0.295	0.325
f_B =	0.08	0.10	0.125	0.16	0.20	0.25	0.30	0.35
$Q_{,}$ =	$0.222D^2$	$0.302D^2$	$0.441D^2$	$0.685D^2$	$1.033D^2$	$1.575D^2$	$2.230D^2$	$2.866D^2$

TABLE SHOWING THE QUANTITY OF WATER $Q_{,}$ IN CUBIC FEET FOR ONE WHEEL, THE HEAD BEING ASSUMED AS 1 FOOT.

D in Inches.	VIII	VII	VI	V	IV	III	II	I
10	0.154	0.210	0.306	0.476	0.717	1.094	1.549	1.990
12	0.222	0.302	0.441	0.685	1.033	1.575	2.230	2.866
14	0.302	0.411	0.600	0.932	1.406	2.144	3.035	3.901
16	0.395	0.537	0.784	1.218	1.835	2.800	3.964	5.095
18	0.499	0.679	0.992	1.541	2.324	3.544	5.017	6.448
20	0.617	0.839	1.225	1.903	2.869	4.375	6.194	7.961
24	0.888	1.208	1.764	2.740	4.132	6.300	8.920	11.46
28	1.209	1.644	2.401	3.729	5.624	8.575	12.14	15.60
32	1.578	2.148	3.136	4.871	7.346	11.20	15.86	20.38
36	1.998	2.718	3.969	6.165	9.297	14.17	20.07	25.79
40	2.467	3.356	4.900	7.611	11.48	17.50	24.77	31.84
48	3.552	4.832	7.056	10.96	16.53	25.20	35.68	45.86
56	4.835	6.577	9.604	14.92	22.50	34.30	48.56	62.41
63	6.119	8.324	12.15	18.88	28.47	43.41	61.46	78.99
72	7.992	10.87	15.88	24.66	37.19	56.70	80.28	103.2
80	9.867	13.42	19.60	30.44	45.91	70.00	99.11	127.4
96	14.20	19.33	28.22	43.84	66.11	100.8	142.7	183.4
120	22.20	30.20	44.10	68.50	103.3	157.5	223.0	286.6
138	29.36	39.94	58.32	90.59	136.6	208.3	294.9	379.0
156	37.52	51.04	74.53	115.76	174.6	266.2	376.9	484.4

Speed of Wheels of the Normal Type. Revolutions per Minute. The number of revolutions per minute of a radial turbine of Diameter D, measured in feet, is determined by the formula:

$$\eta=\frac{60\phi\sqrt{2gH}}{\pi D}=19.1\,\frac{\phi\sqrt{2gH}}{D}, \quad \text{(142)}$$

and if calculated for a head of 1 foot

$$\eta_{\prime}=153.2\frac{\phi}{D}. \quad \text{(143)}$$

If no restriction is placed upon the speed, ϕ is usually chosen as about 0.66 in order to obtain a turbine of the normal types III or IV, which give the most satisfactory results, taking into consideration the various relations. However, it is often necessary to depart from this rule in the case of low or high heads on account of abnormal quantities of water. The normal types of wheels are intended to meet this condition by producing slow-speed wheels with the smallest number of revolutions per minute for high heads and the smallest quantities of water as well as high-speed wheels having the greatest possible number of revolutions per minute for low heads and large quantities of water.

The following table shows in the case of each type the relations between the coefficient of peripheral velocity ϕ, and the R.P.M.

Type.	Coefficient of Peripheral Velocity, ϕ.	Mean Value of ϕ.	Corresponding Number of Revolutions per Minute, $\eta_{\prime}$.	Mean Value of $\eta_{\prime}$.
VIII	$\phi=0.495$ to 0.51	$\phi=0.502$	$\eta_{\prime}=\frac{75.8}{D}$ to $\frac{78.1}{D}$	$\eta_{\prime}=\frac{76.9}{D}$
VII	$=0.51$ to 0.53	$=0.52$	$=\frac{78.1}{D}$ to $\frac{81.2}{D}$	$=\frac{79.7}{D}$
VI	$=0.53$ to 0.56	$=0.545$	$=\frac{81.2}{D}$ to $\frac{85.8}{D}$	$=\frac{83.5}{D}$
V	$=0.56$ to 0.60	$=0.580$	$=\frac{85.8}{D}$ to $\frac{91.9}{D}$	$=\frac{88.9}{D}$
IV	$=0.60$ to 0.66	$=0.630$	$=\frac{91.9}{D}$ to $\frac{101.1}{D}$	$=\frac{96.5}{D}$
III	$=0.66$ to 0.74	$=0.70$	$=\frac{101.1}{D}$ to $\frac{113.4}{D}$	$=\frac{107.2}{D}$
II	$=0.74$ to 0.84	$=0.79$	$=\frac{113.4}{D}$ to $\frac{128.7}{D}$	$=\frac{121.0}{D}$
I	$=0.84$ to 0.96	$=0.90$	$=\frac{128.7}{D}$ to $\frac{147.1}{D}$	$=\frac{137.9}{D}$

In this formula Q may be expressed in terms of the diameter D by means of Eq. (145), i.e.

$$Q = f_B \pi D^2 k_{c_p} \sqrt{2gH}$$

If in the last formula we substitute for D its value in terms of H and η by the use of Eq. (144) we obtain

$$P = 0.114\varepsilon H f_B k_{c_p} \pi \frac{\phi^2 \overline{2gH}^{\frac{3}{2}}}{\frac{\pi^2 \eta^2}{60^2}} = 0.114 \frac{\overline{60}^2}{\pi} \overline{2g}^{\frac{3}{2}} \varepsilon f_B k_{c_p} \phi^2 \frac{H^{\frac{5}{2}}}{\eta^2}.$$

Noting that ε varies with the diameter of the turbine as well as with the head, it will be seen that the coefficient of $\frac{H^{\frac{5}{2}}}{\eta^2}$ is a constant for a given type of turbine. It may be expressed by K_n, which thus becomes the "characteristic coefficient" or "specific speed" of the wheel.

We then have

$$K_n = 130.6\,\overline{2g}^{\frac{3}{2}} \varepsilon f_B k_{c_p} \phi^2$$

and

$$P = K_n \frac{H^{\frac{5}{2}}}{\eta^2},$$

or

$$K_n = \eta^2 \frac{P}{H^{\frac{5}{2}}},$$

K_n is then a measure of the speed of the wheel, a large value indicating a high speed and a small value a low speed. If, for example, we assume a runner diameter of 39 inches and a total head of 33 feet, we may consider $\varepsilon = 0.81$ for Type I, $\varepsilon = 0.85$ for Type IV, and $\varepsilon = 0.80$ for Type VIII. Then using for f_B, k_{c_p} and ϕ, the values hereinbefore given as characteristic of the several types, we find the following values for K_n:

For Type	I	$-K_n = 5030$
" "	IV	$-K_n = 933$
" "	VIII	$-K_n = 120$

Calculation of any Radial Turbine. In case it is not desirable to make use of the tables hereinbefore given and assuming that the speed is fixed as well as H and Q (thus giving Q_1 and η_1), we may proceed as follows:

First ϕ is chosen and the runner diameter obtained from the formula

$$D = \frac{\phi\sqrt{2gH}}{\frac{\eta}{60}} = 153.1\frac{\phi}{\eta_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (144)$$

Then the runner height B in feet is

$$B = f_B D = \frac{Q}{\pi D \cdot k_{c_p} \sqrt{2gH}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (145)$$

An arbitrary assumption may now be made as to the relative values of f_B and k_{c_p}. The most satisfactory result will be attained (see Fig. 114) by making

$$f_B \cong k_{c_p}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (146)$$

We then have

$$k_{c_p} = \sqrt{\frac{Q}{\pi D^2 \cdot \sqrt{2gH}}} = \frac{0.199\sqrt{Q_I}}{D}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (147)$$

and by Eq. (145)

$$B = f_B D = k_{c_p} D = 0.199\sqrt{Q_I}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (148)$$

In all of the above formulæ B and D are expressed in feet and Q and Q_I in cubic feet.

Assuming a right-angled discharge from the runner and omitting consideration of the thickness of the distributor vanes, we may then by fixing ε obtain the entrance angle α by the following equations already developed:

$$\tan \alpha = \frac{2k_{c_p}\phi}{\varepsilon},$$

and

$$k_{c_0} = \frac{\varepsilon}{2\phi \cos \alpha},$$

as well as

$$\tan(\beta - 90°) = \frac{\phi - k_{c_0} \cos \alpha}{k_{c_p}}.$$

Determination of the Vane Angles α and β for the Normal Types Under the Condition of a Leading Discharge Diagram. Final Choice of these Angles.— The relations between the angles α and β when the runner discharges at right angles (i.e. $\delta = 90°$) have already been considered in the previous chapters describing the normal types. It now remains for us to apply our calculations to the case of a turbine having a leading discharge diagram (i.e. δ greater than 90°) at full gate opening.

In the latter case the angles α and β may be so selected that the entrance will be free from impact when the turbine is wide open, or on the other hand they may be so fixed that the entrance will be free from impact when the water leaves the runner at right angles; that is, when the turbine is not wide open. The latter method is preferred in practice and gives the highest efficiency at some partial gate opening. It is usually so arranged that such maximum efficiency is obtained at that position of the distributor which corresponds to about three-quarters of the maximum power output.

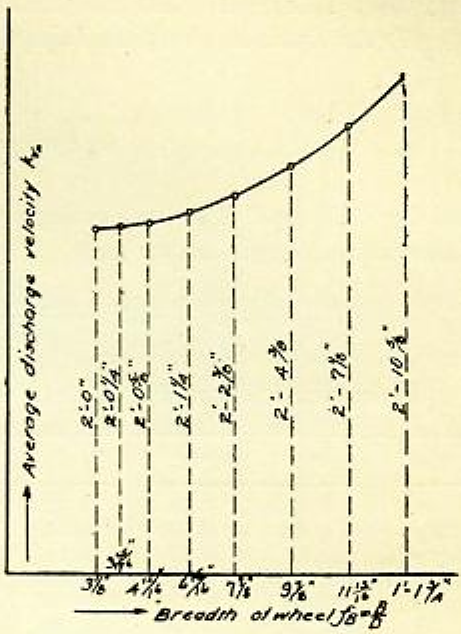


Fig. 116.

In order to apply the fundamental Eq. (95) to cases of normal types where the discharge is not at right angles, it is necessary to make special assumptions in regard to the value k_{v_a}, i.e., the coefficient of peripheral velocity at the point of discharge—and k_{c_a}'', the coefficient of velocity in the direction of the normal water thread at discharge.

The following table shows the coefficient of peripheral velocity, k_{v_a}, at the discharge on the middle water thread, as a function of the coefficient of entrance peripheral velocity, the values given in the table corresponding approximately with the graphical representation in Fig. 118.

Type.	VIII	VII	VI	V	IV	III	II	I
k_{v_a}	$0.61\ \phi$	$0.615\ \phi$	$0.625\ \phi$	$0.64\ \phi$	$0.67\ \phi$	$0.72\ \phi$	$0.79\ \phi$	$0.88\ \phi$

These values are shown as a function of the runner height in Fig. 116.

In regard to k_{c_a}'' we will make the assumption—which is approximately correct—that

$$k_{c_a}'' \simeq k_{c_p},^* \qquad (149)$$

that is to say, the values of the radial inflow are similar to those of the discharge in the direction of the normal water-thread. We may then fix the values of k_{c_a}'' as follows for the several types:

Type.	VIII	VII	VI	V	IV	III	II	I
$k_{c_a}''=$	0.11	0.12	0.14	0.17	0.205	0.25	0.295	0.325

* If it is desired to obtain exact values of k_{c_a}'', see the table on page 179, prepared from Fig. 118. However, the angles α and β may be determined with sufficient accuracy under the assumption stated in Eq. (149).

With the values now given we may determine the angles α and β by means of Eqs. (95), (104*a*), (137), and (138) under the following conditions:

(1) *An Entrance Free from Impact at Full Gate Opening.* Then by Eq. (95)

$$k_{c_0} \cos \alpha = \frac{\frac{\varepsilon}{2} + k_{v_a} k_{c_a}'' \cot \delta}{\phi},$$

and by (137) and (138) as developed from Fig. 113, the thickness of the distributor vanes being disregarded, we have

$$\tan (\beta - 90°) = \frac{\phi - k_{c_0} \cos \alpha}{k_{c_p}} \quad \text{and} \quad \cot \alpha = \frac{k_{c_0} \cos \alpha}{k_{c_p}}.$$

Fixing ε as 0.80 and δ as $90° + 10° = 100°$, the following table has been prepared:

Type.	VIII	VII	VI	V	IV	III	II	I
ϕ	0.502	0.52	0.545	0.580	0.630	0.70	0.79	0.90
k_{v_a}	0.306	0.320	0.341	0.371	0.422	0.504	0.624	0.792
$k_{c_a}'' = k_{c_p}$	0.11	0.12	0.14	0.17	0.205	0.25	0.295	0.325
$k_{c_0} \cos \alpha$	0.785	0.756	0.719	0.671	0.610	0.539	0.465	0.394
β	21° 10′	27° 00′	38° 50′	61° 50′	95° 40′	122° 50′	137° 50′	147° 20′
α	8° 10′	9° 00′	11° 00′	14° 10′	18° 30′	24° 50′	32° 20′	39° 30′

(2) *An Entrance Free from Impact at the Exact Degree of Gate Opening at which a Right-Angled Discharge is Obtained (in the Direction of the Normal Water Thread)*

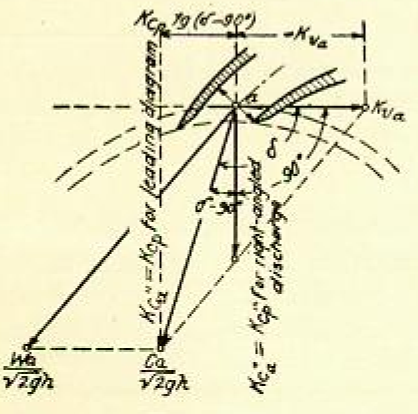


FIG. 117.

The discharge angle δ being greater than 90°, c_a'' has a certain value corresponding to a full gate opening, and as the distributor vanes are gradually closed this value is reduced to that which corresponds to a right-angled discharge, as shown on Fig. 117. At the same time the impact component is also reduced to a new and smaller value c_p''. By retaining the assumption on which Eq. (149) is based, we may determine from Fig. 117 the new value of k_{c_a}'' and the equal value of k_{c_p}'' by the equation

$$k_{c_p}'' = k_{c_p} \frac{k_{v_a}}{k_{v_a} + k_{c_p} \tan (\delta - 90°)}. \qquad (150)$$

In this equation k_{c_p} represents the capacity factor and δ the selected discharge angle which characterizes the turbine at full gate opening.

If, for example, we assume δ as 100°, we obtain the following values of k_{c_p}'' for the eight normal types of turbines.

Type.	VIII	VII	VI	V	IV	III	II	I
k_{c_p}''	0.104	0.113	0.131	0.157	0.189	0.230	0.272	0.303

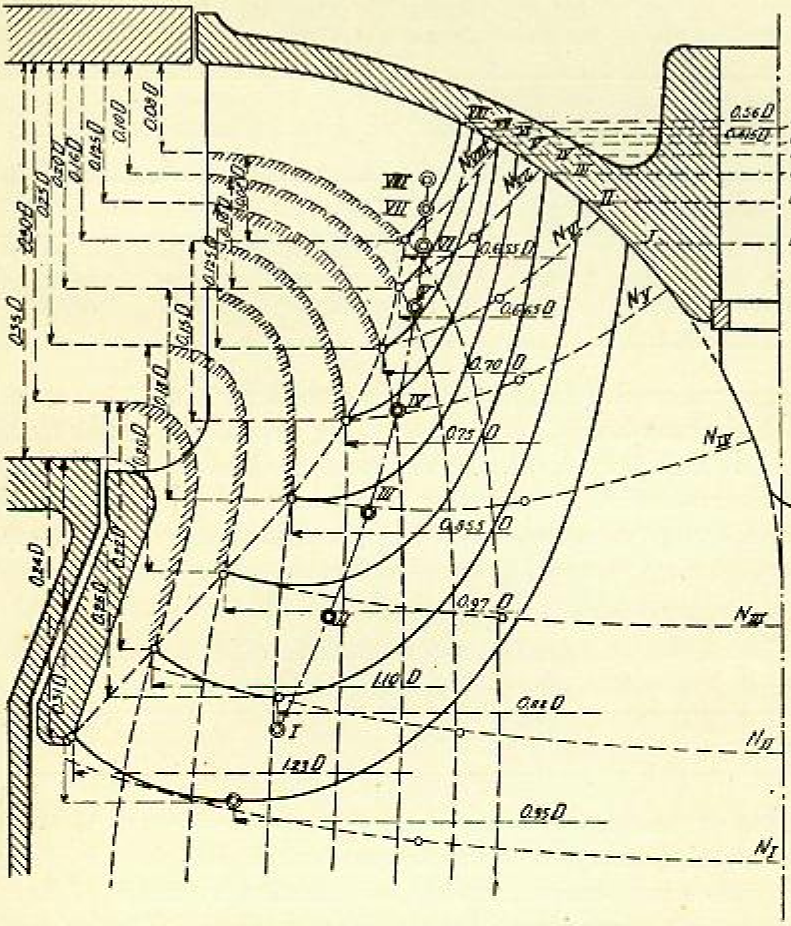


FIG. 118.

Assuming as before that $\varepsilon = 0.80$ and designating by $k_{c_0}'' \cos \alpha''$ the value of $k_{c_0} \cos \alpha$ which obtains when the distributor vanes are so set as to give a right-angled discharge, we have by the fundamental Eq. (97)

$$k_{c_0}'' \cos \alpha'' = \frac{\varepsilon}{2k_{c_0}},$$

Finally α'' and β'' may be found by the use of Eqs. (137) and (138) by substituting for k_{c_p} and $k_{r_0}\cos\alpha$ respectively, k_{c_p}'' and $k_{c_0}''\cos\alpha''$.

We thus obtain the following:

Type.	VIII	VII	VI	V	IV	III	II	I
$k_{c_0''}\cos\alpha''$	0.797	0.769	0.734	0.690	0.635	0.571	0.506	0.444
β''	19° 30′	24° 30′	34° 40′	55° 00′	88° 30′	119° 30′	136° 10′	146° 20′
α''	7° 20′	8° 20′	10° 10′	12° 50′	16° 40′	22° 00′	28° 20′	34° 20′

If $\delta = 110°$ a similar method of calculation will give the following values:

Type.	VIII	VII	VI	V	IV	III	II	I
k_{c_p}'''	0.0973	0.106	0.122	0.146	0.174	0.212	0.252	0.283
β'''	18° 20′	23° 00′	32° 50′	53° 00′	88° 20′	121° 20′	138° 20′	148° 10′
α'''	7° 00′	7° 50′	9° 30′	12° 00′	15° 20′	20° 20′	26° 30′	32° 30′

The calculated values which we have found for the angle β are laid out in Fig. 115 as a function of the runner height, the entrance being assumed to be free from impact. Considering the fact that the thickness of the vanes produces its most harmful effect upon the entrance cross-section of the runner when the angle β approaches either zero or 180°, and remembering that a small impact loss is of practically little importance, we have chosen the following values of β in round figures and have plotted the same on Fig. 115.

Type.	VIII	VII	VI	V	IV	III	II	I
β	25°	30°	40°	60°	90°	115°	130°	138°

Upon Fig. 115 are shown the values of α calculated for the several normal types both for a right-angled discharge with a wide-open turbine and also for the case when $\delta = 100°$. These values are, however, subject to correction by means of Eq. (134), as the thickness of the distributor vanes have not been considered. The values of α corrected for the case of $Z = 16$ and $s = \dfrac{D}{125}$ are also shown on Fig. 115. After the angle α has been calculated it is necessary for the designer to accurately determine the distributor cross-section so that it will equal the quantity of water divided by the entrance velocity. The best way to proceed is to calculate the entrance velocity from the fundamental Eqs. (95) and (97), using the value of α as already determined. Having the cross-section of the distributor, we may assume the height and thus obtain the clearance required. The thickness of the distributor vanes having been graphically assumed, we may then obtain a corrected value of the angle α and from it a new and more accurate determination of the entrance velocity and so on.

In order to insure the capacity of the distributor for the maximum amount of water it will ever be required to handle it is desirable to make its cross-section about 10 per cent greater than it would be determined by calculation. On the other hand the cross-section of the runner should be maintained strictly in accordance with the calculated area. (See page 139).

Graphical Representation of the Normal Types. In order to provide a guide for choice of the discharge edge of the runner buckets, we have shown on Fig. 118 certain curves which are applicable to the eight normal types, the radius being assumed as 3.28 feet. The drawing shows that the following approximate relations exist between the capacity factors k_{c_p} and the several values of k_{c_a} measured just behind the discharge edge of the runner.

Type.	VIII	VII	VI	V	IV	III	II	I
When k_{c_o}=	0.11	0.12	0.14	0.17	0.205	0.25	0.295	0.325
Then $k_{c_a''}\cong$	0.13	0.14	0.16	0.19	0.23	0.26	0.28	0.29

It will be noticed from the above table that with Type VIII the discharge velocity in the direction of the normal water thread is somewhat greater than the entrance velocity, while in the case of Type I the opposite condition obtains. In Types I, II, III and IV this velocity component increases considerably about half way through the runner, because at that point the area of the normal intersection surface becomes a minimum. In order that this increase may not be too great it seems well, at least in Types I and II, to increase the diameter of the outer crown so that its least diameter shall be greater than the diameter of the runner at the entrance edge. This form of construction is frequently found in American wheels and appears to have been discovered by experiment. Photographs of runners of Types I to V built in this way are shown in Fig. 119 to 122.

Finally it may be pointed out that Types VIII to II inclusive are adapted for regulation by means of movable vanes, while Type I is especially fitted for the use of a cylinder gate. However, either a cylinder gate or any other form of regulation may be used with any of the types.

Number of Vanes for Radial Turbines. The number of vanes depends first upon the diameter and second upon the vane angle α for the distributor and β for the runner.

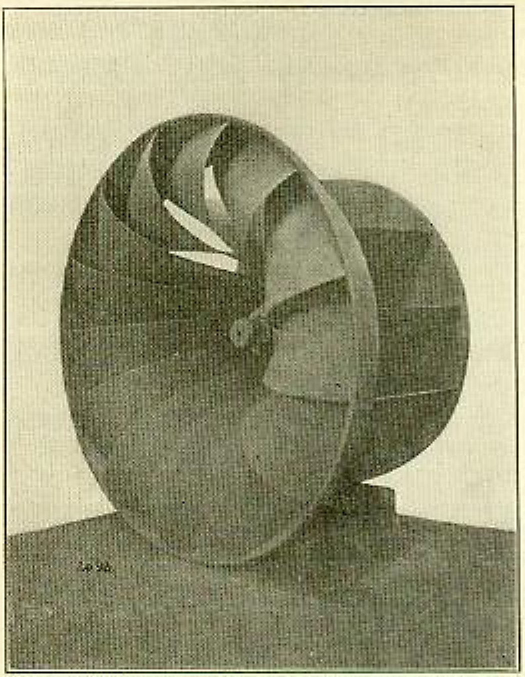


Fig. 119.—Type I.

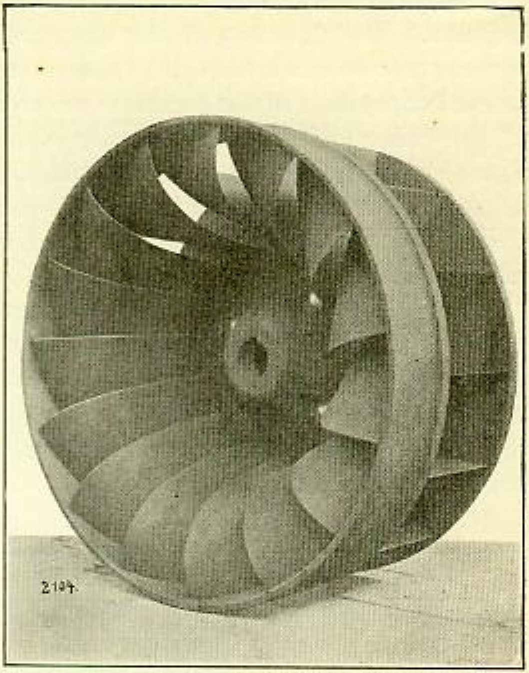


Fig. 120.—Type III.

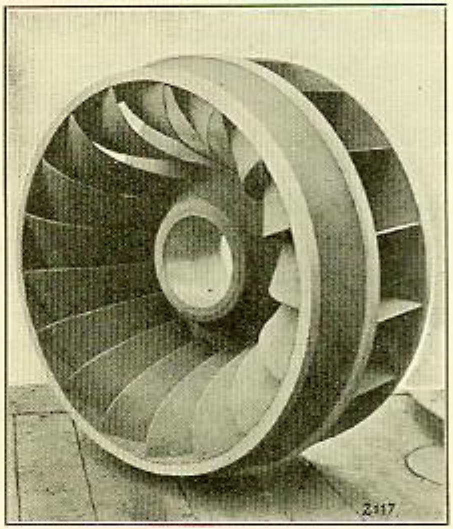


Fig. 121.—Type IV.

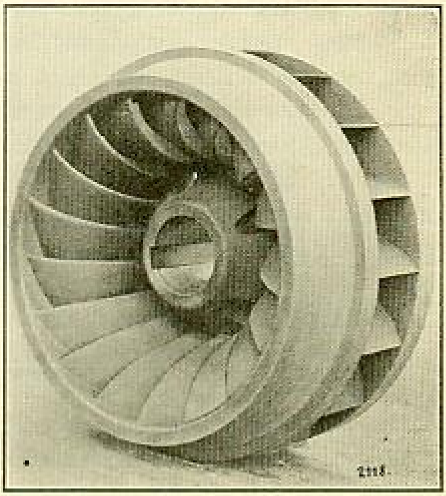


Fig. 122.—Type V.

Type.	Entrance Cross-section of the Runner.	Velocity at Entrance to Runner c_p.	Area of the Normal Intersection Surface, N.	Right-angled Discharge Velocity, $c_{a''}$.
VIII	$0.08\pi D^2$	$0.11\sqrt{2gH}$	$0.067\pi D^2$	$\sim 0.13\sqrt{2gH}$
VII	$0.10\pi D^2$	$0.12\sqrt{2gH}$	$0.086\pi D^2$	$\sim 0.14\sqrt{2gH}$
VI	$0.125\pi D^2$	$0.14\sqrt{2gH}$	$0.110\pi D^2$	$\sim 0.16\sqrt{2gH}$
V	$0.16\pi D^2$	$0.17\sqrt{2gH}$	$0.140\pi D^2$	$\sim 0.19\sqrt{2gH}$
IV	$0.20\pi D^2$	$0.205\sqrt{2gH}$	$0.180\pi D^2$	$\sim 0.23\sqrt{2gH}$
III	$0.25\pi D^2$	$0.25\sqrt{2gH}$	$0.240\pi D^2$	$\sim 0.26\sqrt{2gH}$
II	$0.30\pi D^2$	$0.295\sqrt{2gH}$	$0.320\pi D^2$	$\sim 0.28\sqrt{2gH}$
I	$0.35\pi D^2$	$0.325\sqrt{2gH}$	$0.390\pi D^2$	$\sim 0.29\sqrt{2gH}$

When the distributor vanes are movable it seems best to use a number that is divisible by four, while for the runner vanes prime numbers are usually employed.

As the result of experience we would fix the number of vanes in the distributor as follows:

	Types IV–VIII $\alpha \leqq 20°$.	Types II–IV $\alpha \begin{smallmatrix} <33° \\ >20° \end{smallmatrix}$	Types I–II $\alpha > 33°$.
$D =$ 8 in. – 24 in.	10	12	16
= 26 in. – 38 in.	12	16	20
= 40 in. – 56 in.	16	20	24
= 60 in. – 84 in.	20	24	28
= 87 in. – 114 in.	24	28	32
= $\geqq$118 in.	..	32	36

In the runner, however, the number of vanes depends upon the angle β and should be as follows:

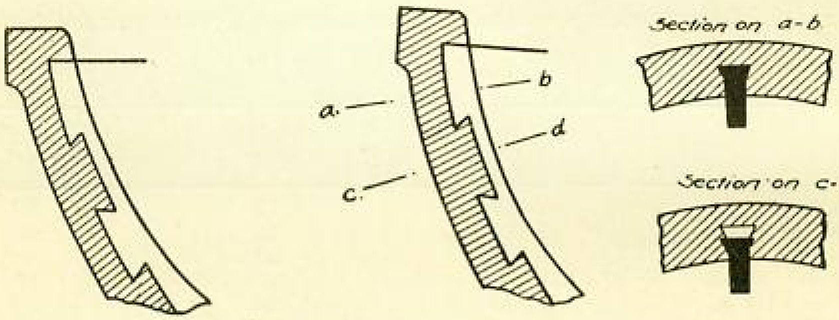
	Types VI–VIII $\beta \leqq 40°$.	Type V $\beta \cong 60°$.	Type IV $\beta \lesseqgtr 90°$.	Type III $\beta \lesseqgtr 115°$.	Types I–II $\beta \lesseqgtr 130°$.
$D =$ 8 in. – 24 in.	15–17	15	13	11	9
= 26 in. – 38 in.	19–21	19	15	13	9
= 40 in. – 56 in.	23–25	21	17	15	11
= 60 in. – 84 in.	27–29	25	19	15	11
= 87 in. – 114 in.	31–33	29	23	17	13
= $\geqq$118 in.	......	..	25	19	13

Material for the Vanes and Crowns. Cast iron is usually employed for fixed distributor vanes used under low and medium heads. In order to resist the wear, or pitting, which takes place at high velocities, especially where the water contains sand, it is best to use cast steel or bronze for high heads.

When movable vanes are used the material should be cast steel for heads exceeding 13 feet. The stronger material is used not so much for the purpose of resisting the water pressure as it is to prevent the breaking of the vanes, whether moved automatically or by hand, should an obstacle become wedged between two adjacent vanes.

Only cast steel or cast bronze vanes should be considered for the runners of Types V to VIII inclusive. In the case of these four types it must be considered that the changes in the relative velocity as well as the absolute velocity take place according to a fixed law, which for these types results in producing vanes having heavily curved backs and in drawing the crowns toward each other. (See Fig. 125.)

Steel-plate vanes may be used to advantage for runners of Types I–IV, such vanes being formed by hammering them against a matrix or by subjecting them to

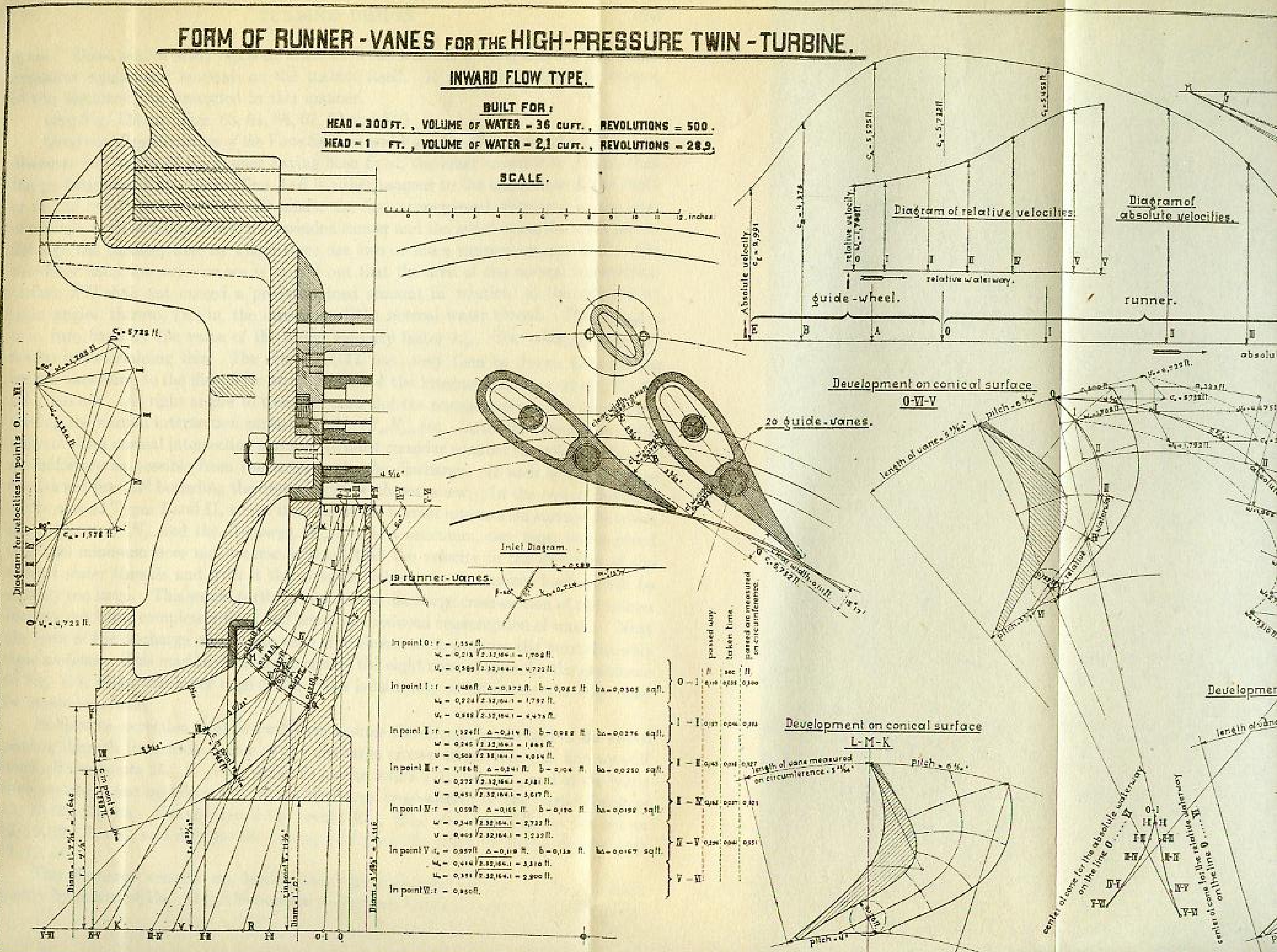


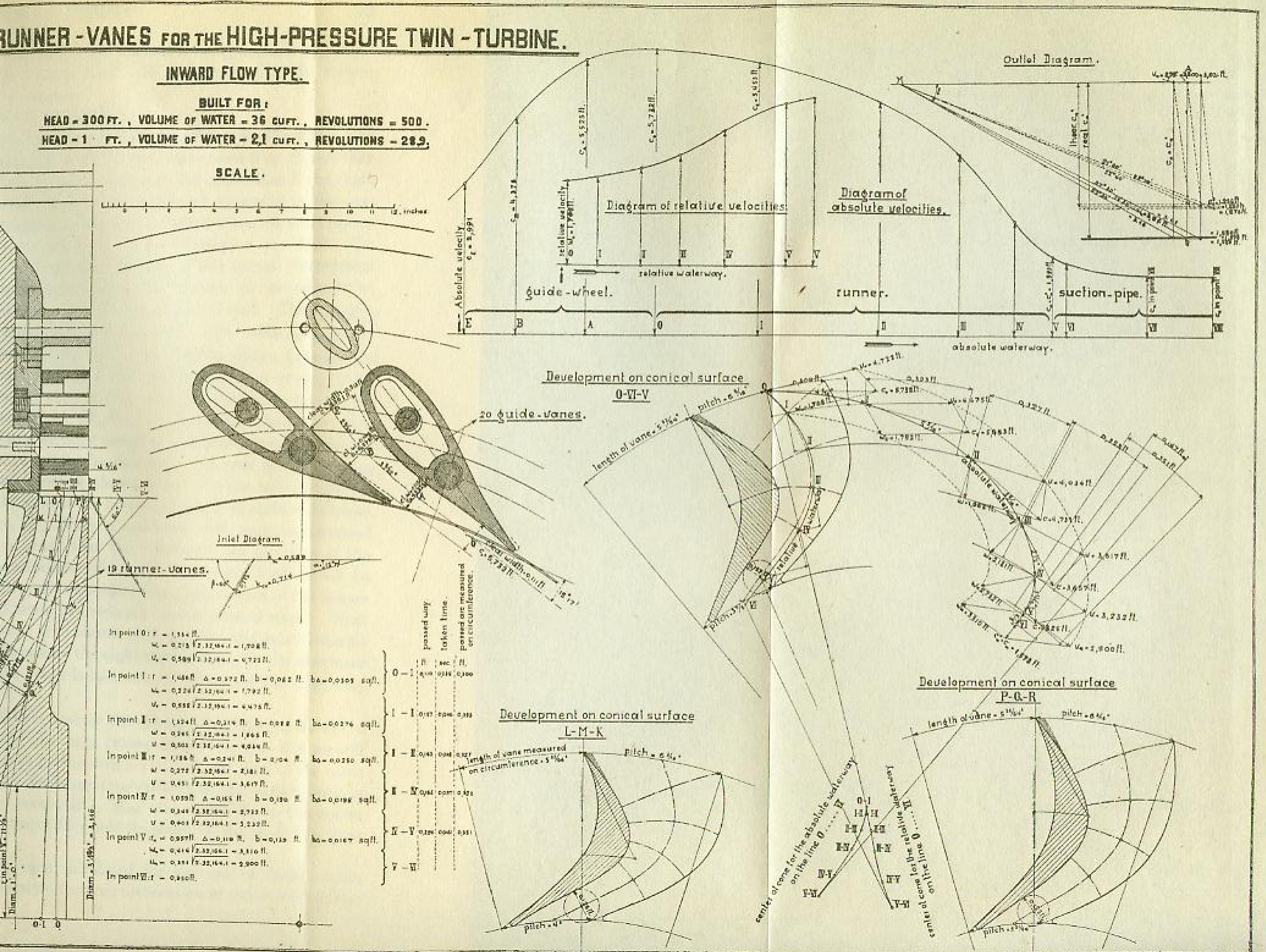
FIGS. 123 and 124.—Methods of Securing Runner Vanes to Crowns.

hydraulic pressure between a matrix and a form. This form of construction does not prevent the crowns from being cast from iron, bronze, or steel in one piece with the vanes in order to increase the strength.

Plate vanes must be set into the crowns at least five-eights of an inch and be provided with dovetails or upset edges in order to prevent them being torn away from the crowns. The writer prefers the method of fastening shown in Fig. 124 to that in Fig. 123.

The runner and distributor crowns should be made of a material having a toughness equal to that of the material in the vanes, especially where high effective heads are concerned. In the latter case special attention should be given to the clearance space between the moving runner and the stationary distributor because the wearing action of the water is apt to be most pronounced adjacent to this space. Even though the discharge of the water through it is prevented so far as possible by repeated changes in the direction of the flow, it is well, nevertheless, to take special precautions against wear by using renewable filler pieces of the toughest material adjacent to the clearance


FORM OF RUNNER-VANES FOR THE HIGH-PRESSURE TWIN-TURBINE.
INWARD FLOW TYPE.
BUILT FOR:
HEAD = 300 FT., VOLUME OF WATER = 36 CU.FT., REVOLUTIONS = 500.
HEAD = 1 FT., VOLUME OF WATER = 2,1 CU.FT., REVOLUTIONS = 28,9.
SCALE.
inches
Diagram of relative velocities.
Diagram of absolute velocities.
Absolute velocity
relative velocity
relative waterway.
guide-wheel.
runner.
Diagram for velocities in points 0.....VI.
20 guide-vanes.
19 runner-vanes.
Inlet Diagram.
Development on conical surface
0-VI-V
Development on conical surface
L-M-K
length of vane measured on circumference
passed way
taken time
passed arc measured on circumference
center of cone for the absolute waterway on the line 0.....VI
center of cone for the relative waterway on the line 0.....VI


RUNNER-VANES FOR THE HIGH-PRESSURE TWIN-TURBINE.
INWARD FLOW TYPE.
BUILT FOR:
HEAD = 300 FT., VOLUME OF WATER = 36 CUFT., REVOLUTIONS = 500.
HEAD = 1 FT., VOLUME OF WATER = 2,1 CUFT., REVOLUTIONS = 28,9.
SCALE.
12. inches
Outlet Diagram.
Diagram of relative velocities.
Diagram of absolute velocities.
Absolute velocity
relative velocity
relative waterway.
absolute waterway.
guide-wheel.
runner.
suction-pipe.
20 guide-vanes.
19 runner-vanes.
Inlet Diagram.
Development on conical surface O-VI-V
Development on conical surface L-M-K
Development on conical surface P-Q-R
passed way
taken time.
passed arc measured on circumference.
center of cone for the absolute waterway on the line 0 VI
center of cone for the relative waterway on the line 0 VI

space. These can be easily replaced when worn out without it being necessary to make extensive repairs and renewals on the turbine itself. It is desirable that the crowns of the distributor be protected in this manner.

(See Fig. 125 and Figs. 63, 64, 66, 67, 73 and 74.)

Graphical Representation of the Vane Sections and Vane Lengths. (See Fig. 127.) The diameter and height of the runner having been fixed, the inner crown is so chosen that the prolongation of the curved line A–B is either tangent to the center line of the shaft or to the cylindrical surface of the shaft, the first arrangement obtaining in the case of a single, vertical turbine with a suspended runner and the latter when the shaft passes through the turbine, that is, where there are two or more runners on one shaft. On the other hand the outer crown is so laid out that the area of the normal intersection surface FH shall not exceed a predetermined amount in relation to the velocity at right angles thereto, i.e., in the direction of the normal water thread. This velocity is in turn fixed by the value of the radial capacity factor k_{cp}. The table on page 167 assists in determining this. The rotors 00, 11, etc., may then be drawn between the crowns according to the directions on page 149, and the intermediate lines 01–01, 12–12, etc., laid out. At right angles to these we then plot the normal intersection lines representing the normal intersection surfaces $N'_{,}N_{,,}$ $N'_{,,}N_{,,,}$ etc. After the calculation of the areas of these normal intersection surfaces we must consider whether the velocity changes as uniformly as possible from the entrance to the discharge. If such is not the case the curved line EF bounding the crown must be chosen anew. In the case of turbines of the normal Types I and II, where the area of the normal intersection surfaces between the entrance $N'_{,}N_{,,}$ and the discharge, $N'_{,x}$ $N_{,x}$ is a minimum, care must be exercised that this minimum does not become so small that the velocity in the direction of the normal water threads and with it the absolute velocity in the runner buckets will be entirely too large. This would further result in the discharge cross-section of the runner buckets not being completely filled and hence in a reduced consumption of water. Next the form of the discharge edge of the vane is so chosen as to produce the most desirable vane sections. This may be done in the case of the eight normal types by the assistance of Fig. 118, where the lower edge of the vanes is laid out, or otherwise the edge may be drawn free hand.

Subject to correction later, we will then assume the circular projection of the curve passing through the middle points of the discharge cross-section. On this line we will mark off the points M_1, M_2, etc., which lie on the rotors 1–1, 2–2, etc., and at the same time represent the middle points of the discharge cross-section of the partial buckets 12, 12–23, 23, etc. We also mark the points M_{12}, M_{23}, etc., which lie on the rotors 12–12, 23–23, etc., and indicate the corresponding middle points of the partial buckets 11–22, etc.

The discharge velocity w_a, having been calculated for any one of these middle points by means of Eq. (114), the necessary effective cross-section of the corresponding partial bucket is readily obtained in square feet. For example for the partial bucket 2, 2–3, 3 we have

$$b_{23}\Delta_{23}=\frac{\frac{Q}{mz}}{w_{a_{23}}},$$

where there are z vanes and m partial buckets. From Eq. (35) we may then determine b_{23}, the length of $l_{23}=M_2M_3$ being scaled from the drawing, the angle k_{23} being measured on the same and γ being calculated from Eqs. (123) and (124) or found by the construction of the discharge diagram with the aid of c''_{a23}. The value of b_{23} having been found, we may divide by it the area of the cross-section of the partial bucket and thus obtain the clearance Δ_{23}. In a similar manner we may obtain the remaining values of Δ, that is Δ_{10}, Δ_1, Δ_{12}, etc. By laying off the half values of these clearances to the right and left of the middle line we obtain the total cross-section revolved into the drawing plane. As thus shown it will of course be distorted in the direction of the middle line. (Compare "Graphical Representation of the Effective Cross-Section," *B* (7), page 108.)

In order to make a drawing of the vane sections we choose surfaces which may be developed in a plane and for this purpose select conical surfaces having as their axes the axis of the turbine. Any desired number of these surfaces can be passed through the surfaces of the vanes, one for example being shown as drawn through the points *C* and *D*. The cross-section of the vane and bucket lying on this conical surface is determined by the pitch and clearance at the point of discharge *D* as well as by the pitch at the point *C* and the clearance at entrance, the latter being determined by the aid of w_e. Instead of the latter the entrance angle β may also assist in the preparation of the drawing, this angle being determined for the oblique section either diagrammatically or by calculation from the equation

$$\tan(\beta_\theta-90^\circ)=\tan(\beta-90^\circ)\cos(90^\circ-\theta)$$

or

$$\cot\beta_\theta=\cot\beta\sin\theta.$$

In the above equation β_θ denotes the new vane angle taken on the conical section and θ the angle which the generating line of the cone, *C–O*, makes with the axis.

The values just obtained determine the runner bucket laid out on the developed conical surfaces. It is evident that care must be exercised to cause the clearance to change at a uniform rate from the entrance to the discharge. It is also necessary that the water threads should be parallel to the rotors at the points where they pass through the discharge cross-sections.

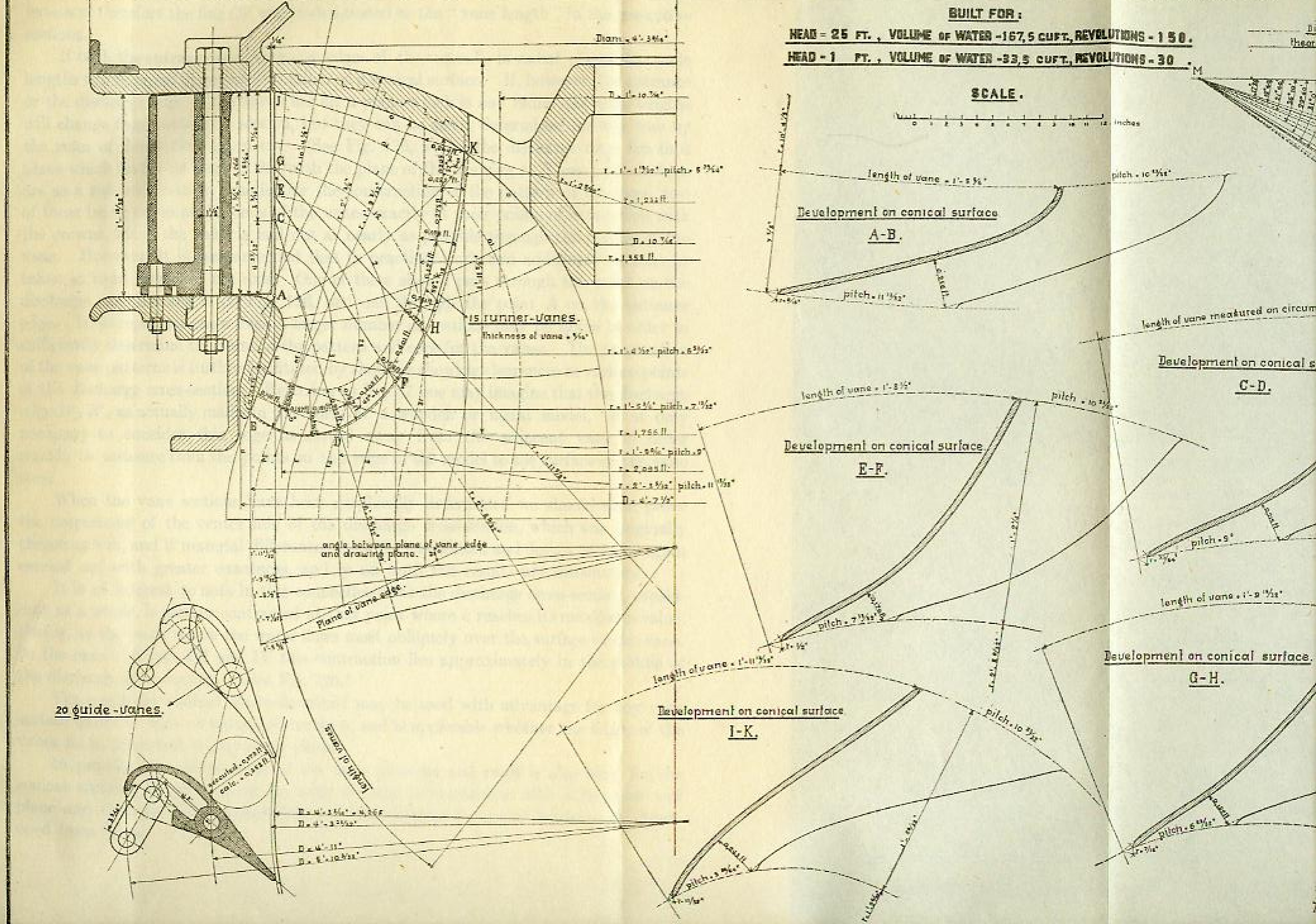
Considering now the development of the conical surface on the section *CD*, if we prolong the radius passing through the point *D* on the discharge edge it will intersect

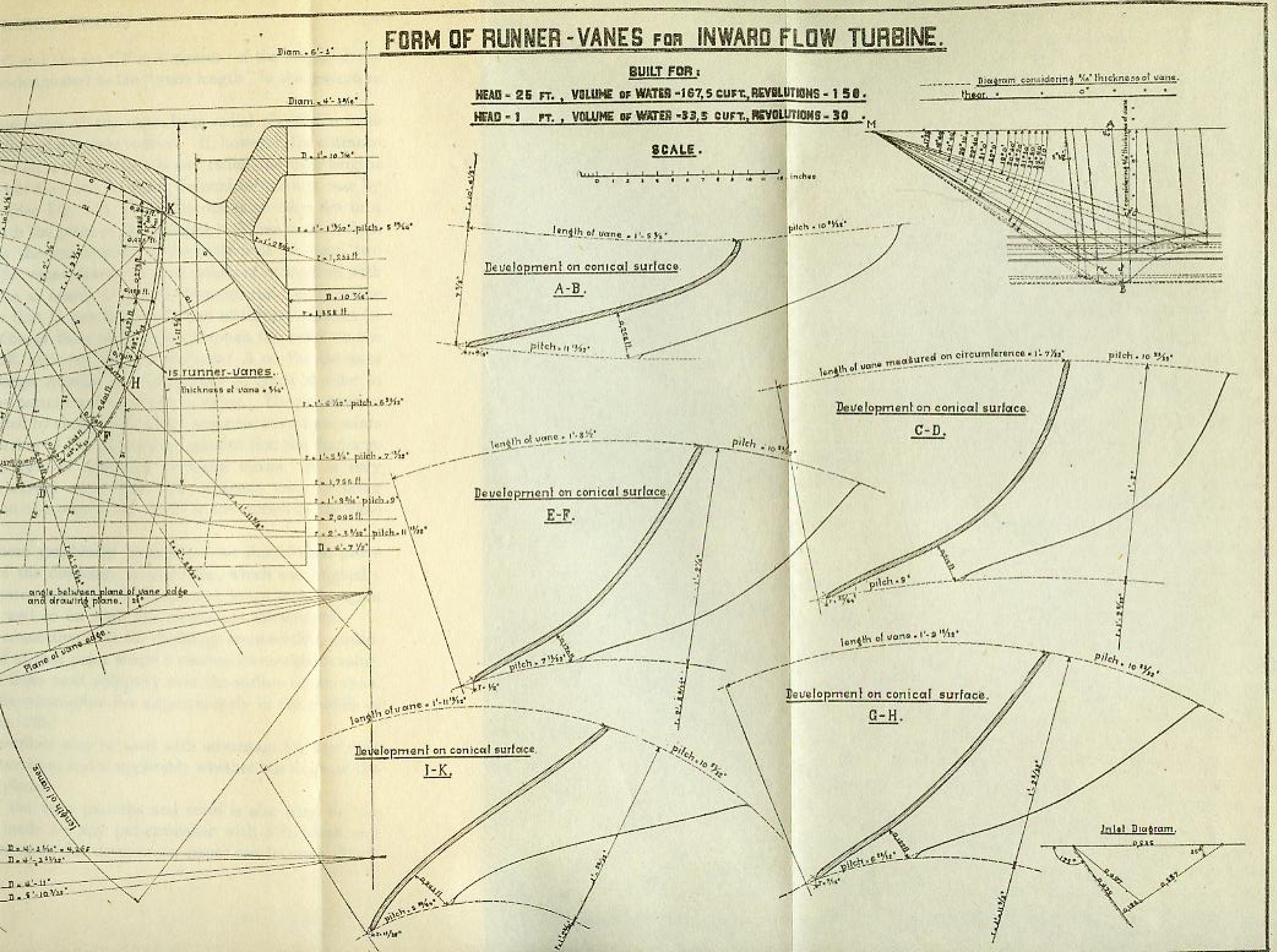
FORM OF RUNNER-VANES FOR INWARD FLOW TURBINE.

BUILT FOR:

HEAD = 25 FT., VOLUME OF WATER - 167,5 CUFT., REVOLUTIONS - 150.

HEAD - 1 FT., VOLUME OF WATER - 33,5 CUFT., REVOLUTIONS - 30.




FORM OF RUNNER-VANES FOR INWARD FLOW TURBINE.
BUILT FOR:
HEAD - 25 FT., VOLUME OF WATER - 167,5 CUFT., REVOLUTIONS - 150.
HEAD - 1 FT., VOLUME OF WATER - 33,5 CUFT., REVOLUTIONS - 30.
SCALE.
inches
Diagram considering 5/16" thickness of vane.
theor.
15 runner-vanes.
thickness of vane = 5/16"
Development on conical surface. A-B.
Development on conical surface. C-D.
Development on conical surface. E-F.
Development on conical surface. G-H.
Development on conical surface. I-K.
length of vane measured on circumference = 1'-7/32"
angle between plane of vane edge and drawing plane.
Plane of vane edge.
length of vanes.
Inlet Diagram.

the outer arc of the development at C and the arc CV is a measure of the length of the vane and therefore the line CV will be designated as the "vane length" in the respective sections.

If both the entrance and discharge edges of the vanes lie in radial planes the vane lengths will be equal in any section taken on a conical surface. If, however, the entrance or the discharge edge of the vanes lies on a plane which is not radial the vane lengths will change from section to section, but they can be easily determined for any case by the rules of descriptive geometry. (See Fig. 126, where the discharge edge lies in a plane which makes an angle of 25° with the plane of the drawing.) Three such sections are as a rule sufficient in practice for the construction of the pattern of the vane, two of these being taken at the ends of the vanes exactly at their points of connection with the crowns, while the third is laid out as nearly as possible through the middle of the vane. However, it is recommended that in practical work two additional sections be taken at right angles to the axis. One of these should pass through the point on the discharge edge marked K on Fig. 126, and one through the point A on the entrance edge. In exceptional cases even a larger number of sections may be taken in order to sufficiently determine the form of the pattern and core for the vanes. The preparation of the vane patterns is further facilitated by sketches showing clearances in various points of the discharge cross-section. Referring to Fig. 87 one may imagine that the discharge edge $B'_2\,B'_3$ as actually made in the form of a wooden or metal model. It is only necessary to consider this edge as moved away from the adjacent vane in order readily to measure from the points on this edge in the model to the surface of the vane core.

When the vane sections have been graphically determined we should then prove the correctness of the center line of the discharge cross-section, which was originally chosen at will, and if material differences exist the calculation and design must be again carried out with greater exactness, and so on until the results are satisfactory.

It is of interest to note in this connection that the discharge cross-section, considered as a whole, is always contracted at that point where k reaches its maximum value, that is, at the point where the water flows most obliquely over the surface of the vane. In the case of Types III and IV this contraction lies approximately in the middle of the discharge cross-section. (See Fig. 126.)

The graphical method above described may be used with advantage for any vane surface either of high- or low-speed turbines, and is applicable whether the edges of the vanes lie in diametral or any other planes.

In practice the construction of the vane patterns and cores is also easy, for the conical surfaces in question can be made by any patternmaker with a bandsaw and plane and then the vane sections drawn by the designer on paper can be easily developed from them.

Numerical Examples

Example 1 (see Fig. 125). The problem is to design a Francis turbine of 2000 effective horse-power for an effective head of 300 feet and a water consumption of 72 second-feet, the maximum guaranteed efficiency to be 81.2 per cent at full gate opening. The turbine is to be directly connected with an alternator making 500 revolutions per minute.

In order to overcome the axial thrust let us decide to build a double turbine. The data for the design of each wheel are then as follows:*

$$H=300 \text{ feet},\qquad Q=36 \text{ cubic feet},\qquad n=500,\qquad \omega=\frac{\pi n}{30}=52.4,$$

or, for a head of 1 ft.,

$$H=1 \text{ foot},\qquad Q_{,}=2.1 \text{ cubic feet},\qquad n_{,}=28.9,\qquad \omega_{,}=\frac{\pi n_{,}}{30}=3.03.$$

In order to obtain a higher reaction in connection with a larger angle β and the smoothest possible form of vanes, we choose a runner with a diameter of 3.116 feet, having a peripheral velocity of

$$v_0=r_0W'=1.558\times 3.03=4.722 \text{ feet}=.589\sqrt{2g\cdot 1}.$$

Therefore

$$\phi=0.589.$$

We will choose the wheel height as 0.1640 foot and we then have

$$f_B=\frac{0.1640}{3.116}=0.0526,$$

from which we calculate by Eq. (129) the radial entrance velocity to be

$$c_p=\frac{2.1}{\pi 3.116\times 0.164}=1.306 \text{ feet},$$

corresponding to a requirement factor for the turbine of

$$k_{c_p}=\frac{1.306}{\sqrt{2g\cdot 1}}=0.163.$$

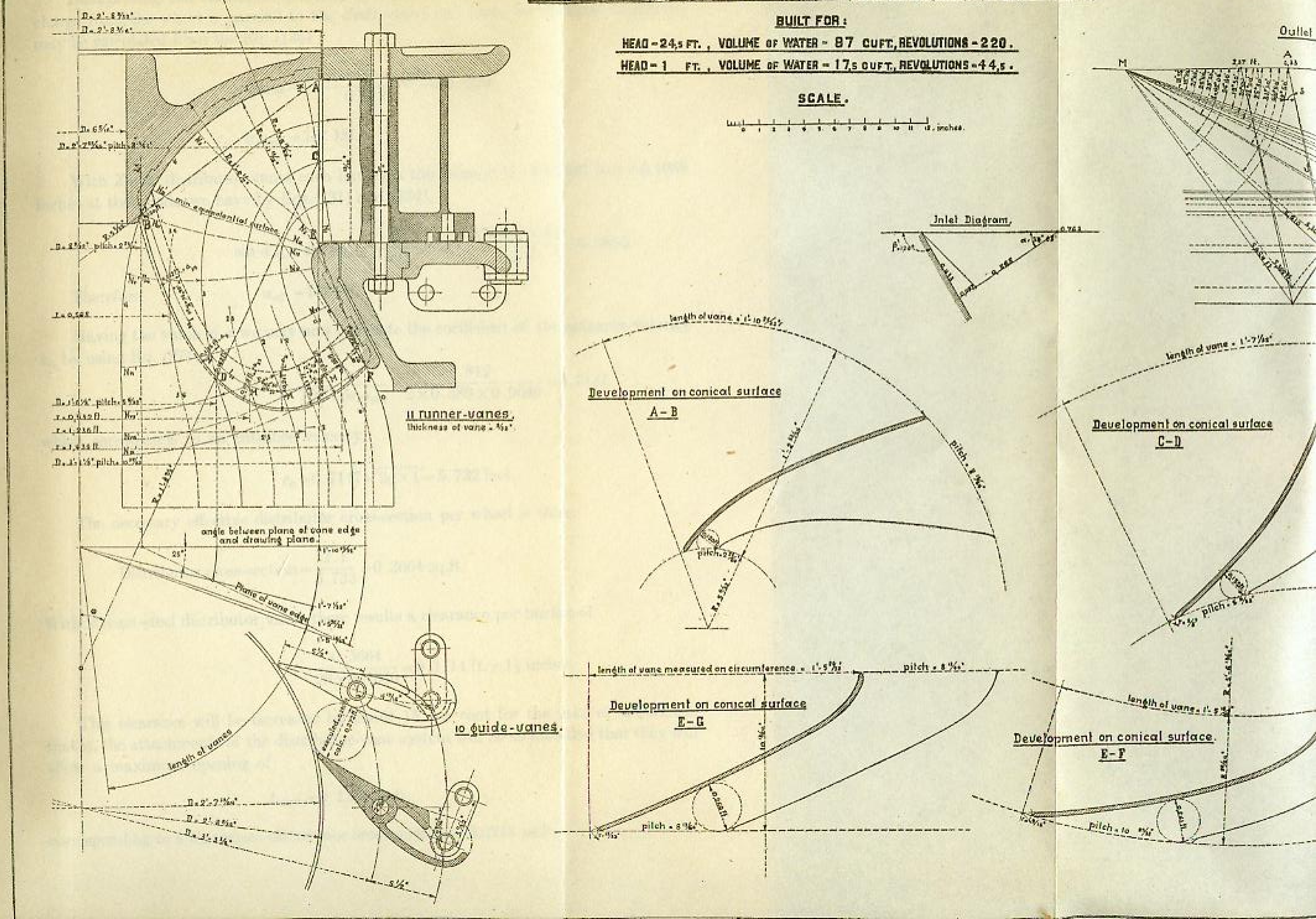
* A double wheel with a diameter of 2.788 feet would suit a normal type, the height being 0.279 ft. or $f_B=0.100$. The relative circumferential velocity of the wheel would be expressed by the factor $\phi=0.53$ (corresponding to $v_0=0.53\sqrt{2gH}$ and the requirement factor k_{c_p} would be 0.107 instead of 0.12 as normal. This proves that the resulting normal type would be too large by the allowable amount of 10 per cent.

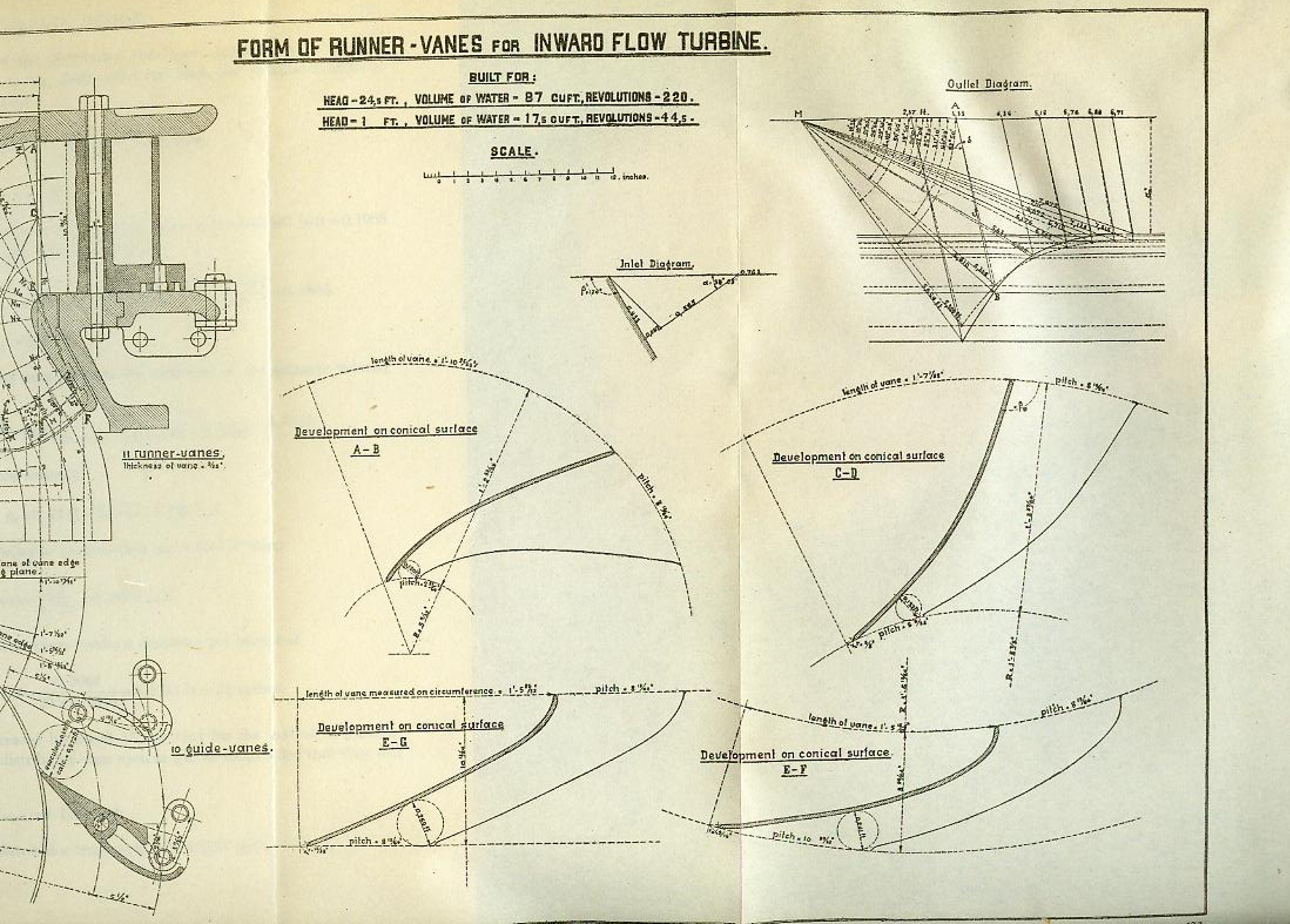
FORM OF RUNNER-VANES FOR INWARD FLOW TURBINE.

BUILT FOR:

HEAD - 24,5 FT., VOLUME OF WATER - 87 CUFT., REVOLUTIONS - 220.

HEAD - 1 FT., VOLUME OF WATER - 17,5 CUFT., REVOLUTIONS - 44,5.




FORM OF RUNNER-VANES FOR INWARD FLOW TURBINE.
BUILT FOR:
HEAD - 24,5 FT., VOLUME OF WATER - 87 CUFT., REVOLUTIONS - 220.
HEAD - 1 FT., VOLUME OF WATER - 17,5 CUFT., REVOLUTIONS - 44,5 -
SCALE.
inches.
Outlet Diagram.
Inlet Diagram.
11 runner-vanes.
Thickness of vane
10 guide-vanes.
Development on conical surface
A-B
Development on conical surface
C-D
Development on conical surface
E-G
Development on conical surface
E-F
length of vane measured on circumference

Disregarding the thickness of the distributor vanes, and assuming that the discharge from the wheel is parallel to the shaft under full load, the distributor angle α may be calculated from the Eq. (135)

$$\tan \alpha = \frac{2 \times 0.163 \times 0.589}{0.812} = 0.2365.$$

$$\therefore \quad \alpha_{\text{theor.}} = 13° \ 18'.$$

With $Z=20$ distributor vanes each having a thickness of $S=0.01640$ foot $=0.1968$ inches at their ends, we have by Eqs.(131) and (134)

$$\sin \alpha_{\text{eff.}} = \sin \alpha_{\text{theor.}} + \frac{ZS}{\pi D} = .230 + \frac{20 \times 0.0164}{3.116} = 0.2635.$$

Therefore $\qquad \alpha_{\text{eff.}} = 15° \ 17'.$

Having the value of α we may now calculate the coefficient of the entrance velocity k_{c_0} by using Eq. (97):

$$k_{c_0} = \frac{\varepsilon}{2\phi \cos \alpha_{\text{eff.}}} = \frac{.812}{2 \times 0.589 \times 0.9646} = 0.7147.$$

which corresponds to an entrance velocity

$$c_0 = 0.7147\sqrt{2g \times 1} = 5.732 \text{ feet}.$$

The necessary effective distributor cross-section per wheel is then:

$$\text{Distributor cross-section} = \frac{2.1}{5.732} = 0.3664 \text{ sq.ft.}$$

With 20 cast-steel distributor vanes there results a clearance per bucket of

$$\Delta = \frac{0.3664}{20 \times 0.1640} = 0.1114 \text{ ft.} = 1\tfrac{1}{3} \text{ inches}.$$

This clearance will be increased by about 15 per cent for the sake of assurance, that is, the attachments of the distributor-vane system will be so installed that they will allow a maximum opening of

$$\Delta_{\text{max.}} = 0.1281 \text{ sq.ft.},$$

corresponding to a maximum distributor cross-section of 0.4214 sq.ft.

Assuming an entrance free from impulse, we may obtain from the entrance diagram a vane angle β for the runner of 61° 57′. We will call this in round figures

$$\beta = 60.°$$

The relative velocity w_o, as calculated by Eq. (105) or determined from the entrance diagram, is

$$w_0 = 0.213\sqrt{2g \times 1} = 1.708 \text{ ft.}$$

The loss of pressure head which results from the change of the angle β from 61° 57′ to 60° is practically zero, as it is accurately only 0.00005 per cent.

With an assumed loss in the draft tube of 4 per cent, the effective head at the entrance to the runner is for a head of 1 foot

$$\varepsilon + .04 - \frac{c_0^2}{2g} + \frac{w_0^2}{2g} = 0.852 - 0.511 + 0.045 = 0.386.$$

The loss in the runner itself is not considered as additional.

The size of the adjacent draft tube is determined by the condition that the velocity therein should be somewhat smaller than the radial entrance velocity. We therefore choose a draft tube having a clearance of 1.6404 feet at its entrance cross-section. This will then correspond to a velocity of

$$c_a^{\text{VIII}} = \frac{2.1}{1.7226} = 1.219 \text{ ft.} = 0.152\sqrt{2g \times 1}.$$

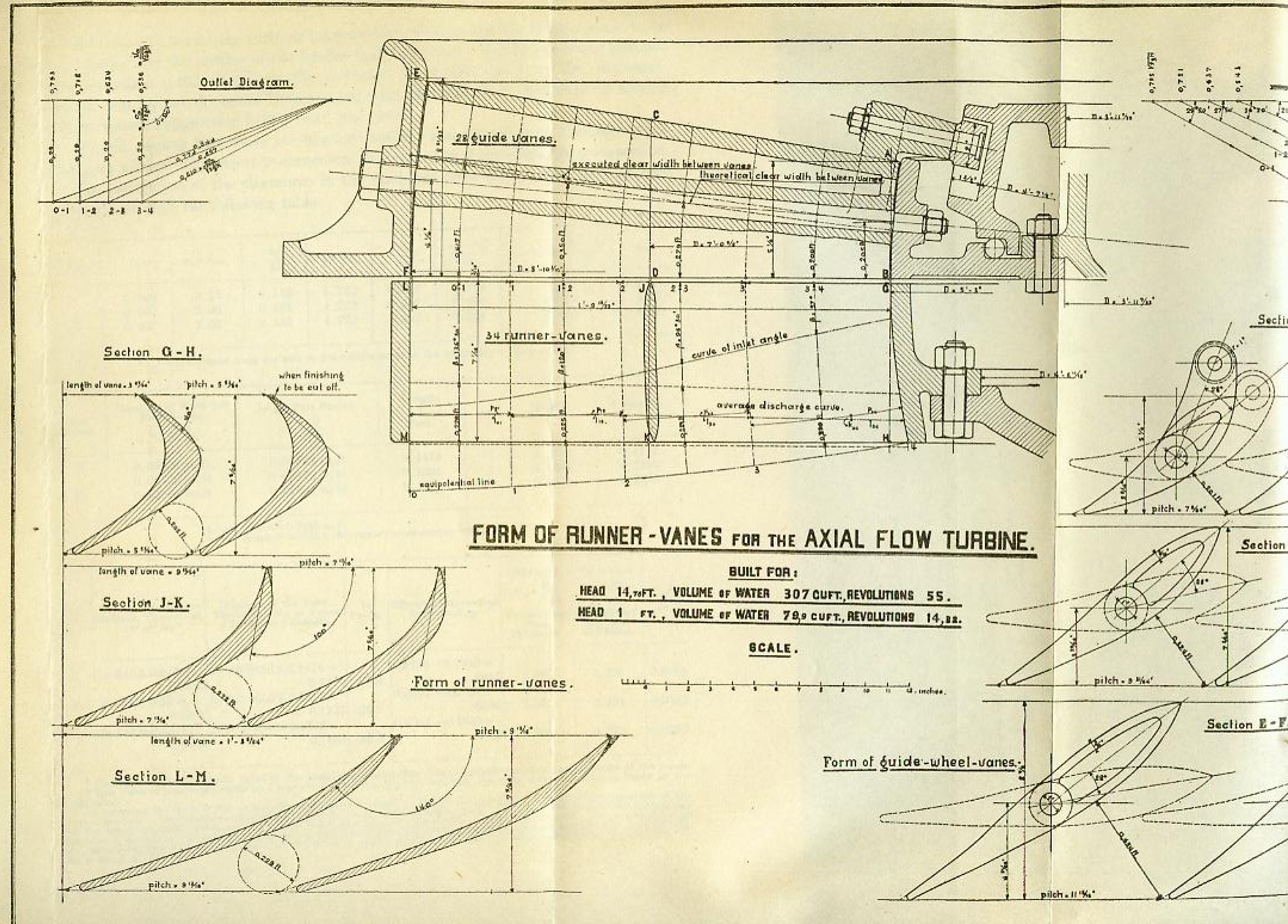
The unobstructed cross-section of the draft tube, 1.7226 sq.ft. is obtained by deducting from the total area 2.1134, the area of cross-section of a shaft 0.7054 foot in diameter, that is

$$\frac{\pi}{4}(\overline{1.6404}^2 - \overline{.7054}^2) = \frac{\pi}{4}(2.1134 - 0.3908) = 1.7226 \text{ sq.ft.}$$

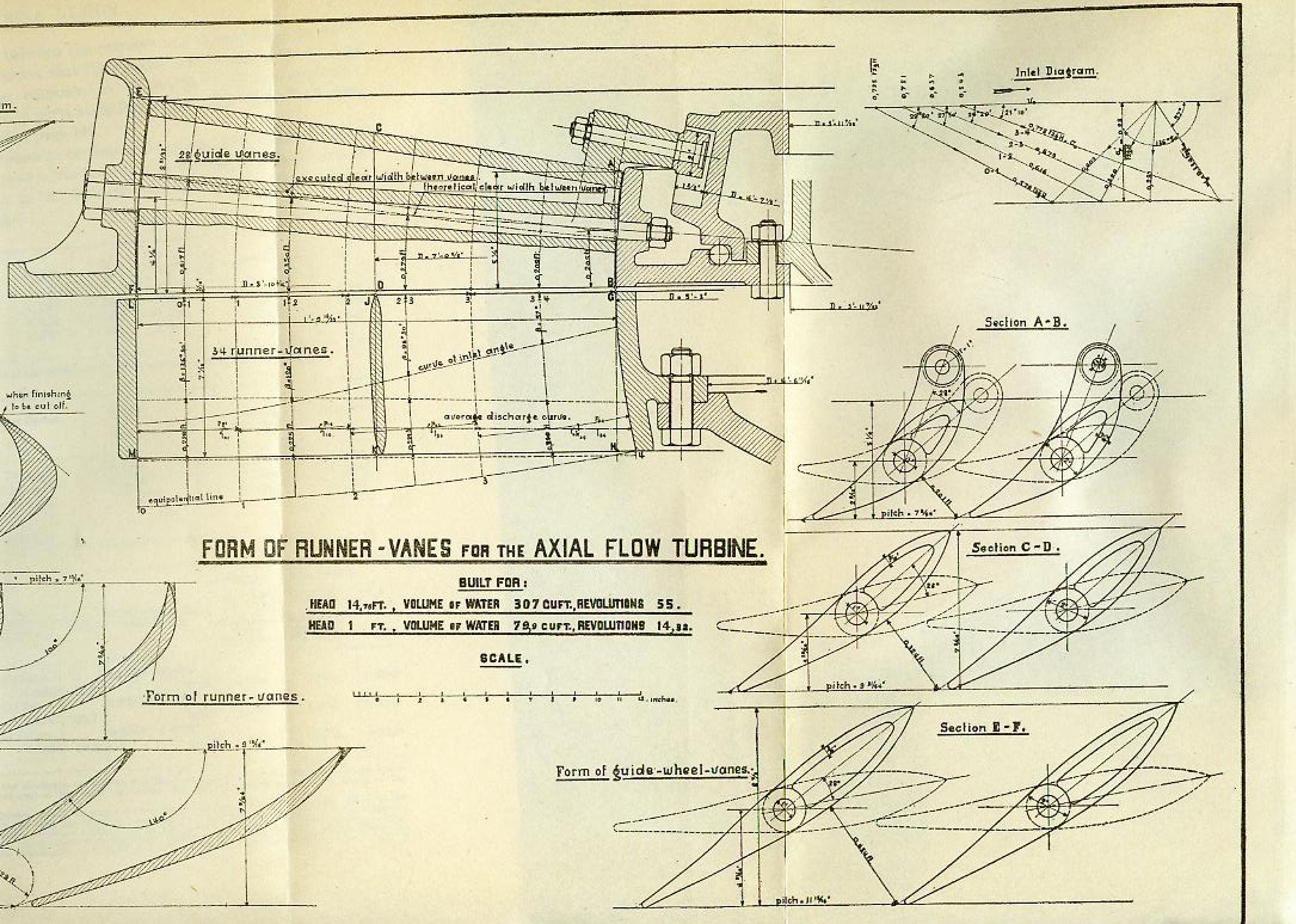
The outline of the runner must be so chosen that it will form a proper transition between the distributor and the draft tube. Immediately behind the ends of the runner vanes the area of the normal intersection surface is 1.6523 sq.ft., so that the velocity at right angles to this (i.e., in the direction of the normal water threads at the middle) is equal to

$$c'''_a = \frac{2.1}{1.6523} = 1.271 \text{ ft.} = 0.158\sqrt{2g \times 1}.$$

This does not take into consideration the reduction of this cross-section due to the ends of the vanes.



FORM OF RUNNER-VANES FOR THE AXIAL FLOW TURBINE.
BUILT FOR:
HEAD 14,70 FT., VOLUME OF WATER 307 CUFT., REVOLUTIONS 55.
HEAD 1 FT., VOLUME OF WATER 79,9 CUFT., REVOLUTIONS 14,32.
SCALE.
inches.
Outlet Diagram.
28 guide vanes.
executed clear width between vanes.
theoretical clear width between vanes.
34 runner-vanes.
curve of inlet angle
average discharge curve.
equipotential line
Section G-H.
when finishing to be cut off.
Section J-K.
Section L-M.
Form of runner-vanes.
Form of guide-wheel-vanes.
Section E-F.


FORM OF RUNNER-VANES FOR THE AXIAL FLOW TURBINE.
BUILT FOR:
HEAD 14,70FT., VOLUME OF WATER 307 CUFT., REVOLUTIONS 55.
HEAD 1 FT., VOLUME OF WATER 79,9 CUFT., REVOLUTIONS 14,32.
SCALE.
inches.
22 guide vanes.
executed clear width between vanes.
theoretical clear width between vanes.
34 runner-vanes.
curve of inlet angle
average discharge curve.
equipotential line
when finishing to be cut off.
Inlet Diagram.
Section A-B.
Section C-D.
Section E-F.
Form of runner-vanes.
Form of guide-wheel-vanes.

The normal intersection surfaces between the entrance and the discharge are to be so determined by the outline of the bucket that the velocity will be gradually changed from the value $0.163\sqrt{2g\times 1}$ at the entrance to $0.158\sqrt{2g\times 1}$ at the discharge. We may now lay out the entire bucket in parts according to the "Directions for Laying out the Normal Intersection Lines," etc., page 149.

We will assume that there are nineteen buckets and consequently nineteen vanes. Finally we will lay out, subject to correction, the center line of the discharge cross-section.

The calculation of the clearances in the discharge cross-section may then proceed in accordance with the following table:

Part Turbine.	r_a*	$v_a = r_a\omega$,	$\frac{v_a^2}{2g}$	$v_0 = r_0\omega$,	$\frac{v_0^2}{2g}$	$\frac{v_0^2 - v_a^2}{2g}$	$\frac{w_a^2}{2g} = 0.386 - \frac{v_0^2 - v_a^2}{2g}$	w_a
01	0.909	2.75	0.118	4.722	0.347	0.229	0.157	3.18
1	0.957	2.90	0.131	4.722	0.347	0.216	0.170	3.31
12	0.997	3.02	0.142	4.722	0.347	0.205	0.181	3.41

* r_a = the distance from the axis to the middle point of the respective cross-sections.

Part Turbine.	Quantity of Water per Part Bucket. $Q = \frac{2.1}{2\times 19}$	$b\Delta$ per Part Bucket. $= \frac{Q}{w_a}$	Length, l Feet.	$\cos \epsilon$	Breadth, $b = l \cos \epsilon$	Clearance, Δ
01	0.05526 cu.ft.	0.01738 sq.ft.	0.1476	1	0.1476	0.1177
1	0.05526 cu.ft.	0.01672 sq.ft.	0.1394	1	0.1394	0.1199
12	0.05526 cu.ft.	0.01621 sq.ft.	0.1312	1	0.1312	0.1235

$b\Delta$ per bucket = 0.0335 sq.ft.
$b\Delta$ per runner = 0.0335×19 = runner cross-section = 0.6365 sq.ft.

Part Turbine.	Normal Intersection Surfaces, Theoretical. $2\pi r' p = N_t$.*	Influence of the Vane Thickness = No. of Vanes × Thickness × Breadth.	In Per Ct.	Effective Intersection Surfaces. N_{eff}.	Effective $c''_a = \frac{\frac{1}{2}Q}{N_{eff}}$ Considering Vane Thickness.	Theoretical $c''_a = \frac{\frac{1}{2}Q}{Nt}$ Not Considering Vane Thickness.	$\frac{c''^2_a}{2g}$ †
01	$2\pi\cdot 0.915.0.1476 =$ 0.8485	$19\times 0.0656\times 0.1476 =$ 0.1840	21.8	$0.8485 - 0.1840 =$ .6645	1.580	1.236	0.0388
1	$2\pi\cdot 0.958.0.1394 =$ 0.8391	$19\times 0.0656\times 0.1394 =$ 0.1737	20.7	$0.8391 - 0.1737 =$ .6654	1.578	1.251	0.0387
12	$2\pi\cdot 1.000.0.1312 =$ 0.8244	$19\times 0.0656\times 0.1312 =$ 0.1635	19.8	$0.8244 - 0.1635 =$ .6609	1.589	1.273	0.0393

* Passing through the middle point of the discharge cross-section without consideration of the thickness of the vanes.
† This value is practically equal to 4 per cent of the head, as we assumed, the same being expended in losses in the draft tube.

In designing the form of the vanes reference should be made to the section entitled, "Graphic Representation of the Bucket Sections. "It should be especially noted that the absolute velocities as a function of the area of the passageway for the water gradually decreases throughout the entire turbine. This effect may be augmented by a proper choice of the form of the vanes. In order to obtain the correct view of the absolute waterway the conical surfaces 0–I, I–II, II–III, etc., should be developed in the plane of the drawing in such a manner that the several generating lines thereof meet at the points I, II, III, etc.

Example 2 (see Fig. 126). The mean head available for a certain turbine installation is 25 feet. There are a number of units to be installed, each of 1500 H.P. The quantity of water consumed by each unit is in round figures 670 second-feet. Considering the previous directions in regard to the proper number of revolutions of the alternator which is to be directly connected with the turbine shaft, it is evident that the speed should be 150 R.P.M. and the quantity of water used per unit should therefore be divided among four wheels, that is, a four-part turbine should be built. The construction data per wheel are therefore as follows:

$$H=25 \text{ feet}, \ Q=\frac{670}{4}=167.5 \text{ cu.ft.}, \quad n=150, \ \omega=\frac{\pi n}{30}15.7,$$

or calculated for a head of 1 foot,

$$H=1 \text{ foot}, Q_{,}=33.5 \text{ cu.ft.}, \quad n_1=30, \quad \omega_{,}=\frac{\pi n_{,}}{30}=3.14;$$

ε is about to be about 80 per cent.

We will choose a runner having a diameter of 4.265 feet and a height of $B=0.25D=1.066$ feet.

Then
$$f_B=0.25,$$

that is, it conforms to our normal Type III. On the other hand the coefficient of circumferential velocity is not in accordance with Type III, for

$$\phi=\frac{2.1325\times3.14}{\sqrt{2g\times1}}=0.835,$$

while it should be between 0.66 and 0.74, and the requirement factor

$$k_{c_p}=\frac{33.5}{\pi 4.265\times1.066\times\sqrt{2g}\times1}=0.292,$$

while it should be 0.25 for a normal wheel. A corresponding wheel of our normal series would belong to Type II and would have a diameter $D=3.937$; $B=0.3D=1.181$; and $Q_{,}=34.5$ cu.ft. with $n_{,}=30.65$. The resulting factors will then be $\phi=0.793$ and $k_{c_p}=0.295$.

The number of distributor vanes will be chosen, as

$$Z=20,$$

and their thickness measured at the ends as

$$S=0.03281 \text{ feet}=\tfrac{3}{8}\tfrac{}{} \text{ inch}.$$

Disregarding the thickness of these vanes and assuming a runner discharge parallel to the shaft, we have by Eq. (135):

$$\tan \alpha = \frac{2k_{c_p}\phi}{\varepsilon} = \frac{2\times 0.835\times 0.292}{0.80} = 0.6095,$$

therefore $\alpha_{\text{theor.}} = 31° \ 22'$.

As the turbine is required to have its highest efficiency between three-quarters and full gate opening, a discharge diagram with a "lead" of 5° 30′ is required; i.e.,

$$\delta = 95° \ 30'.$$

With this value of δ in mind, and considering the thickness of the distributor vanes, we will assume, subject to correction later, that α is somewhat larger, i.e.

$$\alpha_{\text{eff.}} \cong 35°.$$

After choosing the outline of the discharge edge of the vanes and projecting the center line of the cross-section (see Fig. 126), we find the circumferential velocity of the center point of the discharge cross-section:

$$v_a \cong 0.715 v_0 = 0.715\times 0.835\sqrt{2g\times 1}$$

$$\cong 4.788 \text{ feet}$$

and $k_{v_a} = 0.597$.

(Compare Fig. 116) and the discharge velocity in the direction of the normal water thread

$$c_a'' \cong 0.33\sqrt{2g\times 1} = 2.647,$$

i.e., $k_{c_a}'' = 0.33$.

(Compare table for calculation of $k_{c_0'}$.)

Using these values we may obtain k_{c_0} by means of Eq. (95) as follows:

$$k_{c_0} = \frac{\varepsilon}{2\cos 35°\times 0.835} + \frac{0.597\times 0.33 \cot 95° \ 30'}{0.835 \cos 35°} = 0.5571.$$

Therefore

$$c_0 = 4.468 \text{ ft.} \quad \frac{c_0^2}{2g} = 0.310.$$

The corrected value of α may now be calculated by means of Eq. (134) as follows:

$$\sin \alpha_{\text{effic.}} = \frac{.292}{.557} + \frac{20 \times .0328}{\pi 4.265} = .5732,$$

or $$\alpha_{\text{effic.}} = 34° \; 58'$$

or practically 35° as assumed.

The necessary effective distributor cross-section of each wheel is then

$$\text{Distributor cross-section} = \frac{33.50}{4.468} = 7.498 \text{ sq.ft.}$$

With 20 buckets and a runner height of 1.066 feet the clearance in each bucket will be Δ

$$= \frac{7.498}{20 \times 1.066} = 0.3517 \text{ ft.} = 4\tfrac{7}{32} \text{ in.}$$

In the development of the design we will consider this value as increased so that

$$\Delta_{\max} = 0.379 \text{ ft.},$$

corresponding to a maximum distributor cross-section of 8.080 square feet or an addition of practically 6.5 per cent.

By laying out the entrance diagram, using for k_{c_0} a mean value of 0.557, we obtain the value of

$$w_0 = 0.497\sqrt{2g \times 1} = 3.986 \text{ ft.},$$

or $$\frac{w_0^2}{2g} = 0.247.$$

For an entrance free from impulse we may calculate the value of β by Eq. (137) as shown below, or it may be obtained graphically.

$$\tan(\beta - 90°) = \frac{0.835 - 0.557 \cos 35°}{0.292} = 1.297,$$

$$\beta = 142° \; 22'.$$

To improve the efficiency of the turbine at three-fourths gate opening and to allow for an impact loss at full gate opening we will select 125° as the value of β instead of that found above.

$$\beta = 125°.$$

From Eq. (107) we find the component of impact loss due to this change to be as follows:

$$c_n = 4.468 \sin (125° - 35°) - 6.696 \sin 125°,$$

$$= -1.017 = -.127\sqrt{2g \times 1},$$

$$\frac{c_n^2}{2g} = 0.01613 = 1.61 \text{ per cent},$$

corresponding to a full gate opening.

The water flows along the runner vane at its entrance with the velocity

$$w'_0 = 4.468 \cos (125° - 35°) - 6.696 \cos 125°,$$

$$= 3.842 = 0.479\sqrt{2g \times 1}.$$

Selecting the number of runner vanes from the table, page 167, we have

$$z = 15.$$

We will make the vanes of sheet steel having a thickness $s = 0.02625$ ft. = (approx.) $\frac{5}{16}$ inch.

We will consider the total losses in a riveted steel draft tube entering the tail water at right angles to be 5 per cent. The effective head available for producing the relative motion of the water in the moving wheel is therefore:

$$(\varepsilon + 0.05) - \frac{c_n^2}{2g} + \frac{w_0^2}{2g} - \frac{c_0^2}{2g} = .80 + 0.05 - 0.016 + .247 - .310 = 0.771.$$

After the normal intersection lines have been drawn the calculation of the discharge cross-section of the runner takes the following form:

Part Turbine	r_a Feet.	v_a	$\frac{v_a^2}{2g}$	v_0	$\frac{v_0^2}{2g}$	$\frac{v_0^2 - v_a^2}{2g}$	$\frac{w_a^2}{2g} = 0.771 - \frac{v_0^2 - v_a^2}{2g}$*	γ† Preliminary	$\sin \gamma$
01	1.233	3.87	0.232	6.70	0.698	0.466	0.305	33° 30′	0.522
1	1.276	4.01	0.249	6.70	0.698	0.449	0.322	32° 30′	0.537
12	1.358	4.26	0.282	6.70	0.698	0.416	0.355	31′ 0″	0.515
2	1.526	4.79	0.355	6.70	0.698	0.343	0.428	27° 30′	0.462
23	1.756	5.50	0.468	6.70	0.698	0.230	0.541	23° 30′	0.399
3	1.942	6.10	0.576	6.70	0.698	0.122	0.649	20° 30′	0.350
34	2.096	6.58	0.668	6.70	0.698	0.030	0.741	19° 20′	0.331

* Disregarding the sub-vacuum.

† These values are obtained from the first draft of the discharge diagram and are subject to correction later.

Part Turbine.	k *	$\sin k$	$\cos k$	p † Feet.	$\sqrt{\frac{1-\sin^2\gamma\sin^2 k}{\sin\gamma\cos k}}$ ‡	$zps\sqrt{\frac{1-\sin^2\gamma\sin^2 k}{\sin\tau\cos k}}$ § Square Feet.	In Per Cent of the Theoretical Surfaces of Intersection. Per Cent.		Area of the Theoretical Surfaces of Intersection. $N_{theor}=2\pi r'p$. Square Feet.	N_{eff} Sq. Ft.
01	37° 30′	0.609	0.793	0.417	2.15	0.3530	11.0		$2\pi 1.226\times 0.417=3.2122$	2.8588
1	47° 30′	0.737	0.676	0.403	2.53	0.4015	12.5		$2\pi 1.269\times 0.403=3.2133$	2.8116
12	58° 0′	0.848	0.530	0.380	3.29	0.4922	15.1		$2\pi 1.351\times 0.380=3.2257$	2.7386
2	65° 0′	0.906	0.423	0.354	4.66	0.6496	19.2	Mean = 12.6	$2\pi 1.519\times 0.354=3.3786$	2.7290
23	43° 0′	0.682	0.731	0.335	3.29	0.4339	11.7		$2\pi 1.749\times 0.335=3.6814$	3.2479
3	20° 30′	0.350	0.937	0.318	3.03	0.3793	9.7		$2\pi 1.935\times 0.318=3.8662$	3.4869
34	0	0	1.000	0.302	3.02	0.3591	9.0		$2\pi 2.088\times 0.302=3.9620$	3.6054

* These values of k are taken from the drawing.
† These values of p are also directly scaled from the drawing "Area of Normal Surfaces of Intersection."
‡ Compare Eq. (52).
§ Compare Eq. (54); these values represent the amounts by which the area of the theoretical intersection surfaces are reduced to obtain the effective area, the reduction being due to the thickness of the vanes.

Part Turbine.	c''_a *	$\frac{c''^2_a}{2g}$	Sub-vacuum, $\frac{c''^2_a}{2g}-0.05$	$\frac{w_a^2}{2g}$ †	w_a	γ corr. ‡	$\sin\gamma$	$\sin\epsilon=\sin\gamma\sin k$	ϵ	$\cos\epsilon$	$1-\cos\epsilon$ in per cent
01	2.929	0.1333	0.0833	0.389	5.02	35° 10′	0.576	0.351	20° 32′	0.937	6.3
1	2.988	0.1388	0.0888	0.411	5.14	34° 40′	0.569	0.419	24° 48′	0.908	9.2
12	3.058	0.1454	0.0954	0.450	5.39	33° 50′	0.557	0.472	28° 11′	0.881	11.9
2	3.069	0.1464	0.0964	0.524	5.80	31° 30′	0.522	0.473	28° 14′	0.881	11.9
23	2.578	0.1033	0.0533	0.594	6.18	24° 20′	0.412	0.281	16° 19′	0.960	4.0
3	2.402	0.0897	0.0397	0.689	6.66	20° 40′	0.353	0.124	7° 06′	0.992	0.8
34	2.323	0.0840	0.0340	0.775	7.06	19° 00′	0.326	0.000	0° 00′	1.000	0

* See Eq. (56).
† See Eqs. (113) and (114). The sub-vacuum is taken into consideration.
‡ The corrected values of γ are obtained by the construction of the broken-line discharge diagram shown on Fig. 126 by the use of the values of v_a, w_a, and c''_a. Strictly speaking, the areas of the effective planes of intersection should be corrected by the use of these new values of γ, but the resulting corrections would be immaterial and they are therefore neglected.

Part Turbine.	Lengths, l in Feet.*	$b=l\cos\epsilon$ †	Q per Partial Bucket $=33.5+(4\times 15)$ cu.ft.	$bJ=\frac{Q}{w_a}$ sq.ft.	J Calculated Ft.	J Modified Ft. ‡
01	0.523	0.496	0.5583	0.1112	0.225	0.225
1	0.596	0.540	0.5583	0.1086	0.201	0.198
12	0.722	0.635	0.5583	0.1036	0.163	0.161
2	0.709	0.625	0.5583	0.0963	0.154	0.154
23	0.499	0.479	0.5583	0.0903	0.189	0.197
3	0.348	0.345	0.5583	0.0838	0.243	0.236
34	0.303	0.303	0.5583	0.0791	0.261	0.249

* Measured from the drawing.
† See Eqs. (29) and (35).
‡ This unimportant change in the subdivision of the cross-section is made to obtain a better form of vane.

In order to clearly show the influence of the vane thickness upon the discharge diagram, we subjoin the similar calculations with $s=0$.

Part Turbine.	c''_a for $s=0$.	$\frac{c''^2_a}{2g}$	Sub-vacuum, $\frac{c''^2_a}{2g}-0.05$	$\frac{w_a^2}{2g}$ Corresponding Sub-vacuum.	w_a
01	2.613	0.1061	0.0561	0.361	4.81
1	2.612	0.1061	0.0561	0.378	4.93
12	2.596	0.1048	0.0548	0.410	5.14
2	2.479	0.0955	0.0455	0.474	5.52
23	2.275	0.0805	0.0305	0.572	6.07
3	2.166	0.0729	0.0229	0.672	6.48
34	2.114	0.0695	0.0195	0.761	7.00

The full-line discharge diagram on Fig. 126 has been plotted by the use of the last values of c_a'' and w_a, together with the original values of v_a. It will clearly be seen from this diagram that the vane thickness exerts a harmful influence of considerable amount.

The design of the buckets is carried out in the manner before described. In order to obtain a better shape for the buckets the vertical projection of the discharge edge of the vanes is considered to make an angle of 25° with the drawing plane. The resulting lengths of the vanes, as well as the other features of their design are shown on Fig. 126.

Example 3 (see Fig. 127). We have the following data for a double turbine of the high-speed type:

$$H=24.5 \text{ feet}, \qquad Q=87 \text{ cu. ft.}, \qquad n=220, \qquad P\cong 400 \text{ H.P.}$$

We then have per wheel for 1 foot head

$$H=1 \text{ ft.}, \qquad Q_I=17.5 \text{ cu. ft.} \qquad n_I=44.5, \qquad \omega_I=\frac{\pi n_I}{30}=4.66.$$

The efficiency will be considered as $\varepsilon=0.80$. We will choose a type which lies between our normal types I and II and find the following values suitable:*

$$D=2.6263 \text{ ft.}=2 \text{ ft. } 7\tfrac{33}{64} \text{ inches},$$

$$B=0.8828 \text{ ft.}=10\tfrac{19}{32} \text{ inches}.$$

* In considering the normal types we have a choice of either

Type I $D=2.6263$ with $Q_I=19.57$ $n'=52.6$

or

Type II $D=2.9527$ with $Q_I=19.36$ $n'=41.1$.

It is evident that unless the customer will allow the speed to be changed the turbine cannot be built in accordance with either type.

We then have the resulting factors

$$f_B = 0.337 \simeq \tfrac{1}{3}\phi = \frac{1.3132 \times 4.66}{\sqrt{2g \times 1}} = 0.763,$$

$$k_{c_p} = \frac{17.5}{\pi 2.6263 \times 0.8828} \div \sqrt{2g \times 1} = 0.300,$$

$$v_0 = 1.3132 \times 4.66 = 6.120 \qquad c_p = 2.406.$$

For a wheel discharge parallel to the shaft and a thickness of the distributor vanes $= o$ we may determine $\tan \alpha$ by Eq. (135) as follows:

$$\tan \alpha = \frac{2 \times 0.300 \times 0.763}{0.80} = 0.572.$$

Therefore for this case

$$\alpha_{\text{theor.}} = 29^\circ\ 46'.$$

Now if the best efficiency of the turbine is to be obtained at three-fourths gate opening we require a discharge diagram with a lead of 15° when the turbine is wide open. That is, we choose

$$\delta = 90^\circ + 15^\circ = 105^\circ.$$

This requirement as well as that of the actual vane thickness requires that $\alpha_{\text{eff.}}$ should be larger than $\alpha_{\text{theor.}}$. Subject to correction we will then make

$$\alpha_{\text{eff.}} = 33^\circ.$$

The outline of the runner bucket, the discharge edge of the vanes and the projection of the center line of the cross-section having been chosen, we estimate by practical conformity with the drawing and by means of the data before given that

$$k_{v_a} = 0.78\phi = 0.595,$$

$$k_{c_a}'' = 0.295.$$

(A slight error in the choice of k_{v_a} and k_{c_a}'' would have little influence upon the determination of k_{c_0}.)

k_{c_0} may then be calculated by Eq. (95)

$$k_{c_0} = \frac{0.80}{2 \cos 33^\circ \times 0.763} + \frac{0.595 \times 0.295 \cot 105^\circ}{\cos 33^\circ \times 0.763} = 0.551.$$

With $Z=16$ cast-steel distributor vanes, each having a thickness of $S=0.01968$ ft. = $\frac{15}{64}$ in. at its end, the corrected value of α will be

$$\sin \alpha_{\text{eff.}} = \frac{0.295}{0.551} + \frac{16 \times 0.01968}{\pi 2.6263} = 0.5736,$$

or

$$\text{corr. } \alpha_{\text{eff.}} = 35°.$$

A new determination of k_{c_0} with the corrected value of α gives us finally

$$k_{c_0} = 0.565; \ c_0 = 4.531; \ \frac{c_0^2}{2g} = 0.319,$$

from which the corresponding value of α is finally obtained as

$$\alpha = 34° \ 05'.$$

The necessary effective distributor cross-section for each wheel is then

$$\text{Distributor cross-section} = \frac{17.5}{4.531} = 3.862 \text{ sq.ft.}$$

With 16 distributor vanes and a wheel height of 0.8858 ft., this value corresponds to a clearance

$$\Delta = \frac{3.862}{0.8858 \times 16} = 0.2724 \text{ ft.}$$

In the execution of the design this value will be increased for safety to

$$\Delta_{\max} = 0.295,$$

corresponding to a maximum distributor cross-section of 4.167 sq.ft., an increase of 7.8 per cent.

We may then calculate w_0 as follows:

$$w_0 = \sqrt{2g \times 1}\sqrt{0.763^2 + 0.565^2 - 2 \times 0.763 \times 0.565 \cos 34° \ 05'}$$

$$= 0.433\sqrt{2g \times 1}$$

$$\frac{w_0^2}{2g} = 0.1873.$$

Eq. (104*a*) then gives

$$\sin\beta=\frac{0.565 \sin 34^\circ\ 05'}{0.433}=0.731,$$

$$\beta=90^\circ+46^\circ\ 59'=136^\circ\ 59'.$$

In order to obtain the best efficiency at three-fourths gate opening we will make the vane angle

$$\beta=120^\circ.$$

The component of impact loss with a wide-open turbine will then amount to

$$c_n=4.531 \sin (120^\circ-34^\circ\ 05')-6.12 \sin 120^\circ=-0.7804$$

$$=-0.097\sqrt{2g\times 1}.$$

$$\frac{c_n^2}{2g}=0.0094, \text{ or practically 1 per cent.}$$

Along the moving runner vane the water flows with a velocity of

$$w_0'=4.531 \cos (120^\circ-34^\circ\ 05')-6.12 \cos 120^\circ=3.475 \text{ ft.}$$

$$=0.433\sqrt{2g\times 1}.$$

From the table, page 167, we will choose 11 as the number of vanes for the runner and will make the thickness of the steel plate vanes 0.0234 ft.

$$z=11; \quad s=0.0234 \text{ ft.}=\tfrac{9}{32} \text{ inch.}$$

We will consider that the total amount of head lost in the concrete draft tube to be 4 per cent, that is .04 foot.

The pressure head available for producing the relative velocity in the runner

$$=(\varepsilon+0.04)-\frac{c_n^2}{2g}+\frac{w_0^2}{2g}-\frac{c_0^2}{2g}=0.80+0.04-0.01+0.187-0.319$$

$$\cong 0.70 \text{ ft.}$$

After the normal intersection lines have been plotted on the runner cross-section the calculation of the discharge cross-section may proceed as follows:

Part Turbine.	r_a Feet.	v_a Feet.	$\frac{v_a^2}{2g}$	v_0	$\frac{w^2}{2g}$	$\frac{w^2-v_a^2}{2g}$	$\frac{w_a^2}{2g}$ *	p †	Theoretical Intersection Surface, $N_{\text{theor}}=2\pi r'p$. ‡ Square Feet.	Assumed Reduction for Vane Thickness. § Per Cent.	Effective Area Intersection Surfaces, N_{effec} ‖
34	0.508	2.37	0.088	6.12	0.582	0.494	0.206	0.400	$2\pi 0.465\times 0.400=1.169$	13% = 0.152	1.017
3	0.715	3.33	0.173	6.12	0.582	0.409	0.291	0.331	$2\pi 0.696\times 0.331=1.447$	13% = 0.188	1.259
23	0.932	4.34	0.293	6.12	0.582	0.289	0.411	0.302	$2\pi 0.924\times 0.302=1.753$	13% = 0.228	1.525
2	1.106	5.15	0.412	6.12	0.582	0.170	0.530	0.272	$2\pi 1.099\times 0.272=1.878$	13% = 0.244	1.635
12	1.236	5.76	0.516	6.12	0.582	0.066	0.634	0.248	$2\pi 1.233\times 0.248=1.921$	13% = 0.250	1.671
1	1.348	6.28	0.613	6.12	0.582	0.031	0.731	0.233	$2\pi 1.345\times 0.233=1.969$	13% = 0.256	1.713
01	1.439	6.71	0.699	6.12	0.582	0.117	0.817	0.218	$2\pi 1.439\times 0.218=1.971$	13% = 0.256	1.715

* Without consideration of the sub-vacuum: $\frac{w_a^2}{2g}=0.70-\frac{w^2-v_a^2}{2g}$.

† Taken from drawing in Fig. 126.

‡ The values of r' are scaled from the drawing.

§ Estimated, subject to correction later.

‖ Subject to correction.

Part Tubrine.	c''_a Subject to Correction. Feet.	$c''^2_a\div 2g$ Subject to Correction.	Loss in Draft Tube.	Sub-vacuum, $\frac{c''^2_a}{2g}-0.04$.	Effective, $\frac{w_a^2}{2g}$, Subject to Correction.*	Effective, w_a, Subject to Correction.	γ, Subject to Correction.†	k ‡	$\sqrt{\frac{1-\sin^2\gamma\sin^2 k}{\sin\gamma\cos k}}$	$zpe\sqrt{\frac{1-\sin^2\gamma\sin^2 k}{\sin\gamma\cos k}}$ § Sq.ft.	In Per Cent of N_{theor} ‖
34	4.30	0.287	0.04	0.247	0.453	5.399	52° 50′	66° 30′	2.148	0.2212	18.9
3	3.47	0.187	0.04	0.147	0.438	5.308	40° 50′	64° 10′	2.834	0.2414	16.7
23	2.87	0.128	0.04	0.088	0.499	5.665	31° 10′	43° 30′	2.489	0.1934	11.0
2	2.68	0.112	0.04	0.072	0.602	6.223	25° 30′	26° 30′	2.541	0.1779	9.5
12	2.62	0.107	0.04	0.067	0.701	6.715	23° 00′	12° 10′	2.610	0.1665	8.7
1	2.55	0.101	0.04	0.061	0.792	7.139	20° 50′	0	2.809	0.1685	8.5
01	2.55	0.101	0.04	0.061	0.878	7.515	19° 50′	−6° 0′	2.964	0.1660	8.4

* Taking into consideration the sub-vacuum: Eff. $\frac{w_a^2}{2g}=\frac{w_a^2}{2g}+$sub-vacuum.

† These values of γ are taken from the first draft of the discharge diagram, shown in broken lines in Fig. 127, for which v_a, c''_a, and eff. w_a are used.

‡ Obtained from the drawing.

§ The final amount by which N_{theor} is to be reduced on account of the vane thickness.

‖ Mean value is 11.7 per cent for a vane thickness of $\frac{3}{16}''$.

Part Turbine.	Effective Area Intersection Surface, $N_{eff.}$*	c''_s	$\frac{c''_s{}^2}{2g}$	Draft Tube Loss.	Sub-vacuum (Final).	Eff. $\frac{w_s^2}{2g}$	Eff. w_s	γ †	sin γ	sin ε ‡	ε	cos ε	$1-\cos\varepsilon$ in per cent
34	0.948	4.615	0.331	0.04	0.291	0.497	5.654	54° 40′	0.816	0.748	48° 27′	0.663	33.7
3	1.206	3.625	0.204	0.04	0.164	0.455	5.410	42° 05′	0.670	0.603	37° 05′	0.798	20.2
23	1.560	2.805	0.122	0.04	0.082	0.493	5.631	29° 50′	0.498	0.343	20° 02′	0.939	6.1
2	1.700	2.574	0.103	0.04	0.063	0.593	6.176	24° 40′	0.417	0.186	10° 40′	0.983	1.7
12	1.753	2.496	0.098	0.04	0.058	0.692	6.672	21° 00′	0.358	0.075	4° 18′	0.997	0.3
1	1.800	2.413	0.092	0.04	0.052	0.786	7.111	19° 50′	0.339	0	0	1.000	0
01	1.805	2.424	0.091	0.04	0.051	0.868	7.472	18° 55′	0.324	−0.034	−2° 00′	0.999	0.1

* The following values are final.
† The discharge diagram with the new values of c''_s and w_s is plotted in Fig. 127.
‡ sin ε = sin γ sin k.

Part Turbine.	Lengths L Feet.*	Breadths $b = l\cos\varepsilon$ Feet.	$b\Delta = \frac{Q_s}{44} \div$ eff. w_s Sq.ft.†		Δ Calculated.	Δ Modified.	$b\Delta$ Modified.‡	
34	0.869	0.5761	0.07034	Calculated $b\Delta$ per	0.122	0.122	0.07034	Modified $b\Delta$ per bucket
3	0.682	0.5142	0.07352	bucket = 0.2538	0.135	0.135		= 0.2551 sq.ft.
23	0.453	0.4254	0.07063	sq. ft.	0.166	0.172	0.07317	Modified $b\Delta$ per wheel
2	0.308	0.3028	0.06440	Calculated $b\Delta$ per	0.213	0.211		= 2.8051 sq.ft., i.e.,
12	0.254	0.2532	0.05961	wheel = 2.7919	0.235	0.233	0.05900	about 0.8 per cent
1	0.235	0.2350	0.05593	sq. ft. = runner	0.238	0.239		larger than calcu-
01	0.221	0.2208	0.05323	cross-section.	0.241	0.238	0.05255	lated.

* Scaled from drawing.
† $Q_s \div 44$ = quantity of water per part turbine per bucket. $b\Delta$ is therefore per part turbine.
‡ In order to prove that a turbine so constructed fulfills with sufficient accuracy the requirements of the fundamental equation not only in the middle water thread of the entire turbine, but also in each part turbine, we have obtained the values $v_0c_0\cos\alpha - v_ac_a\cos\delta$ = constant $(r_0w_{u_0} - r_aw_{ua})$ in accordance with the footnote to Eq. (88) and have tabulated the same as follows:

Part Turbine.	$v_0c_0\cos\alpha$	$v_ac_a\cos\delta$	$v_0c_0\cos\alpha - v_ac_a\cos\delta = (r_0w_{u_0} - r_aw_{ua})$ Const.	Maximum Difference.
34	+23.257	−2.22	21.04	$\frac{20.80-20.54}{20.80}$ =
23	+23.257	−2.50	20.75	0.011
12	+23.257	−2.70	20.56	or 1.1%
01	+23.257	−2.40	20.85	
			Ave. 20.80	

Full Radical Turbines with Outward Discharge—Fourneyron Turbines

The methods of calculation for radical turbines with inward discharge, as above set forth, are applicable to Fourneyron turbines without further amplification.

It is not apparent why a properly constructed Fourneyron turbine should not be as good as a Francis turbine (see Figs. 54, 55 and 75). Although the discharge relations in the draft tube do not seem so favorable with the former as with the latter, yet these may be materially improved by the form of construction; as, for example, by the use of spiral-draft chests and other means.

The Fourneyron turbines are in many cases to be even preferred to Francis turbines on account of their simplicity of design and consequent economy of construction (for example see Fig. 54).

Full Axial Turbines (Jonval)

Turbines of this class may be divided into normal types in a manner entirely similar to that employed for Francis turbines and may be treated accordingly.*

The following will serve as an

Example Showing the Calculation of an Axial Turbine. The following data are given for the calculation of an axial turbine provided with movable vanes (see Figs. 58, 59, and 128).

$$H=14.76 \text{ feet};\ Q=307 \text{ sec. ft.};\ n=55;\ \omega=\frac{\pi n}{30}=5.766.$$

$$P=390 \text{ H.P.}$$

$$H_{,}=1 \text{ foot};\ Q_{,}=79.91 \text{ sec.ft.};\ n_{,}=14.32;\ \omega_{,}=1.500.$$

We will choose as the capacity factor of the turbine

$$k_{c_p}=0.25,$$

so that

$$c_p=0.25\sqrt{2gH}=7.703.$$

We also desire that the peripheral velocity at the inner crown shall not be less than $0.49\sqrt{2gH}$, that is

$$k_{v_i}=0.49 \text{ and } v_i=0.49\times 30.81=15.10.$$

* For a more detailed description of the theory and construction of axial turbines see the German edition of "Turbines and Turbine Installations," published by Julius Springer, Berlin, 1905.

Therefore $$\frac{D_i}{2}=\frac{v_i}{\omega}=\frac{15.10}{5.766}=2.619 \text{ feet},$$

and $$D_i=5.238 \text{ feet, say 5 ft. 3 in.}$$

The area of the surface at the entrance to the runner—through which c_p flows at right angles—has the value

$$\frac{Q}{c_p}=\frac{307}{7.703}=39.85 \text{ sq.ft.}$$

We may then determine the diameter of the outer runner crown from the equation

$$\frac{(D_0^2-D_i^2)\pi}{4}=39.85 \text{ sq.ft.}$$

or $$D_0=8.849 \text{ ft.}=8 \text{ ft. } 10\tfrac{3}{16} \text{ in.}$$

The peripheral velocity at the outer crown will then be

$$v_0=\frac{8.849}{2}\times 5.766=25.51=0.828\sqrt{2gH}.$$

This value is allowable and exactly corresponds to a turbine of our normal Type II.

The total breadth of the runner is 1.799 feet, and consequently the value of

$$f_B=1.799\div\frac{5.25+8.849}{2}=0.255.$$

By subdividing the turbine into four parts, each of which consumes $\frac{Q}{4}=76.75$ sec.ft., we may obtain the following conditions at entrance to the runner:

Part Turbine	r_0 Ft.	v_0 Ft.	$k_{v_0}=\frac{v_0}{\sqrt{2gH}}$	$\frac{v_0^2}{2g}$	$b=b_0$ * Ft.	$2\pi r_0 b_0$ † Sq.ft.	Reduction for Vane Thickness ‡	$2\pi r_0 b_0 (1-0.12)$ Sq.ft. §	c'_p ‖
01	4.249	24.50	0.795	9.331	0.374	9.985	12%	8.787	$8.627=0.28\ 2gH$
12	3.852	22.21	0.721	7.668	0.413	9.996	12%	8.796	$8.627=0.28\ 2gH$
23	3.405	19.64	0.637	5.996	0.469	9.994	12%	8.795	$8.627=0.28\ 2gH$
34	2.900	16.72	0.543	4.346	0.518	9.985	12%	8.787	$8.627=0.28\ 2gH$
					1.804	39.960		35.165	

* The breadth of the several part turbines at the entrance to the distributor. The scaled lengths b are the same at the entrance as the bucket breadths b_0.

† The area of the part turbines without any reduction for the thickness of the vanes.

‡ Determined from the drawing.

§ Area of the part turbine after deducting the thickness of the vanes.

‖ Capacity factor, taking into consideration the thickness of the vanes.

The diameter of the center of the entrance to the runner has the value

$$D_m=7.054 \text{ ft.}=7 \text{ ft. } 0\tfrac{5}{8} \text{ in.}$$

The velocity of revolution at this point is therefore

$$v_m = 20.337 = 0.66\sqrt{2gH},$$

or

$$k_{v_m} = 0.66.$$

By curving the inner runner crown toward the shaft and thus increasing the runner area toward the discharge we tend to avoid as much as possible a reduction in the value of c''_a as well as to improve the discharge into the draft tube. This being done we have the following values, which obtain at the discharge from the runner:

Part Turbine.	r_a Feet.	v_a Feet.	$k_{v_a} = \frac{v_a}{\sqrt{2gH}}$	$\frac{v_a^2}{2g}$	p* Feet.	$N_{\text{theor.}} = 2\pi r_a p$ Sq. Ft.	Reduction for Thickness of Vanes.†	$N_{\text{eff.}}$‡ Sq. Ft.	$c''_a = \frac{Q}{N_{\text{eff.}}}$	$\frac{v_0^2 - v_a^2}{2g}$ Feet.	$\frac{w_a}{\sqrt{2gH}}$§	w_a Ft.	$\frac{w_a^2}{2g}$
01	4.239	24.44	0.793	9.285	0.381	10.14	17%	8.667	$\sim 8.93 = 0.29\ 2gH$	0.046	0.844	26.01	10.52
12	3.839	22.14	0.718	7.620	0.420	10.14	18%	8.593	$\sim 8.93 = 0.29\ 2gH$	0.048	0.774	23.85	8.84
23	3.389	19.54	0.634	5.935	0.476	10.14	18%	8.593	$\sim 8.93 = 0.29\ 2gH$	0.061	0.697	21.49	7.18
34	2.867	16.53	0.536	4.248	0.564	10.14	19%	8.521	$\sim 8.93 = 0.29\ 2gH$	0.098	0.610	18.80	5.49
						40.56		34.374					

* Breadth of the normal intersection surfaces.
† Calculated directly from the drawing.
‡ $N_{\text{eff.}} = N_{\text{theor.}}$ minus the deduction for the thickness of the vanes.
§ Taken from the discharge diagram.

In reference to the discharge diagram it will be noted that δ is made 90°, corresponding to the highest efficiency when the turbine is wide open. By this condition with the values of both v_a and c''_a given the discharge diagram is fixed.

Now in order to use a fundamental equation such as (113) to ascertain the values of c_0 and w_0 it may be set down as follows:

$$\frac{c_0^2}{2g} - \frac{w_0^2}{2g} = \epsilon_0 H + \frac{c_a''^2 - c_a^{n2}}{2g} - (t_6 + t_7)H - \frac{v_0^2 - v_a^2}{2g} - \frac{w_a^2}{2g}.$$

Before we can use this formula it is necessary to insert the following values:

$$c_a^n = \text{velocity at the point of discharge from the draft tube} = \frac{Q}{\text{area draft tube}}$$

$$= \frac{307}{61.892} = 4.960 \text{ ft.} = 0.161\sqrt{2gH}.$$

c''_a = velocity at the point of entrance to the draft tube $= 8.93 = 0.290\sqrt{2gH}$.

The last-named value takes into consideration the thickness of the runner vanes, of which there are 34.

The pressure head corresponding to the change in velocity in the draft tube amounts to

$$\frac{c_a''^2}{2g}-\frac{c_a^{n2}}{2g}=0.084H-0.026H=0.058H.$$

The draft tube loss will be considered as

$$(l_6+l_7)H=0.035H=3.5 \text{ per cent.}$$

The value of the sub-vacuum then becomes

$$\frac{c_a''^2-c_a^{n2}}{2g}-(l_6+l_7)H=0.058H-0.035H=0.023H=2.3 \text{ per cent.}$$

We will consider the total efficiency as

$$\varepsilon=0.815$$

and

$$\varepsilon_0=0.815+0.035=0.85.$$

With the values now given, we may next obtain $c_0^2-w_0^2$. As the capacity factor c_p' is also known (with due regard to the thickness of the vanes) the entrance diagram is thereby fixed. We then find the values shown in the following table:

Part Turbine.	$\frac{c_0^2-w_0^2}{2g}$	$k_{c0}=\frac{c_0}{\sqrt{2gh}}$	c_0 Feet.	$\frac{w_0}{\sqrt{2gh}}$	w_0 Feet.
01	2.319	0.570	17.56	0.409	12.60
12	3.997	0.616	18.98	0.329	10.13
23	5.644	0.679	20.92	0.281	8.66
34	7.297	0.778	23.97	0.334	10.29

From the above we obtain the following dimensions for the distributor:

Part Turbine.	$b_0 a_0=\frac{Q\div 4}{c_0}$, Square Feet.	b_0, Feet.	a_0 Calculated.* Feet.	a_0 Adjusted.* Feet.	α	β	z
01	4.371	0.374	0.417	0.420	29° 20′	136° 50′	28
12	4.044	0.413	0.350	0.364	27° 00′	—	—
23	3.669	0.469	0.279	0.308	24° 20′	—	—
34	3.202	0.548	0.209	0.234	21° 10′	57° 00′	—

15.286 = total cross-section of distributor

* The calculated and adjusted values of a_0 are not in exact agreement because in the case of regulation by movable vanes the vanes must be so formed as to fit tightly when closed.

For the runner we obtain the following values

Part Turbine.	Lengths l, Feet.	γ	k	$\cos \varepsilon$ *	$b = l \cos \varepsilon$ Feet.	$b\Delta = \frac{Q}{4w_2}$ Square Feet.	Δ Calculated and Final. Feet.
01	0.381	20° 05′	3° 00′	0.9999	0.381	2.951	0.228
12	0.420	22° 00′	4° 00′	0.9996	0.420	3.218	0.225
23	0.476	24° 35′	5° 30′	0.9992	0.476	3.572	0.221
34	0.574	28° 25′	8° 30′	0.9975	0.573	4.082	0.210

Total runner cross-section 13.823

* $\cos \varepsilon = \sqrt{1 - \sin^2 \gamma \sin^2 k}$ as per Eq. (29).

The reader is referred to Fig. 128 in regard to the construction of the vanes.

Pelton Wheels and Spool Wheels

General. Wheels of this form are designed for high heads and a small quantity of water and they may be classified as radial turbines operating under water applied to the outer periphery. In these wheels the row of stationary buckets (i.e., the distributor) is replaced by nozzles which are set at a given angle with the periphery of the wheel. As a rule the number of these nozzles does not exceed three per wheel and in the more usual types only one nozzle is used. The regulation of the area of the cross-section of the nozzle, and hence of the quantity of water impinging upon the wheel, is accomplished either automatically or by hand.

Pelton wheels may be classified as follows, according to the form of the cross-section of the nozzles, which may be circular, quadrangular, or square, viz.:

(1) With circular nozzles.
- (*a*) Nozzle closed by means of a sliding cut-off. (See Fig. 129.)
- (*b*) Nozzle closed by means of a needle placed within the nozzle and moving in the direction of its axis (needle valve). (See Figs. 132 and 133).
- (*c*) Nozzle closed by means of an outer sheath surrounding a stationary needle and moving in relation to it. (See Fig. 130.)

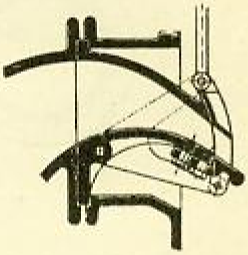


FIG. 129.—Circular Nozzle Closed by Sliding Cut-off.

(2) With nozzles square or rectangular in cross-section.
- (*a*) Nozzles closed by means of a sliding cut-off, just as in the case of circular nozzles.

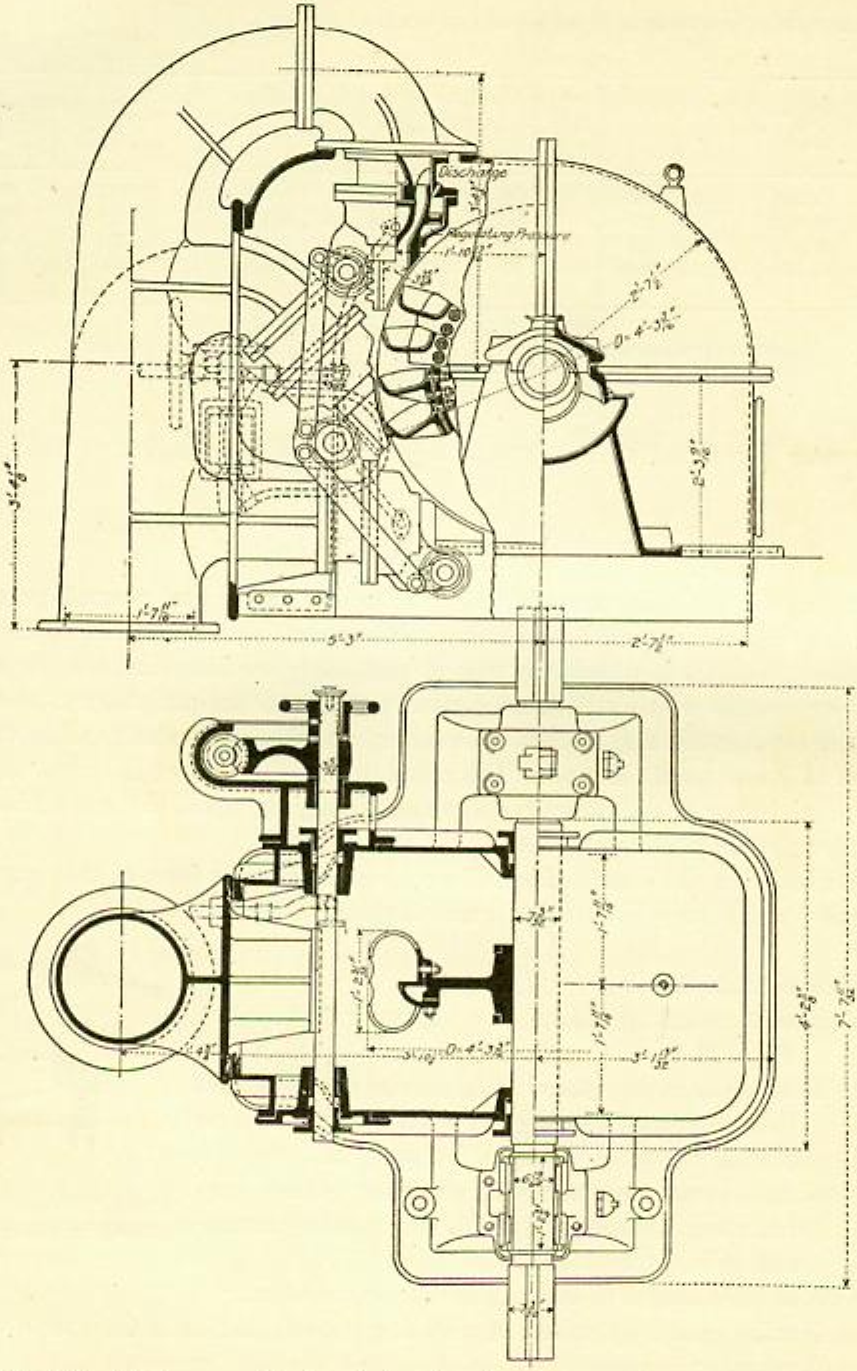


FIG. 130.—High-pressure Turbine having Circular Nozzles with Stationary Needles.

(*b*) Nozzle closed by a contraction of the thickness of the jet, this being usually accomplished by revolving the upper side of the nozzle around a pivot (see Fig. 138), or a parallel motion in guides (see Fig. 136).

(*c*) Nozzle closed by a contraction of the width of the jet, the two sides of the nozzle being moved toward each other either by swinging them or by their parallel motion, the upper and lower sides of the nozzle remaining stationary.

The methods of regulation described under (*b*) and (*c*) are to be preferred because the control of the jet is assured under all gate openings, while the method denoted by (*a*) produces an abrupt contraction of the cross-section and deflects the jet from the desired

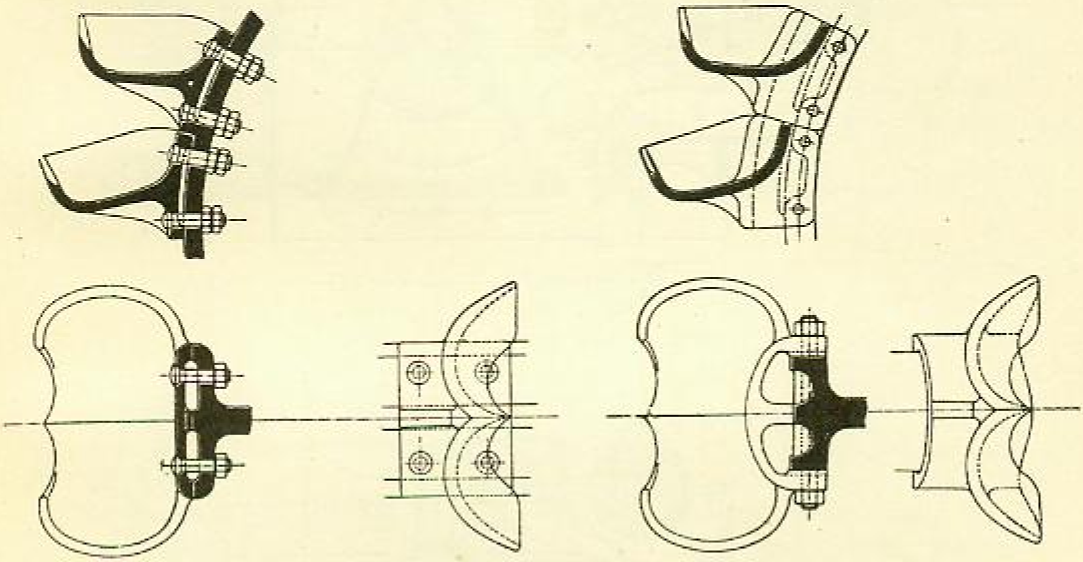


FIG. 131.—Pelton Buckets and Methods of Securing same to Rim.

direction. Both of these results are objectionable, as they lower the efficiency at partial valve openings.

As to the choice of the form of the nozzle, whether circular or rectangular, it may be said that a circular nozzle is to be preferred on theoretical grounds, for it offers the least friction on the bounding surfaces and hence a smoother jet due to a more uniform velocity of the several particles of water while, on the other hand, with a rectangular jet the particles of water which move in the angles are subject to a considerably higher resistance than those which pass along the middle of the sides of the nozzle, and this breaks up, or roughens, the jet.

In respect to ease of regulation, nozzles having a rectangular cross-section have by far the advantage. Nevertheless, the authors recommend the form of construction

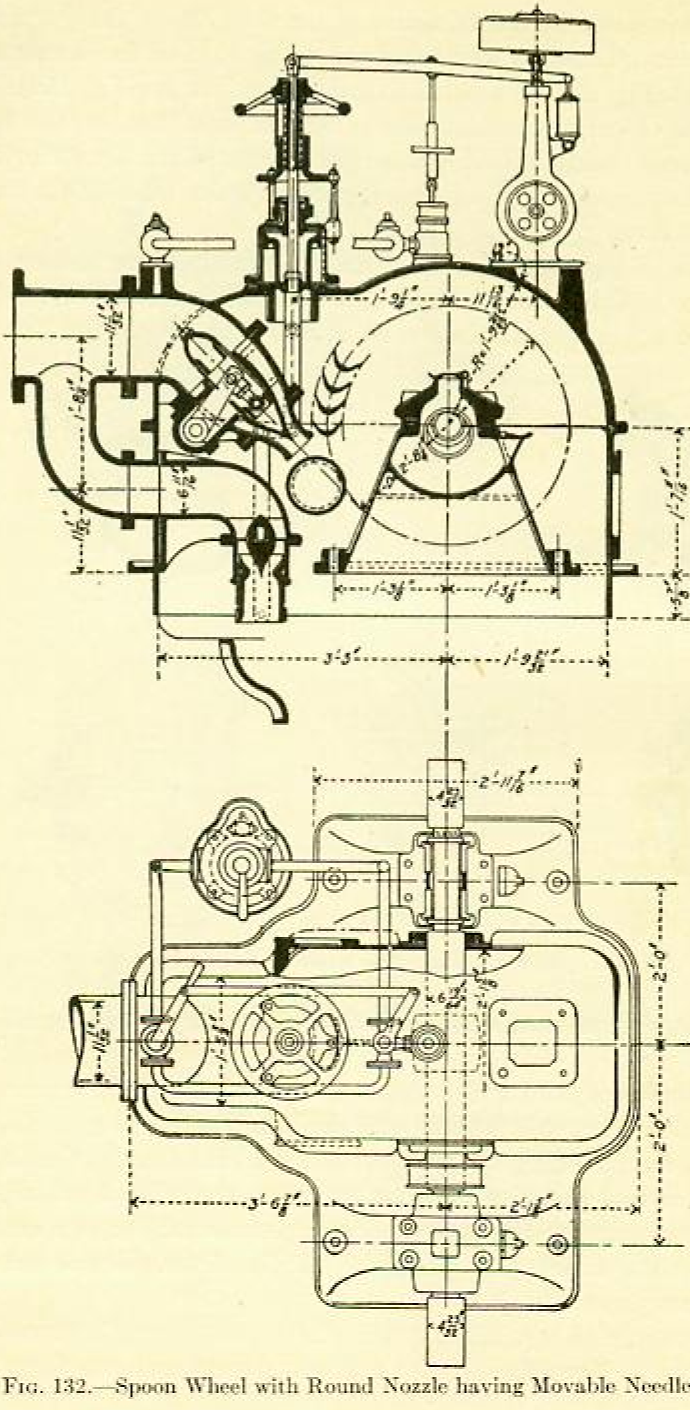


FIG. 132.—Spoon Wheel with Round Nozzle having Movable Needle.

shown in Fig. 138 and in Fig. 132, or the design customary in the United States as shown in Fig. 133.

The runner, or moving row of buckets, is formed by a circle of scoops, or cups,

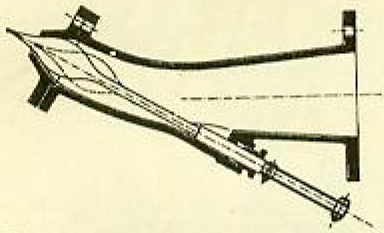


Fig. 133.—Circular Nozzle with Long Needle passing through Sheath.

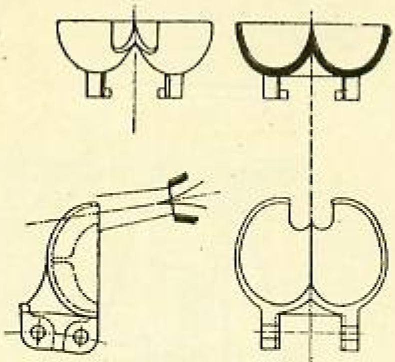


Fig. 134.—Details of Pelton Buckets.

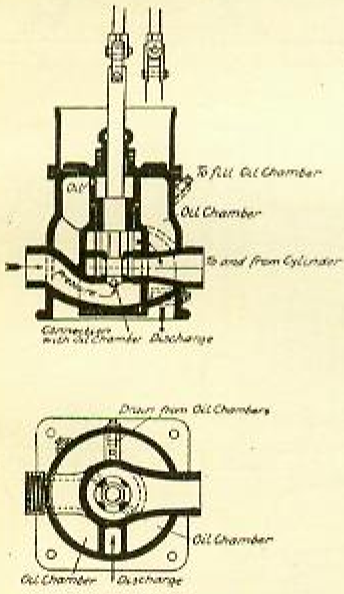


Fig. 135.—Regulating Valve for Spoon Wheel shown on Fig. 132.

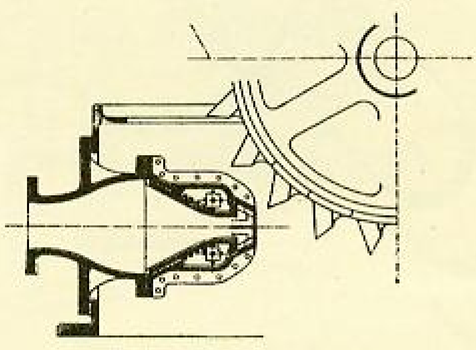


Fig. 136.—Rectangular Nozzle Closed by Parallel Motion of Inner Sheath.

cast on a rim or separately secured to it. In order to properly balance the wheel the buckets are arranged symmetrically in respect to the middle plane of the rim and the buckets must be of the same size and equal weight.

The shaft of the wheel is horizontal in almost all cases. It must have this position

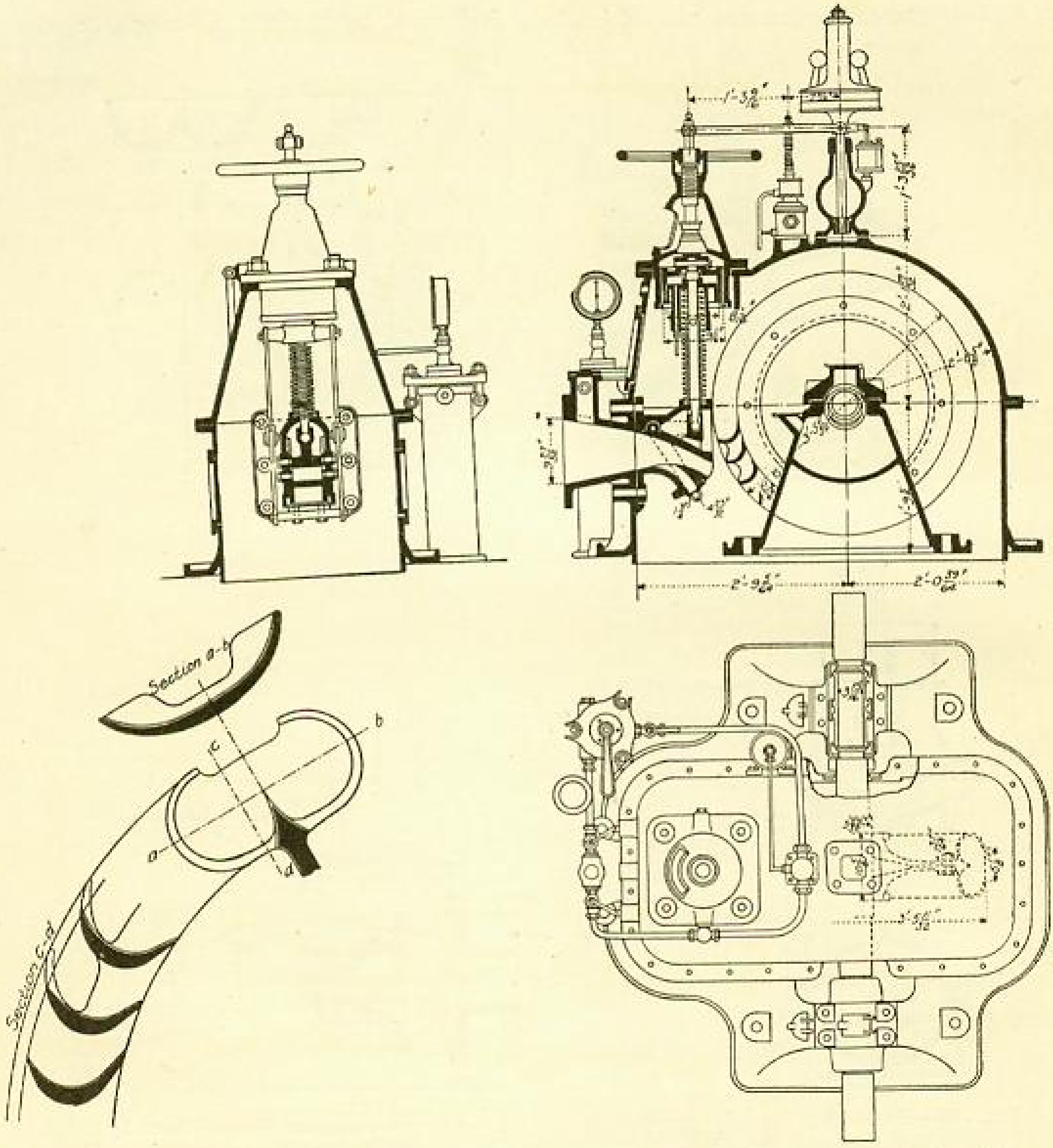


FIG. 137.—High-pressure Turbine with Adjustable Square Nozzle and Automatic Pressure Regulator.

if the water is to leave the buckets readily on both sides. If a vertical shaft is used, the water which flows from the upper half of the buckets interferes with the movement

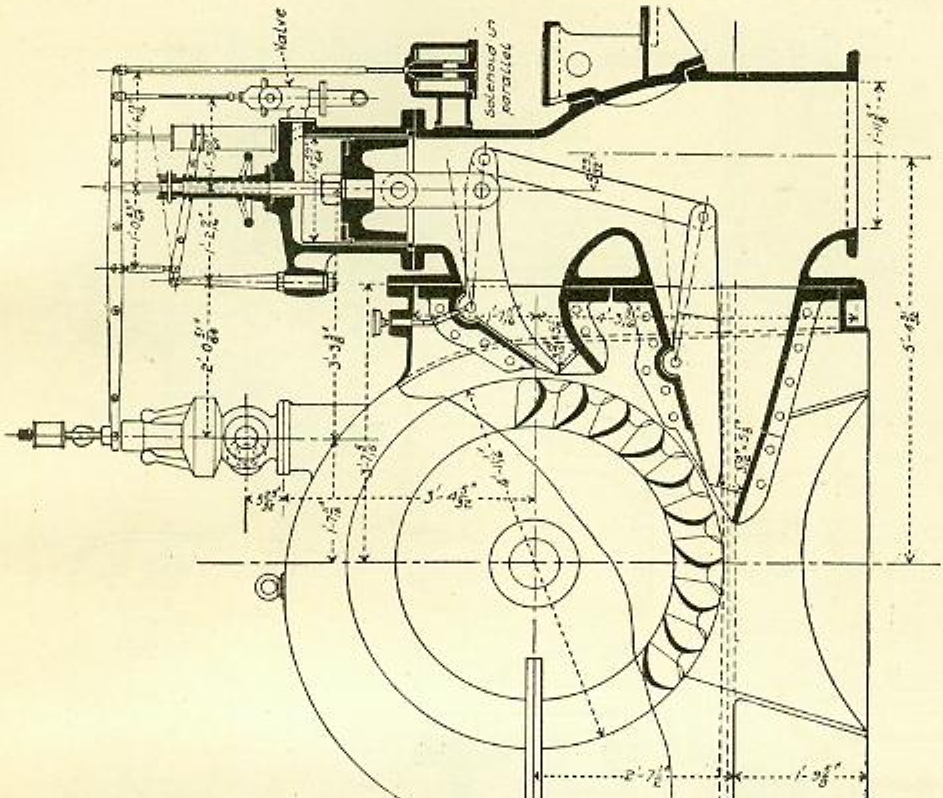


Fig. 138.—Turbine Installed by Escher, Wyss & Co.

Head = 382.9 ft. Capacity = 1250 H.P.
Volume = 38.13 sec.-ft. Speed = 300 r.p.m.

of the wheel unless special precautions are taken to prevent it from falling back on the buckets.

As in the case of Francis turbines, two or more impulse wheels may be secured to

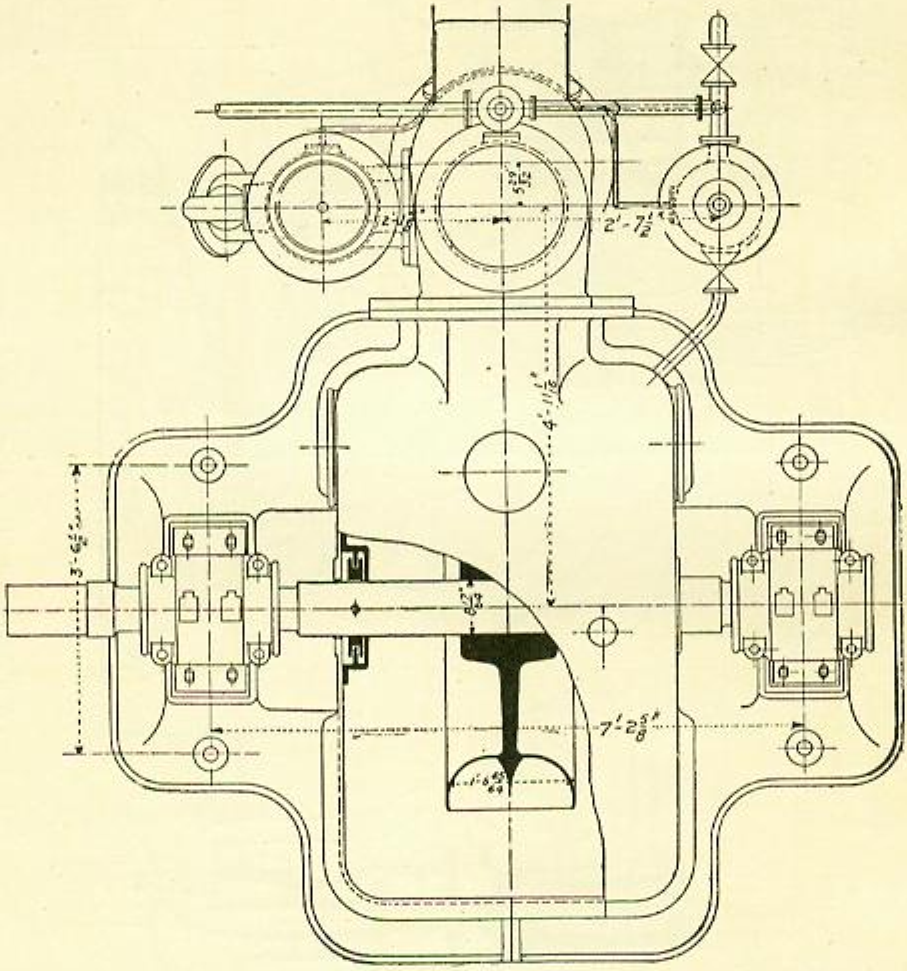


FIG. 139.—Plan View of Wheel shown on Fig. 38.

one shaft so as to subdivide the quantity of water used. This arrangement may be desirable:

(a) To obtain a higher speed with a given quantity of water and a given head;

(b) To increase the power of the unit where the speed is fixed.

The characteristic distinction between a spoon wheel and a Pelton wheel is made by the position of the entrance edge of the bucket. In the former it lies parallel to the

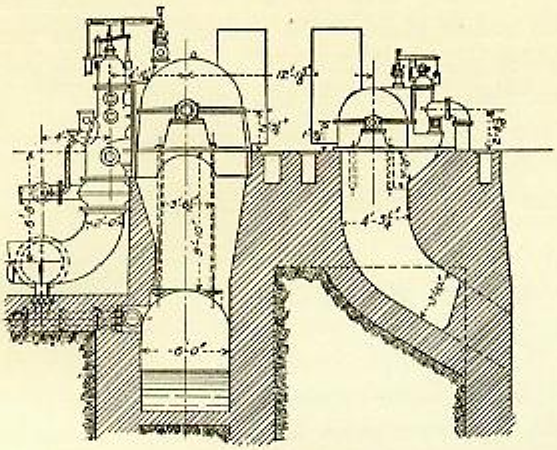


Fig. 139a.

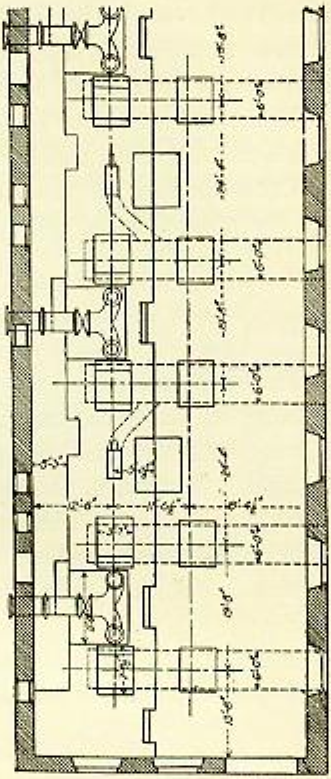


Fig. 139c.

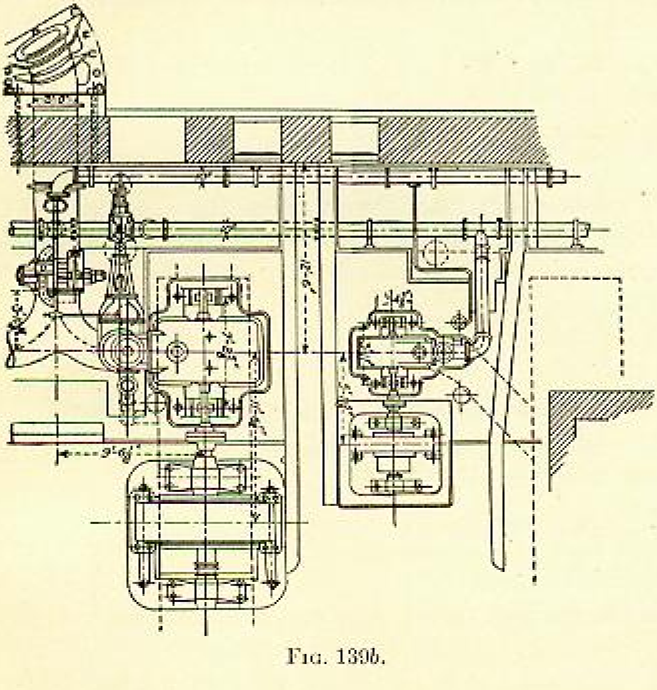


Fig. 139b.

shaft as in a Francis turbine and the water is discharged freely, not only sidewise, but also toward the inside of the wheel. (See Fig. 137.) In a Pelton wheel, however, the entrance edge of the bucket is on the middle plane of the rim and the water can leave the bucket only at the sides as the discharge toward the middle of the wheel is prevented either wholly or in great part by a partition. (See Figs. 130 and 131.)

Both forms may be combined in a single bucket, but no increase in efficiency will be obtained thereby. (See Figs. 138 and 139.)

Cross-Section of the Nozzle and the Diameter of the Wheel. The jet issuing from the nozzle has a velocity of

$$0.95\sqrt{2gH_0} \text{ to } 0.98\sqrt{2gH_0};$$

that is

$$k_{c_0}=0.95 \text{ to } 0.98.$$

This corresponds to a loss of 4–10 per cent in the apparatus connected with the nozzle. The head H_0 should be measured with a pressure gauge at the level of the runner axis, the gauge being attached to a pipe through which the veolcity of the water does not exceed $0.1\sqrt{2gH}$. Otherwise, the pressure head corresponding to the velocity must be added to the gauge reading. The head usually lost from the center line of the runner to the surface of the tail water is to be added to H_0 in order to obtain the true efficiency. The last-named loss may amount to 2 per cent or more, depending on the position of the wheel.

In order to obtain a discharge at right angles to the circumference of the wheel the peripheral velocity varies from

$$0.45\sqrt{2gH_0} \text{ to } 0.50\sqrt{2gH_0},$$

corresponding to $\phi=0.45$ to 0.50, depending upon the value of the entrance angle. We are thus in a position to determine the diameter of the wheel by the first condition as expressed in the following equation:

$$\frac{D\pi n}{60}=\phi\sqrt{2gH_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (152)$$

Practice shows that it is best to choose the value of the nozzle cross-section—or the diameter d of a circular nozzle—as having a determined relation to the diameter of the runner D. This relation may be expressed as follows:

$$d=1.2D-0.0366D^2+0.2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (153)$$

in which d is expressed in inches and D in feet.

The area of the cross-section of the nozzle is then

$$\frac{\pi d^2}{4}.$$

This nozzle cross-section is the largest one which gives a high efficiency for a wheel with diam D, but for urgent reasons it may be reduced by one-half or even more. If the nozzle is unnecessarily large the total efficiency is reduced in several ways. An increase in the cross-section of the nozzle rapidly decreases the efficiency because the water does not have sufficient space in which to flow freely from the spoons or buckets and, consequently, is dammed back in the wheel housing. It will also be seen that at the point where the water strikes the wheel an impact loss will occur in the case of the elements of water which move on the outside of the jet because the entrance can no longer be maintained free from injurious impact. By an increase in the width of the wheel and the wheel housing we may overcome this loss of efficiency, but this in turn gives rise to unnatural construction relations. It is better in such cases to divide the water among two or more nozzles of normal dimensions. If we desire to use a nozzle having a square cross-section whose sides have a length of d_1 then

$$d_1 = d\sqrt{\frac{\pi}{4}}. \qquad (154)$$

If a rectangular nozzle cross-section has a breadth b which is twice the thickness c, then

$$c = d\sqrt{\frac{\pi}{8}}, \qquad (155)$$

and

$$b = 2c. \qquad (156)$$

The quantity of water in cubic feet which flows against the wheel from each nozzle is

$$q = \frac{\pi d^2}{576} k_{c_0} \sqrt{2gH_0}, \qquad (157)$$

in which d is expressed in inches and H_0 in feet. If Z is the number of nozzles and Q the total quantity of water in cubic feet, then

$$Z = \frac{Q}{q}. \qquad (158)$$

It may be added that nozzles having circular or square cross-sections are especially adapted for Pelton wheels, while those with rectangular cross-sections are used exclusively for spoon wheels.

The following corresponding values of d, d_1, b and c have been calculated for wheels of various diameters:

Diameter of Wheels D in Inches.	Diameter of Circular Nozzles d in Inches.	Length of Sides of Square Nozzles d_1 in Inches.	Dimensions of Rectangular Nozzles with $b=2c$.		Maximum Cross-Section of Nozzles.
			b in Inches.	c in Inches.	$\frac{d^2}{4}$ in Square Ins.
8	0.984	0.872	1.234	0.617	0.7605
12	1.363	1.208	1.709	0.854	1.459
16	1.753	1.553	2.198	1.099	2.413
20	2.098	1.859	2.630	1.315	3.457
24	2.454	2.174	3.078	1.539	4.730
28	2.801	2.482	3.512	1.756	6.162
32	3.140	2.782	3.938	1.969	7.744
36	3.471	3.075	4.352	2.176	9.446
39	3.714	3.291	4.656	2.328	10.834
44	4.108	3.640	5.152	2.576	13.253
48	4.414	3.911	5.534	2.767	15.302
51	4.639	4.110	5.816	2.908	16.902
60	5.285	4.683	6.626	3.313	21.937
72	6.082	5.389	7.626	3.813	29.051

Spoon Wheels—Dimensions. For a given wheel having a diameter D, measured in feet across the entrance edges, the bucket should have a height h in inches measured in a radial direction as given by the following empirical formula:

$$h=1.8D-0.11D^2+0.2. \qquad (159)$$

This value corresponds approximately to

$$h=1.25d. \qquad (160)$$

The breadth of the bucket b, in inches, measured in an axial direction, is found by the equation

$$b=3.0D-0.0732D^2+1.18, \qquad (161)$$

or approximately

$$b \simeq 3d.$$

The wheel housing should be made at least three times the breadth of the buckets in order to be sure that the water has ample room to flow away from the wheel.

In determining the entrance angle we must meet the condition that the water must surely enter the bucket. If the regulating system is such that the middle water thread remains unchanged in its position during the period of regulation, then the said thread must be tangent to a circle having a diameter $\leq D-h$. In the extreme case the entrance angle α will, therefore, have the following minimum value:

$$\cos\alpha=\frac{D-h}{D}. \qquad (162)$$

If, however, a method of regulation is employed which diverts the jet from the wheel by the closing of the nozzle (which, for example, is the case when the jet is changed by a contraction from the upper side as in Fig. 137, then the middle water thread should be tangent at full gate opening to a circle having a maximum diameter of $D-\frac{1}{2}h$. This corresponds to a mean entrance angle α', whose value is expressed by the equation

$$\cos\alpha'=\frac{D-\frac{1}{2}h}{D}. \qquad (163)$$

The following table contains the values of h, B, b, α and α', calculated in accordance with the foregoing directions, for wheels of various diameters D:

Diameter in Inches.	Diameter of Circular Nozzles d in Inches.	Bucket Height h in Inches.	Bucket Breadth b in Inches.	Breadth of Housing B in Inches.	α	α'
8	0.984	1.351	3.147	9.50	33° 50′	39° 10′
12	1.363	1.890	4.107	12.50	32° 30′	37° 50′
16	1.753	2.404	5.050	15.25	31° 50′	36° 50′
20	2.098	2.894	5.977	18.00	31° 10′	36° 10′
24	2.454	3.360	6.887	20.75	30° 40′	35° 30′
28	2.801	3.801	7.782	23.50	30° 10′	35° 00′
32	3.140	4.218	8.660	26.00	29° 50′	34° 30′
36	3.471	4.610	9.521	28.75	29° 20′	34° 00′
39	3.714	4.888	10.157	30.50	29° 00′	33° 40′
44	4.108	5.321	11.196	33.75	28° 30′	33° 00′
48	4.414	5.640	12.009	36.25	28° 00′	32° 30′
51	4.639	5.863	12.608	38.00	27° 50′	32° 10′
60	5.285	6.450	14.350	43.25	26° 50′	31° 00′
72	6.082	7.040	16.545	54.25	25° 30′	29° 40′

The number of buckets and their spacing must be such as to meet the following requirements:

(*a*) There must be a sufficient discharge cross-section between the several spoon buckets so that the loss due to the discharge may not become excessive.

(*b*) The buckets on which the jet impinges simultaneously must be placed sufficiently near to each other so that the edges of the jet may enter the buckets as free as possible from injurious impact.

In order to satisfy both conditions we make the spacing

$$t=0.4 \text{ to } 0.45\frac{d}{\sin\alpha} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (164)$$

and the number of buckets z is then

$$z=\frac{\pi D}{t}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (165)$$

By the use of the numerical values given in the last table we may then obtain the following:

TABLE SHOWING THE NUMBER OF BUCKETS FOR SPOON WHEELS

Diameter d in Inches.	Number of Buckets z.		Diameter d in Inches.	Number of Buckets z.	
	From	To		From	To
8	31	35	36	36	40
12	32	37	39	36	40
16	33	37	44	36	40
20	34	38	48	36	40
24	35	39	51	36	40
28	35	39	60	36	40
32	36	40	72	36	40

For the graphical representation of a spoon wheel see Fig. 137.

The Discharge Cross-Section and the Relation between the Angles. The entrance angle of the runner bucket β measured on the periphery of the wheel is evidently given by the following equation, the entrance being assumed to be free from wasteful impact:

$$\frac{w_0}{v_0}=\frac{\sin\alpha}{\sin(\beta-\alpha)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (166)$$

in which w_0 may be obtained by the equation

$$w_0=\sqrt{v_0^2+c_0^2-2v_0c_0\cos\alpha}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (167)$$

As v_0, c_0 and α are considered as known from the fundamental equation for right-angled discharge

$$c_0v_0\cos\alpha=g\varepsilon H,$$

so w_0 and β may be calculated and the entrance diagram drawn.

The velocity of the discharge, w_a, measured in the direction of the flowing water, is given for any middle point of the discharge cross-section by the equation

$$w_a=\sqrt{w_e^2-(v_e^2-v_a^2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (168)$$

in which, subject to connection later, we may place

$$w_e \simeq w_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (169)$$

Further, v_e and v_a, the circumferential velocities, measured respectively at the middle of the entrance and discharge cross-sections, may be calculated The loss in the runner is to be assumed as heretofore from the value of c_0.

If the loss in the discharge is given a mean value of $0.05\ H_0$, it corresponds to a discharge velocity of

$$c_a \simeq 0.225\sqrt{2gH_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (170)$$

Assuming said loss to be known we may now determine the discharge diagram from the values of v_a, w_a and c_a. If we now find that c_a is not at right angles to the peripheral velocity, as we expected, then we must change the speed corresponding to v_0, v_e and v_a in order to sufficiently fulfill the requirements. With these new values of v_0, v_e and v_a the calculation may be repeated and at the same time w_e may be finally interpolated between w_0 and w_a.

w_e and w_a are thus determined for the size of the entrance and discharge cross-sections. With a right-angled discharge the resulting mean discharge angle varies on the one hand with the diameter of the runner and on the other hand with the distance from the axis to the several points on the discharge edge. For example, in the case of a wheel having a diameter of 20 inches we find, by reference to the tabulated values, that the discharge angle at the outer periphery of the bucket is $\gamma \simeq 24°$; while at the inner periphery it is $\sim 30°$, and half way between is $\sim 27°$. For wheels having a diameter less than 20 inches this mean value is somewhat greater, while for wheels more than 20 inches in diameter it is somewhat smaller.

Practical Construction. We may proceed as follows in the practical design of the form of the bucket: We choose the curve of the bucket in such a manner that the apparent water thread is as nearly as possible at right angles to the discharge edge, although of course this can be accomplished only approximately. Then we design the cross-section of the bucket in such a way that it may be easily built—somewhat like Fig. 137, and draw out on this the form of the bucket with due regard to the angles β and γ, which have been calculated. One must not lose sight of the fact that the angle γ is referred to the direction of the flowing water and that the section of the vane does not lie in this direction. This makes a reconstruction necessary to determine the discharge angle γ as it appears in the section already taken.

In order to determine the area of the discharge cross-section we must estimate those lengths of the discharge edge which effect the discharge of the water, and which amount to about two-thirds of the total lengths of the buckets. Strictly speaking we should substitute for the lengths of the discharge edge the somewhat shorter lengths of the center line of the discharge cross-section.

With the width of the discharge tentatively fixed by the drawing we may then determine the discharge cross-section and by multiplying this area by the discharge velocity w_a we will have the quantity of water which may be used in each bucket. The quantity thus determined should be only about 10 per cent to 20 per cent greater than that which issues from the nozzle and flows upon one bucket under the most unfavorable conditions. This case arises, for example, when with a circular nozzle the middle water thread falls on the middle of the space between two buckets.

If the discharge cross-section does not satisfy this condition nothing remains but to change the curve of the bucket, or if possible to adapt the cross-section of the nozzle to the discharge cross-section already found, by a change in the number of nozzles. The dimensions of the wheel as stated in the tables must then be made to agree with the nozzle cross-section last assumed.

As in the case of freely radiating turbines this surplus in the discharge cross-section is necessary in order to prevent the contact of the jet with the back of the buckets, otherwise a back pressure easily arises in the discharge cross-section, which is not only transmitted backward to the nozzle, thus lowering the coefficient of discharge, but also acts as a kind of a dam and thus fills the wheel housing. The latter action naturally has an extremely unfavorable effect upon the efficiency of this kind of a freely radiating turbine.

Pelton Wheels—Dimensions. The distinctive feature of this form of construction consists in the fact that the tangential jet impinging upon the wheel is divided into two equal parts by a middle partition on the bucket which lies in the symmetrical plane of the wheel and then flows from the bucket toward the sides in halves. (See Fig. 131.) The absolute path of the several particles of water therefore falls approximately tangentially to the wheel. It is inherent to the form of bucket that little or no water can be discharged toward the inside of the wheel. Consequently the necessary discharge cross-section must be obtained by broadening the bucket and increasing its height as compared with spoon wheels. Using designations similar to those employed for spoon wheels we express the bucket height, h, measured in inches in a radial direction as

$$h=2.4\,D-0.1465\,D^2+0.2, \quad \ldots \ldots \ldots \quad (171)$$

where D is measured in feet across the parting edges, or knives, of the buckets.

The breadth of the bucket, b, should be

$$b=3.6D-0.0914D^2+1.18. \quad \ldots \ldots \ldots \quad (172)$$

These equations are based upon the assumption that the maximum cross-section of the nozzle used for any wheel of the diameter D has the area $\frac{\pi d^2}{4}$, where d is determined by the equation

$$d = 1.2D - 0.0366D^2 + 0.2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (173)$$

The following are approximate values:

$$h \cong 1.7\,d. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (174)$$

$$b \cong 3.3\,d. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (175)$$

If for a wheel of the diameter D we choose a smaller nozzle having a diameter $d_0 < d$, it is evident that in constructing the wheel we must use a smaller bucket whose dimensions shall correspond to the smaller nozzle.

The breadth of the housing should be $\cong 3b$; i.e., greater than in the case of a spoon wheel of equal capacity.

The entrance angle, i.e., the angle due to the position of the jet—is determined from the condition that the water thread at the center of the jet is tangent to a circle whose diameter is $D - \frac{4}{3}h$, it being understood that the jet maintains its position unchanged during the period of regulation (for example, when the nozzle is regulated by a needle valve), on the contrary if the jet is forced away from the wheel by closing the nozzle outwardly, that is, away from the bucket, then when the wheel is under full load the jet should be tangent to a circle having the diameter $D - h$.

It may be said in reference to the number of buckets that Pelton wheels require fewer buckets than spoon wheels. This is due to the fact that even though the spacing with the former is greater, yet the entrance relations remain essentially unchanged, no matter what position the bucket may occupy in respect to the jet. In the table below are given the number of buckets for Pelton wheels which satisfy the necessary conditions.

The impinging jet should strike at right angles against the knife, or cutting, edge of a bucket, at the moment when the quantity of water entering that bucket is a maximum.

In order that the entrance to a Pelton bucket should be free from impact the entrance angle β should be zero, as c_0 and v_0 fall in one direction. Although it is not practicable to accomplish this we should be careful to make β as small as possible for all the water threads; that is, b should be chosen as large as possible in relation to the nozzle diameter d. From this point of view it is evident that a Pelton bucket must be larger than a spoon bucket under conditions otherwise similar.

In accordance with the foregoing equations and directions, the following table has been prepared to give the data necessary for Pelton wheels of various diameters:

TABLE SHOWING DIMENSIONS FOR PELTON WHEELS

Diameter of Wheel D. Inches.	Diameter of Round Nozzle d. Inches.	Length of Sides of Square Nozzle d. Inches.	Area of Cross-section of Nozzle. Sq. Ins.	Height of Bucket h. Inches.	Breadth of Bucket b. Inches.	Breadth of Wheel Housing B. Inches.	Angle Indicating Position of Jet for $\cos\alpha = \frac{D-h}{D}$	Angle Indicating Position of Jet for $\cos\alpha' = \frac{D-\frac{3}{4}h}{D}$	Number of Buckets
8	0.984	0.872	0.7605	1.735	3.539	10.75	38° 30′	33° 10′	16
12	1.363	1.208	1.459	2.454	4.689	14.25	37° 20′	32° 10′	16
16	1.753	1.553	2.413	3.140	5.818	17.50	36° 30′	31° 30′	18
20	2.098	1.859	3.457	3.793	6.923	21.00	35° 50′	31° 00′	18
24	2.454	2.174	4.730	4.414	8.014	24.25	35° 20′	30° 30′	18
28	2.801	2.482	6.162	5.002	9.082	27.25	34° 50′	30° 00′	20
32	3.140	2.782	7.744	5.558	10.130	30.50	34° 20′	29° 30′	20
36	3.471	3.075	9.446	6.081	11.157	33.50	33° 50′	29° 10′	20
39	3.714	3.291	10.834	6.453	11.915	35.75	33° 30′	28° 50′	20
44	4.108	3.640	13.253	7.030	13.151	39.50	32° 50′	28° 20′	22
48	4.414	3.911	15.302	7.456	14.118	42.50	32° 20′	28° 00′	22
51	4.639	4.110	16.902	7.754	14.829	44.50	32° 00′	27° 40′	22
60	5.285	4.683	21.937	8.538	16.895	50.75	31° 00′	26° 40′	22
72	6.082	5.389	29.051	9.326	19.390	58.25	29° 30′	25° 30′	24

Computing the Discharge Cross-Section. For each half of a Pelton bucket the jet may be considered as having a cross-section in the form of a half circle. The middle water thread, which may be imagined as drawn through the center of gravity of the half circle, enters the entrance cross-section at the point P (see Fig. 140), where there occurs a deflection equal to the angle β. The size of this angle β would be the same as one-half of the knife angle at A if the jet were uniformly divided in all directions at the point of entrance. This is not actually the case, especially as immediately after its entrance the jet is spread out in the direction of the parting edge so that the particles of water on the exterior of the jet are subject to an abrupt diversion, not at Q but at Q'. The angle of such diversion is β', an angle large thanr β, and hence corresponding to an increase impact loss. Although it may not be possible to accurately calculate the angle β for a given particle of water, yet it is clear that if the jet is broad in proportion to the bucket then the entrance loss due to impact is undoubtedly

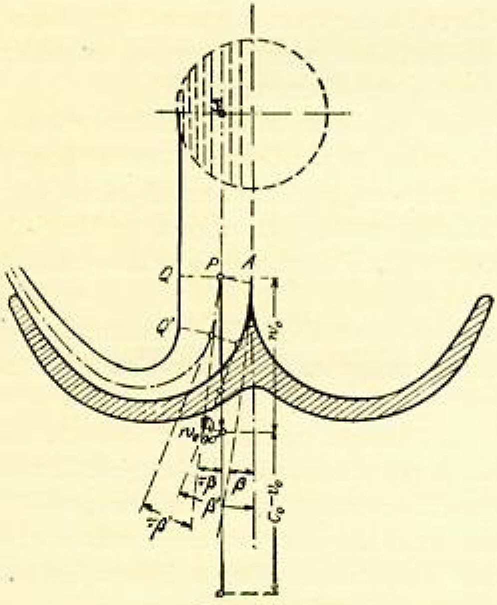


Fig. 140.

increased, and so must adversely effect the efficiency of the turbine. It is therefore apparent

(*a*) That the jet should be narrow in proportion to the width of the wheel;

(*b*) The knife angle β should be as small as possible.

The velocity w_0 relative to the moving wheel, as measured in the direction of the jet, is

$$w_0 = c_0 - v_0. \qquad (176)$$

The useful part of this velocity is

$$w_e = w_0 \cos \beta, \qquad (177)$$

while $w_0 \sin \beta$ is lost in useless impact. Strictly speaking, we should consider the subdivision of v_0 in accordance with the position of the several water threads, which is possible by a separation into partial buckets.

We may then proceed exactly as in the case of spoon turbines; that is, we first ascertain the discharge angle γ and then draw out the form of the bucket from the data above given. The resulting discharge cross-section must then be made to agree with its proper value.

Practical Construction. Pelton buckets may be cast in one piece with the hub of the wheel, as a separate rim, as segments of the rim, or individually. In the last three cases there must be a solid connection with the hub. The method used in constructing the wheel and securing the buckets depends upon the size of the wheel, the quality and kind of material employed and upon the requirements of using as few bucket models as possible. The last consideration requires each bucket to be made separately and bolted individually to the rim of the wheel. Two well-known methods of fastening the buckets in place are shown in Fig. 131.

Finally in order to obtain a high efficiency it is necessary to polish the working faces of the buckets so as to overcome the roughness of the castings.

PART III

NOTABLE TURBINE INSTALLATIONS

The canal, wheel pit, and tunnel in the plant under consideration were completed in 1904 to provide for a development of 100,000 E.H.P., while at that date only one-half of the power house was constructed to shelter machinery having a capacity of 50,000 E.H.P.

Owing to the necessity of comforming to æsthetic requirements, the plant is located so that the center of the wheel pit is practically 360 feet from the river shore and is at such a distance above the falls that the discharge tunnel has a length of 2164 feet. The reader will note that, had conditions permitted, a cheaper and more

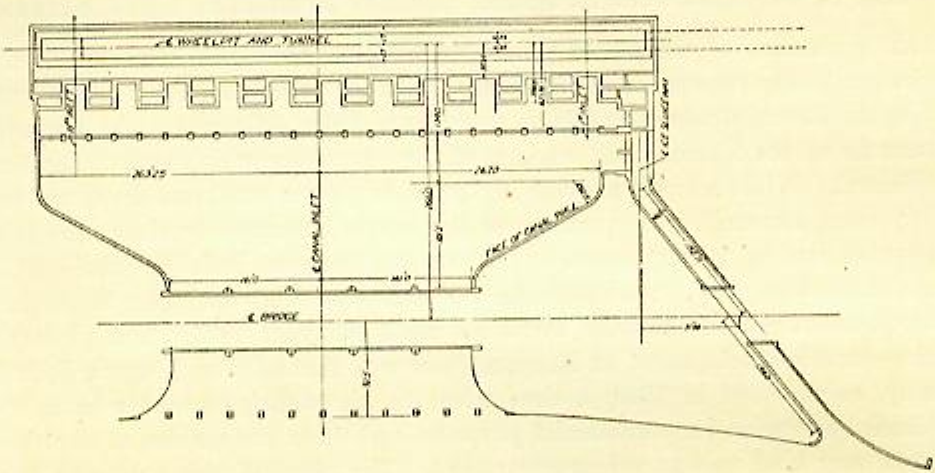


FIG. 141.—Map showing Canal, Wheelpit and Ice Run of Canadians Niagara Power Co.

satisfactory development could have been made upon the bank of the river at a point near the crest of the falls.

On Fig. 141 is shown a general plan of the canal, forebay, and wheel pit. The entrance to the canal is in the midst of the rapids above the falls, and a depth of 15 feet at mean water is provided. The area of waterways is so selected as to give an entrance velocity of 2.35 feet per second, increasing to 3 feet per second at the bridge. The general form of the canal was predetermined by the length of the wheel pit, which, in turn, was selected so as to give a spacing of 48 feet between the centers of the 10,000 E.H.P. units, a size materially in advance of anything theretofore adopted. The use of these larger units represented a material advance in hydro-electric development and greatly reduced the cost of construction.

In common with all developments located in a northern climate, particular reference was made in the design of the plant to protection against ice, which protection is especially necessary in the plant under consideration, as it is located below a part of the rapids of the Niagara River, a location to be avoided wherever possible. To divert the ice from the canal a rack of 2-inch round bars is attached to the piers shown at the bottom of the Plate. On account of frazil, the rack could not be carried to the bottom of the river, but projects only 4.8 feet below mean water. To further protect the forebay from ice the outer wall of the power house is carried on a series of submerged arches shown on Fig. 142, a photograph taken before water was admitted to the canal. In order to remove from the canal such ice as may there accumulate, the channel shown on Fig. 141 is built to connect with the river at a point about 500 feet below the center line of the canal. On account of the rapid fall in the river sufficient velocity is thus obtained for the discharge of ice through the gates separating

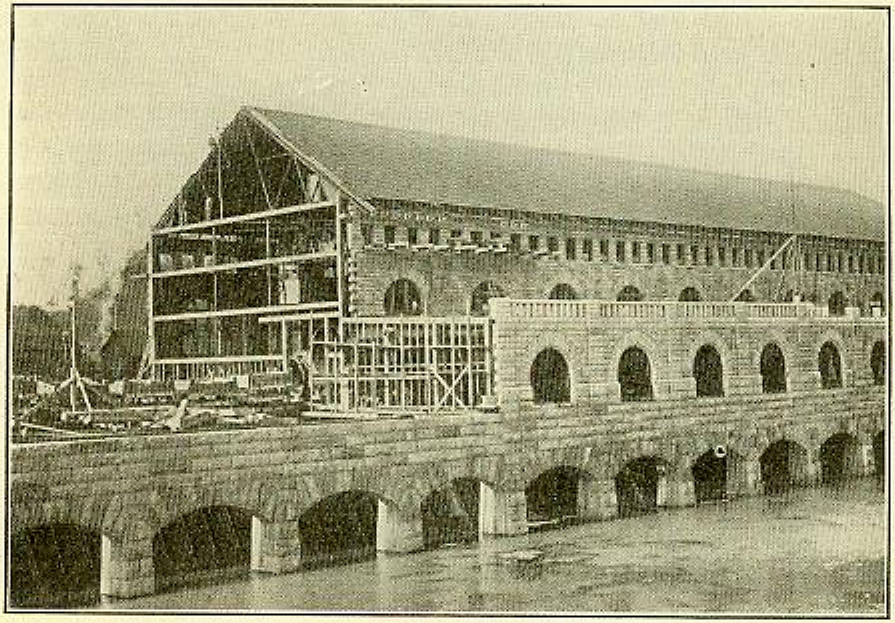


Fig. 142.—Submerged Arches in Outer Wall of Forebay of Canadian Niagara Power Co.

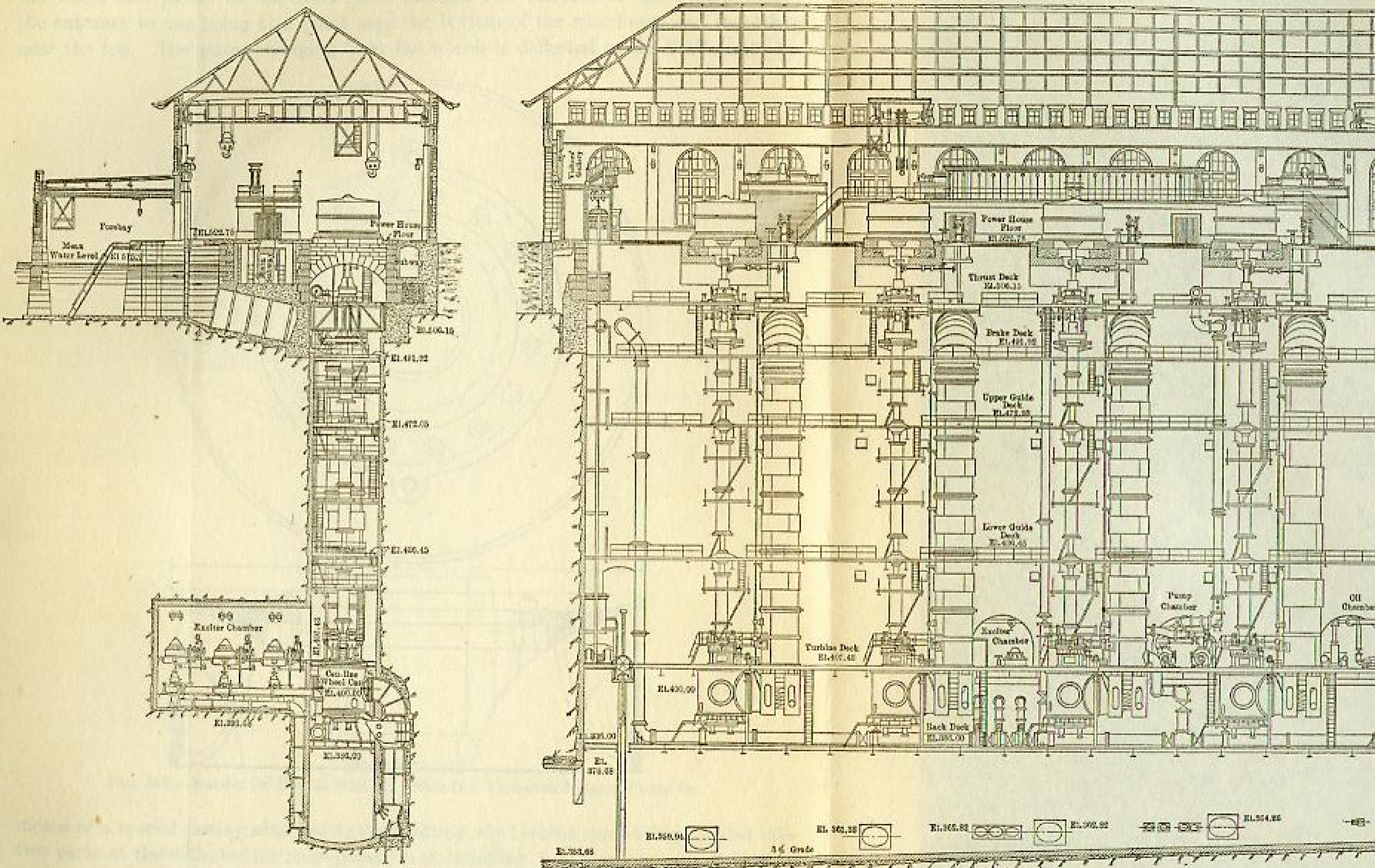
the sluiceway from the canal. A continuous rack of inclined bars further protects the entrance to the inlets leading to the penstocks.

The wheel pit consists of an excavation 567 feet long, 20.5 feet wide, with a maximum depth of 171 feet, such excavation being made through limestone and shale with the exception of about 14 feet of earth covering. The maximum depth was determined by the size and gradient of the discharge tunnel, which had been fixed in the early part of the development. The bottom of the wheel pit is given a grade of 3 per cent to approximately equalize the velocity of the discharge water at sections on center lines of the several units. The walls are lined with brick to protect the machinery and operators from the water contained in the rock strata through which the excavation is made. It was known that the rock walls of the wheel pit would move inwardly and, therefore, provision was made for the attachment of all floor beams, girders, and machinery in such a manner as to prevent their distortion or destruction from such movements. After moving inwardly from 3 to 4 inches, and thus relieving the shrinkage stresses in the earth's crust, the walls have reached a condition of equilibrium where only slight movements take place due to difference in temperature.

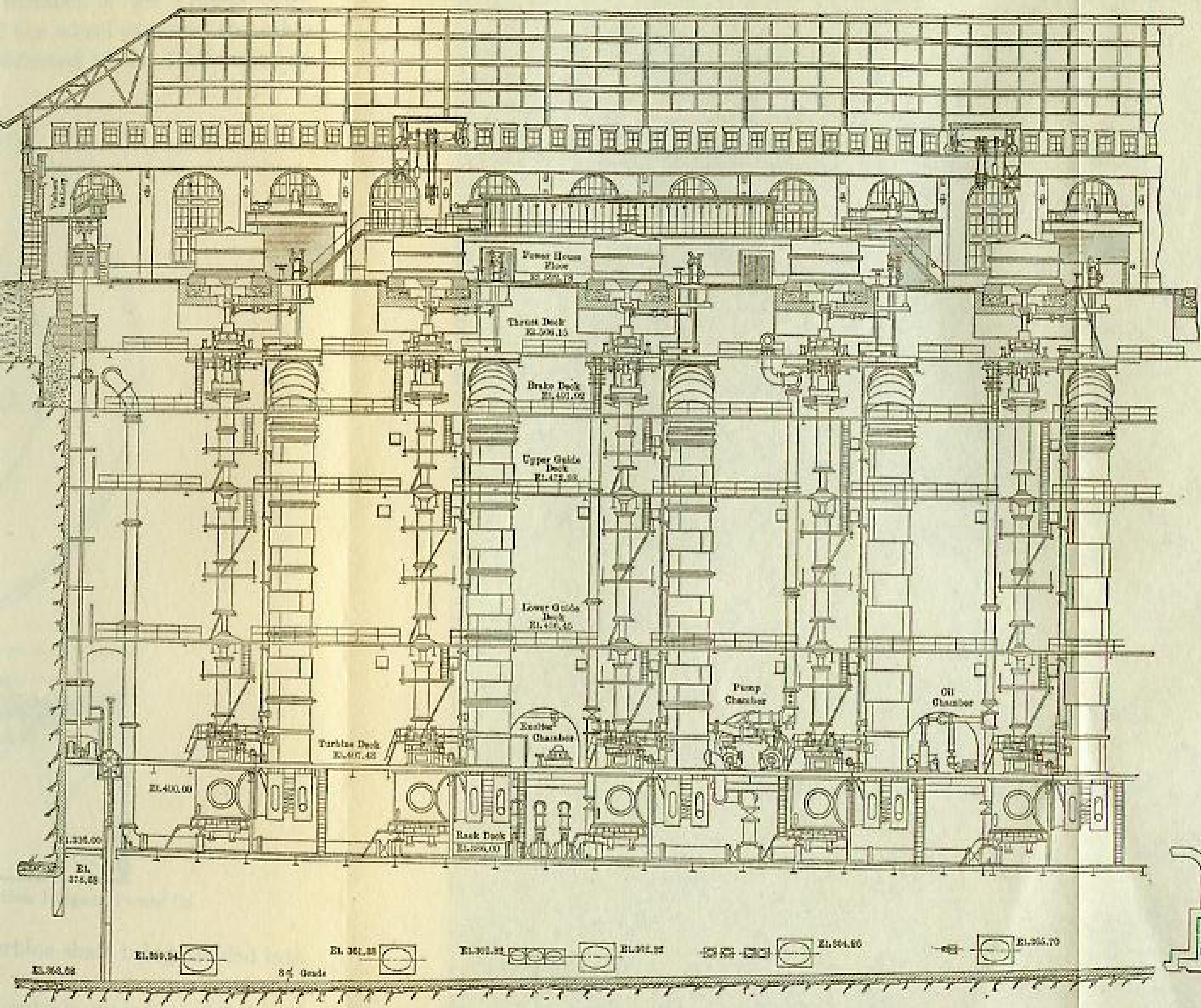
The discharge tunnel is of horseshoe form, having a height of 25 feet, a maximum span of 18 feet 10 inches, and a waterway of 404.8 feet, the gradient being 7 feet per thousand to a point 103 feet from the portal, whence the fall is 11.2 feet in a reverse vertical curve to the point of discharge. The ogee form adopted at the portal reduced the total available head for power generation, but has the advantage of affording an opportunity for inspection of the tunnel when the plant is not in operation. The tunnel roof is timbered, the arch being composed of four courses of brickwork, while the remainder of the section is lined throughout with concrete protected by a facing of one course of vitrified brick. As the velocity of the water when the tunnel is operating to its full capacity will be 31.4 feet per second, it was decided that concrete alone would not be sufficient to withstand the erosion, and experience has proven the wisdom of this decision.

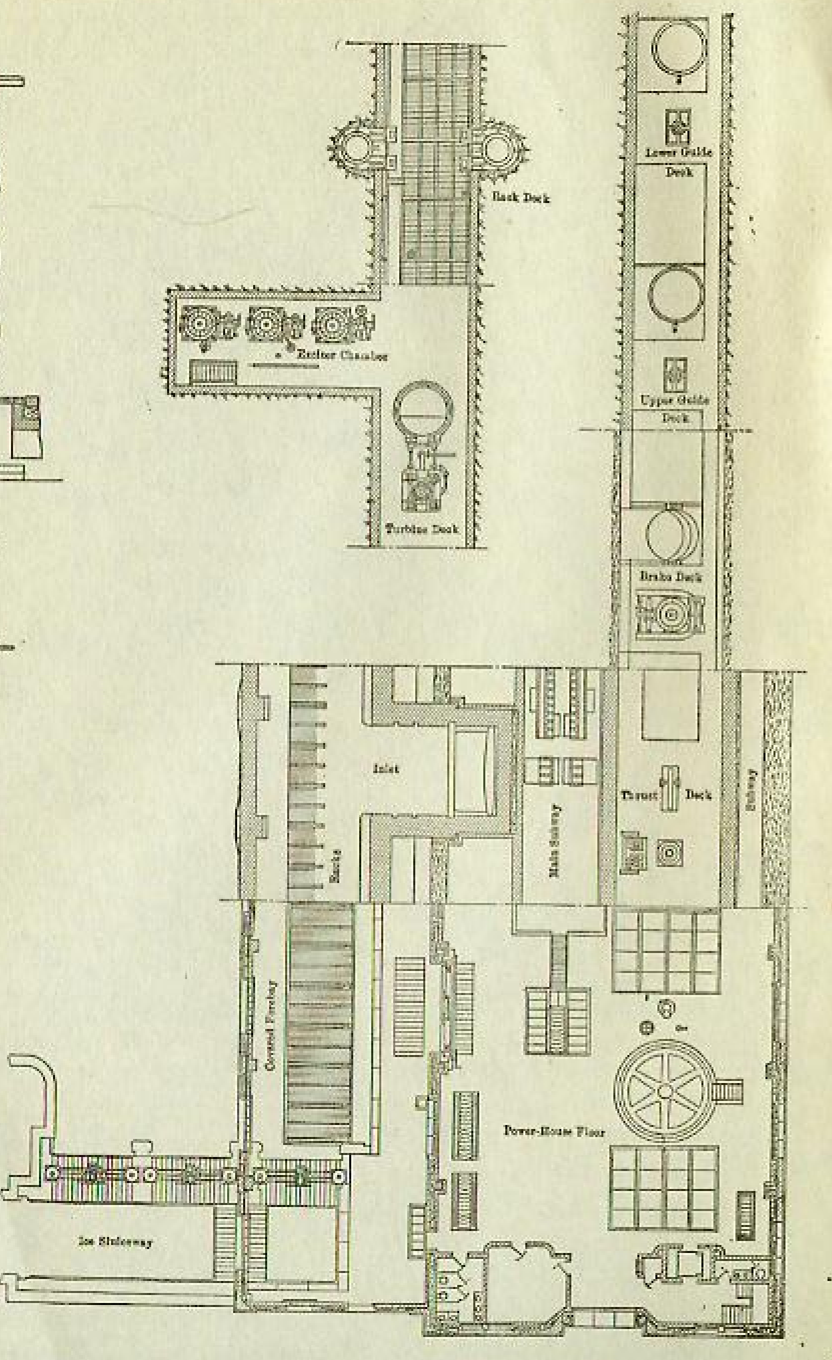
The size and gradient of the tunnel determined the elevation of the water in the bottom of the wheel pit, and hence the maximum head under which the turbines would operate, such head being normally 136 feet, which will be reduced to 130.5 feet under extreme conditions. An interesting question arose as to whether such head should be utilized by means of horizontal, or vertical-shaft turbines, the decision in favor of vertical shafting being made prior to a determination of the form and size of the wheel pit. While a construction with horizontal shafts offers many advantages in the way of simplicity of construction, the vertical-shaft units were adopted largely for the convenience thereby afforded for operation and maintenance and because it was not known in advance whether the wheel pit could be made sufficiently dry to allow the installation of electrical machinery therein.

The general form of the unit finally adopted is shown on Fig. 143. It will be seen that the motor water from the canal passes through masonry inlets to penstock mouth-pieces located beneath the power-house floor, whence it descends the wheel pit and enters the single wheel case; whence, after passing through the wheels, it is discharged by means of draft tubes into the tailrace. The entrance to the penstock is protected by motor-operated gates 19 feet 2 inches, in width and 17 feet 10¾ inches high. The diameter of the penstock is 10 feet 2 inches, the velocity of the water being 11.2 feet per second with maximum load on the unit. Through a cast-steel elbow weighing over 90,000 pounds the water enters a steel-plate wheel case 13 feet in diameter and 14 feet in height, the form of the same, together with its relation to the turbine machinery and draft tubes, being shown on Fig. 144. The wheel case as well as



SECTIONS OF POWER-HOUSE AND WHEEL-PIT
SHOWING MACHINERY INSTALLED
CANADIAN NIAGARA POWER COMPANY
FIG. 143.





SECTIONS OF POWER-HOUSE AND WHEEL-PIT
SHOWING MACHINERY INSTALLED
CANADIAN NIAGARA POWER COMPANY
FIG. 143.

[To face page 216.

the penstock elbow is supported on bracket castings so arranged as to allow adjustment of the parts in their correct position. It will be seen that the water having entered the wheel case passes to the draft chest through two turbines of the Francis type, the entrance to one being at a point near the bottom of the wheel case and the other near the top. The water emerging from the wheels is deflected to the draft tubes by

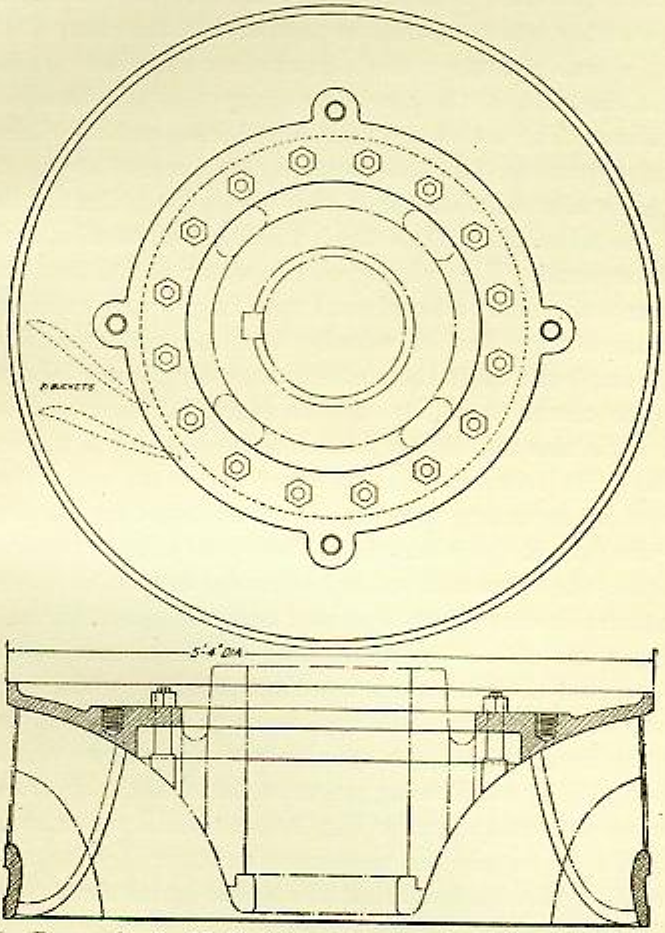


FIG. 145.—Runner for Double Turbine, 10000 H.P. Canadian Niagara Power Co.

means of a special casting attached to the shafting, the turbine shaft being divided into two parts at the deflector for convenience in dismantling.

As shown on Fig. 145, the turbine wheels have an outside diameter of 5 feet 4 inches and a height of 1 foot $11\frac{1}{2}$ inches, the form of the buckets being indicated on the drawing. The distributors have an outside diameter of 6 feet 9 inches and between the distributors and the wheels is provided a space for the cylinder gates which furnish the regulation, a close fit being provided for said gates. Both the wheels and distributors are made of bronze containing 85 per cent Cu, 10 per cent Sn, 3 per cent Pb, and 2 per cent Z. The necessary power could have been developed by the use of a single wheel, but to have done so would have reduced the velocity of the revolving part, thus increasing the size and cost of the alternator. Further it would have been difficult to have provided a satisfactory draft tube for the single wheel. As the plant under consideration was to operate in parallel with the plant of The Niagara Falls Power Company, it was necessary that the number of cycles in the current should agree with that in the plant of the parent company, and this consideration materially affected the situation. The use of a second wheel increased the difficulties of arranging the governing apparatus, but the problem was successfully met.

The discharged water from the draft chest enters two draft tubes built into the opposite walls of the wheel pit. These draft tubes are of peculiar design, the diameter being uniform throughout. The velocity of the water at full load is unusually high, being 21 feet per second. This arrangement may be properly criticised on the ground that there is an unnecessary loss in velocity head as well as in frictional resistance. However, to have employed draft tubes of theoretically correct form would have been difficult owing to the character of the rock in which the excavations therefor would have been made. It is also true that a portion of the velocity of the water issuing from the bottom of the draft tube is useful in imparting to the water flowing toward the tunnel a portion of its necessary velocity, the draft tubes discharging at an angle of 45 degrees with the axis of the wheel pit. Though a tailrace regulating gate is provided, it is found that the vacuum is not lost when the draft-tube outlets are uncovered, and that the plant may be successfully operated under these conditions.

The revolving parts of the hydraulic and electrical machinery weigh 240,000 pounds, and the support of this mass is a serious problem. With the design adopted this weight is supported in the following manner: (*a*) By full penstock pressure admitted below the lower wheel, the space above the upper wheel communicating with the draft chest; (*b*) by a balancing piston attached to the shaft about 6 feet above the top of the wheel case as shown on Fig. 144, a special water connection from the canal being provided to supply the necessary pressure; (*c*) by an oil-step bearing attached to the shaft about 15 feet below the power-house floor, such bearing consisting essentially of two discs, one being stationary and the other attached to the shaft, a film of oil being maintained between said discs at all times by special high-pressure pumps.

The speed regulation is afforded by oil-pressure governors supplied by separate high-pressure pumps, the governors being located on the power-house floor for convenience of operation and communicating by means of levers, racks, and pinions to the gate rods shown on Fig. 144. The governor is single acting, the gates closing by gravity. As with a short circuit the entire load of 10,000 E.H.P. is liable to be instantly

CROSS-SECTION
OF 10000 H. P. TURBINE
CANADIAN NIAGARA POWER
COMPANY

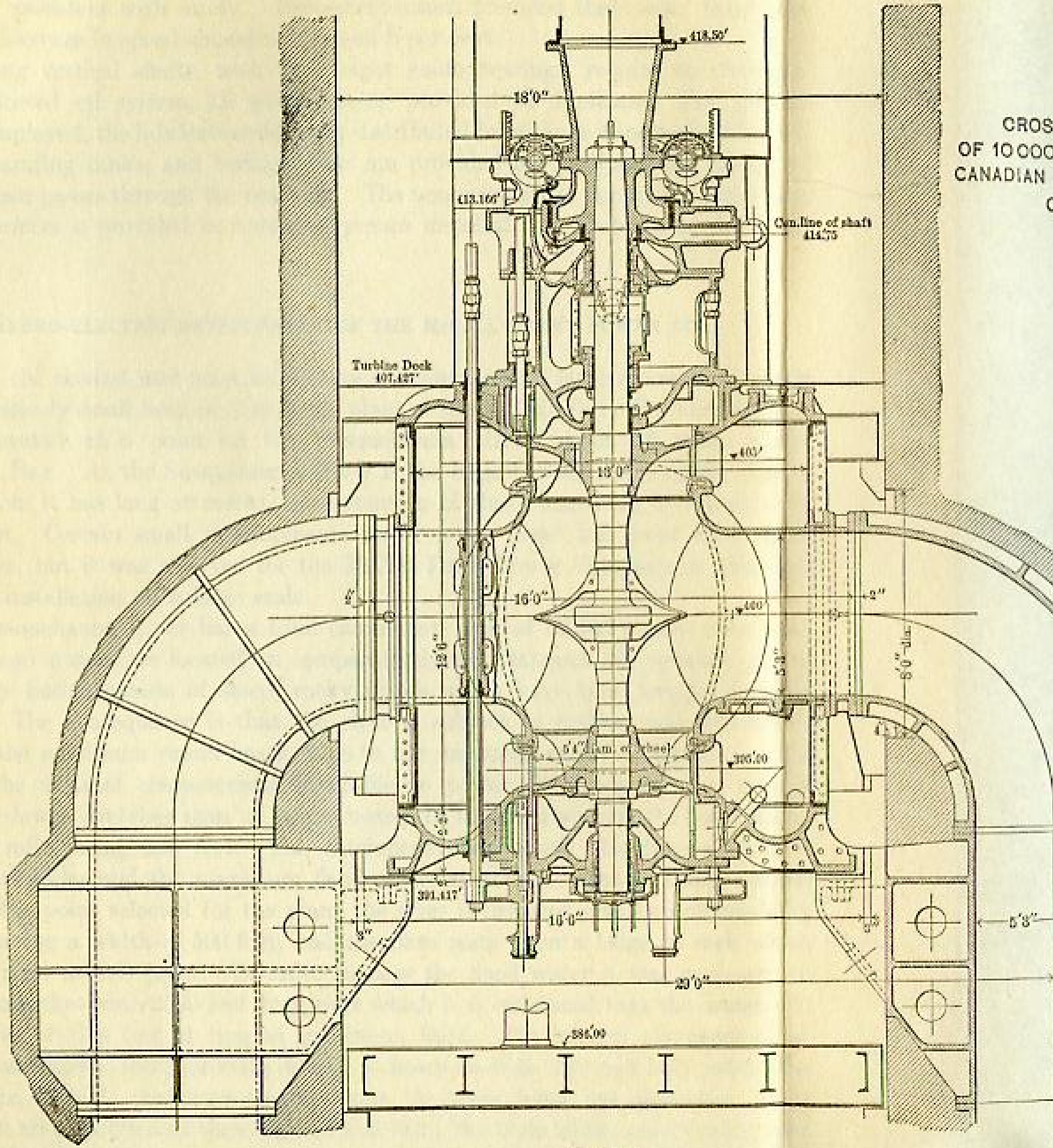


FIG. 144.

[To face page 218.

cut off from the units, it is necessary that the action of the governors should be as rapid as is consistent with safety. The specifications provided that under these conditions the increase in speed should not exceed 8 per cent.

The long vertical shafts, with their eight guide bearings, require an extensive and complicated oil system, all pumps being provided in duplicate. The gravity system is employed, the lubricating oil being distributed from tanks in power house roof. Filters, separating tanks, and boiling tanks are provided for purifying the return oil before it again passes through the bearings. The necessary water supply for machinery and transformers is provided by means of pumps installed in a chamber as shown on Fig. 143.

HYDRO-ELECTRIC DEVELOPMENT OF THE McCALL FERRY POWER CO.

One of the boldest and most interesting developments of large amounts of power under a relatively small head is that of the plant of the McCall Ferry Power Company, which is located at a point on the Susquehanna River about 20 miles above Chesapeake Bay. As the Susquehanna River is the largest of all rivers of the Appalachian system it has long attracted the attention of those interested in water-power development. Certain small developments under a low head had been made from time to time, but it was reserved for the McCall Ferry Power Company to plan and execute an installation on a large scale.

The Susquehanna River has a total catchment area of 27,400 square miles and, while the head waters are located on comparatively flat plateaux, the greater part of the drainage basin consists of steep, rocky slopes which have been largely denuded of timber. The consequence is that the river is subject to sudden and destructive floods and the maximum runoff has a ratio to the minimum runoff of 300 to 1. The river has the unusual characteristic favorable to power development of a steeper slope in the lower stretches than at points nearer to the head waters, the total fall in the last 43 miles being 220 feet. The catchment area above the site selected is 26,766 square miles and the maximum flood recorded at this point is about 730,000 sec.ft. At the point selected for the plant the river is diverted into two channels by an island having a width of 500 feet, and the dam rests upon a ledge of rock which crosses the river at this point. In order to pass the flood water it was necessary to provide an overflow weir 2500 feet long over which it is estimated that the water will have a depth of 17.5 feet at time of maximum flood. The general arrangement of the dam, power-house, and protective devices is shown on Figs. 146 and 147, which does not, however, indicate the numerous islands in the river below the dam site. It is evident from an inspection of these figures and from the facts above enumerated that very difficult problems were presented in the construction of this plant. An examination of the stream-discharge records which were commenced at this point in 1902 and which had been kept at Harrisburg since 1889 indicated that if the work was to progress with reasonable speed it would be necessary to make provision for the passage of at least 50,000 sec.ft. at frequent intervals and that larger volumes must be passed without destroying the construction plant. These considerations led to the building of a construction bridge immediately below the dam and extending from shore to shore. This bridge was 50 feet in width, supported on arches 50 feet in the clear,

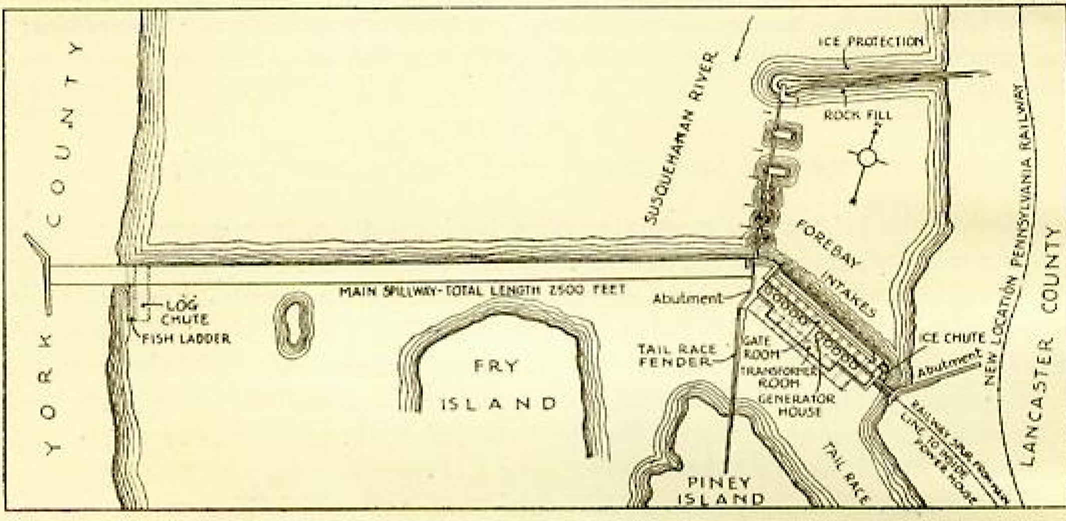


Fig. 146.—Map showing Location of Power Plant of McCall Ferry Power Co.

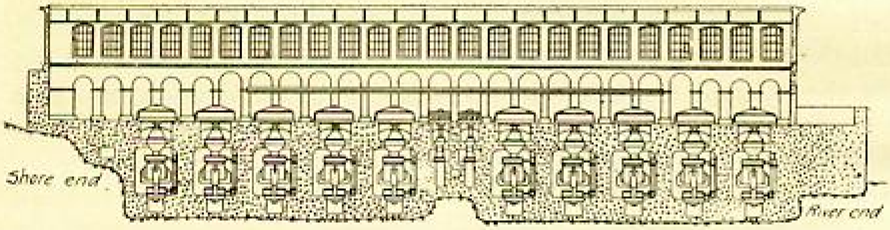


Fig. 147.—Longitudinal Section of Power House, McCall Ferry Power Co.

and formed the support for four lines of railway track and for the rails on which ran specially built travelers used in the construction of the dam and power house. The top of this bridge is clearly shown on Fig. 148, a photograph taken at time of flood.

Another novel feature in the construction of the dam consisted in the use of steel forms by which the dam was erected in alternate sections 40 feet in width. The usual method of passing flood water by means of openings under the power house or by gates in the dam was inadequate to provide for the flood water, and it was for this reason that the dam was constructed in alternate sections. The construction plant throughout was of the most substantial character and of great capacity, so that rapid progress might be made in all branches of the work. The total quantity of concrete to be deposited in the dam was 174,000 cubic yards and the concrete plant had a capacity of 2000 cubic yards per day. The presence of Fry Island was an aid, as the construction of a short cofferdam of a height sufficient to provide for the ordinary flow of the river enabled such flow to be turned into the York channel while the dam was being constructed across the Lancaster channel, and when this portion of the work was completed the York channel was similarly dammed and the river diverted through

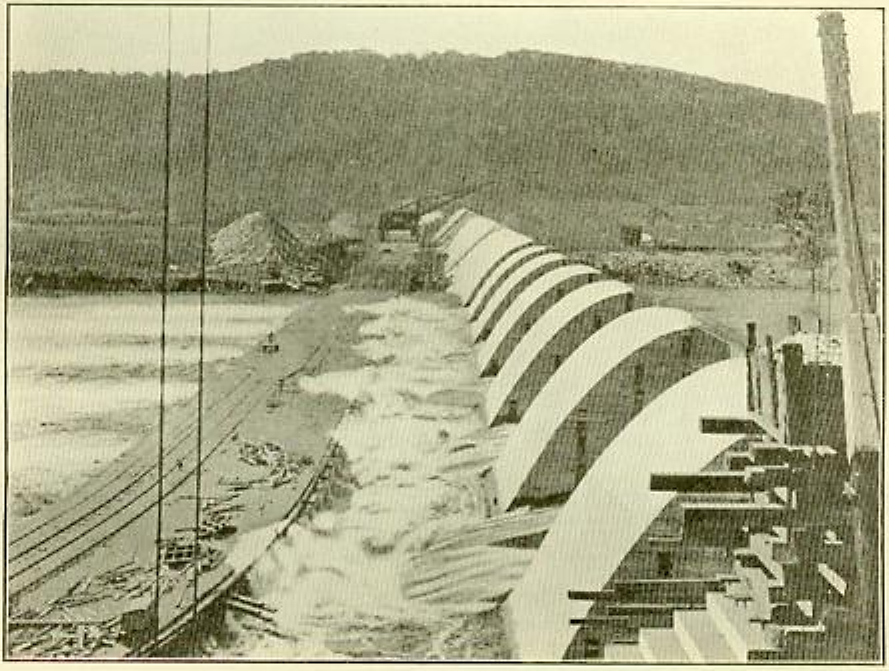


Fig. 148.—Construction Bridge and Partially Completed Dam, High Water, McCall Ferry Power Co.

the openings left in the dam across the Lancaster channel. In all cases, however, the dam was constructed continuously to a level 40 feet below the top of the dam. The open sections were then closed by building them up 5 feet at a time by the use of portable stop curtains.

By reference to Fig. 146 it will be seen that the power-house is located at an angle of 42 degrees with the axis of the dam and is placed at a point adjacent to the Lancaster shore, where the original channel of the river between Piney Island and the river bank forms a tailrace excavated by nature in the rock bottom, such tailrace being two-thirds of a mile long. This tailrace is protected from overflow from the river below the spillway by certain fender walls, one of which is shown in the illustration. The forebay is protected against ice and débris by a rock-filled ramp about 550 feet long at right angles to the shore and a line of defenses extending from its outer end to the junction of the power house and dam. The latter contains three submerged arches, each with a span of 68 feet, and floating booms 3 feet deep. The crowns of the arches are 2 feet below low water, but should any material pass under them it may be discharged into the tailrace by an ice chute 40 feet wide adjacent to the shore.

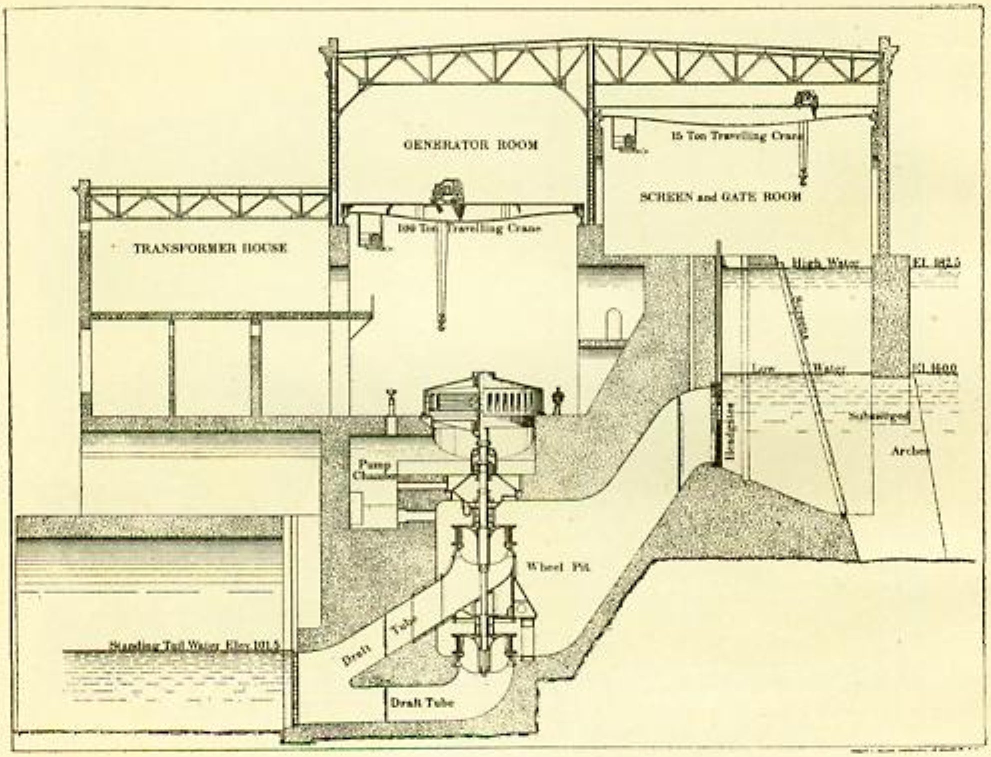


FIG. 149.—Cross-section of Power House, McCall Ferry Power Co.

As the maximum velocity of water entering the forebay is only 1.5 feet per second, little, if any, floating material will be diverted into it from the river. Further protection to the wheels is, however, afforded by the eleven submerged arches on which rests the outer wall of the power house. Should any material reach the space below the screenroom it may be discharged into the tailrace by a second ice chute. In flowing to the penstock mouthpieces the water passes through the racks shown on Fig. 149. These racks are removable and have the remarkable depth of 55 feet. In considering the passages provided for the flow of the water it must be kept in mind that at peak load and with minimum head each unit will consume over 3000 sec.ft. The need for large openings is thus apparent, and the designers have made ample provision for obtaining low entrance and draft-tube velocities. From the space behind the racks the water enters by four openings 6 feet by 16 feet, which merge into a common channel which leads to the wheel case. The latter is remarkable not only for its great dimensions, being 30 feet wide and 33 feet high, but also because it is constructed entirely of non-reinforced concrete. The draft tubes are made of the same material and are of unusually complicated forms, as may be seen from Fig. 150. The one lead-

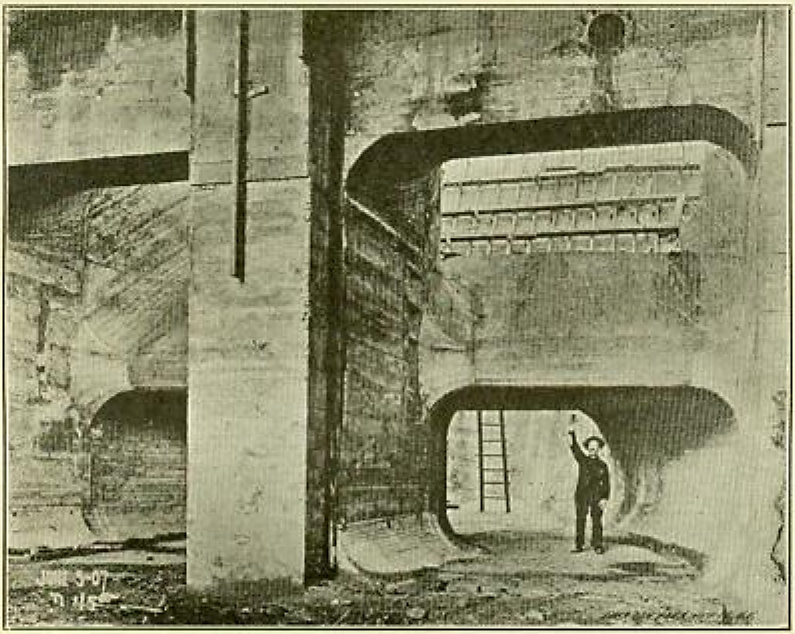


FIG. 150.—View of Concrete Draft Tubes looking toward Turbine, McCall Ferry Power Co.

ing from the lower wheel divides into passages 7.5 feet by 13 feet until joined about 20 feet from the wheel case by the draft tube from the upper wheel, which is also forked. Each unit thus finally discharges into the tailrace by means of two passageways, each 13 feet by 15 feet, resulting in a maximum draft-tube velocity of 7.7 feet. These complicated waterways were built with the use of steamed bass-wood forms, which were broken up and removed after the concrete had set. Fig. 151 shows two of the main tubes in the right-hand side of the foreground, while the construction bridge appears on the left and the dam in the background. The power house, whose section is shown on Fig. 149, is a concrete and steel structure 500 feet long, containing ten main units spaced 40 feet on centers and two exciter units. It is divided longitudinally into three rooms—a screen and gate-room 56 feet 8 inches wide; a generator room 48 feet wide; and a transformer room resting on the construction bridge, 50 feet wide. The transformer room contains two stories, the transformers being located in inclosed compartments in the lower story, while the switching apparatus is placed in the upper story. The dam has a gravity section built of concrete without reinforcement, the proportions of material being 1 part Portland cement, 3 parts sand, and 5 parts broken stone, 18 per cent of the volume consisting of large stones so laid as to bind

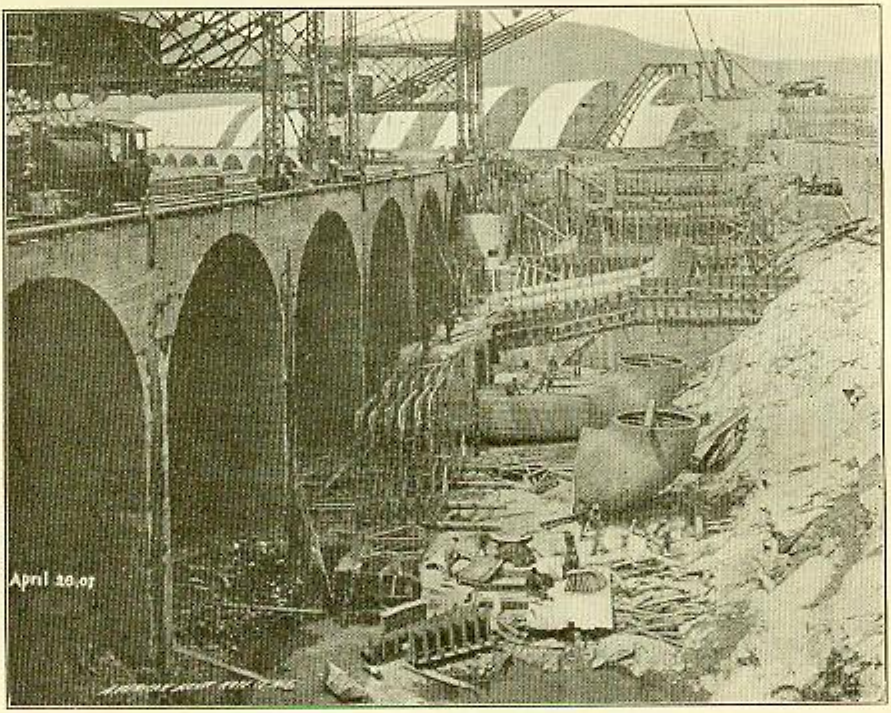


FIG. 151.—Construction Bridge and Forms for Draft Tubes, McCall Ferry Power Co.

the mass thoroughly together. The height of the dam from crest to original surface of river bottom is 53 feet, and from crest to base, 59 feet. The width at base is 65 feet, the form of the overflow section being shown on Fig. 152. The section has a factor of safety against overturning of $2\frac{1}{2}$ with water 17.5 feet deep over the crest, the weight of the water over the spillway being neglected. Where the alternate 40-foot sections of the dam are united expansion joints are formed 3 feet in depth around the periphery by the use of tar paper, the thickness used depending on the temperature at time of laying concrete. This provision for expansion is a most important one that has frequently been neglected in the design of long concrete dams.

The turbine machinery for this plant is remarkable for the boldness of its design. Each unit has a capacity of 13,500 H.P. at .8 gate opening under a head of 53 feet, or 12,000 H.P. at full gate opening under a head of 43 feet, the minimum anticipated at time of greatest flood discharge in the river. To obtain this amount of power in a single unit it was necessary to use two turbines on the shaft, each having a diameter of 10 feet and revolving 94 rev. per min. The wheels are of the Francis type, and regulation is effected by movable vanes in the distributors. The upper wheel of each unit discharges into a short metal draft tube connecting with the concrete waterway,

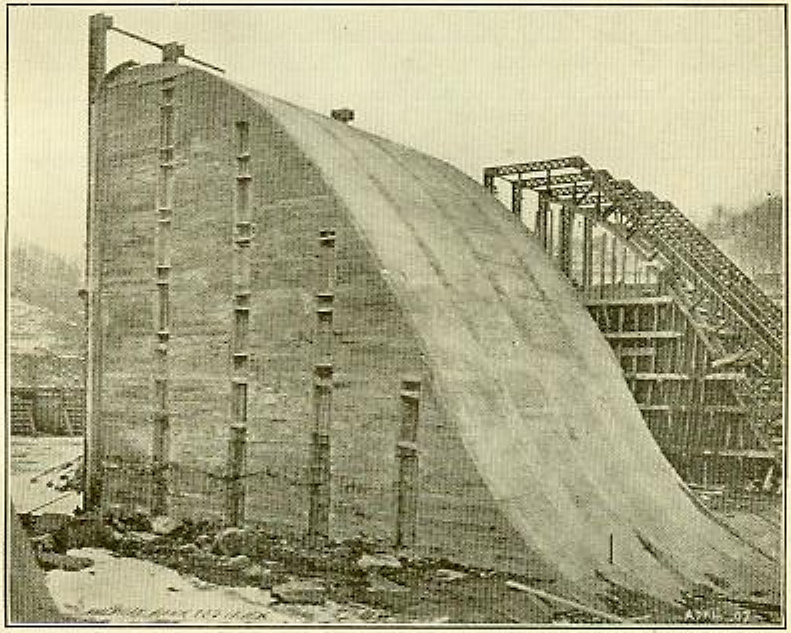


FIG. 152.—Section of Dam and Steel Forms, McCall Ferry Power Co.

while the lower wheel discharges directly into the concrete opening. The turbine machinery for each unit weighs about 700,000 lbs., a weight in keeping with the massive character of the concrete surrounding it.

The rotor weighs 194,000 lbs. and is supported on an oil-step bearing of the same general design as that used by Canadian Niagara Power Company. Oil pressure at 250 lbs. per square inch is supplied from separate pumps at each unit and these pumps may be operated either by turbines, electric motor, or direct drive from the main shaft. Every effort is thus made to insure continuity of service.

As the head under which the turbines will operate will vary so widely between low and flood water, it was considered desirable to carefully investigate the probable performance of the wheels under all conditions, and accordingly the engineers of the I. P. Morris Company, contractors for the turbine machinery, made careful studies to determine the power and efficiencies at full and partial gate openings under heads ranging from 50 feet to 70 feet and with a speed of 107 rev. per min. While the speed and heads are not those finally employed, yet the analyses are interesting and are therefore given in some detail, the diagrams and information being derived from I. P. Morris Company's Bulletin No. 2.

It was assumed that each unit was to develop 13,000 H.P. at full gate opening under a head of 65 feet with ϕ equal to .7. One point on curve No. 1, Fig. 153, was

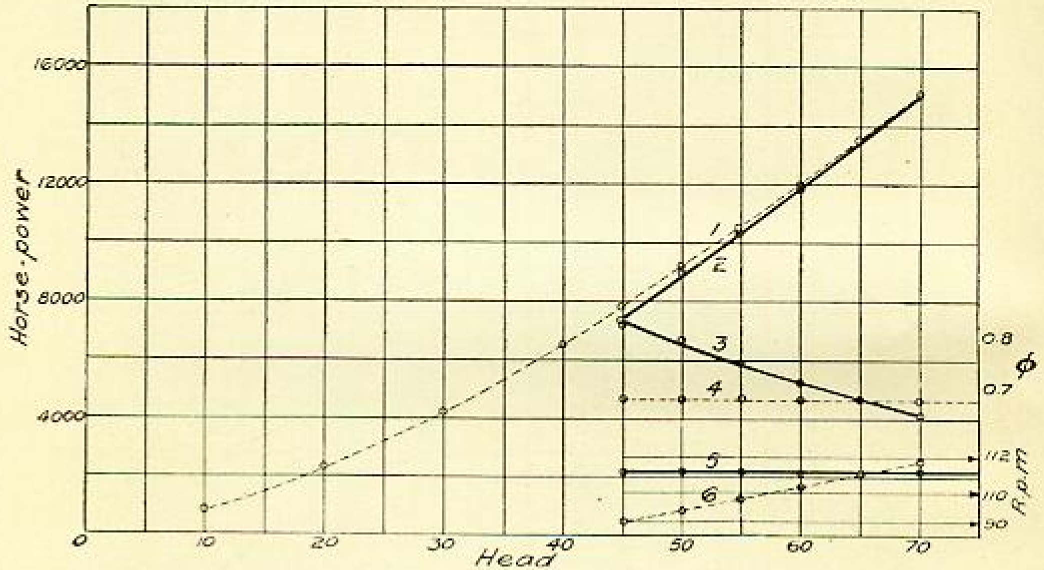


FIG. 153.

thus fixed and other points were located on the basis that the horse-power developed will vary as the three halves power of the head and that if ϕ remains constant the efficiency will not vary with a given gate opening. Curve No. 3 indicates the manner in which the value of ϕ will vary if the speed and diameter of wheel is held constant while the head changes, ϕ being .7 under a head of 65 feet. Curves Nos. 4 and 5 indicate respectively constant values of ϕ equals .7 and rev. per min. equals 107. Curve No. 6 shows the speed required under the several heads in order that the value of ϕ may be kept at .7.

As it was necessary to keep the speed of the rotor constant under all heads the next step was to determine the power lost by conforming to this condition. Accordingly the curves shown, Fig. 154, were plotted from the results of the actual performance of other wheels designed and built by I. P. Morris Company. Curve No. 1, for example, shows the results of tests made on a wheel designed to develop 10,500 H.P. under a head of 135 feet at 180 rev. per min. at full gate. The power developed was

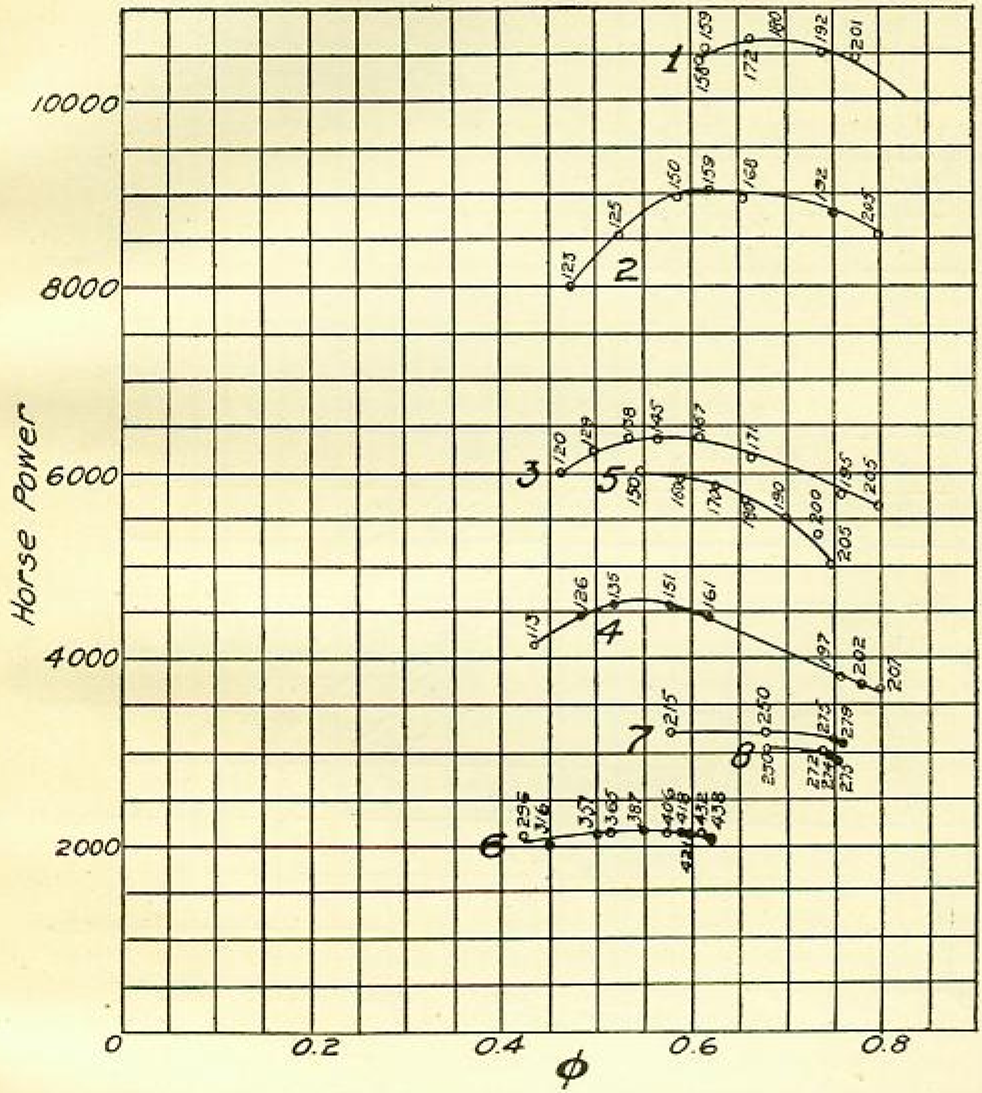


Fig. 154.

determined from the output of a generator loaded with a water rheostat. By changing the field current, speeds from 158 rev. per min. to 201 rev. per min. were obtained. Curves 2, 3, and 4 were similarly obtained at partial gate openings for the same wheel while curves 5, 6, 7, and 8 show the results of tests made upon other wheels at full gate opening. With these curves of actual results as a basis, the power to be developed

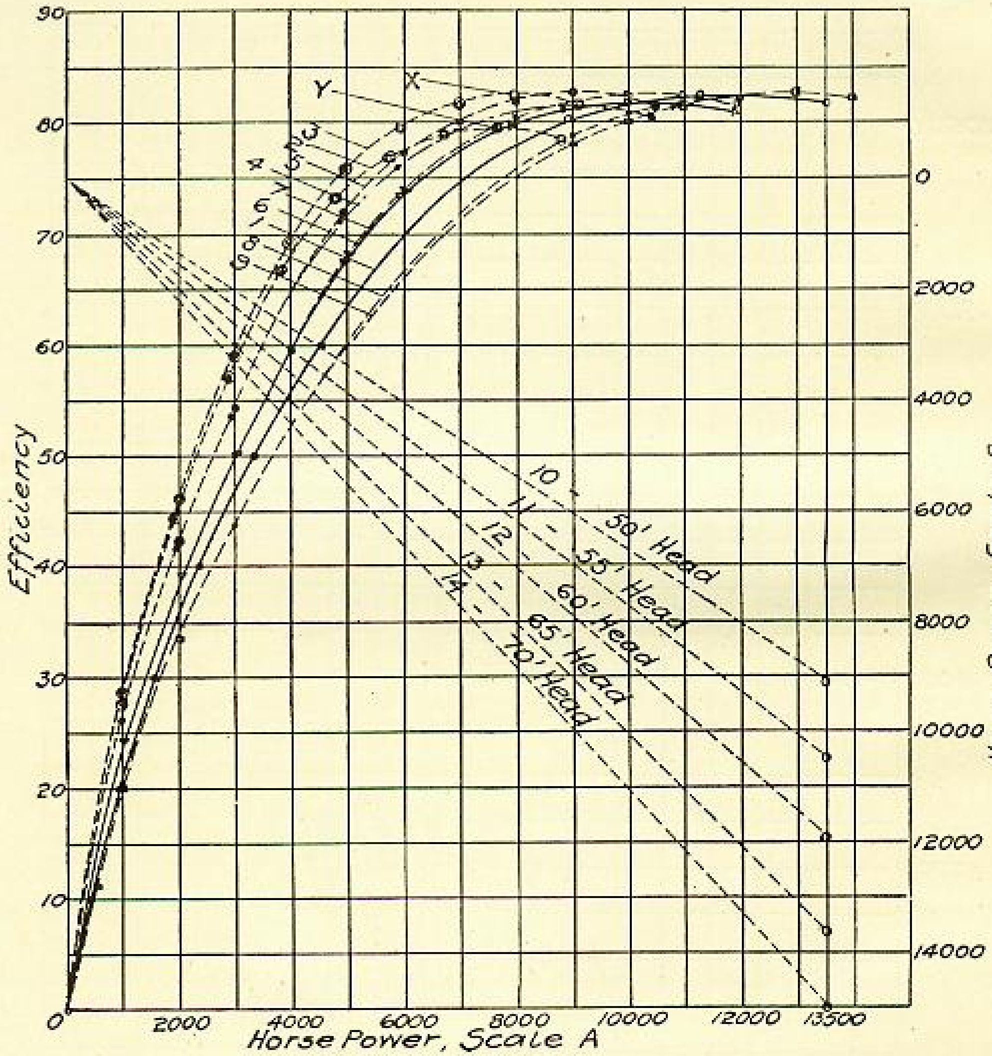


FIG. 155.

by the wheels at McCall Ferry under the several conditions of head may be approximated.

Referring to curve No. 3, Fig. 154, it will be seen that with a speed of 107 rev. per min. under a head of 50 feet the value of ϕ is .8. Curve No. 1, Fig. 154, shows that as

154, it will be seen that the power varies as the speed varies, and curves Nos. 2, 4, 6, and 8, Fig. 155, are plotted on the assumption that the efficiency varies as the power, or, in other words, that the quantity of water does not vary with the speed for a given setting of the gate, an assumption probably correct within ½ of 1 per cent within the limits of the full gate experiment. In the manner heretofore discussed it is found that at full load the wheel under full gate opening at 50 ft. head at 107 rev. per min. will give 3.3 per cent less power than at 94 rev. per min., and assuming the drop in efficiency to be the same we may plat a point on curve No. 2, Fig. 155, 3.3 per cent below point X. Now by referring to curves 1, 2, 3, and 4 on Fig. 154, it will be observed that the slope of these curves between ϕ equals .7 and ϕ equals .8 is practically the same, and hence we know that the power and efficiency of the wheel under design will be approximately 3.3 per cent less at 107 rev. per min. than at 94 rev. per min. under 50 ft. head at all gate openings. Curve No. 2, Fig. 155, may therefore be plotted by reducing the efficiencies on curve No. 3 at all points by 3.3 per cent, and a curve thus obtained which will give the efficiency that may be anticipated from the unit when operating under 50 feet head at 107 rev. per min.

By the same method as above described curves 5, 7, and 9 are plotted and curves 4, 6, and 8 deduced therefrom. The general result of the investigation as above outlined is to show that the efficiency and power of the units would not be seriously affected by operating at a speed which varies even 13 rev. per min. from that best suited to the head, but, on the other hand, that the power available is largely affected by the operating head, as is clearly shown by the position of the right-hand end of the curves in Fig. 155, read on scale A.

While the above method of wheel analysis is not strictly accurate, it is of value as giving with comparatively little work the performance which may be expected from a wheel designed on the same general lines as others already in service. It would, of course, have been more satisfactory if the wheels whose power and efficiency curves were used had operated under heads more nearly approaching that available at McCall Ferry.

Each turbine shaft is direct connected with a 7500-KW., 11,000-volt, 25-cycle generator mounted on a casting set in the massive concrete of the power-house floor. The field current is supplied by two 1000-KW. exciters driven by independent turbines. The exciters operate at 240 rev. per min. and furnish current at 250 volts. The exciters are located at the center of the power house, the forms for their draft tubes being shown on Fig. 151.

The plant is admirably located in respect to a market for power, being 65 miles from Philadelphia, 40 miles from Harrisburg and Baltimore, 43 miles from Wilmington, and about 50 miles from Washington. The total amount of steam power in use in these cities and in other smaller towns within easy reach of the plant is said to aggregate 750,000 H.P.

PLANT No. 3 OF THE NIAGARA FALLS HYDRAULIC POWER AND MANUFACTURING COMPANY AT NIAGARA FALLS, N. Y.

The hydro-electric development of the Canadian Niagara Power Company at Niagara Falls, Ontario, described in an earlier chapter, was constructed between the years 1901 and 1906, and is therefore one of the latest of the plants deriving their power from the falls of the Niagara River. The utilization of the energy of this river had begun many years before on the American side, and in fact the early settlers had operated a sawmill on the rapids above the falls as early as 1725. In this chapter will be described the latest hydro-electric development at Niagara, which, nevertheless, is the property of the company which first utilized the power of the falls on a large scale. If the existing restrictions as to the use of the water from the Niagara River shall continue in force indefinitely the plant to be described will probably be the last to be constructed at Niagara Falls, and as such is of especial interest.

The general plan under which The Niagara Falls Hydraulic Power & Manufacturing Company (frequently called "The Hydraulic Company") is operating was laid out as early as 1847 by Mr. Augustus Porter, a citizen of Niagara Falls whose family owned not only all of the land on the American side adjacent to the falls and the upper rapids, but also most of the property on which the existing city of Niagara Falls is built. All of the early mills had been built on the banks of the river and the islands adjacent to the rapids above the falls, upon lands now included in the State Reservation, and used only small heads on undershot or overshot wheels. Mr. Porter sought a form of development which would leave the surroundings of the falls unimpaired for future generations and thus conceived the idea of a canal connecting the upper river, 5000 feet above the crest of the falls, with a forebay 2300 feet below that line. Those who for purposes of self-advertisement now so blatantly oppose the use of the great national resource contained in the power of Niagara Falls appear to overlook the fact that even from the middle of the last century there were public-spirited and far-seeing men who considered and successfully solved the problem of combining the practical and the æsthetic, of creating wealth for the community while still preserving the majesty of the falls, this action being the more commendable in view of the fact that it lay within their power to make a power development at the minimum cost on Goat Island or the land now occupied by Prospect Park. From first to last the power developments at Niagara have shown but little evidence of the greed of commercialism, and it is unfortunate that those who have furthered these enterprises in such a way as to promote not only engineering progress, but the public interests, should have incurred the opposition of the ignorant and the unthinking. While the canal was commenced in 1852, financial difficulties prevented its completion until 1861, when it was extended to the forebay with a width of 36 feet and a depth of 8 feet. Owing to the effects of the Civil War this canal was unused until 1872, when a grist mill using 150 H.P. under a head of 25 feet was erected on the bank of the gorge. No further progress was made in the development until the purchase of the property in 1877 by the present owner, The Hydraulic Power & Manufacturing Company. The earlier wheels installed by this company and its tenants operated in all cases under heads of from 25 feet to 86 feet, the water discharging over the bank. It was not considered practicable at that date to utilize greater heads of water and it is significant of the state of the art of hydraulic development that it was not until 1886 that The Hydraulic Company thought it desirable to secure the lower talus slope of the gorge. As it is not the purpose of this chapter to describe the earlier work of the development made by the above company, it will be sufficient to state that in 1881 they built Plant No. 1 at the top of the bank, containing two wheels with a total capacity of 1500 H.P. operating under a head of 75 feet to drive dynamos furnishing power for commercial purposes, and in 1895 they installed on the river bank in the gorge Station No. 2, containing sixteen turbines operated under approximately 212 feet head, to actuate thirty-two dynamos which are principally low-tension, direct-current machines having a total capacity of 34,000 E.H.P. This is believed to be one of the earliest installations employing reaction wheels under so great a head, the operating head at the earlier plant of The Niagara Falls Power Company being only 136 feet.

In addition to the above-mentioned hydro-electric developments various tenants of the Niagara Falls Hydraulic Power & Manufacturing Company, whose mills are located between the forebay and the river, develop approximately 5500 H.P. by means of vertical-shaft wheels operating under heads not exceeding 100 feet, the tail-race from each wheel pit discharging from the face of the cliff along the gorge.

It is, however, the purpose of this chapter to describe the last station built by the Hydraulic Company, and known as Station No. 3. This station is of interest not only because of its large capacity, i.e., 130,000 H.P., but also because its design is the result of experience gained in the earlier stations, and it therefore represents the later thoughts of the engineers who have long been connected with The Hydraulic Company. The installation was planned and constructed by Mr. John L. Harper, member American Society Civil Engineers, to whom the authors are indebted for the drawings and photographs accompanying this description.

On Fig. 156 is shown the location of the works of The Hydraulic Power & Manufacturing Company both in reference to the upper and the lower river, and also to the other power developments at Niagara on both sides of the river. It will be seen that water is supplied to the plant by a canal having a length of 4860 feet from its connection with the river at Port Day to the entrance to the forebay. This canal has been excavated to a width of 100 feet and a depth of flow of 14 feet, the bottom being about 33 feet below the original surface. When a total load of 150,000 H.P. is carried on the combined plants the velocity of flow will be 5.00 to 5.50 feet per second, which while higher than usual is amply justified by the character of the solid rock through which the excavation is made. The bottom has been given a grade of 2 feet in its length, which would appear ample, as Bazin's formula would require only 1.5 feet if the bottom is well cleared off.

As indicated on the general map, the forebay supplying Station No. 3 is at the northerly end of the original forebay, about 500 feet from the lower end of the canal. The general form of the forebay is shown on Fig. 157, which also indicates the head-

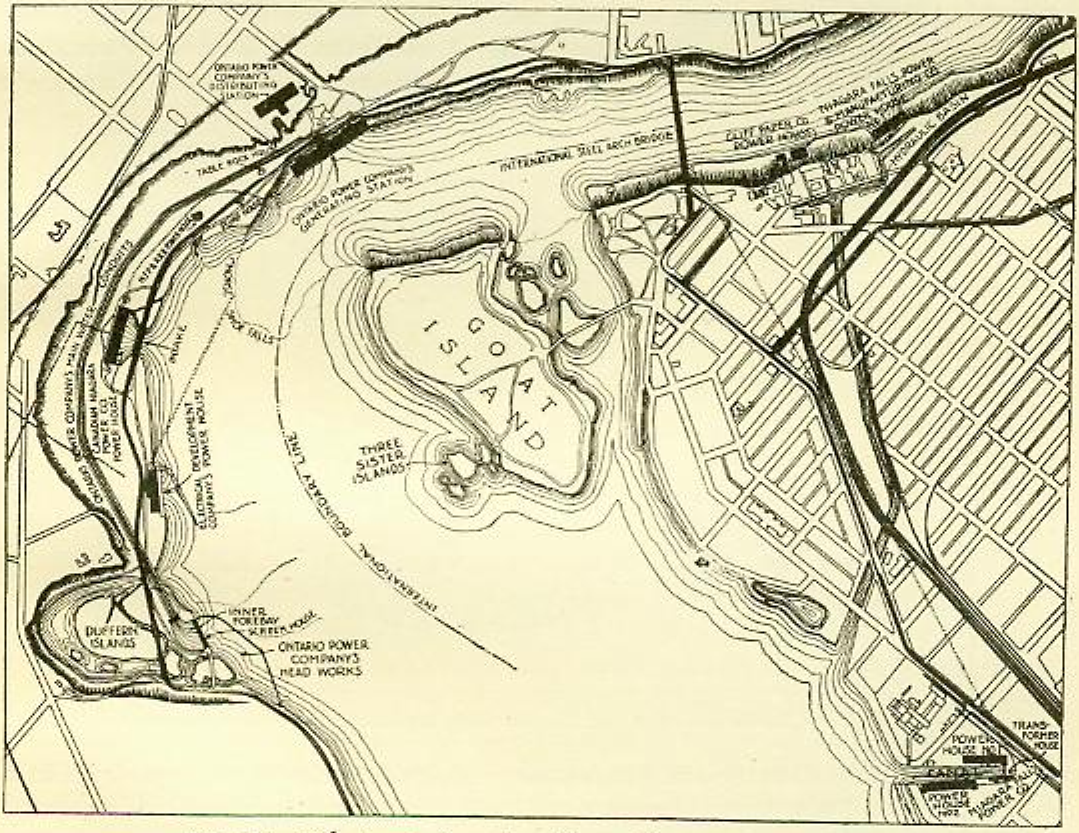


FIG. 156.—Map showing Location of Power Plants at Niagara Falls.

house, bridge, crane, etc., in their relations to the power house below the bank. The entire forebay is built in excavation, the rock being removed to give a normal depth of water of 25 feet, a variation of 3 feet above and below this figure occurring at high and low water. The westerly side of the forebay is formed by a massive concrete wall $24\frac{1}{2}$ feet thick at the bottom and $17\frac{1}{2}$ feet thick at the top. On the other sides the rock is faced with concrete while the floor is composed of the natural rock covered with 18 inches of concrete.

The serious difficulties in operation occasioned by ice in the Niagara River have already been referred to in the description of the plant of the Canadian Niagara Power Company. Under the conditions which usually prevail in the Niagara River the ice problem is more serious to the plants on the American than on the Canadian side of the river, and a description of the means employed by the Hydraulic Company to solve the difficulty may be of service to others designing power plants in a northern climate. The northerly or American channel of the Niagara River is quite shallow compared with the southerly or Canadian channel, only about 5 per cent of the total volume of the river passing over the American Falls. A reef extends upstream from Goat Island to a point opposite to Port Day, the entrance to the Hydraulic Company's

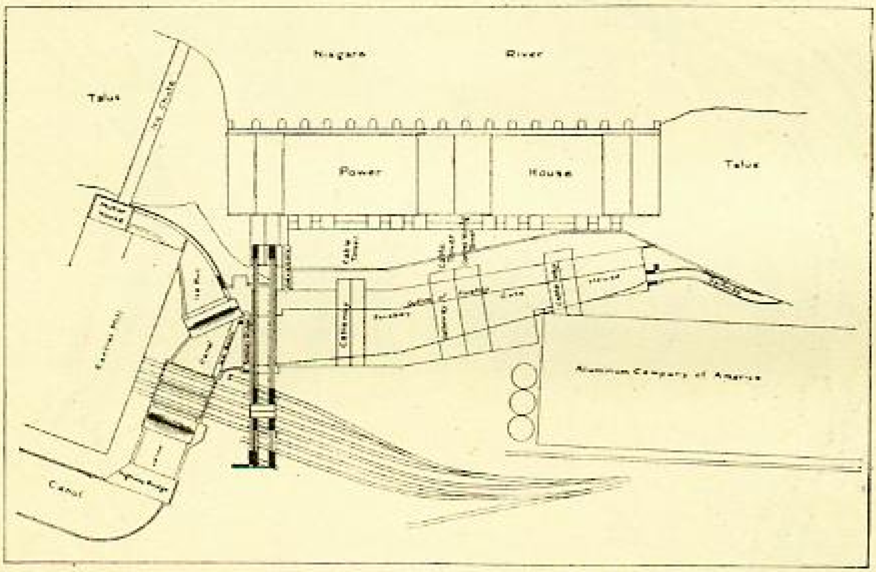


FIG. 157.—Forebay and Power House of The Niagara Falls Hydraulic Power and Manufacturing Co.

canal. Above this entrance are two islands connected to the shore by a dyke built by The Niagara Falls Power Company, and still further up the river are large reefs far out in the river. As the prevailing wind is from the southwest the ice coming from Lake Erie is driven on the shallows toward the south side and thus form nuclei from which the ice fields gradually spread until they extend into the river as much as 2100 feet from Port Day at right angles to the shore. Through these fields only a few small channels find their way. At times frazil is abundant, and by adhesion to the under side of the ice fields greatly reduces the waterway leading to the canal. As the natural depth of the river for 2000 feet or more from Port Day is but 4 to 6 feet, the consequence of such obstruction of the channel is serious. An additional difficulty arises from the large masses of ice carried into the channel by the current under certain conditions. The problem is therefore two-fold, i.e., to provide increased waterway and to prevent the entrance of ice to the canal. The solution is shown on Fig. 158, a map showing the construction carried out by the two American power companies. A large amount of dredging has been done not only in the immediate vicinity of Port Day, but also for some distance upstream, a depth of water of 14 feet to 20 feet being thus secured. To prevent the entrance of ice to the canals a series of timber cribs with concrete tops have been built as shown. The Hydraulic Company has a double

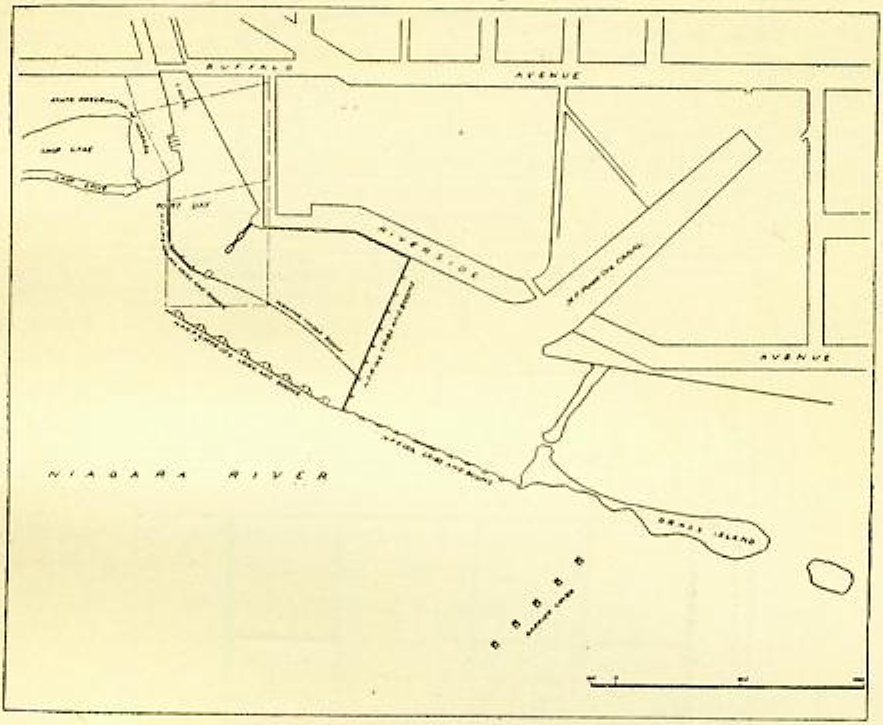


Fig. 158.—Cribs and Booms for Protection against Ice, Niagara Falls, N. Y.

line of defense, the outer line consisting of nine cribs extending 850 feet downstream from the cribs of The Niagara Falls Power Company, in such a direction as to divert the floating ice downstream past the canal entrance. The inner protection consists of a rock-filled ramp with two openings extending 300 feet into the river, together with a line of cribs and booms connecting its outer end with the shore west of Port Day. The openings between the cribs are closed by timber booms extending 3 feet below the water, hinged at the inner side so as to automatically raise and lower with the changes in the river level. These booms may also be raised by means of winches, so that when it is desired to clear the ice from the entrance a given quantity may be allowed to enter the canal. At the lower end of the canal provision is made for the disposal of the ice by two ice sluiceways connecting the forebay with the top of the high bank. One of these discharges directly over the cliff north of Plant No. 2, while the second one connects with the southerly end of the forebay for No. 3 Plant as shown

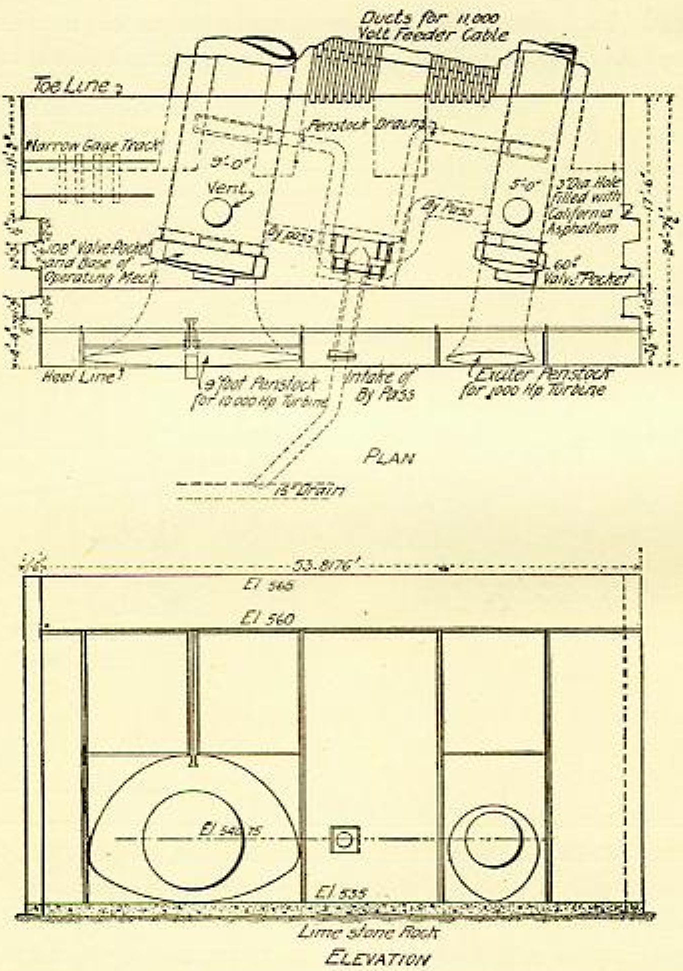


FIG. 159.—Plan and Elevation of Forebay Walls, The Niagara Falls Hydraulic Power & Mfg. Co.

on Fig. 157. Considerable difficulty was experienced from the large masses of ice formed by spray from the first ice sluiceway when the water fell freely over the bank, and therefore the recent design provides for a smooth gradual descent of the water to the top of an unused wheel pit through which it falls protected from the wind. In the first sluiceway there has been employed an ingenious device consisting of a large motor-driven paddle wheel which pushes the ice along the sluiceway and thus prevents clogging.

In addition to the boom which diverts the ice, the entrance to the penstocks is protected by a line of steel racks extending the entire length of the forebay near the westerly wall. These racks are set well out from the wall and are inclined at an angle of 60° with the horizontal. Because of the great depth of the water a very large surface is presented to the water flowing to each penstock. The loss of head is thus reduced and effects of local clogging is minimized.

An interesting feature of the plant is the concrete wall built on the west side of the forebay. With 25 feet of water behind it, standing at the top of a cliff 200 feet above the power house the consequences of any failure in this wall would be indeed disastrous. It has, therefore, been constructed in a most substantial manner, as is indicated both by Figs. 159 and 160, and by Fig. 161, a photograph taken during the early construction of the wall. Its total length is about 400 feet, and as it is necessary that it should be absolutely watertight, an ingenious form of expansion joints was employed, openings of the design shown on Figs. 159 and 160 being placed at intervals of 53.8 feet. It will be seen that pipes are placed at each of the offsets. During the first winter after construction and before water was turned behind the wall those pipes were withdrawn and the space filled with asphalt. The effect of subsequent expansion of the wall was thus to compress the asphalt and squeeze it into the openings. The design has been successful in preventing leakage, although one face of the wall is directly exposed to the sun. The attention of the reader is called to the form of entrance to the penstocks, the design being such as to uniformly accelerate the velocity. Tests show that the loss of head at entrance is extremely small. On the drawing will be seen the guides by which temporary gates may be used to close the mouth of the penstocks in case repairs to the main valve are necessary.

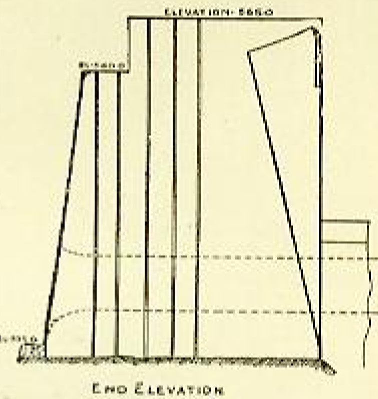


Fig. 160.—Section of Forebay Wall, The Niagara Falls Hydraulic Power & Mfg. Co.

As an additional safeguard to the entire plant, gates have been so constructed at the entrance to the forebay as to completely close the waterway. The entire forebay may thus be drained in case of necessity. These gates are three in number, each 16 feet wide and 30 feet high, rest against rollers, are made tight by strips of rubber belting and are motor-operated under local control.

Fig. 162 shows the gate house at the top of the cliff and the power house at the water's edge as they will appear when the entire installation is completed. The massive and substantial character of the construction is apparent at a glance. A noticeable feature is the concealment of all penstocks and wire towers by a concrete, stone-faced wall, this feature being adopted on the suggestion of the æsthetic committee appointed by the Federal Government to advise concerning the preservation of the natural beauties of the surroundings of the river.

The construction of the site of the power house involved novel and difficult features, as will be seen from the construction photograph, Fig. 163. The talus slope, composed of rock fallen during ages from the cliff above, extended on a slope of 45°

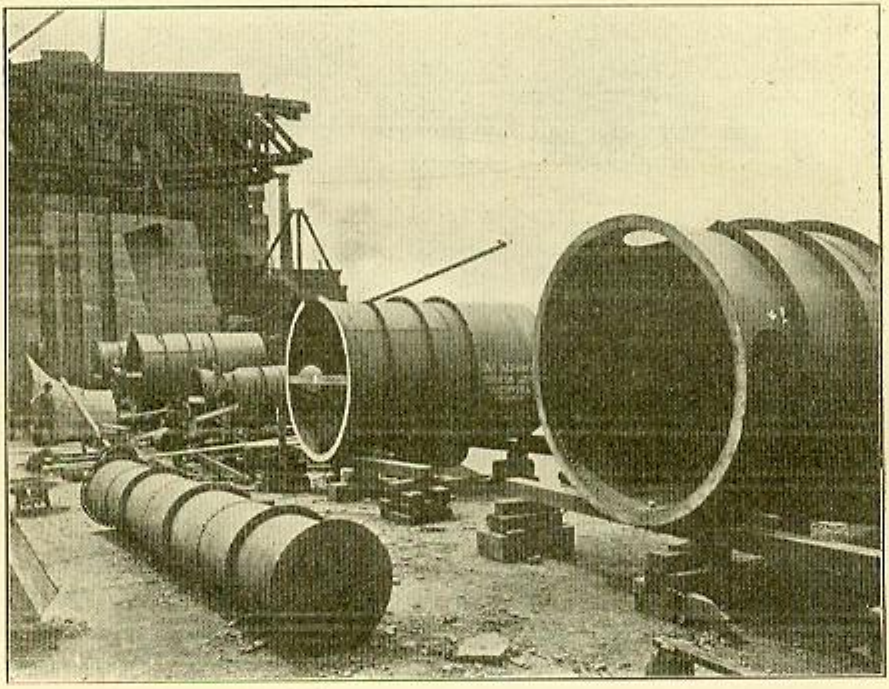


Fig. 161.—Photograph during Construction, Forebay Wall and Penstock Mouthpieces, The Niagara Falls Hydraulic Power & Mfg. Co.

to the river's edge, while the solid strata so projected from the face of the cliff as to leave no room for the building. The talus was first washed into the river by giant monitors supplied with water from the basin above the bank, and the ledge rock was then drilled and blasted away until there was uncovered a level stratum of red sandstone some feet below the normal level of the water in the river, and on this the foundations were placed. As shown on Fig. 164, the building has a total length of 499 feet, a width of 95½ feet and a normal height of 50 feet above floor level. The walls are of rubble, except on the cliff side, the outer wall having a thickness of 4 feet. The inner wall is solid concrete. The building is divided longitudinally into two main rooms, by a reinforced concrete wall, the turbine room on the east having a width of 39 feet 8 inches while the generator room on the west has a width of 48 feet 4 inches. The interior walls are lined with brick, an ornamental wainscoating being carried around each room. It will be seen that the discharge from the draft tubes is carried in reinforced concrete channels beneath the generator room, but the floor of this room is placed at such an elevation as to provide ample space for bus-bar compartments, conductors, etc., between the tops of the arches over the water channels and the bottom of the floor. This elevation of the power-house floor is also desirable because of the extreme height at times

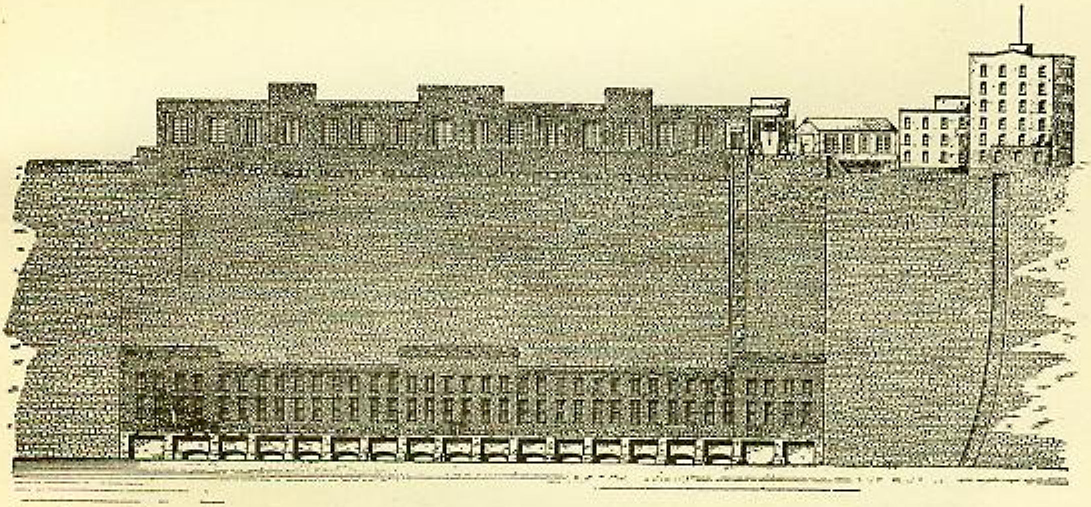


Fig. 162.—View of Completed Power House and Forebay Housing, The Niagara Falls Hydraulic Power & Mfg. Co.

attained by the water in the lower river, the rise in the gorge being about eight times that in the upper river.

An interesting feature of the plant is the traveling crane at the top of the bank which will pick up a freight car with its load of machinery, carry it to the edge of the bank and lower it to the power-house level. The car can then be run into the power house and the load removed by electric cranes which travel the length of both the turbine and generator rooms.

The plant consists of thirteen main units of 10,000 E.H.P. each and two exciters of 850 E.H.P. each. A typical section of the main units is shown in Fig. 164. Each unit is supplied by a separate penstock and operates under a normal static head of 212 feet, and a minimum static head of 208 feet. Each unit is supplied with draft tubes, the center of the turbine being 22 feet above normal water level in tailrace. This length of draft tube was necessitated by the elevation of the machinery floor above high water in river.

The arrangement of the penstocks is clearly shown on Figs. 163 and 165, taken before they were inclosed by the masonry. Emerging horizontally from the forebay wall they descend the cliff vertically, enter the power house horizontally and again turning at right angles enter the wheel case vertically. The diameter of penstocks for main units is 9 feet 0 inches and for exciters is 5 feet, the velocity in main pen-

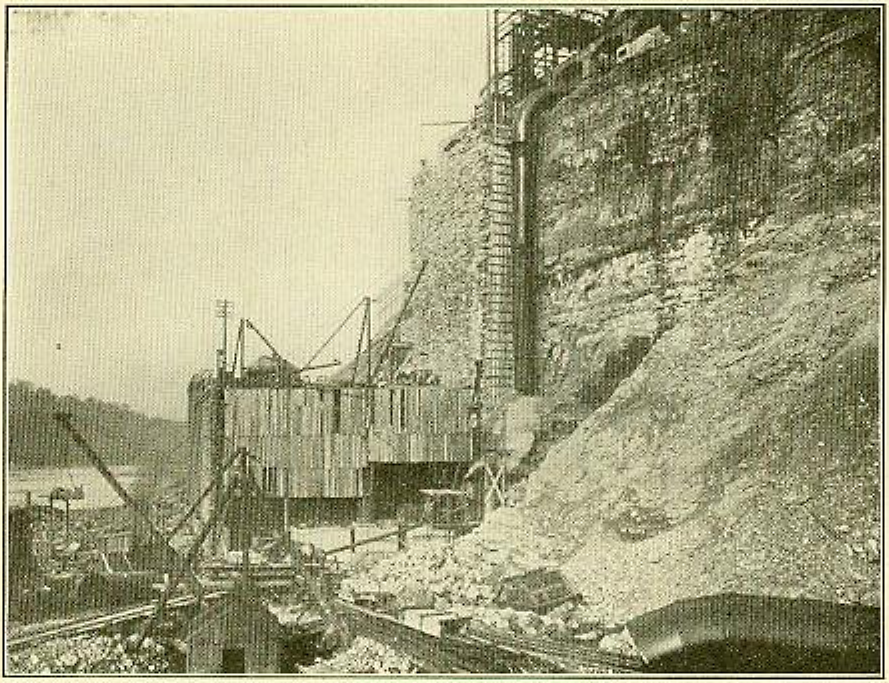


FIG. 163.—Construction Photographs showing Power House and Penstocks, The Niagara Falls Hydraulic Power & Mfg. Co.

stocks being 7.5 feet per second, when the unit is operating under rated full load of 10,000 brake H.P., with water at normal level. While the penstocks are designed to have ample strength in themselves, yet the inclosing concrete is heavily reinforced, so that when in the process of time corrosion shall weaken the steel to such an extent that it can no longer discharge its functions ample security will be afforded by the concrete. It will be seen that the radius of the elbows is large and experiments show that the total loss of head at full load between the forebay and bottom of second elbow, due to friction and entrance is but 1 foot. Aside from the emergency gate already mentioned the entrance of water to the penstock is controlled by a gate valve 9 feet in diameter placed just inside the mouthpiece. This is operated by a 20-H.P. motor, controlled from the switchboard in the power house, and may also be operated by hand. An automatic cut-off is provided so that the current will be cut off from the motor when the gate reaches either limit of its travel, and lamps on the board also give notice of the approach of the gate to extreme positions. A useful device on the penstock consists of a drained depression between the gate and the upper elbow provided for catching such leakage as may pass the gate and which would otherwise annoy men working inside the wheel case. A by-pass is provided to fill the penstock and the usual air vent is placed in each penstock. The diameter of the penstock is reduced in the lower elbow from 9 feet to 8 feet, the entrance to the wheel case being directly below

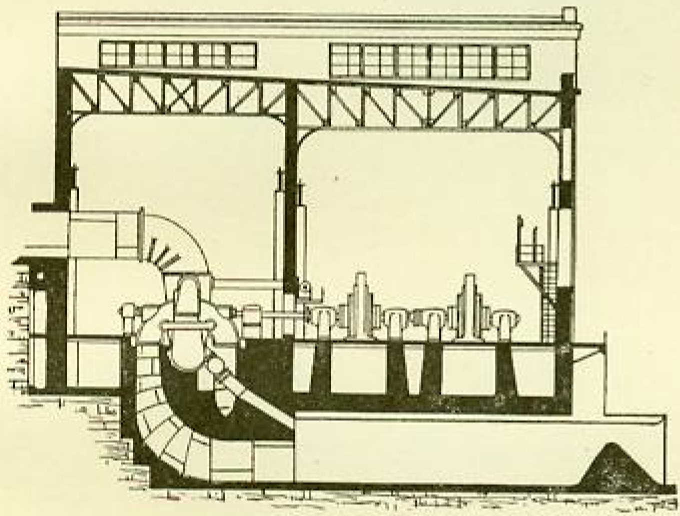


FIG. 164.—Section of Power House, The Niagara Falls Hydraulic Power & Mfg. Co.

such elbow. The wheel case is of spiral form, made in four parts, and an idea of the immense size of the units may be formed from its outside dimensions, i.e., 19 feet 8 inches high, and 22 feet wide, the thickness varying from $1\frac{3}{8}$ inches to $1\frac{3}{4}$ inches. The case was designed with great care in order that the velocities might be maintained constant at all points, sections at right angles to the spiral varying in diameter from 7 feet 7 inches to 3 feet 1 inch. The circular slot forming the entrance to the distributor is 18 inches wide and the case is strengthened adjacent to this slot by eighteen staybolts $3\frac{1}{16}$ inches in diameter. To prevent rupture of the wheel case by high pressure due to inertia of water in the penstock in case of sudden closing of the ports of the distributor, the wheelcase is provided with two outlets each 18 inches in diameter, in each of which is placed a cast-iron plate so designed as to break when the pressure rises a specified amount above the normal. An hydraulically controlled valve is placed between the plate and the wheel case, and the plate can be quickly removed. The discharge from the case through the plate is carried to the channel into which the draft tubes flow. While there are some difficulties to be overcome in the design of the bursting plate this device seems to offer the advantage of more immediate relief of

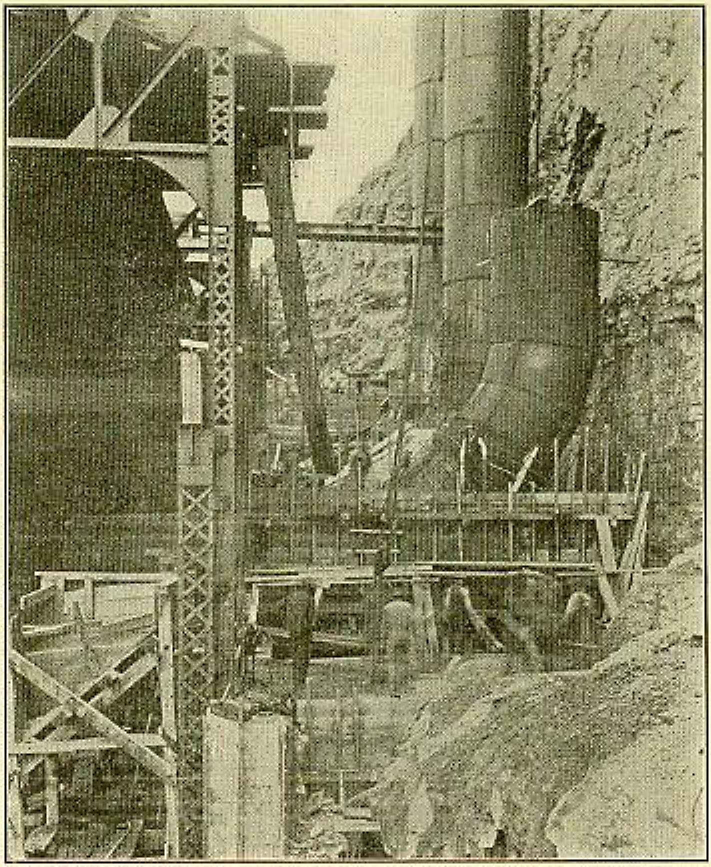


Fig. 165.—Construction Photograph showing Penstock Elbows, The Niagara Falls Hydraulic Power & Mfg. Co.

the pressure than is afforded by poppet valves and of a far greater area of discharge at a given point.

The distributor contains twenty movable vanes operated through links by a movable double ring which is rotated by a connecting rod from a shaft leading to the governing mechanism located in the generator room. The distributor vanes and regulating rings are steel castings, the latter being 6 feet 5 inches in diameter. Ample strength for operating the vanes is furnished by the shaft from the governor mechanism, as it is composed of an 18-inch standard wrought-iron pipe.

The water passing inwardly from the wheel case through the distributor enters the double Francis type bronze wheel, which is 5 feet $\frac{1}{4}$ inch diameter outside of vanes and 5 feet $\frac{11}{16}$ inch in diameter on rim, contains 22 vanes, and is 2 feet 5 inches high. The vanes are 9 inches high less one-half the thickness of the plate separating the wheel into two parts. The runner discharges horizontally east and west into two cast-iron quarter turns leading to the draft tubes, the shaft passing through stuffing boxes therein. These quarter turns are connected by a 16-inch equalizing pipe to prevent excessive end thrust due to possible unequal vacuum in draft tubes, although provision is made for an end thrust of 12,000 pounds. As shown in Fig. 164, the draft tubes are curved in both horizontal and vertical planes, being so arranged as to enter the tailrace parallel to its axis. The longer draft tube is 43 feet $11\frac{1}{4}$ inches from quarter bend to outlet, the diameter increasing from 4 feet to 8 feet. Under maximum load at normal head the velocity of water entering the draft tubes is 20.8 feet per second, while the discharge into the tailrace is only 5.2 feet. Thus practically all of the energy is taken from the water, and this fact, no doubt, contributes largely to the high efficiency reported for the hydraulic installation, an efficiency seldom equaled.

At the outlet of each tailrace is placed a weir whose crest is 8 feet above the bottom of the channel, the depth of flow over the weir at maximum load and normal head being 3.9 feet. Owing to the lack of exact knowledge of the coefficient to be employed for flow over this weir together with the effect of velocity of approach the efficiency of the hydraulic machinery was not determined by the flow over the weir, but by inserting Pitot tubes in the horizontal section of the penstock. By means of a sharp-edged weir placed on the top of the ice-run gates leading from the forebay a comparison was made between measurements made by the Pitot tube and by weir flow and a close agreement was obtained.

Reverting to the turbine machinery it will be seen from Fig. 164 that the energy of the wheel is transmitted to the electrical machinery by a 14-inch shaft passing through the wall separating the turbine and generator rooms, a bearing being provided on each side of the wheel case.

The advantage of horizontal-shaft machines is seen in the avoidance of a complicated oiling system, no oil piping being required for the bearings of either turbines or generators. Self-adjusting oil-ring bearings are used throughout, and though the generator bearings can be lubricated by oil pressure such an arrangement has been found unnecessary.

Each exciter is driven by a Francis type turbine 2 feet $10\frac{1}{2}$ inches in diameter operating under the same head as the main units, the general arrangement being shown on Figs. 166 and 167. The wheel case is spiral, the sections varying from 2 feet $1\frac{1}{4}$ inches to $10\frac{1}{4}$ inches. The movable distributor vanes are 16 in number while the runner contains 22 vanes each 3 inches high. It will be seen from the drawing that the apparatus for operating the distributor vanes is all on the outside of the casing, an excellent arrangement for convenience of inspection and repair.

The exciters are controlled by hand or by a motor operated from the switchboard gallery, but each main unit is regulated by a governing apparatus placed in the generator room on a steel gallery adjacent to the east wall. A portion of this apparatus

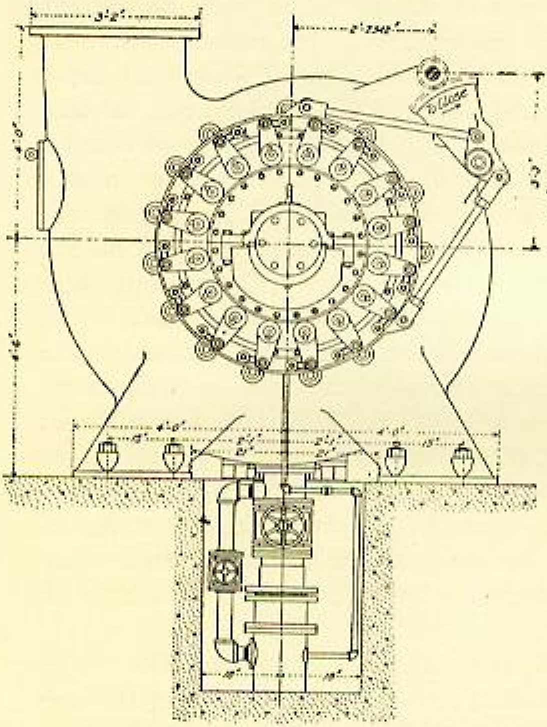


Fig. 166.—Elevation of Exciter, The Niagara Falls Hydraulic Power & Mfg. Co.

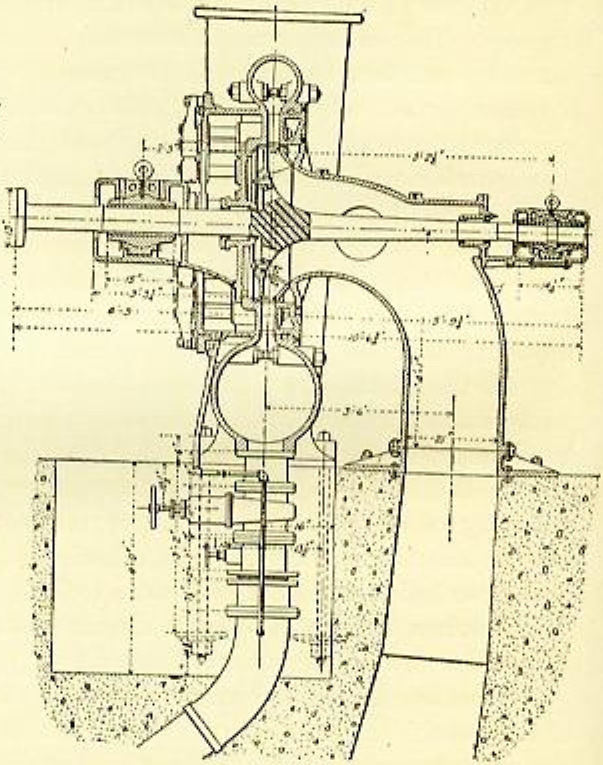


Fig. 167.—Section of Exciter, The Niagara Falls Hydraulic Power & Mfg. Co.

is alike for all units and consists of an hydraulic cylinder containing a piston operated by water from the penstock under a pressure of about 82 pounds per square inch, a slide valve above admitting water to either end of the cylinder. The outer end of the piston rod is formed by a guided rack which engages with the teeth on a quadrant attached to the shaft actuating the regulating ring on the distributor. A large motor is also provided at each unit which will operate the rack directly without the use of hydraulic pressure in the cylinder, but such motor is not usually connected up, as it will be used only under unusual circumstances. To the above apparatus is connected for each of the alternating units the latest type of White automatic governor, which controls the motion of the slide valve above the main cylinder. This governor is of the oil-pressure type, the pump and governor head being driven by separate belts from the main shaft. It is remarkable for the large amount of centrifugal force stored in the revolving head, thus enabling the governor to overcome resistances which might cripple a lighter machine. All the usual adjustments are installed and a system of relays provide for "dead-beat" action. The governor may be set above or below normal when machines are operating in parallel, it may be hand controlled with the governor head in operation or the latter may be cut out of gear and the hydraulic regulating valve operated by oil pressure by hand control. A small motor geared to the spindle controlling the position of the fulcrum may also be operated from the main switchboard. Thus it will be seen that there are not less than six distinct means by which the distributor vanes may be controlled.

Part of the turbines are used to operate the direct-current machines, each electrical unit consisting of two coupled direct-current generators operating at 300 R.P.M., each rated at 3540 KW., at 650 volts, 5446 amperes. Each generator has two bearings and also two commutators, one positive and the other negative. An automatic circuit breaker and a switch are provided for each machine, both being capable of control from switchboard. The field contains 18 shunt-wound poles and 18 series-wound interpoles, the diameter of the armature being 11 feet 8 inches.

The other units are alternators revolving at same speed as D.C. machines, and generating current at 12,000 volts, 3-phase, 25 cycles, 314 amperes. Each unit has a rated capacity of 6500 KW., but will carry a continuous overload up to 7460 KW. without injurious heating. The machines are of the internal-revolving field type and have 10 poles weighing 4660 pounds each, the total weight of the generator, rotor being 95,000 pounds, its outside diameter 10 feet 7 inches and radius of gyration 3 feet 7 inches.

The exciters have a rated capacity of 500 KW., at 250 volts, 450 R.P.M. Either of the two will furnish all of the current required for exciting the fields of the eight alternators, but as an additional safeguard a 150 KW. motor-generator set is to be placed on the power-house floor.

The switchboard for the control of the D.C. current is comparatively simple.

The A.C. conductors are carried in subways beneath the power-house floor and in a single wire tower up the cliff, a third wire tower being reserved for control wiring.

Oil switches are placed on the power-house floor, bus-bar compartments in a subway, and the A.C. switchboard on a steel platform adjacent to the westerly wall. The latter is very complete and while simple in operation gives great flexibility of control. The main feature of the arrangement consists in the ability to supply practically any feeder from any generator. The current from each generator is controlled by one hook switch and three oil switches. Ordinarily it will supply either one or both adjacent feeders, thus segregating the load, but when desired it may be switched to the main bus-bar and thence to any feeder, the machine being synchronized with the other generators connected with the main bus. This bus in turn can be divided into halves so that trouble on any one of four machines will not affect the load carried on the remaining machines. The arrangement requires a total of 12 bus-bars, forming practically two rings. The main switchboard is a combination bench and panel board, while back of it and within reach of the operator is the board containing the field instruments.

The writers have felt that this plant merits the detailed description which has been given, not only because of its large capacity, but also for the reason that its simple and substantial construction contains features worthy of imitation elsewhere.

KERN RIVER PLANT No. 1 OF SOUTHERN CALIFORNIA EDISON COMPANY

To the engineers who have labored in California must be attributed a large part of the progress made in hydro-electric development by means of impulse wheels operating under high heads. The numerous plants built in that section of the country show boldness of design and construction, an adaptibility of means to ends and a departure from conservatism which is in marked contrast to the methods employed in the older and more densely settled portions of this and foreign countries. The necessity for rapid construction and the financial restrictions have in some cases produced plants which, however well adapted for the purpose in hand, nevertheless do not possess the desirable elements of efficiency and durability. Of late years, however, there has been a marked tendency to combine with the courage of the West the conservatism of the East, to build plants bold and even novel in design, which are yet so carefully designed and so substantially constructed as to give promise of large production coupled with a low cost of maintenance. Of such plants none is more notable, not only for the size of the development, but for its many interesting features, than that of the Southern California Edison Company on the Kern River. This plant was the seventh hydro-electric installation made to supply the Edison Company's system, the earliest of these plants dating back to 1892. Prior to the construction of the Kern River plant all of the water power had been derived from streams which find their way to the Pacific from the western slopes of the Sierras, but the increased demand for power made it necessary to seek new sources of water supply, and accordingly five sites were selected over the mountains on the Kern River, which is the most southerly of the large tributaries of the San Joaquin River. The first plant to be constructed, and that which it is the purpose of this chapter to describe, is located at the southermost of the sites selected. The intake works are located one-half mile below Democrat Spring, in Kern county, and the water is returned to the river 8 miles away at a point opposite the power house of the Bakersfield Power, Transit & Light Company, the effective available head at that point being 870 feet.

The catchment area contributing to the intake works is well suited for the main-

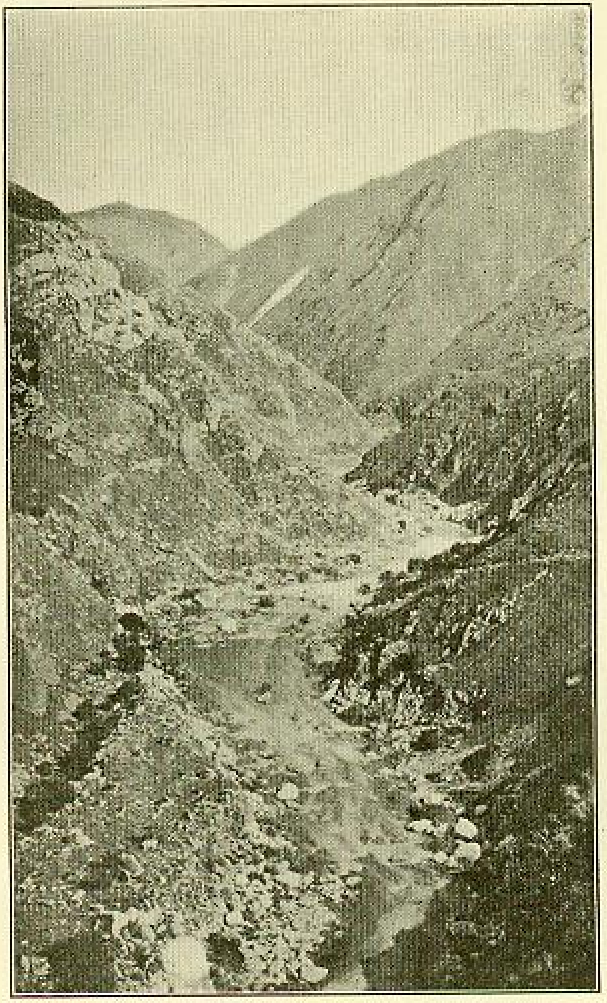


Fig. 168.—Kern River, Cal., and Typical Catchment Area.

tenance of a satisfactory supply. The river rises on the slopes of Mt. Whitney and other lesser peaks where the snow lies throughout the year; the lower slopes and valleys are heavily wooded and the waste land is covered with glacial drift. The conditions thus tend toward a more uniform flow than in the streams first utilized by the Edison

Company and increases the reliability of the service in Los Angeles. The general character of the catchment area is indicated by Fig. 168.

The usual minimum flow is 400 sec.ft. and although the flow for a short time in dry years is only 200 sec.ft., yet the installation was made on the basis of a flow of

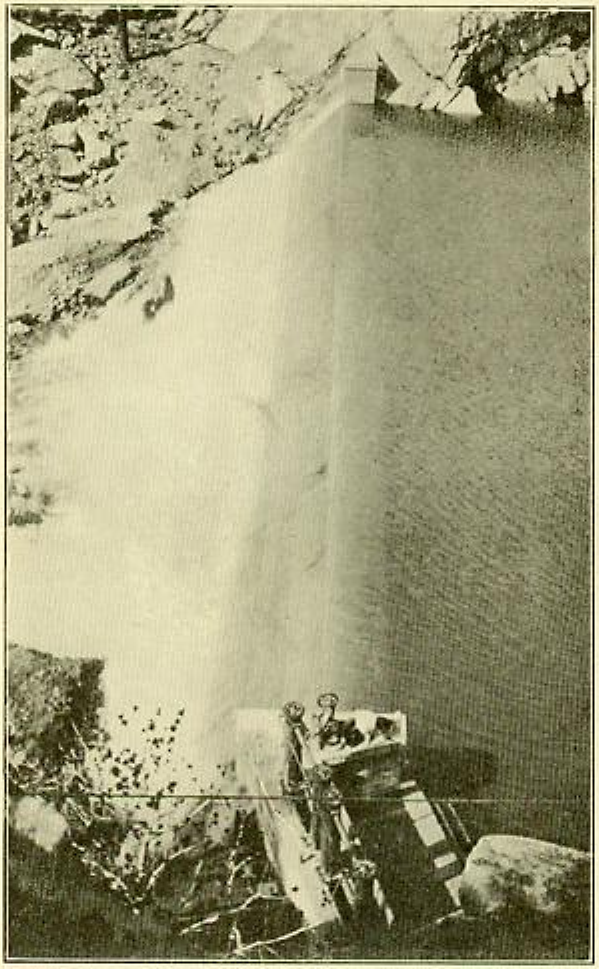


Fig. 169.—Division Dam on Kern River, Southern California Edison Co.

of 470 sec.ft., as the company's system contains a steam auxiliary and the average flow of the river is more than 800 sec.ft.

The diversion dam, located 14 miles above the mouth of the canyon, consists of Cyclopean concrete without reinforcement except for steel rods used for securing it to the bed rock. Solid strata were found at a depth of 35 feet below the bed of the stream and the dam was constructed of sufficient height to give a depth of water of about 30 feet back of the dam. The stream was diverted and the bed rock carefully cleaned, after which it was trenched for additional security. The length of the dam on top is 203.5 feet while the bottom length is only 52.8 feet. The pondage is not great and no provision is made for utilizing it, the crest of the dam being fixed at an elevation just sufficient to give the maximum desired flow to the power house. A view of the completed dam is shown in Fig. 169.

A drainage tunnel 365 feet long not only served to pass the flow of the stream during the construction of the dam, but is now used as a means of cleansing the reservoir from silt. The entrance to it is controlled by two steel gates, each 9 feet by 3 feet 7 inches, operated by hydraulic cylinders under an oil pressure of 200 pounds per square inch, and located in a concrete tower above the dam.

The entrance to the tunnel nearest to the dam is formed by a covered concrete flume 50 feet long tapering from a width of 16 feet on one end and to 8 feet on the other, protected by two hydraulically operated gates each 6 feet $7\frac{3}{4}$ inches by 7 feet, in the clear and by an inclined rack.

The elevation of the crest of the drain was carefully fixed at such an elevation as to provide the necessary velocity and frictional head at entrance so that when the pond is exactly full the water in the gravity tunnel will flow at the exact height desired. Under the ordinary condition of water flowing over the dam it would seem that the regulating gates must be employed. The calculations, however, are interesting and are applicable to similar cases by the use of the following formulæ:

$$W_n = \frac{N_n}{N} W$$

$$F = \frac{V_1^2 - V_2^2}{2g} + \frac{2L}{R_m}\left(\frac{V_m}{C_m}\right)^2.$$

W = width of section N in feet.

N_n = total number of sections into which intake is divided = 8.

N = the number of a given section, the intital section at intake being numbered zero.

W = width at small end of taper.

F = fall in feet in a given section of intake.

V_0 = velocity in feet per second from which to accelerate.

V_1 = velocity in feet per second to which to accelerate.

V_m = mean velocity in feet per second in section considered.

L = length of each section in feet.

g = acceleration of gravity.

r_m = mean hydraulic radius of section.

C_m = mean coefficient of the section in the Chezy formula.

It was considered that the first four of the eight sections into which the intake was divided were omitted because of the excessive width and the velocity at entrance was considered to be 452 feet per sec., or one-half that in the gravity tunnel. The total losses of head at entrance and grade given to tapered section are as follows:

Head lost in passing entrance rack	0.50	
Head lost in passing gate pier	0.18	0.68
Velocity head at entrance	0.32	
Difference of elevation of bottom at ends of tapered section	1.015	1.335
Total		2.015

It will be seen that had the conduit been carried the normal width of 8 feet to the dam and the depth of water increased to provide the requisite velocity head the total head lost in conduit would have been

$$\frac{V^2}{2g}+4SL=\frac{\overline{9.04}^2}{64.4}+\frac{50\times 7.92}{5280}=1.345,$$

or .01 more than above. There were advantages, however, in increasing the width and keeping the depth constant.

In designing the installation at Kern River the most important departure from usual practice consisted in the use of tunnels instead of open flumes wherever possible. The total length of gravity conduit from the entrance to the forebay is 44,944.7 feet, of which 42,910.5 feet is formed by nineteen tunnels cut through the mountains. In fixing the size of the tunnel it was decided that for commercial reasons the velocity should be made as great as possible in order that the size and cost of the conduit might be reduced. From experience elsewhere the engineers considered that the maximum allowable velocity in concrete-lined tunnels was 9 feet per second. The coefficient of roughness in Kutter's formula being taken at .012 for plastered surfaces, it was found that with a depth of flow of 6½ feet the width should be 8 feet (the hydraulic radius being 2.48) to carry 470 second-feet, the quantity determined to be necessary to operate the wheels at 50 per cent overload. Theoretically the width should have been greater in proportion to the height, but the difficulties of driving the tunnel would have been increased. In general the rock section was excavated 8 feet wide, 7½ feet high to the springing line and 9 feet high at the crown of the arch, but where timbering was necessary this section was increased. In general 8 inches of concrete was placed on the sides and 4 inches on the bottom, the roof being supported by an 18-inch concrete arch for a length of 10,000 feet where the character of the roof necessitated it. The section of the water channel having been determined the grade was calculated from Kutter's formula as 7.92 feet per mile. After the completion of the work careful measurements by current meters were made and it was found that the actual results practically agreed with those anticipated; the velocity with a depth of water of $6\frac{1}{2}$ feet being 2.3 per cent greater than that given by the formula, while at a depth of 3

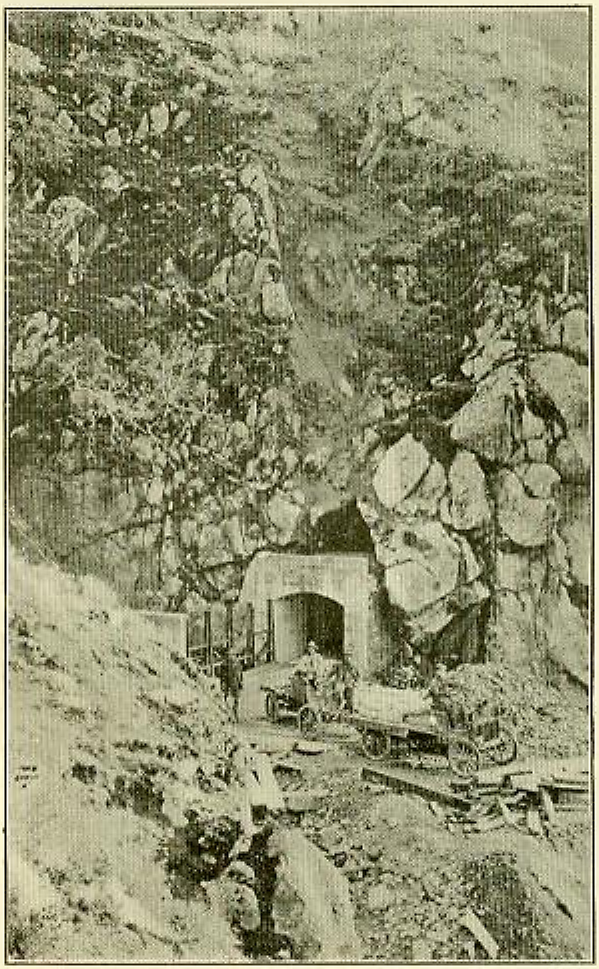


FIG. 170.—Flume between Tunnel Portals, Southern California Edison Co.

feet, it was 1.3 per cent less than calculated. From the stated efficiency and working head of the wheels it would appear that the flow must at times of peak load exceed 470 second-feet, but the finished waterway is 7 feet high, thus giving an opportunity to increase the depth 6 inches above the calculated amount. It is also improbable that the entire plant would be operated under a 50 per cent overload.

While flumes were avoided so far as possible, yet in crossing depressions extending across the tunnel alignment a certain amount of open work was inevitable and six flumes with an aggregate length of 1520.7 feet were therefore constructed. In order that these flumes might approach as nearly as possible to the permanent character of the other portions of the conduit they were constructed with great care. The foundations of the bents were concrete; the supporting timbers of the wooden flumes were heavy Oregon pine and the lumber exposed to the action of the water was carefully selected California redwood. The beveled joints were caulked and were covered on the sides with special battens, behind which was poured asphalt. The section was 7 feet 2 inches by 8 feet and the coefficient of roughness for the battened boards was considered the same as for the plastered surface of the tunnels. A novel form of flume was employed for one span of 49.9 feet, where the supporting framework was made of steel to which was attached two thicknesses of expanded metal covered with 4 inches of concrete. If this form of conduit remains watertight it will doubtless be frequently employed in works where the financial conditions permit the construction of permanent work, for if the supporting steelwork is kept properly painted the life of the conduit should be indefinite. Two adjacent conduit portals with a short section of uncompleted flume is shown in Fig. 170.

Between the ends of the tunnels and the flumes at points where there was danger from falling rocks arched concrete sections were built of the same general form as the covered section of the tunnel, the tops being covered with an earth cushion. Eight of these sections formed a part of the water channel, the total length being 503.5 feet.

Throughout the entire construction of the tunnels and flumes it was necessary to overcome great physical difficulties, and many operations which would under ordinary construction conditions be plain and simple became both difficult and dangerous. Roads had to be constructed on the steep hillsides, cableways built from hill-top to hill-top, and many expedients adopted to suit the untoward conditions. Some of the tunnel portals were inaccessible by roads at all times and many of them were in this condition during the storms of winter. Five of the tunnels were lined with concrete, the cement for which was brought through the lower tunnels in automobiles.

The lowest tunnel emerges from the side of Mt. Breckenridge at a point nearly 900 feet above the water in Kern River, and it would have been desirable to build at this point an equalizing reservoir, but conditions did not permit it, the face of the mountain being composed of broken formations on a grade of about 45°. In order, however, to provide a means of regulating the flow between the gravity tunnel and the pressure main a reservoir 30 feet by 42 feet was excavated in the hillside with its bottom 8 feet below the tunnel grade at its outlet. The reservoir was lined throughout with heavy reinforced concrete and contains the regulating gates and screens through which the water passes to the pressure main connecting with the bottom. At one side of the forebay is a spillway, controlled by flash boards over which the water passes to the waste flumes. This flume is necessary, not only to provide a relief for water flowing in the tunnel at a time when the load on the plant is suddenly reduced, but also to equalize for irrigating purposes the flow in the river below the power house. The upper part of the waste flume is on a comparatively flat slope, and is built of con-

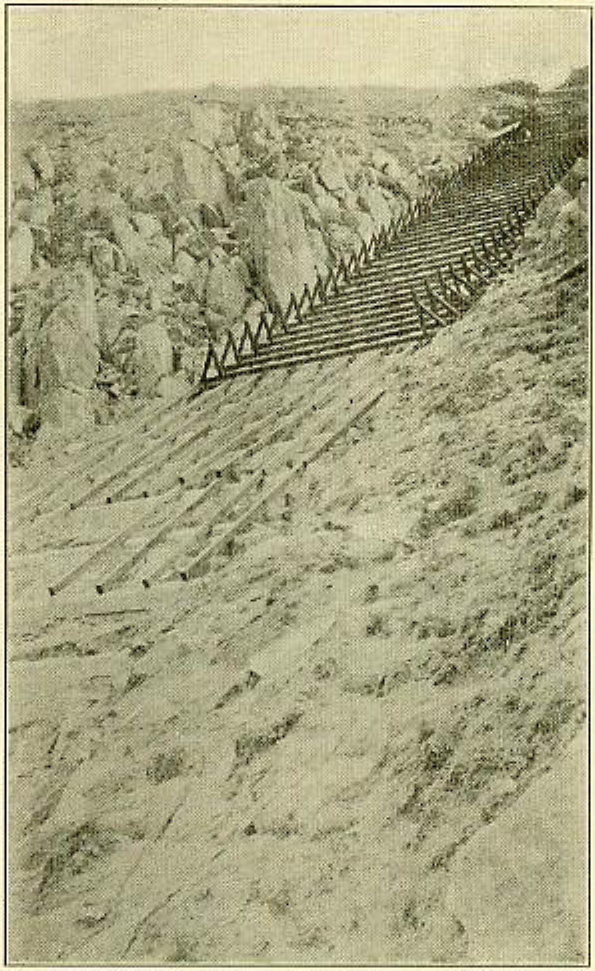


Fig. 171.—Waste Flume, Kern River Plant No. 1, Southern California Edison Co.

crete with a section 8 feet by 8 feet 6 inches. The main portion of the flume leads down the mountain side on a slope of 1 to 1.25 and discharges into Kern River about 750 feet above the power house. On account of the great velocity in this flume it was necessary to construct it of wood and it was decided to make it 20 feet wide, as shown in Fig. 171. An interesting problem arose in fixing the height of the sides, as the accelerated velocity gives rise to standing waves at various points and the height of these waves determined the necessary depth of flume. The wave depth was calculated for all conditions of flow from zero to 470 second-feet by the following formula:

$$D=2\sqrt{d\frac{v^2}{2g}},$$

where D = depth of water at wave·
d = nominal depth;
$v=c\sqrt{ds}$.

A second problem in the waste flume consisted in determining the depth of water on the outside of a curve. It was assumed that the surface of the water on a radial section took the form of a parabola having $\frac{g}{V^2}$ for its parameter, V being the angular velocity.

A radical departure from ordinary practice was made in the construction of the pressure main leading from the forebay to the supply pipes at the power house. The ordinary method of laying the pipe exposed on the hillside was objectional for three reasons: (1) the extremes of temperature to which it would be subjected (145°); (2) the danger of injury by sliding boulders; (3) the great weight and cost of steel required to sustain the pressure. The second objection would have been overcome by burying the pipe in a trench, but such trenching in the loose rock of the steep hillside would be expensive and almost impracticable. It was accordingly decided to excavate a sloping tunnel to the level of the main header at the power house and thence a horizontal header to the face of the mountain. Penstocks have been placed in tunnels elsewhere, as at the plant of the Ontario Power Company; but in the plant under description the rock receives the pressure from the water, the steel lining $\frac{3}{16}$ inch thick being used only for impermeability. The interior of the pipe is 7.5 feet diameter, while the rock was excavated to a circular section 9 feet diameter. The steel pipe was lowered into place in 10-foot sections and the circular seams riveted in place. The space between the exterior of the pipe and the rock was filled with concrete which, while strong enough to take the pressure, was too permeable to use directly exposed to the full pressure of 380 pounds per square inch. Drainage pipes are provided to prevent an accumulation of pressure on the outside of the pipe.* From the point where the horizontal tunnel leading to the power house emerges from the solid rock the size of the pipe is reduced to 5.25 feet, and it is laid loosely in the tunnel, being made of $1\frac{1}{8}$-inch plates and thus self-sustaining. The header pipe back of the power house varies

* This method of construction proved very unsatisfactory. The penstock failed to sustain the pressure some months after the plant was placed in operation and has been reconstructed of steel sufficiently thick to take the entire pressure.

in size from 5.25 feet to 2.33 feet, the reductions in size being made with tapered sections and the branches laid on long radius curves. The supply pipe leading to each main wheel is 28 inches in diameter, and the exciter supply pipes 10 inches in diameter. On each 28-inch pipe are placed two valves, the outer one hand controlled for emergency use and the inner one operated by a motor controlled from the switchboard.

The machinery in the Kern River plant is designed as a harmonious whole, not, as is too frequently the case, as an aggregation of machines which, however satisfactory in themselves, do not possess that correlation so essential for an efficient and

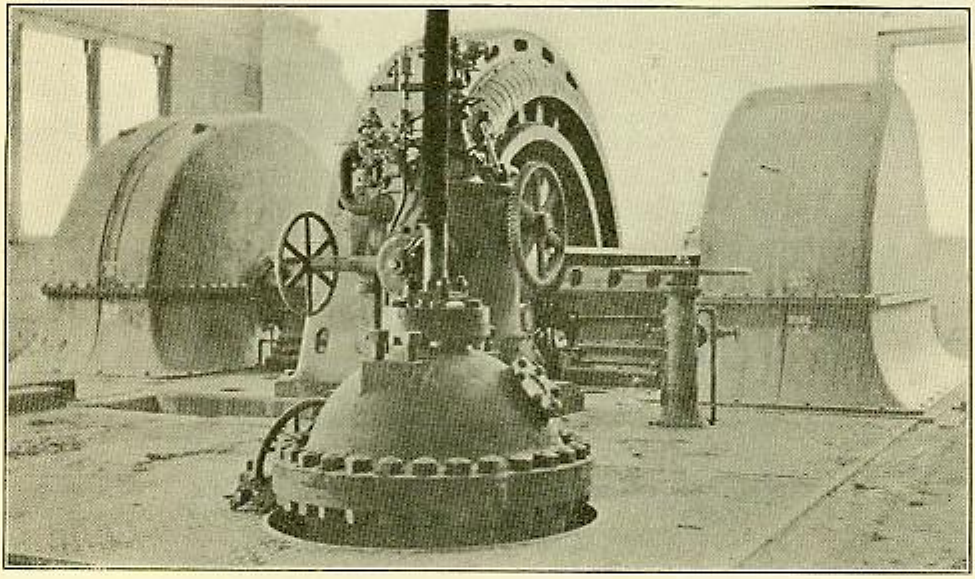


Fig. 172.—Hydro-Electric Unit, Kern River Plant No. 1, Southern California Edison Co.

well-regulated plant. The wheels, generators, and governors were so planned as to give not only the desired power and efficiency, but also a close regulation.

The speed of the main units was fixed at 250 R.P.M., and their capacity under a head of 865 feet is as follows: normal load, 7160 H.P.; 25 per cent overload, 8950 H.P.; 50 per cent overload, 10,750 H.P. The generators were guaranteed to safely carry a 50 per cent overload for several hours and the wheels were therefore designed for a maximum power of 10,750 H.P., or a station capacity for the four units of practically 40,000 H.P. A single wheel to develop the power of each unit would have been objectionable in size and the units therefore consist of two wheels, one on each overhung end of the generator shaft, as shown in Fig. 172. Each wheel is driven by a jet $7\frac{3}{8}$ inches diameter, the diameter of the wheel at the outside of the buckets is 9 feet

8 inches and the bronze buckets, 18 in number, are secured to a cast-steel rim. An interesting view of the nozzle and wheel is shown in Fig. 173.

The buckets were specially designed to obtain not only high specific speed, but also good efficiency. The manufacturer guaranteed 82½ per cent efficiency and it is stated that such efficiency has been approximately obtained. It would appear, how-

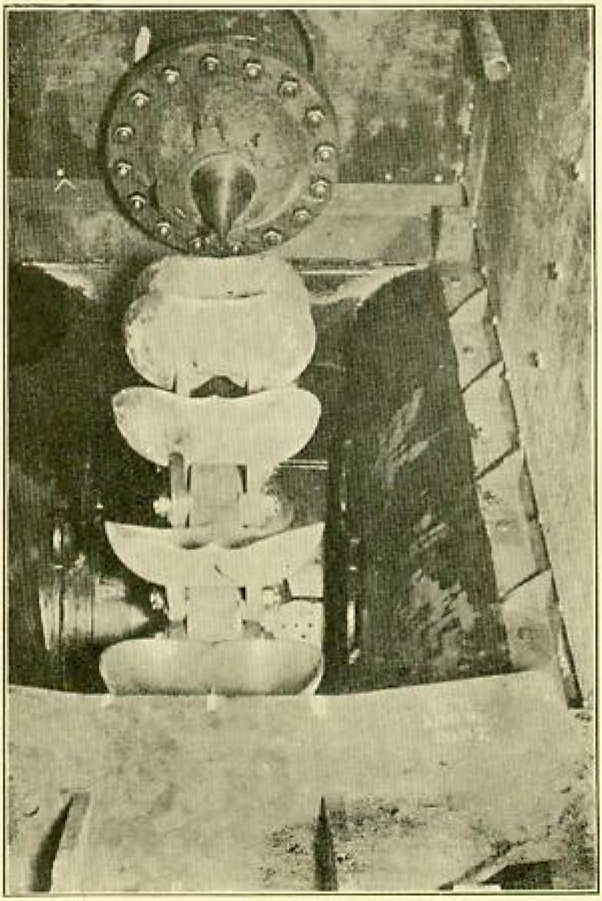
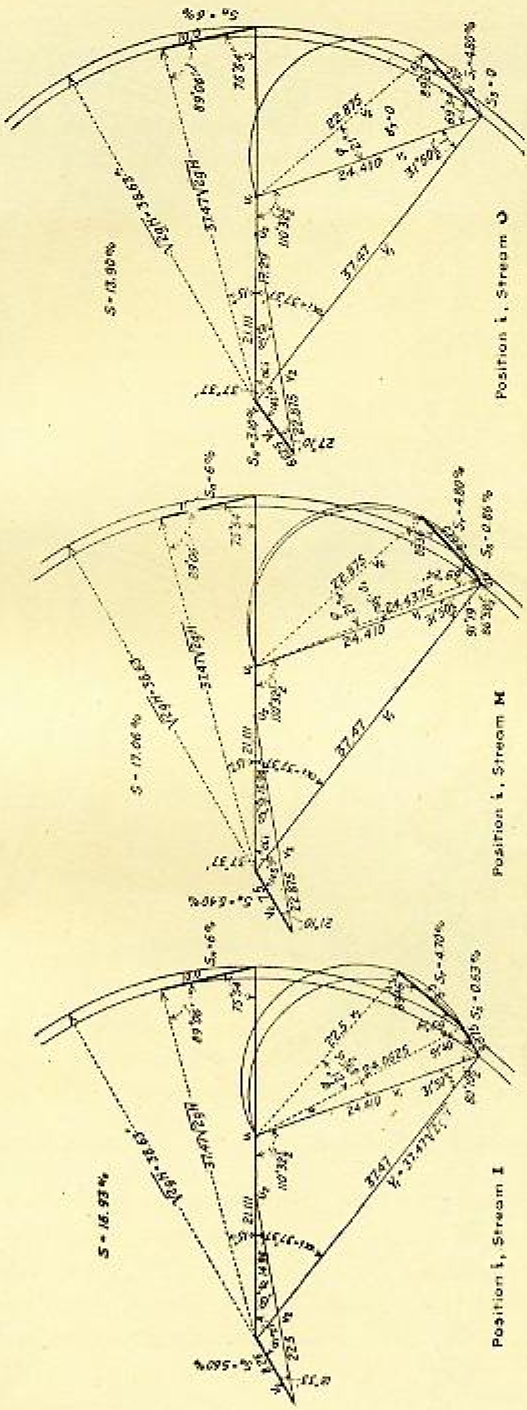


FIG. 173.—Nozzle and Wheel, Kern River Plant No. 1, Southern California Edison Co.

ever, that at peak loads either greater efficiency exists or the flow exceeds 470 second-feet. On Plates I, II and III will be seen the diagrams used in designing the buckets and the graphical determinations of the losses of energy for various positions of the jet. It will be noted that these losses include those in the water from the nozzle to the point of departure from the bucket but not those due to windage, friction on bearings, etc.

PLATE I

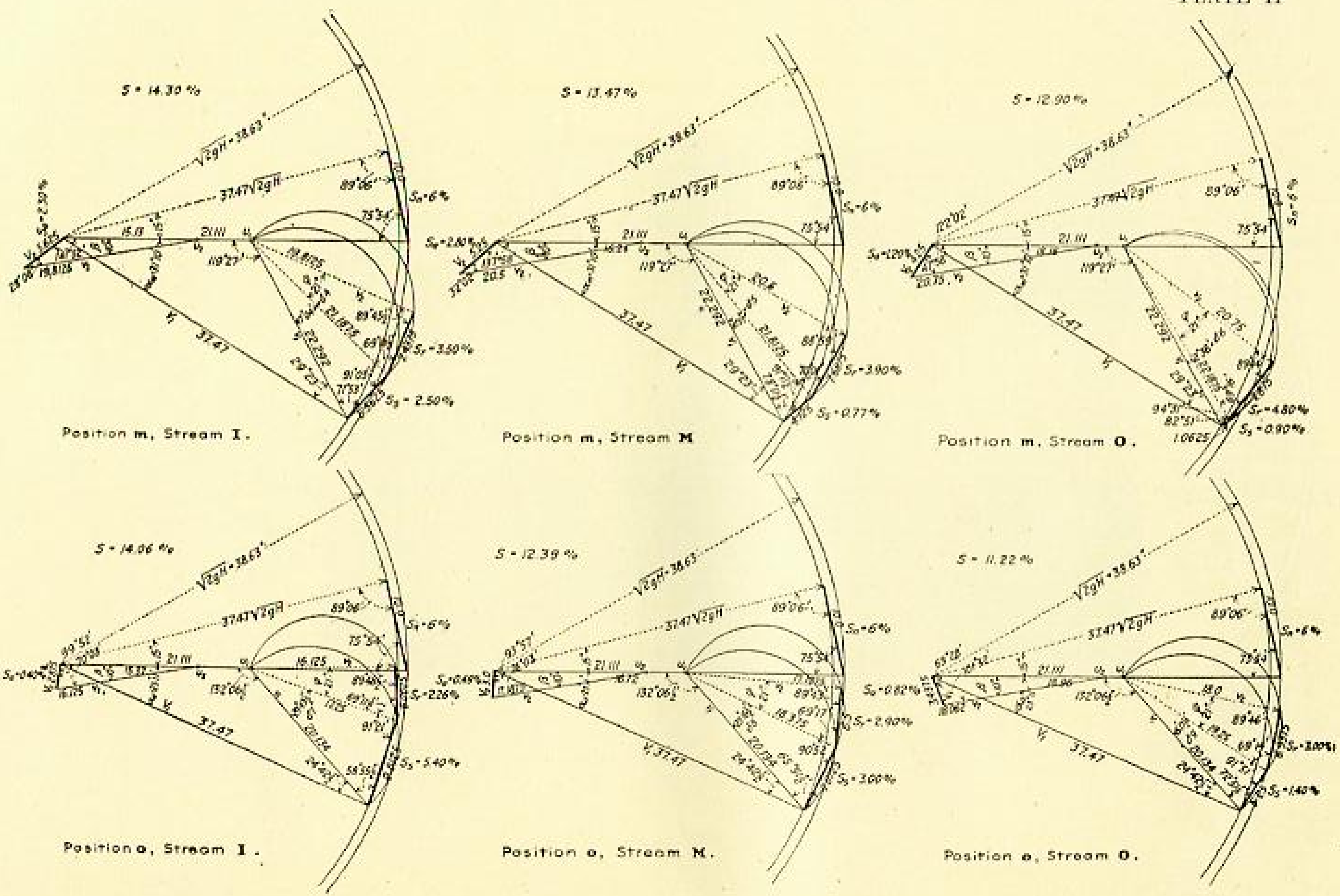


Position I, Stream O

Position I, Stream M

Position I, Stream I

PLATE II



Position m, Stream I.

Position m, Stream M

Position m, Stream O.

Position o, Stream I.

Position o, Stream M.

Position o, Stream O.

PLATE III

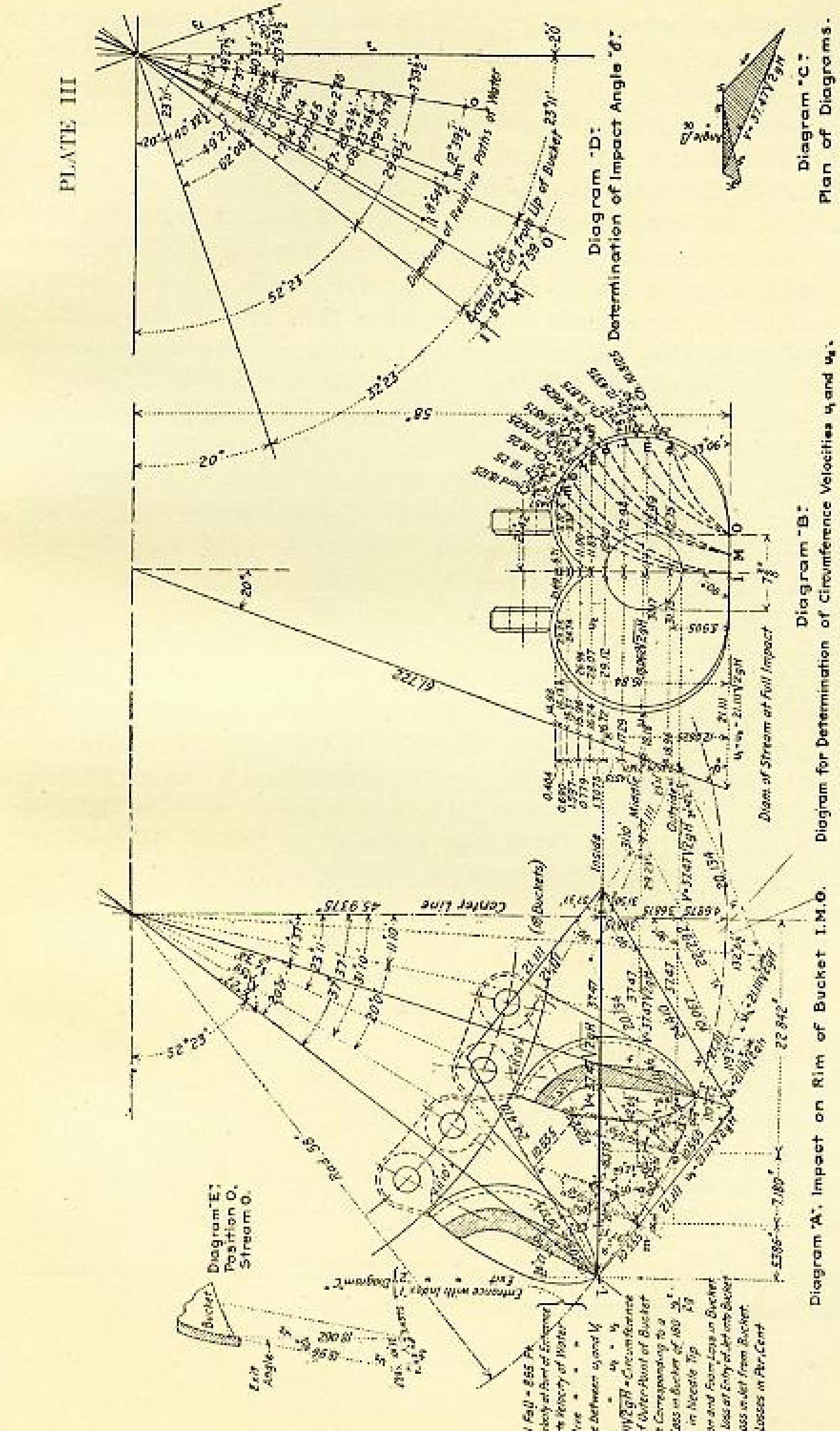


Diagram "A": Impact on Rim of Bucket I.M.O.

Diagram "B": Diagram for Determination of Circumference Velocities u_1 and u_2.

Diagram "C": Plan of Diagrams.

Diagram "D": Determination of Impact Angle "δ".

Diagram "E": Position O, Stream O.

One governor driven from the main shaft controls the nozzles of the two wheels connected with a given generator, the connection to the nozzles being made by a series of rocker arms and shafts and the weight of the nozzles partially carried by an hydraulic cylinder. The governors are arranged for both automatic and hand control in the usual manner.

The generators are of the revolving-field type, furnishing 3-phase, 50-cycle, 2300-volt current, the normal rating of each being 5000 KW. They are guaranteed to operate continuously at 25 per cent overload and for several hours at 50 per cent overload, the maximum capacity of the station thus being 30,000 KW.

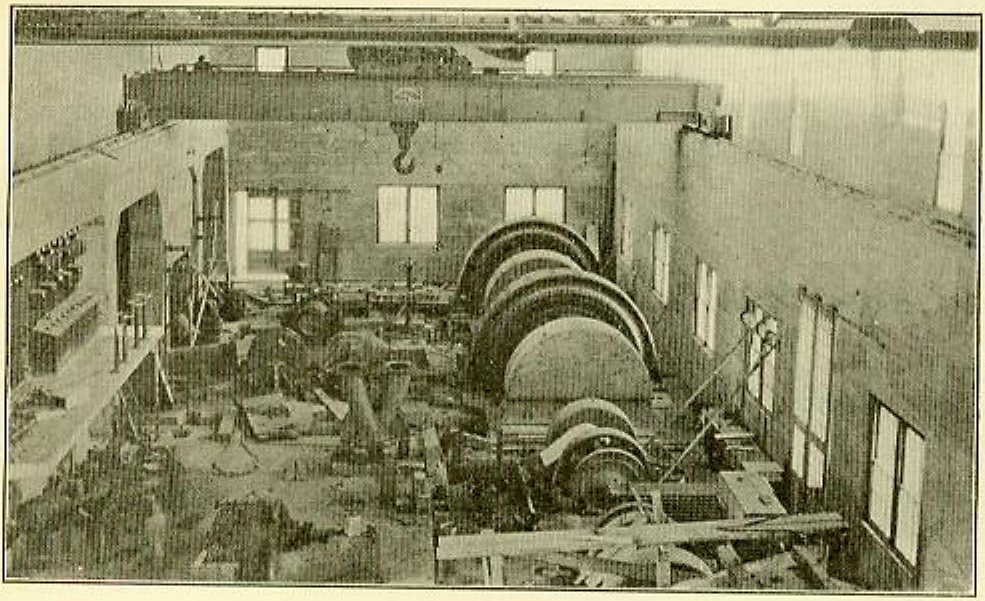


Fig. 174.—Construction View of Power House, Kern River Plant No. 1, Southern California Edison Co.

Owing to the great weight of the rotor the bearings between the revolving pole pieces and the wheels are 16 inches by 48 inches, lubricated with six flooded oil rings, the oil passing over water coils in the pedestals. In addition, there are small openings in the bottom of each bearing connected to a pump producing 1000 pounds pressure per square inch.

The exciters are 250-KW., 125 volt direct-current machines driven at 430 R.P.M. by impulse wheels having automatic oil pressure governors and stationary needles in the nozzles.

A general view of the interior of the power house during construction is shown in Fig. 174.

The station contains four banks of three 1667-K.W. transformers each, and one spare transformer. The transformers are connected Δ to the generators and Y to the transmission lines. The normal secondaries are 75,000 volts, but auxiliary secondaries are provided for 37,500-volt and 56,250-volt leads. Cooling is effected by oil circulation under low pressure, the oil being cooled by circulation through tubes in the forebay.

The control board contains switches for the generators, feeders, bus-bars, and transmission lines, and is a combination bench and panel board. It is mounted on a gallery near the center of the power house where the attendant has a clear view of the

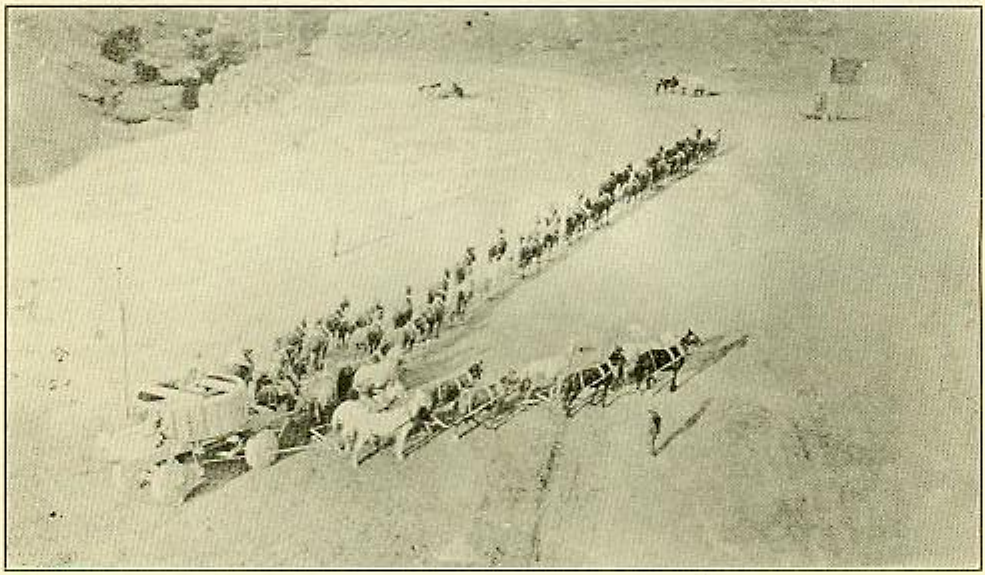


Fig. 175.—Transporting Machinery to Kern River Plant No. 1, Southern California Edison Co.

floor. Oil switches are provided for both the 2300-volt and 75,000-volt current, the latter switches being placed in a concrete-lined room with no other apparatus except the lightning arrester on the same circuit. Special efforts were made to isolate all of the high-tension apparatus so as to reduce the danger of short circuits to a minimum. The high-tension bus-bars are placed in concrete conduits 4 feet square and the outgoing lines leave the building through openings near the eaves to reach the first tower. These lines are equipped with choke coils and lightning arresters.

The power house calls for no special comment, being a concrete building 164 feet long and 66 feet 6 inches wide, including transformer and switch bays. The foundations were carried to rock and cemented boulders and portions were heavily reinforced.

The roof is of corrugated iron on wooden purlines, rather in contrast to the substantial character of the other portions of the plant.

The transmission line, 117 miles long from the power house to receiving station No. 3 at Los Angeles, is arranged for three circuits of three wires each on 6-foot centers, only two of them being erected for the first operation. Each conductor is composed of seven strands and is equivalent to 0000 B. & S. gauge. Hard-drawn copper with an elastic limit of 35,000 pounds per square inch was employed. The towers are of galvanized steel sections and were designed for a stress transverse to the line of 30 pounds per square foot on towers and conductors and for a stress longitudinal to the line due to the breaking of one conductor. The normal span in open, flat country is 700 feet, but much of the line passes through a difficult broken country where it was necessary to place towers on high points and thus cross the intervening hollows. In one case a span of 2250 feet was used with a guide tower at the bottom of the sag. The height of the towers varied from 30 feet to 60 feet. Two telephone circuits were carried on the lower part of the towers and on intermediate wooden poles. Porcelain insulators of the pin type $18\frac{1}{4}$ inches high with four petticoats and a pin protector were employed. There are three switching stations on the line, two of which are also transformer substations.

Fig. 175 will give the reader an idea of the difficulty of transporting heavy machinery to this plant.

A TURBINE INSTALLATION OF 23,500 H.P. AT GRAND FALLS, NEWFOUNDLAND

A large wood-pulp and paper mill driven by turbines of 23,500 H.P. has recently been erected and placed in operation on the Island of Newfoundland at St. John's and Grand Falls. It was built by the "Anglo-Newfoundland Development Company," located at the above-named places.

The conditions existing at St. John's were peculiarly favorable to the constructing of an economical plant and the power for operating it was present at Grand Falls in the form of a waterfall of 106 feet, where a power plant of large capacity could readily be installed. There was at hand an abundance of wood which by the power furnished from the turbines could be ground into pulp and then made into paper. The credit of having recognized the importance of these conditions belongs to Lord Northcliffe of London, the founder and head of an association of the leading London news companies. This association concluded to construct the plant with the further object of transporting the paper thus cheaply made to London, there to be used for printing newspapers. The execution of the plans and the management of the new paper mill was vested in a new company called the "Anglo-Newfoundland Development Co." The management of this company was in the hands of Mr. H. Beeton of London as president, while Mr. George F. Hardy, a consulting engineer in New

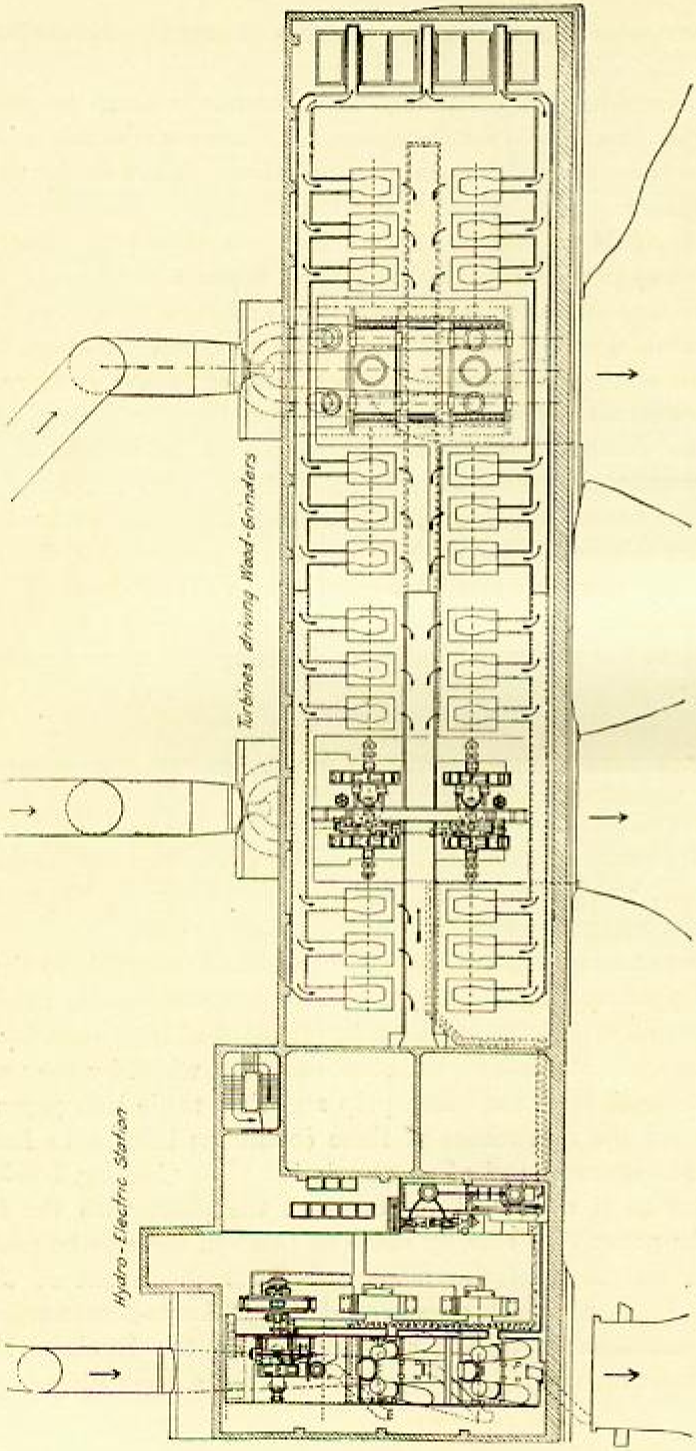


FIG. 176.—Hydro-Electric Station and Pulp Mill, The Anglo-Newfoundland Development Co.

York, acted as advisory engineer on the ground as regards the construction of the plant.

A lively competition for furnishing the turbines required for the power plant arose between the American and European firms engaged in this business, a competition finally decided in favor of Amme, Giesecke & Konegan, the turbine builders of Braunschweig.

In the following pages will be described the turbines and governors furnished by this firm, the apparatus being novel in many respects.

We will omit any extended description of the headworks and the method of conducting the water to the power house. It may be remarked, however, that for the latter purpose there are installed two steel pipes, each 15 feet in diameter and about 2150 feet in length, which are connected near the power house by one header 10 feet in diameter. One of the penstocks is reduced from 15 feet in diameter to 10 feet and supplies the turbines driving the generators. At a point opposite the center of the power house the other penstock is divided into two branches, each 10 feet 9 inches in diameter, and feeds the turbines operating the grinders (See Fig. 176). The thickness of the 10 foot pipe is ½ inch. The circular seams are single riveted and the longitudinal seams are double riveted. In Fig. 177 is shown the particular form of expansion joint used to compensate for changes in the length of the penstock. The size of the penstock was so chosen that at full load on the plant the velocity therein will not exceed 8.2 feet per second.

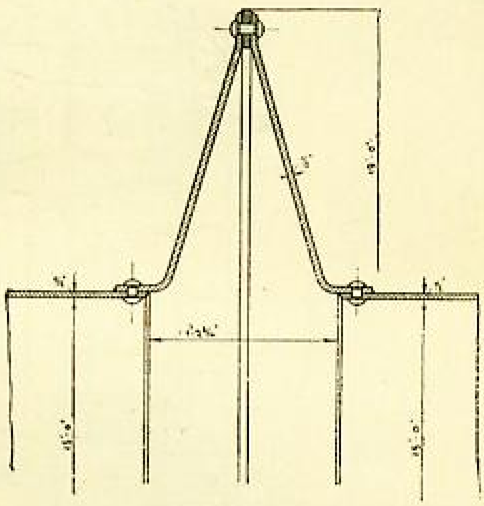


FIG. 177.—Expansion Joint for Penstock.

The turbine plant itself is divided into three principal parts as follows:

(1) The group of turbines used for driving the wood grinders, the same consisting of four turbines of 4000 H.P. each operating at 225 R.P.M. Each turbine is coupled on each side with three powerful grinders each of about 700 H.P., a unit thus consisting of six grinders (see Fig. 176). The conducting of the raw material to the grinders is effected by a channel in the middle of the power house in which the round logs, cut to length, are floated from without so that they may be taken in the most direct way to the presses; on the other hand the pulp produced by the grinders is conducted away in two conduits which extend the length of the power house next to the walls and connect with the grinders by cross-channels.

(2) A group of turbines used for driving the special machinery installed in the

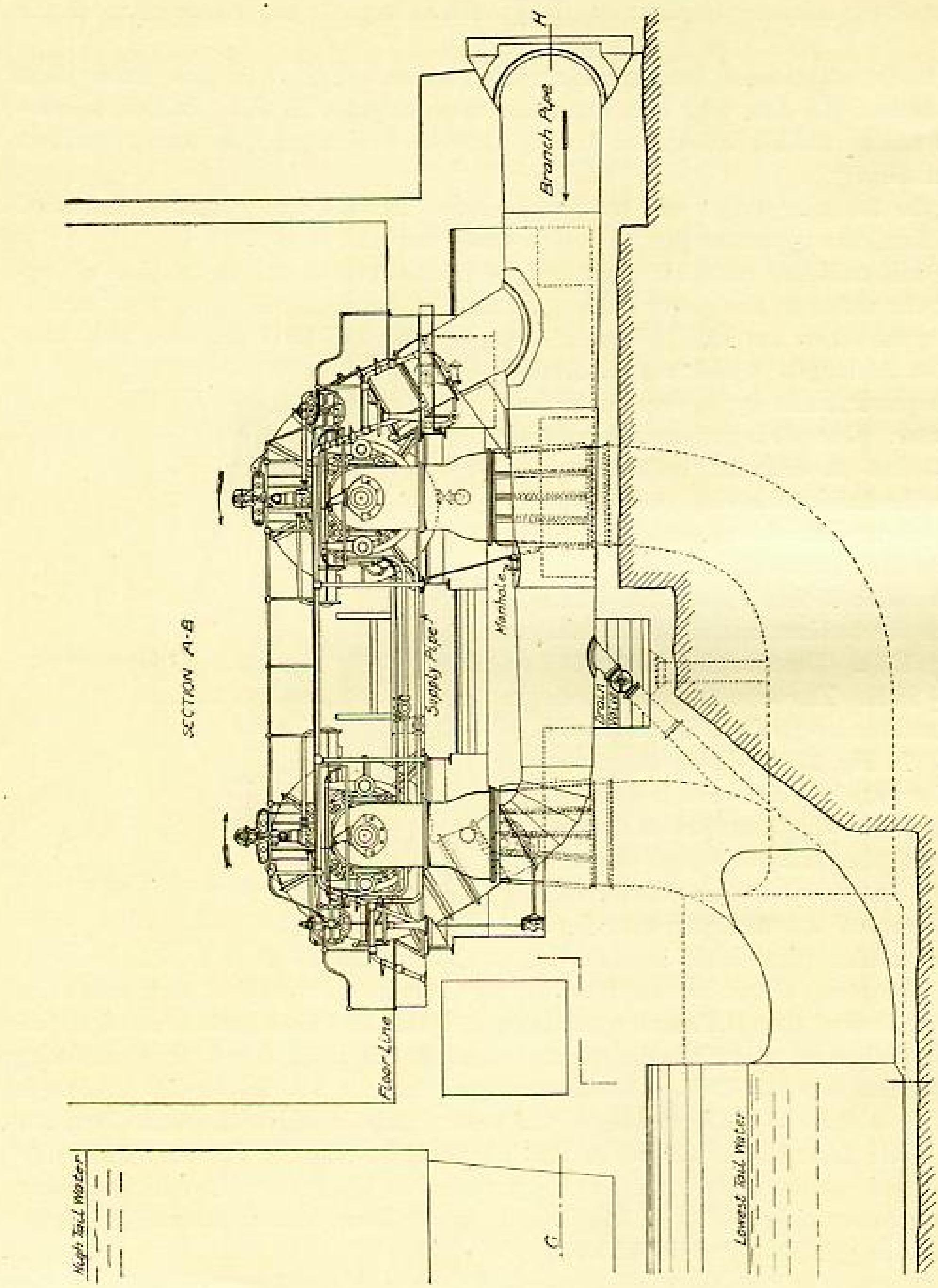


Fig. 178.—End Elevation showing Two 4000-H.P. Double Turbines for Grinding Pulp, The Anglo-Newfoundland Development Co.

paper mill. As this mill is somewhat removed from the power station electric power was chosen instead of mechanical power for transmission thereto and for this purpose

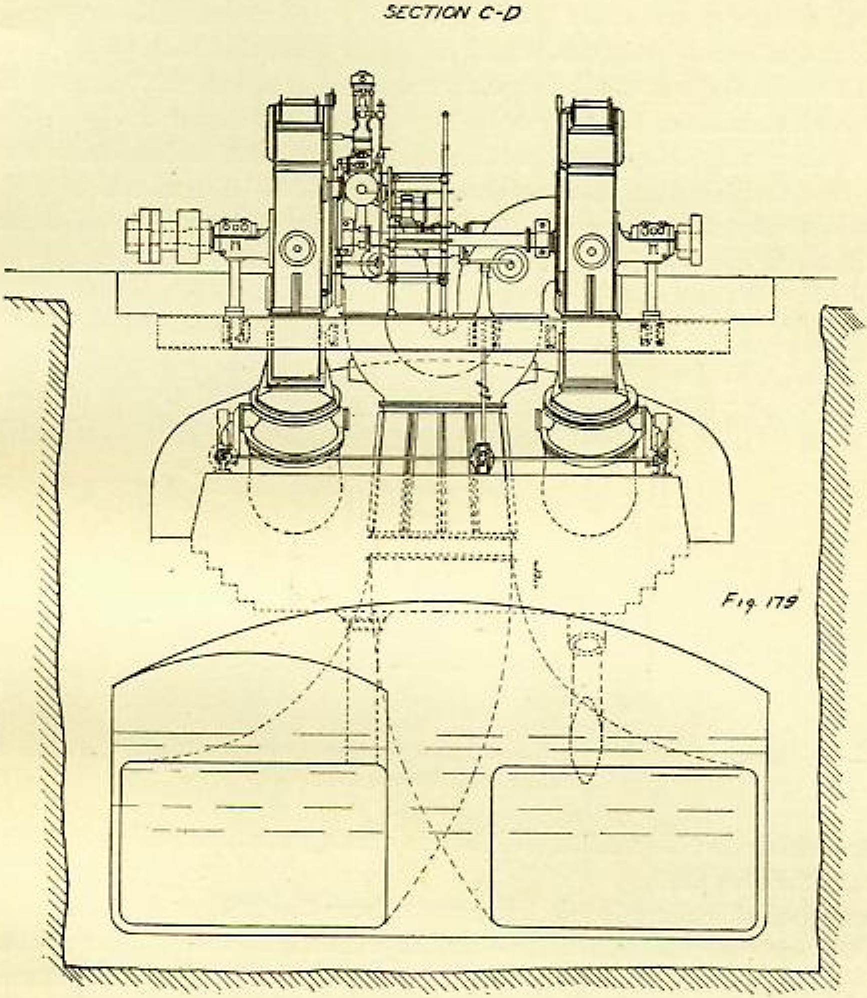


FIG. 179.—Side View of 4000-H.P. Double Turbine for Grinding Pulp, The Anglo-Newfoundland Development Co.

there were placed in the power house three turbo-generator sets each of 2500 H.P., operating at 375 R.P.M.

(3) An installation of oil pressure pumps for providing oil under pressure which serves for the regulation of the grinder turbines as well as those driving the generators.

The several groups of machinery in the power station will now be described in the order in which they are enumerated above:

(1) *Grinder Plant* (see Figs. 178 to 185.)

The distributor pipes which feed the 4000-H.P. turbines are bifurcated and without further description their construction may be readily understood from Figs. 180 and 181. Each main forked pipe is connected with two double turbines each of 4000 H.P., and thus two such distributor pipes are necessary for the four turbines.

The usual practice is followed in the general arrangement of the turbines as double turbines with double spiral cases as shown in Figs. 178, 179 and 182.

Two butterfly valves (see Fig. 183) are used for closing the two branch pipes leading to each turbine, and these are so connected with each other that they open and

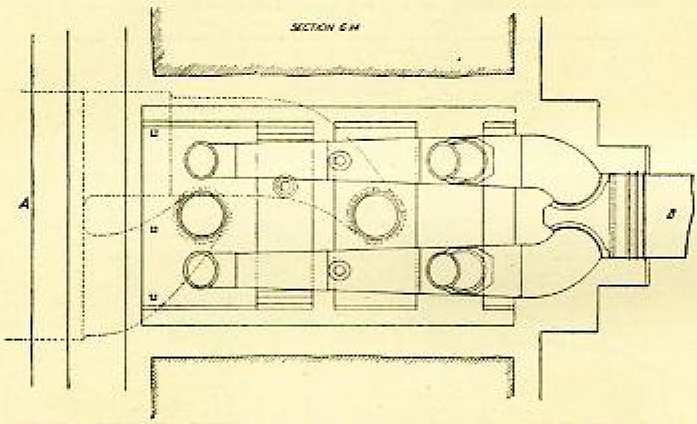


Fig. 180.—Plan showing Penstocks and Draft Tubes for Two 4000-H.P. Double Turbines, The Anglo-Newfoundland Development Co.

close coincidently (see Fig. 179). With this arrangement an axial thrust can never exist in the turbine shaft.

The method of leading the discharge water from the turbines is worthy of attention because the two concrete draft tubes—each of which is attached to its double turbine by a Y piece and two elbows—must pass by each other. This results from the fact that in the case of each pair of the 4000-H.P. turbines of this installation one lies behind the other in respect to the direction of the discharge. The concrete draft tubes must therefore have a doubly curved center line (see Figs. 179 and 180). Of greater interest is the construction of the turbine and especially of the distributor as well as the governor. These were built by the Amme, Giesecke & Konegan Co., and their design is novel in many respects.

(*a*) Description of the Turbine.

The runner of the double turbine revolves at a specified speed of $187 \div \sqrt{2} = 133$

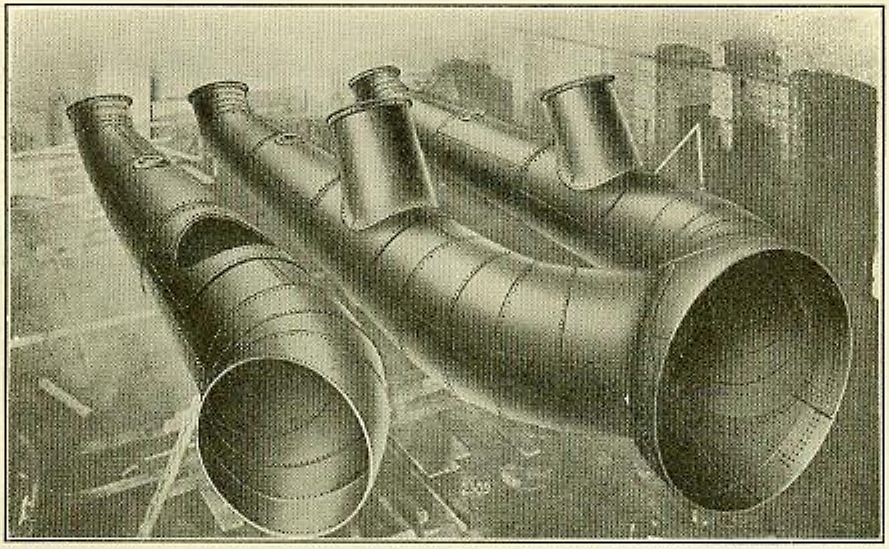


FIG. 181.—Photograph of Penstock Supplying Two 4000-H.P. Double Turbines, The Anglo-Newfoundland Development Co.

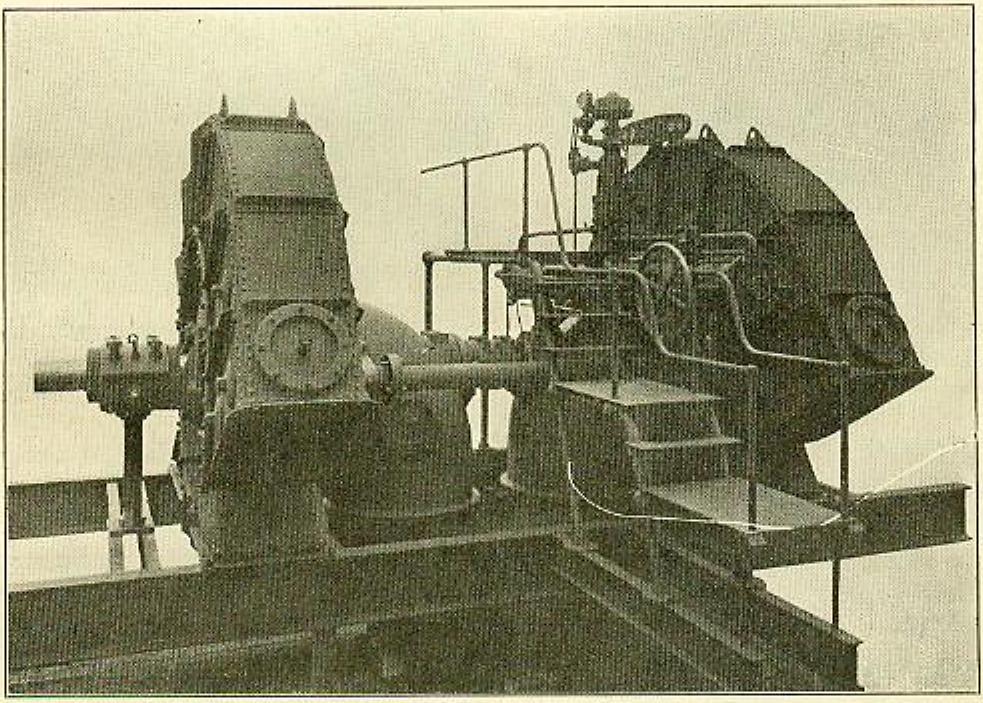


FIG. 182.—Photograph of Assembled 4000-H.P. Double Turbine with Governor, The Anglo-Newfoundland Development Co.

R.P.M. and is a true normal runner, thus warranting a high efficiency. Each runner has 17 sheet-steel vanes, a photograph of the same being shown in Fig. 184.

Surrounding the runner is arranged the distributor with 20 movable vanes which are forged in one piece with their pivots, the latter passing on each side through the distributor crowns (see Fig. 185). Interchangeable bronze bushings are provided in the covers of the housing as bearings for the pins of the revolving vanes. Tight-

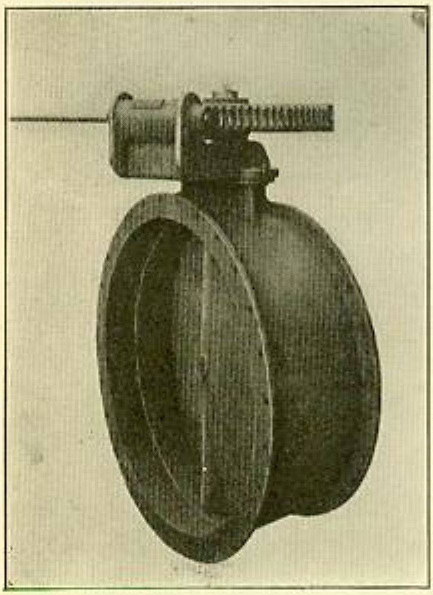


Fig. 183.—Photograph of Butterfly Valve for Branch Penstocks, The Anglo-Newfoundland Development Co.

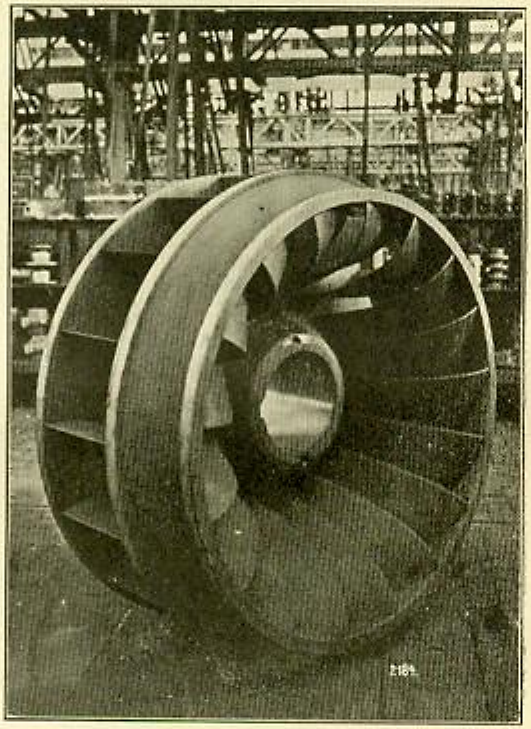


Fig. 184.—Runner for Double 4000-H.P. Turbine, The Anglo-Newfoundland Development Co.

ness is obtained by stop bushings which are provided with lock nuts. At the place where the link from the revolving ring is fastened to the movable vane there is fixed an outer bearing, also bushed, in order to support the reaction pressure on the vane, and this is so constructed that it forms in respect to the inner bearings a single support which is secured to the cover of the housing by two screws. This construction makes it possible to dismantle in an unusually simple manner and to easily renew the bearings and bushings. During operation the inner as well as the outer bearings are lubricated by grease cups. The ends of the levers operating the movable vanes are

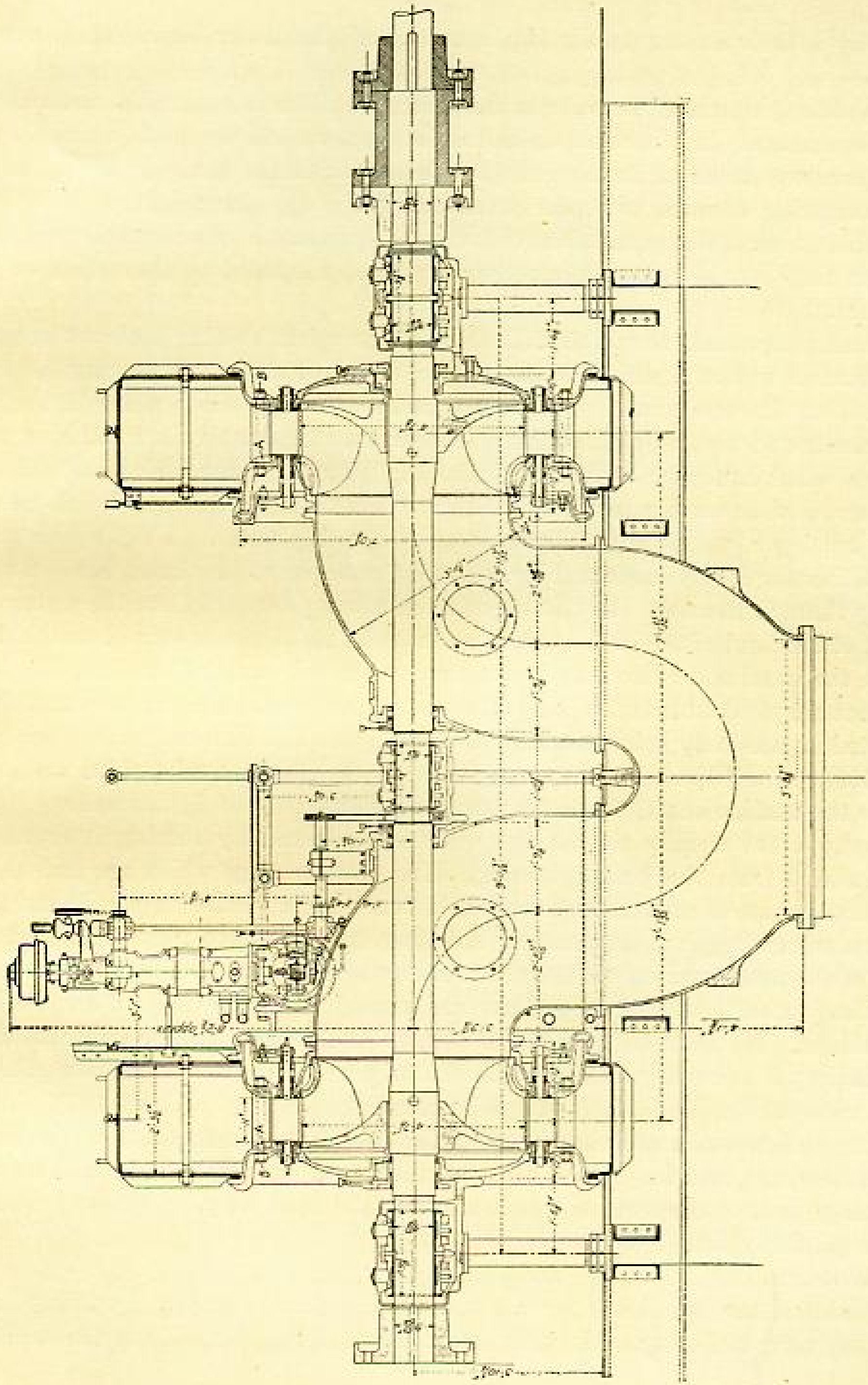


FIG. 185.—Section showing Double 4000-H.P. Turbine Operating Pulp Grinders, The Anglo-Newfoundland Development Co.

connected by links to the regulating ring, a section of the latter having the form of an S.

The regulating ring itself is made in one piece and slides in an axial direction over a cylindrical surface. It is secured against axial displacement by stops which at the same time serve as limits of its movement. The power of the governor is applied to the regulating ring through two pins at diametrically opposite points.

The inner surface of the distributor covers are protected by renewable wrought-iron angular rings at the points where they are most subjected to the action of the flowing water.

It was also important to stiffen both faces of the spiral housing against the inner water pressure as well as against the buckling of the steel plates. This reinforcement was effected, on the one hand, by staybolts passing through the housing, spaced at proper intervals and secured against the water pressure by double nuts and lock nuts and, on the other hand, by a double reinforcing ring (A) lying outside of the distributor vanes and extending from cover to cover. This is provided with fish-formed ribs whose cutting edges lie in the direction of the flowing water. A rigid connection with the distributor covers is effected by bolts which are placed in reamed holes extending entirely through the ribs. In this way the bolts are protected from the water and on account of the form of the inclosing ribs no material resistance is offered to the flow.

Unlike the usual and customary arrangement with this style of turbines, the draft tubes are not simple double bends, but an elbow connects with each turbine and those are connected immediately below the floor with a Y piece and thus united into one pipe (see Fig. 185). The object of this arrangement is the insertion of a third accessible bearing for the turbine shaft, the latter being altogether 24 feet 5¾ inches long. The bearing is supported equally on the two quarter-bends and by turning it around it may be removed from the bearing casting. The two bearings which are fastened to the wheel housing are of normal construction, but each has a special support in the form of a column whose length is adjustable. All three bearings have removable bushings lined with white metal and one of the end bearings is built as a double-thrust bearing.

As already mentioned, three grinders are arranged on each side of each of the double turbines. In order to be able to remove each single runner from its shaft without making it necessary to dismantle the adjacent grinder, the turbine shaft is connected with the grinder shaft by a special coupling (see Figs. 179 and 185). When this coupling is removed a sufficient space is provided between the ends of the shafts so the wheelhousing and the runner may be taken out through it. In this manner the remaining parts of the turbine which encircle the shaft may be removed without taking the grinder apart.

(*b*) Governors (Figs. 182, 185 and 186).

The turbines are controlled by oil pressure governors having revolving balls, valves, pilot valve and relay. However, their construction differs from the ordinary one in respect to the balancing levers, this detail, among others, being covered by U. S. Patent No. 929206. (The inventors of this governor, which has proven very successful in practice, are Messrs. Kugel & Gelpke.)

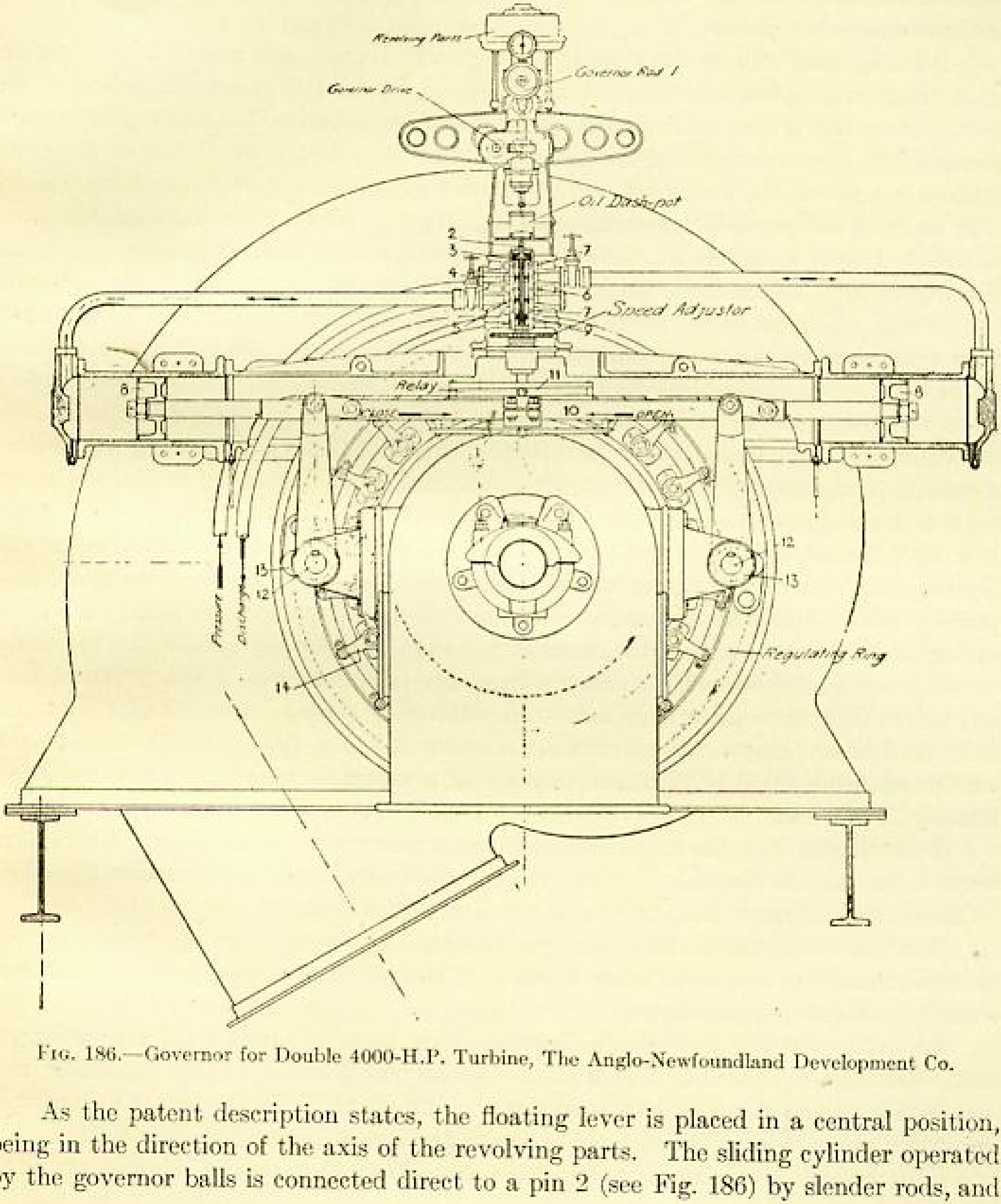


FIG. 186.—Governor for Double 4000-H.P. Turbine, The Anglo-Newfoundland Development Co.

As the patent description states, the floating lever is placed in a central position, being in the direction of the axis of the revolving parts. The sliding cylinder operated by the governor balls is connected direct to a pin 2 (see Fig. 186) by slender rods, and this pin is in turn connected with part 3, which at the same time forms the piston of the pilot valve. This piston is so arranged as to open and close the ports 4 and 5. For example an upward movement of the piston due to the action of the governor balls will connect the port 4 with the pressure chamber 6 while port 5 will be connected with the discharge chamber 7.

Ports 4 and 5 lead to the main operating valve and at the moment when they are open there takes place a movement of the piston 8 which transfers the action of the governor to the levers connected with the regulating ring. A secondary movement produced by the movement of the piston 8 has for its purpose the closing of the ports 4 and 5 in order that the action of the governor may be interrupted at the proper time. This is rendered possible according to the foregoing construction because the ports 4 and 5 are not arranged in a stationary valve, but in a bushing 9, surrounding the floating piston. Therefore this bushing simply requires to be shoved back toward the piston, which has been moved up in order that the ports which have been opened may again be closed by the projections on the pilot valve. This return movement of the bushing is accomplished in a simple manner by attaching an inclined surface 11 to the piston rod 10 of the main operating valve, the end of the bushing being supported upon this surface by a roller bearing or otherwise. The bearing is therefore moved up and down as the piston moves back and forth. Morever, the part connecting the bushing with its bearing is so arranged that its length may be altered either by hand or by an electric motor while the machine is in operation. In this way the operator has a simple means of changing at will the speed of the machine. It is a natural supposition that throughout the entire operation of the governor a part, say one-third, of the vertical movement of the revolving head is not effective, while the remaining two-thirds may be utilized for changes in speed. For example, if the governor head has a total vertical motion of $1\frac{3}{16}$ inches when the speed changes from 100 R.P.M. to 115 R.P.M., then one may obtain a normal minimum speed of 102.5 R.P.M. and a normal maximum speed of 112.5 R.P.M., corresponding to a possible change in speed of approximately 10 per cent.

It is evident that the relay action of the bushing may be accomplished in other ways if desired, but that above described appears to be the simplest method by which to translate a horizontal motion into the desired vertical motion.

Without further discussion it is evident that such a governor must operate with unusual sensitiveness and accuracy because of the omission of nearly every joint and experience shows this to be the case.

As in the construction above described it is best to install such a governor so that its action may be communicated as directly as possible to the regulating ring and so that a true force couple may then be introduced. In the foregoing example this was accomplished in the case of the double turbine as follows (see Fig. 186):

The shell of the main operating valve, consisting of two cylinders and an intermediate casting, is bolted by means of feet against the spiral housing. The automatic pilot valve and revolving parts stand on the casting connecting the two cylinders (see Fig. 186).

The two pistons of the operating valve are connected with each other by a piston rod which is secured by links to the forked ends of the diametrically opposite main regulator levers and in this way revolve the regulating shafts 12. At the ends of these shafts are short crank arms, which are fastened by means of links to pins which are located at opposite ends of the diameter of the regulating ring.

All of the lubricating oil is very carefully collected and is led back to an oil reservoir through the drain pipes.

In consequence of the importance of the installation the governor balls are not driven by a belt, but by a wormwheel in connection with a Morse silent chain.

Besides the automatic regulation two hand-regulating devices are provided, viz.:

(*a*) A controlling arrangement consisting of a three-way cock, which device presupposes the existence of oil under pressure. With this the turbine may be brought from full load to no load in about ten seconds;

(*b*) A regulating device using hand pump. With this the turbine may be opened or closed in about three minutes. It is necessary to use the apparatus only in case of failure of the oil pressure from the power-pumping plant, a contingency which is extremely improbable, as it could happen only by the bursting of the oil piping.

The accessibility of each of the several moving parts of the exposed governor is largely taken advantage of by the provision for an operating platform. All of the operating levers for the control of the machine are so compactly arranged that the machinist has the governor under complete control without leaving the platform. For a similar reason the indicating instruments are assembled in one frame so that the operator without leaving his station may read the tachometer, and the several gauges showing respectively the pressure in the turbine, the pressure in each cylinder of the main operating valve and the vacuum in the draft tube.

As will be seen from Fig. 178, each pair of turbines have a connecting bridge for the purpose of crossing the channel in which the wood is brought to the turbines.

(2) *Turbines Driving the Generators* (see Figs. 187, 188 and 189). The turbines in the electric power house are also double turbines but, in contrast with the 4000 H.P. turbines above described, each of them has only one spiral housing with right- and left-draft tubes. The branch penstock leading to each turbine joins the heads at an angle of 60° and in each branch is a butterfly valve about 4 feet 6 inches in diameter, operated either mechanically or by hand, for the regulation of the water supply.

Each of the cast-iron quarter-bends receiving the discharge from the turbines is connected below to a riveted draft tube. In order to improve the conditions of the

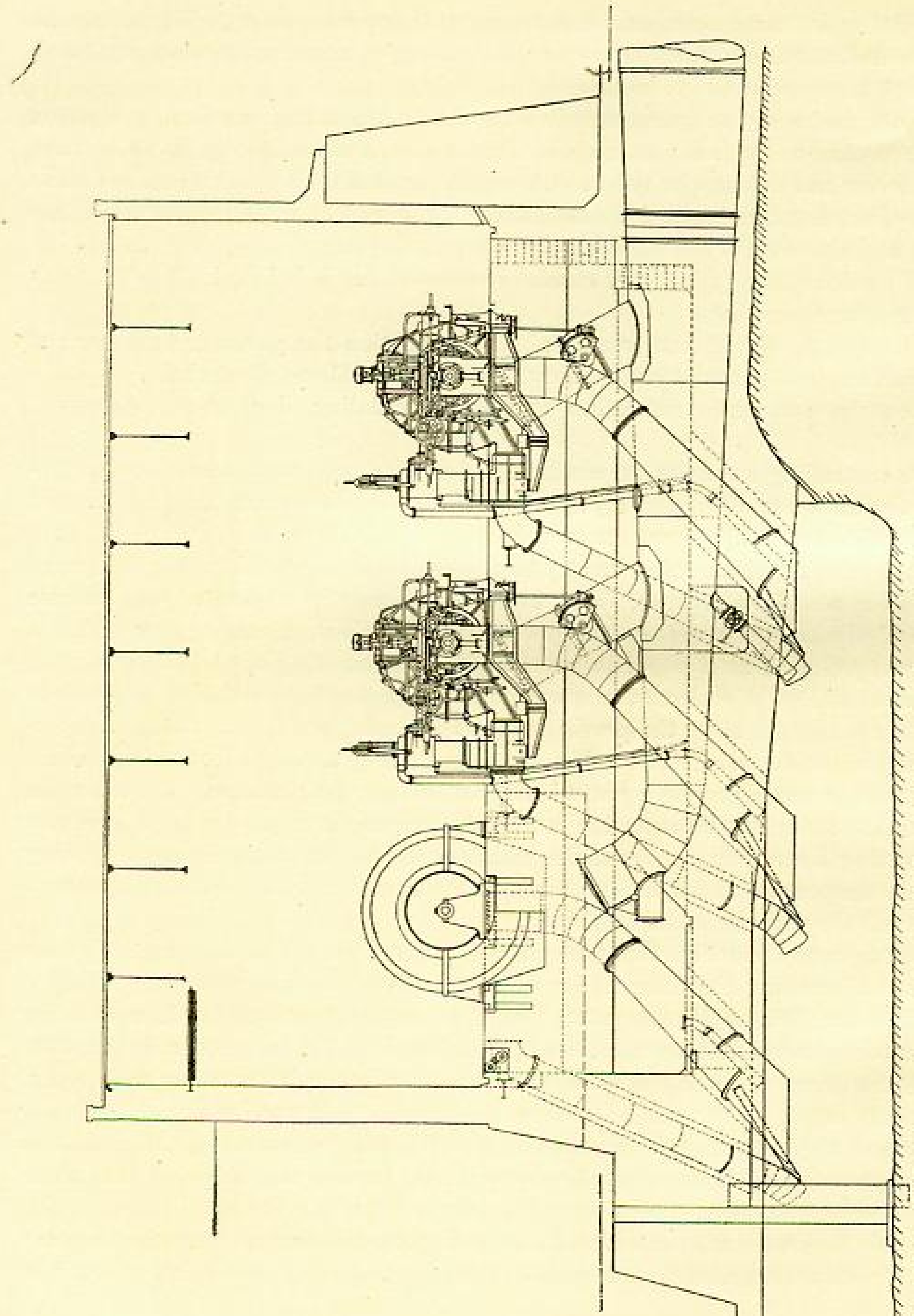


FIG. 187.—2500 H.P. Hydro-Electric Units, The Anglo-Newfoundland Development Co.

discharge into the tail water each of the latter are cut off on an angle at the bottom. (See Figs. 187 and 188.)

The normal speed of the turbines is 375 R.P.M., but by means of the apparatus for changing the speed, as above described, it may be varied at will from 356 R.P.M. to 394, while the machine is in operation. The specified speed of a single wheel of the double turbine was $247 \div \sqrt{2} = 174$.

The WD^2 of the generator rotor amounted to 434,000 ft.lbs., and it was guaranteed

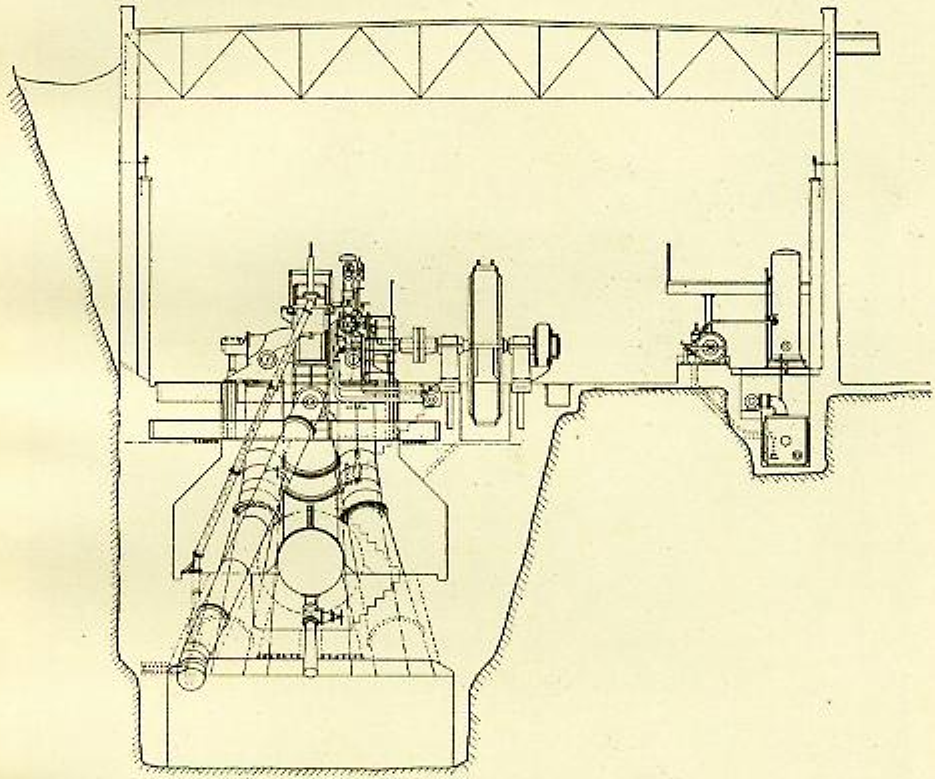


FIG. 188.—Section of Hydro-Electric Power Station, The Anglo-Newfoundland Development Co.

that when the entire load was instantaneously thrown off the increase in speed would not exceed 12 per cent.

The generators, which have a capacity of 1700 KW., were built at the works of Brown, Boveri & Co. at Baden, Switzerland. They are of the revolving field type, 600 volts, 1800 amperes, 50 cycles when the power factor is 0.8. Each generator has an exciter having a capacity of 43 KW. at 115 volts, the current furnished by each being sufficient to excite two generators.

The efficiency of the generators at full load, including friction and windage, is 94 per cent with a power factor of 100 and 93 per cent with a power factor of 80 per cent, while at half load the corresponding efficiency is 90 per cent and 89 per cent,

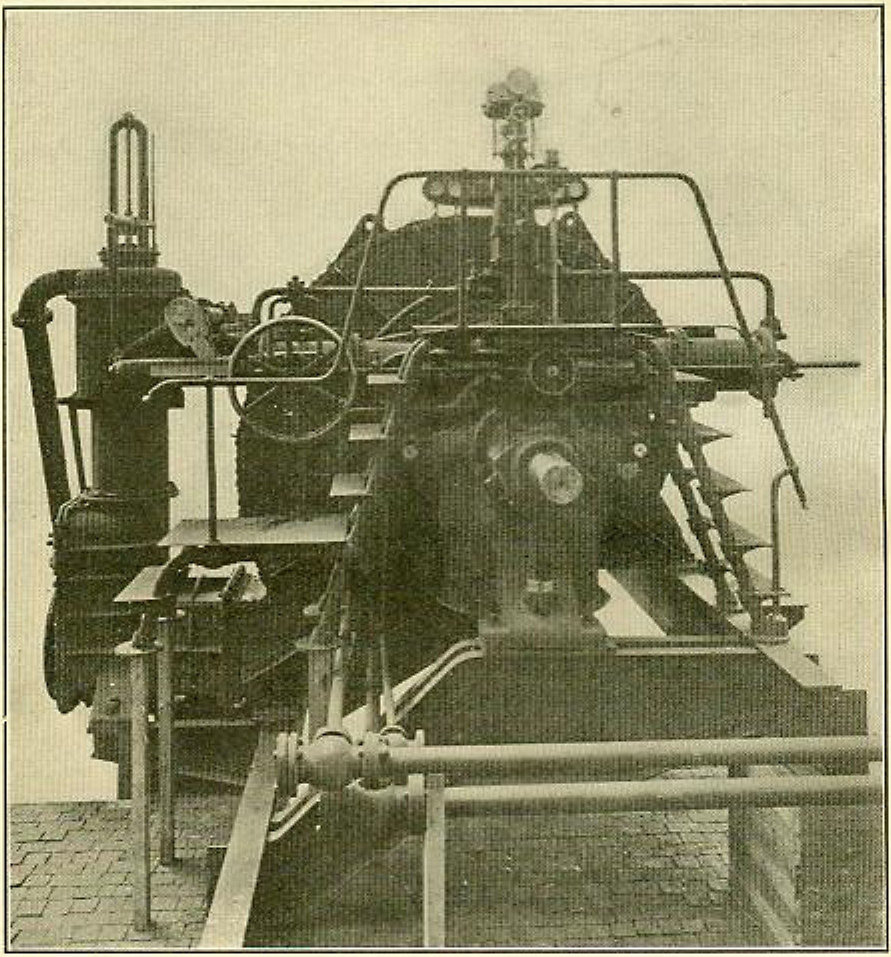


Fig. 189.—Photograph of 2500 H.P. Turbine Assembled with Governor, The Anglo-Newfoundland Development Co.

respectively. The change in tension from full load to no load is 8 per cent when the power factor is 100 per cent and 20 per cent when the power factor is 80 per cent.

The construction of the generator turbines as well as that of the speed governors is practically the same in detail as that of the 4000-H.P. turbines, and the reader is referred to Fig. 189 for a photograph of the assembled machine.

It is worth while to mention the pressure regulator which is provided for each turbine. When the turbine is suddenly closed it is the office of this regulator to provide a free opening for the moving water and to close automatically and slowly after the completed action of the governor. It thus prevents an abnormal rise in pressure in the penstock. The fundamental idea of this device was fully described by Professor Prazil in the Swiss "Bauzeitung" in 1906 and is here carried out on a larger scale and in connection with a pilot valve of special construction. The drain pipes from the pressure regulator discharge into tail water and their lower section is so curved that

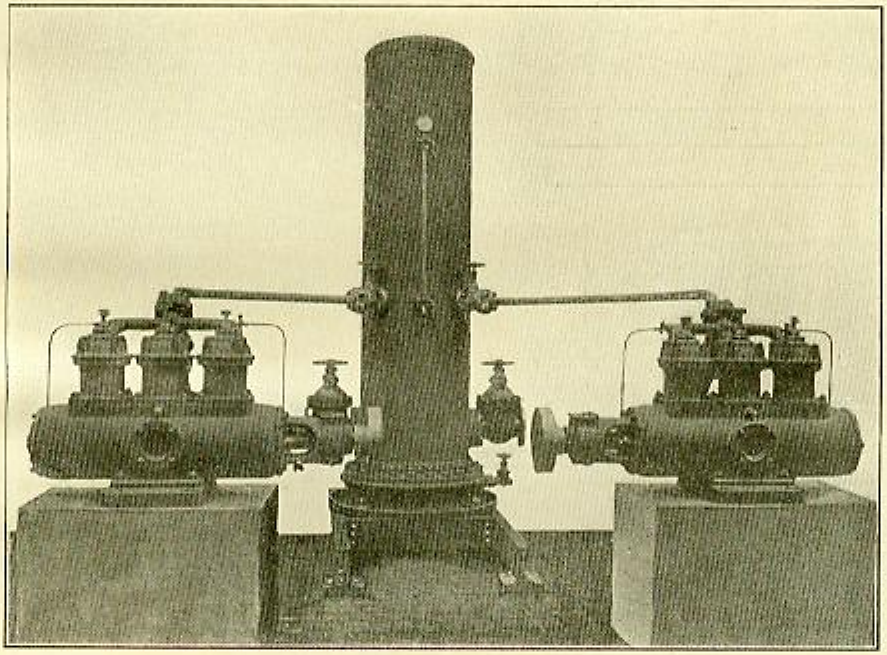


Fig. 190.—High-pressure Oil Pump Unit, The Anglo-Newfoundland Development Co.

the discharge water under high velocity does not injure the walls and bottom of the tailrace.

(3) *The Oil-pump Installation* (see Figs. 190, 191 and 192). Four three-cylinder reciprocating pumps furnish the necessary supply of high-pressure oil for the installation, and the pressure is equalized by two steel air cylinders each 2 feet 4 inches in diameter and 9 feet 6 inches high.

The arrangement of one group of pumps may be seen in Fig. 190.

A steel-plate reservoir with a capacity of 2100 gallons is placed at a low point in order that all of the oil required by the governors, including the lubricating oil, may return to it by gravity. Before its entrance to the reservoir it is cleansed in a separator.

The pressure pipes are welded tubes with the flanges welded on.

The design of the pump is worthy of notice both on account of its compact construction with a correspondingly great capacity and because it is fully inclosed so that all of the moving parts operate in an oil bath (see Figs. 191 and 192).

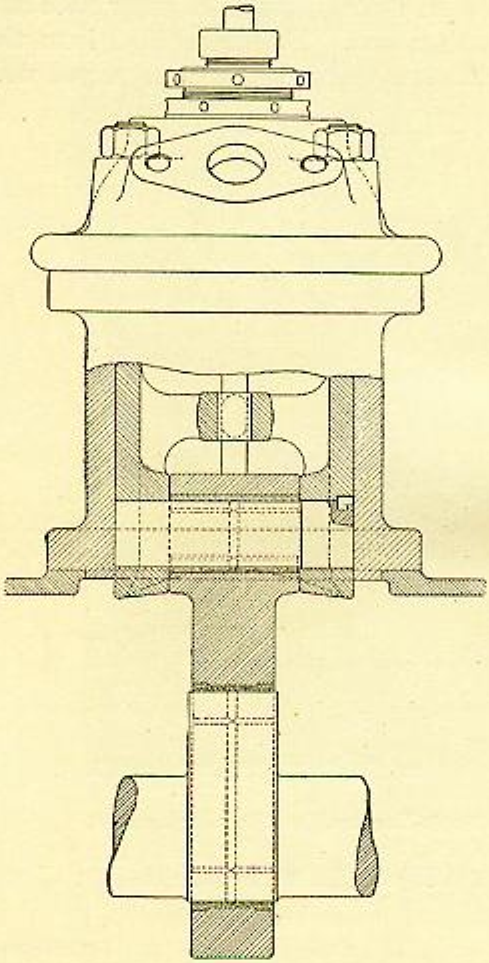


Fig. 191.—Section of Piston and Plunger for High-pressure Oil Pump, The Anglo-Newfoundland Development Co.

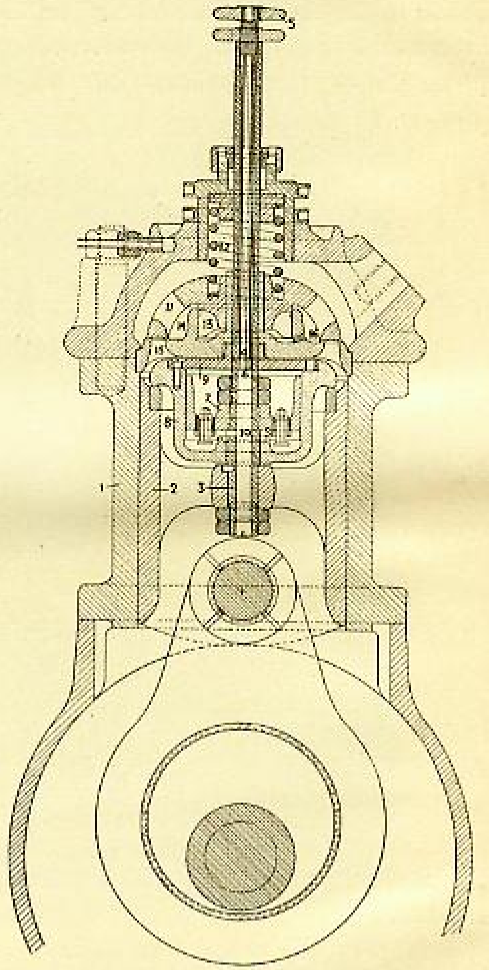


Fig. 192.—Details of Piston and Plunger for High-pressure Oil Pump, The Anglo-Newfoundland Development Co.

The suction valve is built in the piston itself (2) while the pressure valve is secured to the cover of the cylinder (1). Both valves are so controlled by dashpots that at the end of the stroke the piston comes gently to its seat, thus producing a silent operation. In the case of the suction valve this action is obtained as follows: A hollow rod 3 (see Fig. 192), closed below, is rigidly connected with the pump piston. In the bore of this rod is a little piston 4. This piston may be moved back and forth longitudinally by a small handwheel 5 and thus uncover the openings 6. A small dashpot piston 7 is rigidly connected with the hollow rod and thus with the pump piston. Conversely, the case 8, surrounding the dashpot piston, and the cover 9, which together form the suction valve, can slide along the movable rod 3 to an extent corresponding to the permissible valve stroke S. Finally, openings 10 are provided in the piston 7, which communicate through the hollow rod 3 with the openings 6 and in this manner connect the space above the piston with the space below it. Now if the opening 6 is partially closed by the small piston 4 then the flow of the oil through it will be checked

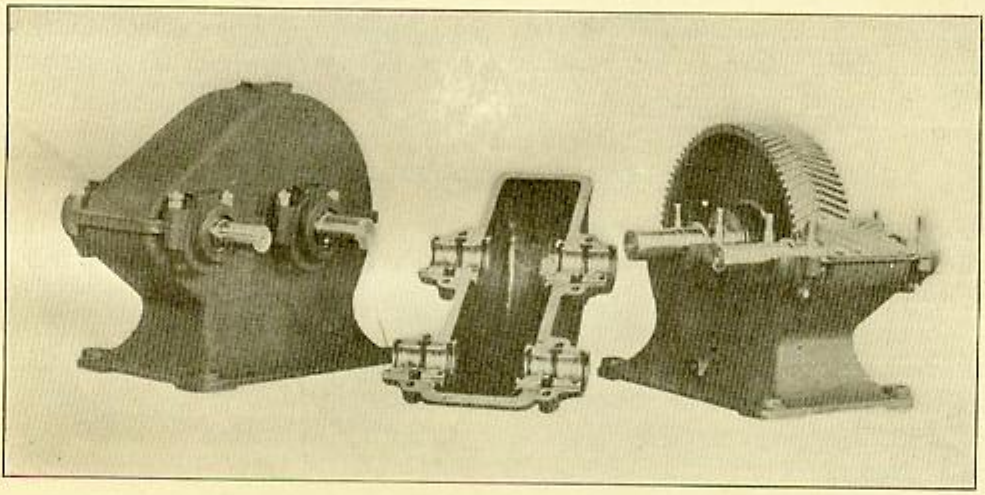


Fig. 193.—Motor Drive for High-pressure Oil Pumps, The Anglo-Newfoundland Development Co.

and the shell 8 can move but slowly in relation to the piston 7; i.e., the piston 7 and the pump piston 2 form a firm whole and the suction valve, which is identical with the shell 8, is seated slowly.

The action of the dashpot used in connection with the pressure valve is effected somewhat more simply. The valve clapper 11 is under the influence of the spring 12 placed above it, and has a piston 13 in its concave base which projects a small fraction of an inch into the casing 14 of the valve seat 15. Moreover, there are small openings 16 in the valve seat 15, by which we may regulate the velocity of the oil compressed during the downstroke of the valve by the piston 13 in the casing 14, and thus the valve clapper is gently seated.

The pumps normally operate at 178 R.P.M., and the working pressure is 150 pounds per square inch. The capacity of one of the three-cylinder pumps is 95 gallons per minute. But the speed can be easily raised to 200 R.P.M. and capacity thus increased to 111 gallons per minute.

The group of pumps is operated by an electric motor whose speed is reduced from 710 R.P.M. to 178 R.P.M. by a spur-wheel drive running in oil. This drive, which has herring-bone teeth, operates almost noiselessly on account of its construction. Each of the pumps is connected by a friction clutch with the shaft of the gear-drive. Thus each pump may be thrown either in or out of service at will while the plant is in operation (see Fig. 193).

The turbines and governors for the installation above described were delivered complete in the spring of 1909 and were placed in operation without delay in the fall of that year. According to the operating results the guarantees as to the performance of the turbines and the variations in speed due to changes of load are more than satisfied. The guaranteed efficiency of 82 per cent has not been questioned by the customer. Nevertheless, the contractor has recommended a test because from his experience he expects to obtain an efficiency of 86 per cent.

AN INSTALLATION OF IMPULSE WHEELS WITH A CAPACITY OF 1200 H.P.

Figs. 139 and 194 to 200 show a power development consisting of two sets of impulse wheels, built by Amme, Giesecke & Konegan Co., and installed in Spain. Each set of turbines is designed for the development of 600 H.P. under a head of 515 feet. In order to obtain a speed of $n=750$ R.P.M.—corresponding to a specific speed of 16.5 R.P.M. for a single wheel with one nozzle—it was necessary to build the impulse wheel as a double wheel, or one with two runners, and to drive each wheel by two nozzles. This arrangement has the further advantage of making it possible to shut down a given wheel or to close off a given nozzle while the plant is in operation. The wheels are gauranteed for an efficiency of 80 per cent and operate as efficiently at 300 H.P. as at 600 H.P. and indeed the efficiency holds to 100 H.P.

The following is to be noted in connection with the description of this power development:

The two nozzles which operate each wheel, as is apparent from Figs. 139 and 198, have a common penstock connecting to the header and in this is placed a gate valve. The branch to each nozzle also contains a gate valve so that each half of the wheel may be placed in operation or shut down by opening or closing the proper valve. The wheels shown in Fig. 200 are built with 18 double ellipsoidal buckets, the diameter on the jet circle being 2 feet $1\frac{1}{2}$ inches.

The nozzles are circular in cross-section and are so arranged that one jet impinges on the wheel horizontally and the other vertically downward. The diameter of each jet is $1\frac{31}{32}$ inches. The construction of the nozzle will appear from Figs. 194 and 195. The design, as shown, is protected by U.S. patent No. 877323, and to a certain extent is the inverse of a construction first developed in this country. The needle (1) is stationary, one end being a rod rigidly secured to (3) while the other end is a point of suitable form, the longitudinal section being so designed as to obtain a jet of circular cross-section with absolutely parallel paths for the several particles of water, both at full and partial openings of the nozzle.

The partial opening of the nozzle is effected as follows: The outer part of the nozzle (2) is a cylindrically shaped piece, usually of hard bronze, movable in the direction of its length. The longitudinal section of the interior of this piece near its outlet is so

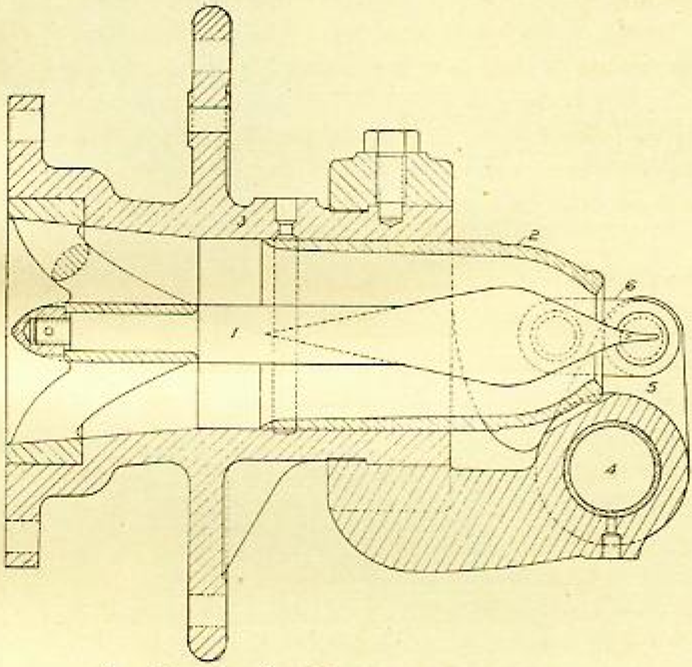


FIG. 194.—Details of Nozzle for Impulse Wheel.

designed as to fulfill the conditions governing the design of the needle. The outside of the nozzle (2) is accurately turned to fit the bored cylinder (3) in which it slides, the latter being bolted to the wheel housing. It is now possible to move the cylinder (3) in two ways: 1st, indirectly by the action of the governor shaft (4) to which is keyed a double crank, consisting of the levers (5) and the links (6) which are secured to the cylinder (2) by pins fitting into the reamed holes (7); 2d, directly as shown in Figs. 129 and 130, by which it will be seen that the moving, cylindrical nozzle (2) is furnished with partial piston (8) which is so built within the cylinder (3) as to form two chambers (9) and (10). Chamber (9) is always connected with the discharge while chamber ber (10) may be connected with the high-pressure water by means of a lever valve when it is desired to close the nozzle. The proper position of the nozzle is obtained in both cases as follows: On account of the pressure of the water in the nozzle the sliding portion (2) tends automatically to move forward and thus open the nozzle, while, on the contrary, the attendant may close the nozzle either by the governor or by the auxiliary piston. He is therefore able to obtain any desired intermediate position of the nozzle in respect to the needle.

When, as in the case under consideration, there are several nozzles to be governed

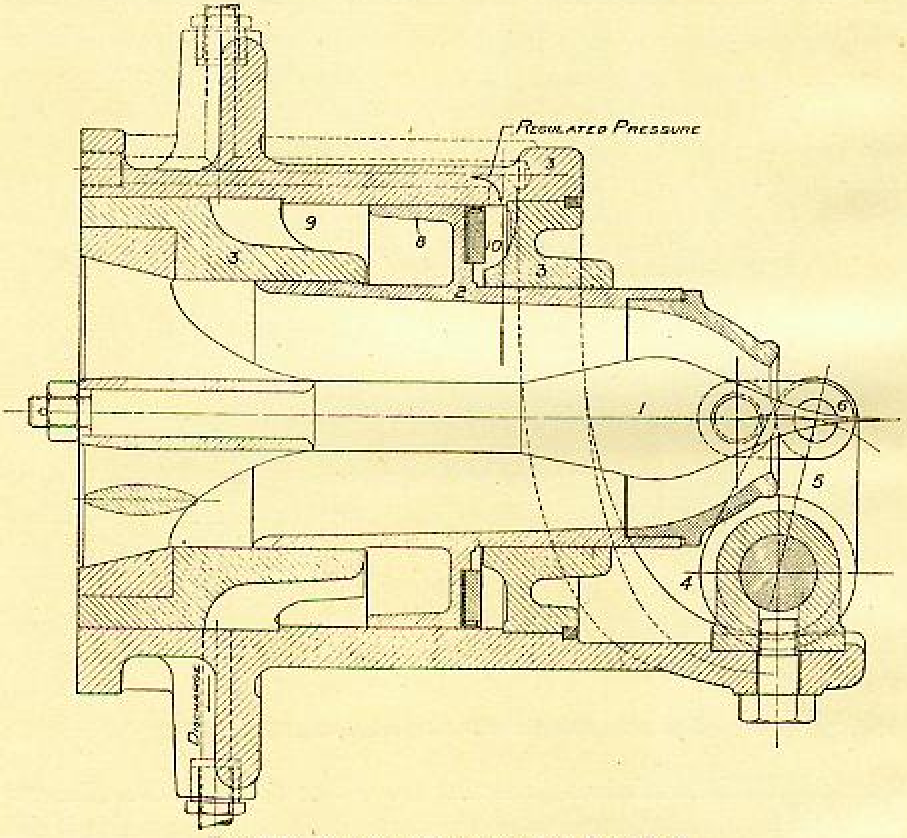


Fig. 195.—Details of Nozzle for Impulse Wheel.

simultaneously, then the indirect method offers a marked advantage, for several controlling shafts (4), see Fig. 124, may then be coupled together by rods (12) and projecting levers (11) and thus all of the nozzles may be regulated at the same time by a driving rod from a governor placed on one side of the unit. This may, for example, be an oil-pressure governor.

This method is employed in the plant we are describing and it is apparent that the regulation of the four nozzles used does not present mush greater difficulty than the regulation of one. A very clear idea of the arrangement may be had from Fig. 198, which is a photograph of the assembled unit.

The reader is referred to page 272, under the previous chapter, for a description

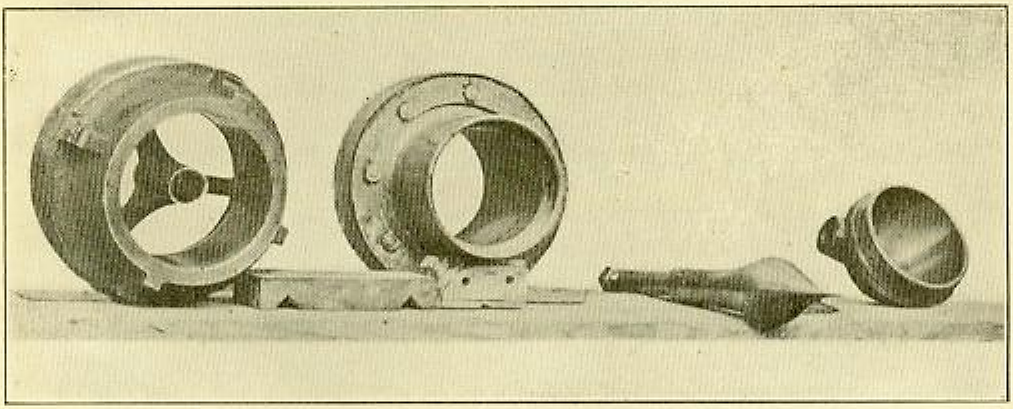


Fig. 196.—Photograph of Separate Parts of Nozzle and Needle for Impulse Wheel.

of the oil-pressure governors. The only difference between the governors used in the plant under consideration and those used in Newfoundland consists in the fact that here

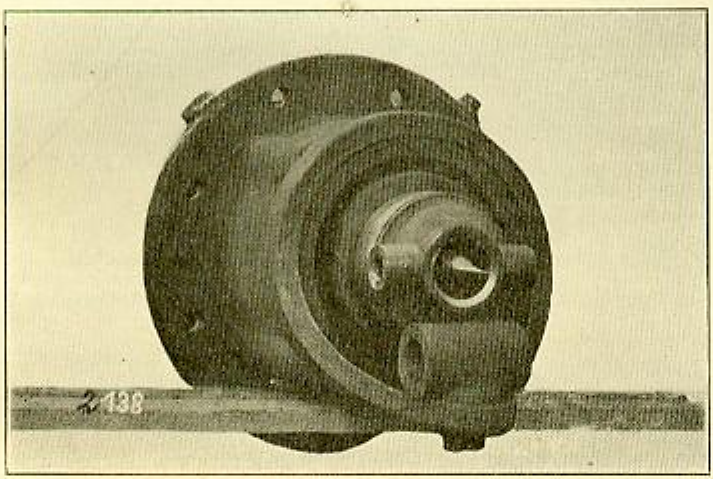


Fig. 197.—Photograph of Assembled Nozzle and Needle for Impulse Wheel.

the main operating valve has only one piston to close the nozzle instead of two, as is necessary for the double Francis turbines.

With the construction just described there is a tendency for the nozzle to open if the governor should fail and the nozzle be free to move. This condition can, however, easily be met. For example, chamber (10) see Fig. 195, may be kept under a constant pressure while chamber (9) receives a regulated pressure. Further, little difficulty is presented in so dimensioning the diameter of the auxiliary piston (8) that the movable nozzle will tend to approach the stationary needle even though opposed by the water pressure in the nozzle. However, the tendency for the nozzle to remain open is easily corrected through the oil-pressure governor and its connecting rod as shown on Fig. 198. The closing side of the main operating valve may be subjected to a constant pressure while it is opposed by a regulated pressure on the opening side.

A small geared rotary pump is provided to operate by hand in case no oil pressure

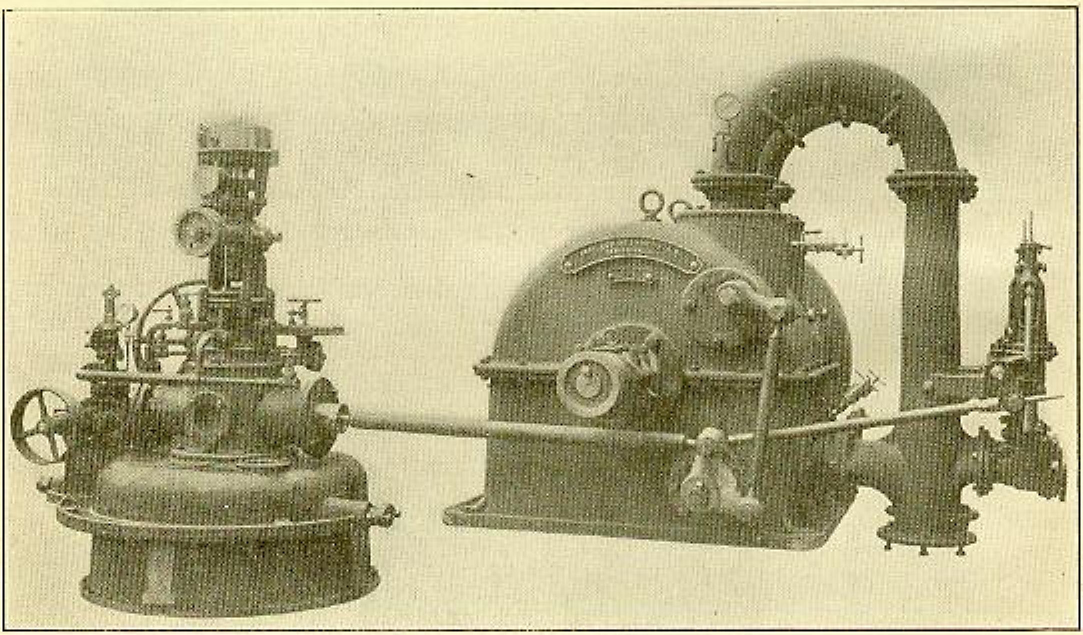


FIG. 198.—Photograph of 600 H.P. Double Impulse Wheel Assembled with its Governor.

is available. The shifting over from automatic regulation to hand regulation may be easily accomplished while the machine is in operation by closing and opening the necessary valves.

Attention may now be called to the fact that on account of the long penstock pressure regulators were provided for each turbine and because of the imposed condition that each half of the unit should be able to operate separately it was necessary to install two such regulators for each unit. Naturally, each pressure regulator is attached beyond the gate valve in the branch penstock, so that if one of the unit is out of service the corresponding pressure regulator will not be operative. The discharge from the pressure regulators takes place directly into the tail-water channel.

In this way there is no opportunity for the discharge with its high velocity and great energy to exert a destructive action on the turbine housing.

These pressure regulators operate on the same principle as those described on page 281, but the form of the construction is essentially different, especially because the natural water pressure is not used for governing and the operation is thus freed from difficulties due to impurities in the water. Instead, the same high-pressure oil is employed as is used for the governors regulating the speed. Consequently, the true regulating apparatus is completely separated from the adjacent part, which is on the discharge pipe, the latter, in this case, being constructed as a gate valve.

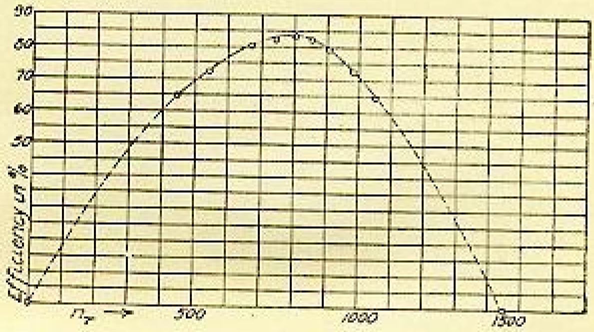


Fig. 199.—Efficiency Curve for Impulse Wheel.

Speed, turbine, r.p.m.	442	542	666	743	800	852	905	977	1050	1445
Speed, pump, r.p.m.	50.9	51.0	51.3	52.1	51.2	50.9	51.0	51.4	51.8	50.7
Volume of water, sec. ft.	0.7034	0.7044	0.7090	0.7203	0.7076	0.7034	0.7044	0.7104	0.7161	0.7002
Pressure head, ft.	246.7	246.0	246.7	245.7	246.3	246.0	246.3	246.7	246.3	246.0
Theoretical H.P.	19.72	19.69	19.88	20.11	19.80	19.67	19.72	19.92	20.04	19.57
Brake H.P.	12.80	14.31	15.98	16.59	16.51	16.21	15.70	14.54	13.03	0
Efficiency in per cent	64.9	72.7	80.4	82.5	83.4	82.4	79.6	73.0	65.0	0

Tested by Doctor Becker in the mechanical laboratory of the Royal Technical High School, Dresden, June 12, 1909.

Experience shows that in spite of this precaution an increase of pressure is not fully avoided, and neither is the decrease in pressure which takes place when the nozzles are suddenly opened. In order to completely obviate the detrimental action which would affect the uniform operation of the impulse wheel, a fly wheel is provided. This completely overcomes the faults of the pressure regulator.

The shaft of the impulse wheels is supported in oil-ring bearings, the one next to the fly wheel receiving additional support from an adjustable column.

The efficiency of such an impulse wheel is shown by the diagram in Fig. 199, which was plotted from tests on a wheel smaller than those used in the installation we are describing, but one otherwise practically similar. The specific speed of the wheel tested was 14.8 R.P.M. instead of 16.5 R.P.M. for the wheels above described, and the diameter of the jet was $1\frac{1}{4}$ inches. The quantity of water used was measured by a piston pump which furnished the water under pressure required for the operation of the impulse wheel. By a preliminary test run lasting half an hour this pump was calibrated by the

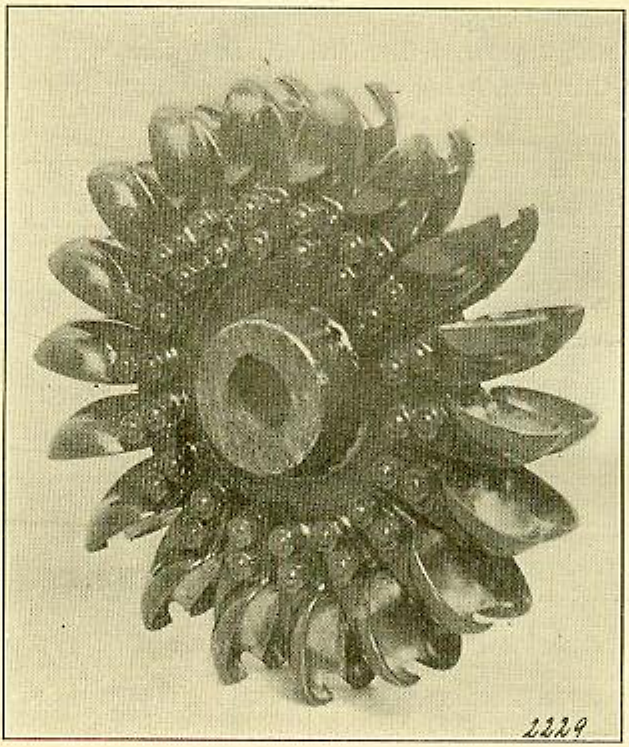


Fig. 200.—Photograph of Assembled Impulse Wheel.

use of a 110-lb. back pressure in connection with a gauged standpipe. The pressure head was read from a differential manometer whose results were further corrected by comparison with a mercury column. The number of revolutions of the impulse wheel and pump were determined by a direct-reading revolution counter. The slide on the wheel housing was open during the test so that there was atmospheric pressure within it.

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governors this peculiarity must be borne in mind—that with every automatic governor operating a ring gate equal motions of the piston rod in closing the gate do not correspond to equal reductions of the water discharge, or, in other words, that unequal movements of the piston are required at different stages of the closing motion to produce the same amount of reduction of the discharge.

Referring to condition (4) it will be seen that a ring gate is easily operated in any case, but the power required may be still further reduced by the use of counterweights

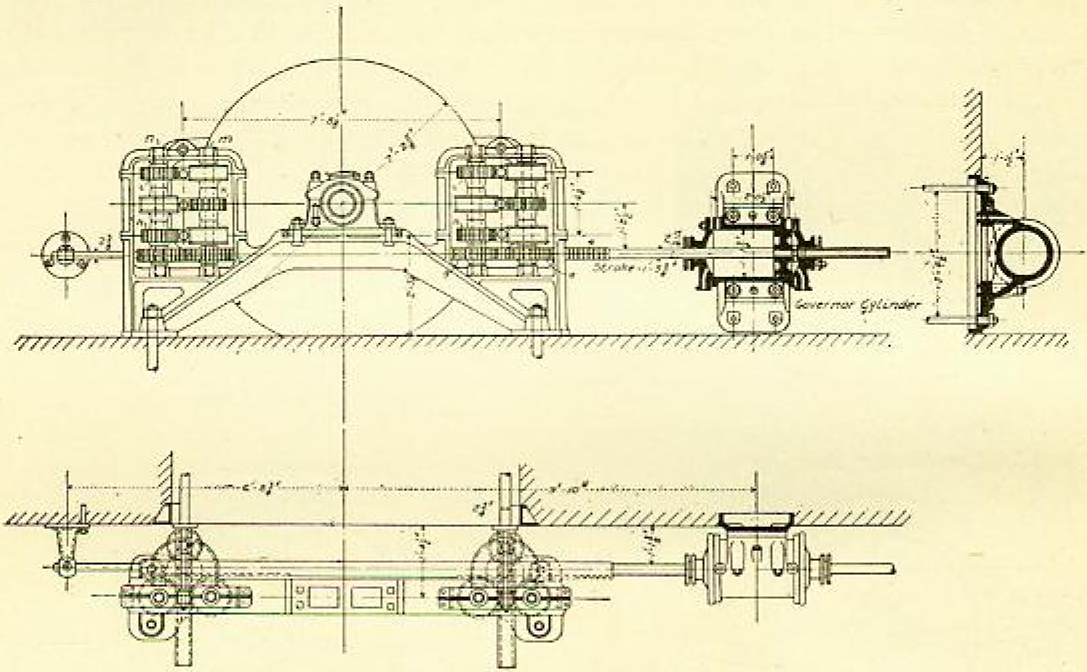


FIG. 72*b*.—Details of Regulating Device for Turbines shown in Fig. 72.

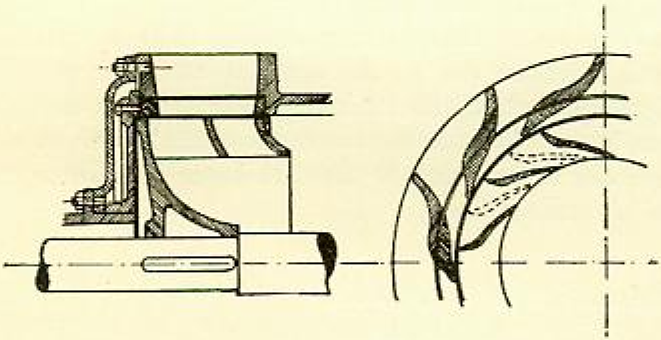
as in Fig. 56, or hydraulically by constructing the upper part of the cylinder gate in the form of a piston as shown in Fig. 71. A further improvement in the guiding of the ring consists in attaching to the gate either two or four stiff rods which move in guiding bushings.

Turbines with vertical shafts are better adapted for regulation by circular gates than turbines with horizontal shafts. The design of the apparatus necessary to operate the circular gates of a turbine installation consisting of more than one wheel is rather complicated. An arrangement similar to that shown in general in Fig. 46, and in detail in Figs. 72, 72*a* and 72*b*, was first adopted at the plant of the Ottawa & Hull Power Company at Hull, Canada. The three operating rods are attached symmetrically to the opposite sides of the two or more turbines. The gates of the turbines having even numbers are operated by the upper and lower rods, while the middle rod moves the gates of the turbines having odd numbers. To accomplish this result the middle rod is rigidly attached to the ring gate by the arm d. The motion of the upper and lower rods, which must be in the opposite direction to that of the middle rod, is transmitted by the clamps a to the tubes b, which are firmly connected to the ring valves of the even-numbered turbines by means of the arm c. It is evident that the middle rod must pass freely through the tube b, but without play. Well-designed and accessible bearings and connections for each valve rod constitute an essential feature of this form of construction. The valve rods are operated by an hydraulic piston as shown in Figs. 72, 72a and 72b. The piston rod is attached to the rack e, which is guided by the rollers f and engages with the gear wheels g, keyed to the shafts m and n. The shafts, in turn, operate the valve rods by the gears h, i and k, engaging with racks on the valve rods. The loose rollers on the shafts m and n guide the rods and prevent their buckling.

For durability, convenience in erection, ease of inspection, and resistance to injury, the cylinder gate is not surpassed by any other method of regulation.

Regulation by Means of a Revolving Ring Valve between the Distributor and Runner (Register Gate)

This method of regulation, originated by the Platt Iron Works, is shown in Figs. 73 and 74. Similar to it is the patented Zodel construction, which, however,



FIGS. 73 and 74.—Arrangement showing Regulation by Means of Register Gate.

uses in addition elastic steel blades which are secured to the back of the stationary distributor vanes and project freely between the vanes of the ring valve. This form of construction has been used in a number of large turbines. In reference to this method of regulation it may be said that conditions (1), (3), (4) and (5) are properly met if the best construction is employed, but regarding condition (2) it will be noted that on account of the necessity of having the distributor vanes so thick that they

From the above table showing the characteristics of the several normal types, we have prepared the following table, indicating the number of revolutions per minute for wheels having various diameters:

TABLE SHOWING NUMBER OF REVOLUTIONS PER MINUTE FOR WHEELS OF NORMAL TYPE, THE HEAD BEING ASSUMED AS 1 FOOT

Diameter in Inches.	VIII	VII	VI	V	IV	III	II	I
10	92.3	95.6	100.2	106.7	115.8	128.6	145.2	165.5
12	76.9	79.7	83.5	88.9	96.5	107.2	121.0	137.9
14	65.9	68.3	71.6	76.2	82.7	91.7	103.7	118.2
16	57.7	59.7	62.6	66.7	72.4	80.4	90.7	103.4
18	51.3	53.1	55.7	59.3	64.3	71.5	80.7	91.9
20	46.1	47.8	50.1	53.3	57.9	64.3	72.6	82.7
24	38.4	39.8	41.7	44.4	48.2	53.6	60.5	68.9
28	32.9	34.1	35.8	38.1	41.3	45.9	51.9	59.1
32	28.8	29.9	31.3	33.3	36.2	40.2	45.4	51.7
36	25.6	26.6	27.8	29.6	32.2	35.7	40.3	46.0
40	23.1	23.9	25.0	26.7	28.9	32.2	36.3	41.4
48	19.2	19.9	20.9	22.2	24.1	26.8	30.2	34.5
56	16.5	17.1	17.9	19.0	20.7	23.0	25.9	29.5
63	14.6	15.2	15.9	16.9	18.4	20.4	23.0	26.3
72	12.8	13.3	13.9	14.8	16.1	17.9	20.1	23.0
80	11.5	12.0	12.5	13.3	14.4	16.1	18.1	20.7
96	9.6	10.0	10.4	11.1	12.1	13.4	15.1	17.2
120	7.7	8.0	8.3	8.9	9.6	10.7	12.1	13.8
138	6.7	6.9	7.3	7.7	8.4	9.3	10.5	12.0
156	5.9	6.1	6.4	6.8	7.4	8.2	9.3	10.6

It is usually the case that in addition to the quantity of water Q and the head H, the number of revolutions per minute η is also specified. We must then determine the values of η_i and Q_i and select from the above tables showing the quantity of water and the speed, that diameter which at the same time satisfies both conditions. We will thus learn whether the required quantity of water can be used by a single wheel, or whether it is necessary to use two or more normal wheels fastened to one shaft to obtain the necessary power. This will determine the number of wheels to be used on one unit.

Characteristics of High-Speed Turbines. In order to characterize, or to differentiate between, the degrees of speed of high-speed turbines it is the custom in practice to use certain characteristic coefficients, and these are obtained as follows:

The horse-power P delivered to the turbine shaft is

$$P = 0.114\ QH\varepsilon,$$

in which Q is expressed in cubic feet per second, H in feet, and ε is the net hydraulic efficiency of the turbine.

PART III

NOTABLE TURBINE INSTALLATIONS

PLANT OF CANADIAN NIAGARA POWER COMPANY AT NIAGARA FALLS, ONTARIO

The utilization of the water power at Niagara Falls has attracted world-wide attention, not only because of the immense potential value contained in the waters of the Niagara River, descending 328 feet between Lake Erie and Lake Ontario, but also because of the number and variety of the plants by which such utilization has been effected. While a small amount of hydraulic power had been developed as early as 1725, when a sawmill was operated on the rapids immediately above the falls, the first development of importance was that of the Niagara Falls Hydraulic Power & Manufacturing Company. Although the operations of that company began in 1861, the development was exclusively hydraulic until the year 1895. The pioneer plant in hydro-electric development at Niagara Falls was that of The Niagara Falls Power Company, commenced in 1890, a time when no alternating-current generators were in successful operation for commercial purposes and when the success of electric transmission of any kind was as yet problematical. The plant of this company has been fully and repeatedly described in the technical press, notably, in the Niagara number of *Cassier's Magazine*, and it is therefore the purpose of this chapter to describe the plant of the Canadian Niagara Power Company, which, though organized under the laws of the Province of Ontario, is closely allied financially with The Niagara Falls Power Company and, being built after the experience gained in the construction and operation of the plants of that company, embodies the best principles of hydro-electric development on a large scale under the conditions affecting power development in the vicinity of Niagara Falls.

In order to understand certain features of the development, it is necessary to remember that all of the land in the vicinity of the Canadian, or Horseshoe, Falls is owned by the Province of Ontario and is used for park purposes. In making a development in such a park æsthetic as well as engineering conditions have an important bearing upon the character of the development. The general plan adopted is similar to that of The Niagara Falls Power Company, the plant consisting of a short headrace canal connecting with the rapids about 1600 feet above the crest of the Horseshoe Falls, a long narrow excavation in which is placed the turbine machinery, and a tunnel carrying the discharged water from the wheels to a point in the river below the falls.

ϕ increases from .7 to .8 the power of the wheel decreases from 10,600 H.P. to 10,250 H.P., or about 3.3 per cent. By deducting 3.3 per cent from the power shown on curve No. 1, Fig. 153, at 50 feet head, we obtain a point on curve No. 2 which point indicates the power actually obtainable under 50 feet head at a constant speed of 107 rev. per min. at full gate opening. Other points on the curve are similarly obtained and the curve drawn through them. As a check on curve No. 1, "sheet B," curves 5, 6, 7, and 8 have been plotted from the results of actual tests from other wheels in service. Wheels at Shawinigan and Niagara Falls are Francis turbines designed with ϕ equal .7, while the wheel at Trenton Falls is a Fourneyron turbine with ϕ equal .5.

The curves on Fig. 155 were prepared to show the efficiency which might be expected from the designed wheel under various heads and gate openings at a constant speed of 107 rev. per min. The fundamental curve is No. 1, which shows the efficiencies obtained by careful tests on a 3500 H.P. wheel working under 210 feet head at 250 rev. per min. at the plant of the Niagara Falls Hydraulic Power & Manufacturing Company. This wheel was used at a basis of efficiencies for the reason that although it operated under a far higher head than the proposed wheel at McCall Ferry would utilize, yet it was the same general type as the wheel under design. It will be seen that the form of the curve is a very satisfactory one for usual conditions, as the maximum efficiency is reached at $\frac{7}{8}$ gate opening and the efficiencies hold up well for partial loads, the usual conditions in a large plant subject to fluctuating demands.

In plotting curve No. 1 in Fig. 155 it was assumed that the efficiency of the wheel under design would be the same at 13,500 H.P. as the tested wheel was at 3500 H.P., and the efficiencies at partial loads would be proportional, this curve showing the conditions when the wheel was working under a head of 65 feet at 107 rev. per min. with ϕ equal .7. In order to obtain curve No. 2, Fig. 155, showing the efficiencies under 50 feet head at 94 rev. per min. the following method was employed: By curve No. 1 in Fig. 153 it is seen that the maximum power of the wheel under 50 feet head at 94 rev. per min. is 9150 H.P. Scale B in Fig. 155 is plotted by making 15,000 H.P. on same coincide with 15,000 H.P. on scale A, decreasing upward at the same rate as scale A decreases toward the left. By connecting point 9150 H.P. on scale B with zero lines on both scales, we have curve No. 10. To obtain a given point, as X, on curve No. 3, we find the intersection of curve No. 10 with the 8000 H.P. line on scale B and note that it corresponds with 11,800 H.P. on scale A. The efficiency at 8,000 H.P. under 50 feet head at 94 rev. per min. is therefore the same as that at 11,800 H.P. under 65 feet head at 107 rev. per min., and we project the point on curve No. 1 corresponding to 11,800 H.P., until it intersects the line from 8000 H.P. on scale A, thus fixing point X. Other points on curve No. 3 are similarly obtained.

But as the wheel is to run at 107 rev. per min. it is necessary to correct curve No. 3 for the loss in power and efficiency due to running at the higher speed, and curve No. 2 is plotted as follows to show such conditions: By referring to curve No. 1, Fig.

The original of these diagrams was prepared by Mr. F. C. Finkle, consulting engineer, who should also be credited with the design of the other features of the Kern River Plant No. 1.

The regulation is effected by the deflection of the jet or an alteration in the position of the needle sliding in the nozzle, the former being normally automatic while the latter is done by hand, the needle being set in proper position for the peak load likely to occur until the time of the next setting. One of the best features in the design is the use of a bifurcated nozzle casting which possesses the following advantages over the older form: (1) Any reaction from the jet upon the needle or nozzle is concentric, for the center of the jet corresponds to the center of the inlet; (2) the needle is not subject to vibration; (3) the jet is smooth, as the water leaves the nozzle concentrically; (4) the needle is not subject to bending. The needle employed is 8 feet long and uniformly 12 inches in diameter except at the tip and stem, being hydraulically balanced by the full pressure acting on the rear. The deflected jet impinges almost tangentially upon a steel plate by which it is deflected nearly to the vertical and thence by a second plate in the bottom of the wheel chamber so that it enters the forebay horizontally. The minimum amount of erosion may thus be anticipated.

As the Kern River Plant No. 1 was to act as the regulator for the entire Los Angeles Edison system it was necessary that the governing apparatus should be efficient and that the entire rotor should be designed with regard to close regulation. Escher-Wyss oil-type governors operating under 125-pound pressure were adopted for the movement of the nozzle, the time for deflection to be one second. It was desired to obtain a speed regulation of 5 per cent when full nominal load was instantaneously thrown off with a corresponding regulation with the addition or deduction of partial loads. Such regulations would be equivalent to a change in speed of 6.36 per cent when 25 per cent overload (8950 H.P.) was thrown off or 7.64 per cent when 50 per cent overload (10,750 H.P.) was lost. To determine the design of the rotor to effect this result the following formula was employed:

$$L=800,000\frac{NT}{r^2\Sigma WR^2}.$$

When L = per cent of change in speed.
N = brake H.P. thrown off.
T = time in seconds for nozzle deflection = 1.
r = revolutions per minute.
W = weight in pounds of revolving parts of unit.
R = radius of gyration of revolving parts.

It was necessary that ΣR^2 should be 1,800,000 lb.ft.^{2}, and this was obtained by increasing the weight of the generator field spider.

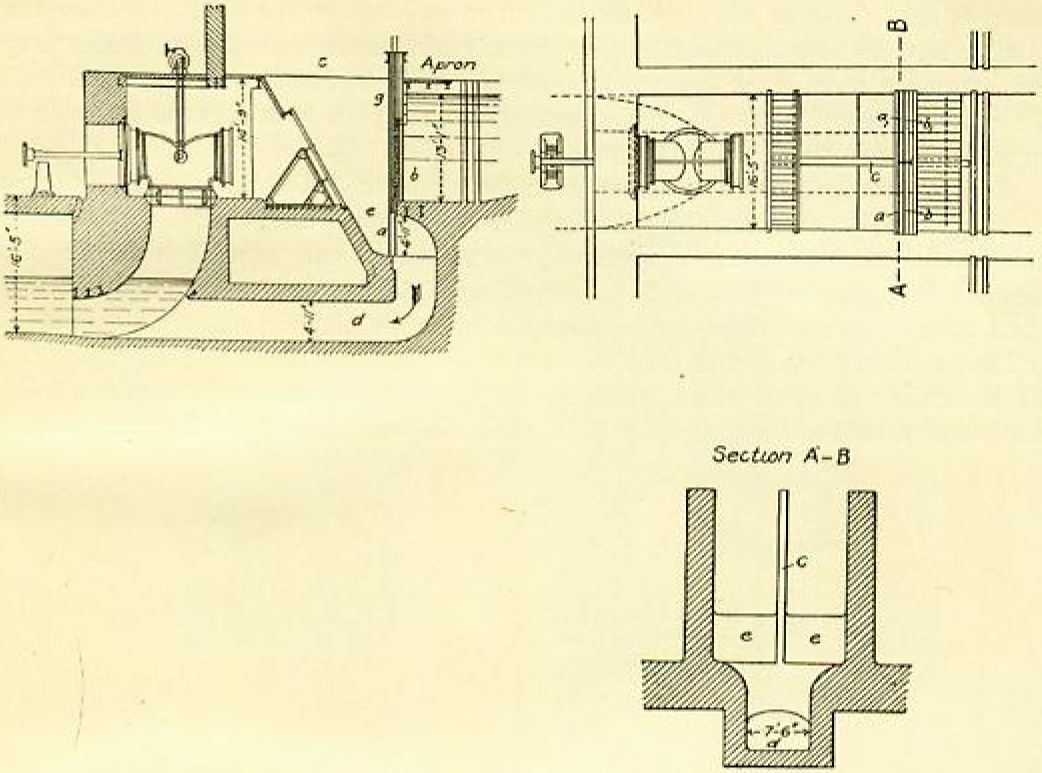


FIG. 14.—Arrangement for Cleaning Racks by Reversing Flow of Water.

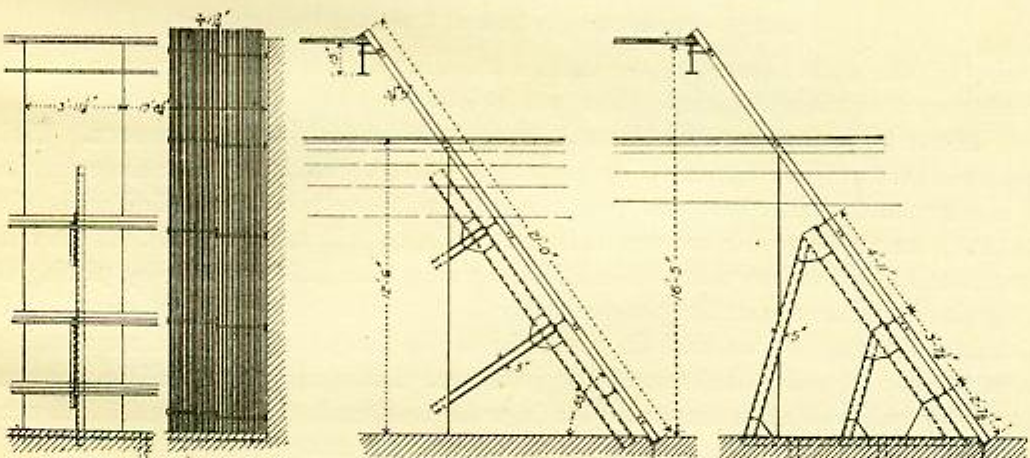


FIG. 15.—Inclined Racks in Forebay.

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