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Cyclopedia

of

Civil Engineering

A General Reference Work

ON SURVEYING, RAILROAD ENGINEERING, STRUCTURAL ENGINEERING, ROOFS AND BRIDGES, MASONRY AND REINFORCED CONCRETE, HIGHWAY CONSTRUCTION, HYDRAULIC ENGINEERING, IRRIGATION, RIVER AND HARBOR IMPROVEMENT, MUNICIPAL ENGINEERING, COST ANALYSIS, ETC.

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THE editors have freely consulted the standard technical literature of America and Europe in the preparation of these volumes. They desire to express their indebtedness, particularly, to the following eminent authorities, whose well-known treatises should be in the library of everyone interested in Civil Engineering.

Grateful acknowledgment is here made also for the invaluable co-operation of the foremost Civil, Structural, Railroad, Hydraulic, and Sanitary Engineers in making these volumes thoroughly representative of the very best and latest practice in every branch of the broad field of Civil Engineering; also for the valuable drawings and data, illustrations, suggestions, criticisms, and other courtesies.

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LOWER LOCK AND POWER HOUSE, ST. MARY'S FALLS CANAL, MICHIGAN

Overcoming rapids in the St. Mary's River between Lake Superior and Lake Huron, Upper Peninsula of Michigan.

Foreword

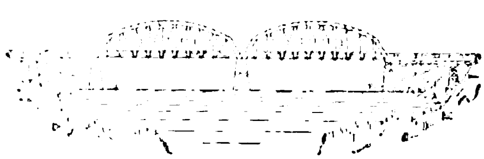
THE marvelous developments of the present day in the field of Civil Engineering, as seen in the extension of railroad lines, the improvement of highways and waterways, the increasing application of steel and reinforced concrete to construction work, the development of water power and irrigation projects, etc., have created a distinct necessity for an authoritative work of general reference embodying the results and methods of the latest engineering achievement. The Cyclopedia of Civil Engineering is designed to fill this acknowledged need.

¶ The aim of the publishers has been to create a work which, while adequate to meet all demands of the technically trained expert, will appeal equally to the self-taught practical man, who, as a result of the unavoidable conditions of his environment, may be denied the advantages of training at a resident technical school. The Cyclopedia covers not only the fundamentals that underlie all civil engineering, but their application to all types of engineering problems; and, by placing the reader in direct contact with the experience of teachers fresh from practical work, furnishes him that adjustment to advanced modern needs and conditions which is a necessity even to the technical graduate.

¶ The Cyclopedia of Civil Engineering is a compilation of representative Instruction Books of the American School of Correspondence, and is based upon the method which this school has developed and effectively used for many years in teaching the principles and practice of engineering in its different branches. The success attained by this institution as a factor in the machinery of modern technical education is in itself the best possible guarantee for the present work.

¶ Therefore, while these volumes are a marked innovation in technical literature — representing, as they do, the best ideas and methods of a large number of *different* authors, each an acknowledged authority in his work — they are by no means an experiment, but are in fact based on what long experience has demonstrated to be the best method yet devised for the education of the busy workingman. They have been prepared only after the most careful study of modern needs as developed under conditions of actual practice at engineering headquarters and in the field.

¶ Grateful acknowledgment is due the corps of authors and collaborators — engineers of wide practical experience, and teachers of well-recognized ability — without whose co-operation this work would have been impossible.



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*For page numbers, see foot of pages.

†For professional standing of authors, see list of Authors and Collaborators at front of volume.

HEADWORKS OF THE ONTARIO POWER COMPANY, NIAGARA FALLS, ONTARIO

View showing outer and inner forebays, with screen house between. Gate house at farther end of inner forebay.

HYDRAULICS.

1. Hydraulics is that branch of Mechanics which treats of the laws governing the pressure and motion of water. *Hydrostatics* is that particular branch of hydraulics which treats of water at rest, and *hydrodynamics* is that branch which treats of water in motion.

2. Units of Measure. The unit of length most frequently used in hydraulics is the foot. The unit of volume is the cubic foot or the United States gallon. The unit of time usually employed in hydraulic formulas is the second, but in many water-supply problems the minute, the hour, and the day are also often used. The unit of weight is the pound, and that of energy the foot-pound.

1 U. S. gallon = 231 cubic inches = 0.1337 cubic foot;

1 cubic foot = 7.481 U. S. gallons;

1.2 U. S. gallons = 1 Imperial gallon.

3. Weight of Water. The weight of distilled water at different temperatures is given in Table No. 1.

The weight of ordinary water is greater than that of distilled water on account of the impurities contained. For ordinary purposes the weight of a cubic foot of fresh water may be taken equal to 62.5 pounds. Sea water will weigh about 64 pounds per cubic foot.

TABLE NO. 1.

Weight of Distilled Water.

Temperature, Fahrenheit.	Weight. Pounds per Cubic Foot.	Temperature, Fahrenheit.	Weight, Pounds per Cubic Foot.
32°	62.42	140°	61.39
39.3	62.424	160	61.01
60	62.37	180	60.59
80	62.22	200	60.14
100	62.00	212	59.84
120	61.72		

As will be seen from this table, water is heaviest at a temperature of about 39.3° F., or as is commonly stated, about 40° F.

4. Atmospheric Pressure. As has already been explained in the papers on Elementary Mechanics, the atmosphere everywhere exerts a pressure upon all objects uniform in every direction; and is itself compressed to the same degree. At sea level the average pressure of the atmosphere is sufficient to balance a column of mercury in a closed tube (a barometer) about 30 inches high, which is equivalent to a pressure of 14.7 pounds per square inch. A corresponding water barometer would be 34 feet high, the weight of water being much less than that of mercury. At points higher than sea level the air pressure is less, and hence the height to which a mercury or water barometer will be raised will be less. Since we depend upon air pressure to raise water into "suction" pipes it is important to know how much this pressure is when designing such pipes.

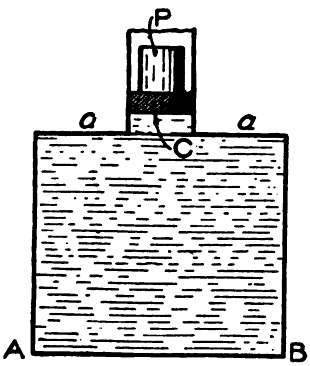


Fig. 1.

The following table gives, for different elevations above sea level, the pressure of the atmosphere, expressed, first, in pounds per square inch, second, in the height of the mercury barometer and, third, in the height of the water barometer:

TABLE NO. 2.

Atmospheric Pressure at Different Elevations.

Elevation above Sea Level. Feet.	Pressure in Pounds per Square Inch.	Height of Mercury Barometer. Inches.	Height of Water Barometer. Feet.
0	14.7	30.00	34.0
500	14.5	29.47	33.3
1,000	14.2	28.94	32.8
2,000	13.7	27.92	31.6
4,000	12.7	25.98	29.4
6,000	11.8	24.18	27.4
8,000	11.0	22.50	25.5
10,000	10.3	20.93	23.7

PRESSURE OF WATER AT REST.

5. Transmission of Pressure. If AB, Fig. 1, be a tight vessel containing water, and a close fitting piston C be heavily loaded with a weight P the entire body of water will be subjected to a pressure corresponding to the weight P. The water will not be compressed into a smaller space as would a gas like air, because water is almost incompressible, but whatever pressure is exerted by the weight P will be transmitted through the water equally in all directions so that the pressure of the water against the walls of the vessel will be the same per square inch as that of the weight P upon the water (neglecting the small effect of the weight of the

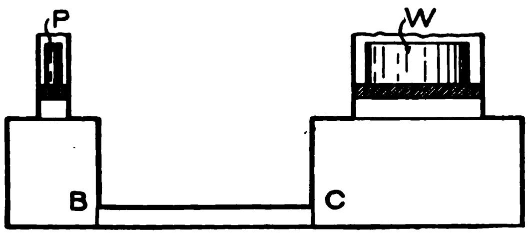


Fig. 2.

water in the vessel). Thus if the area of the piston = 10 square inches and the weight P = 1,000 pounds, the pressure per square inch will be 100 pounds, and this will be the pressure in every part of the liquid and upon the walls of the vessel. Furthermore, the pressure of the water upon the walls of the vessel is perpendicular to the surface at all points. The pressure at *a* is upwards, on the bottom of the vessel it is downwards and on the sides it is horizontal.

As a further illustration of the foregoing principle, let B and C, Fig. 2, be two vessels connected by a pipe, and let P be a loaded piston exerting a heavy pressure in the small vessel B. In accordance with the principle above stated, this pressure will be transmitted equally to the larger vessel where the water will exert the same pressure per square inch upon the vessel and upon any piston W which may be inserted in any opening of the vessel C. By making the area of P small and of W large a small load P will balance a large load W.

If a is the area of the piston P and A that of the piston W, then the pressure per square inch produced by the weight P will be $\frac{P}{a}$. This will also be the pressure per square inch on W, and

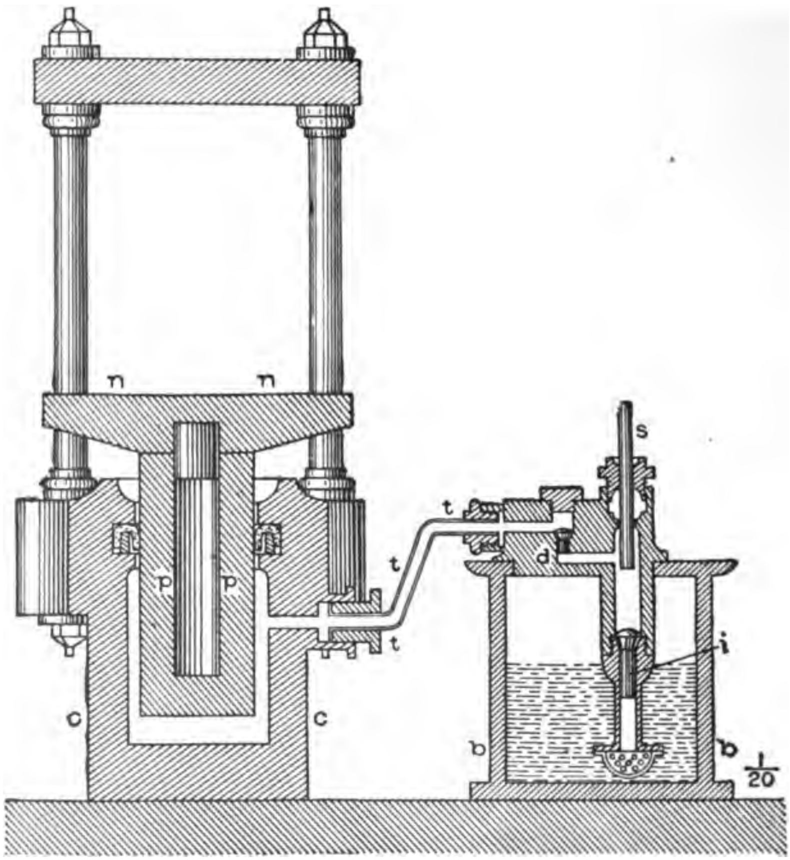


Fig. 3.

hence the weight W which will be sustained will be equal to the area A multiplied by the pressure per square inch, $\frac{P}{a}$, or

$$W = A\frac{P}{a}. \qquad (1)$$

The principle above stated is utilized in the hydraulic press shown in Fig. 3. In this apparatus a pump on the right with small plunger feeds a large plunger p underneath the movable plate of the press on the left. The pump plunger corresponds to the piston P in Fig. 2, and the press to the piston W. By making the pump very small and the plunger under the press very large, enormous pressures can be exerted even by means of a hand pump. The pressure produced is given by formula (1) above. It is to be noted that the pressure per square inch on the interior of the apparatus, the pump, piping and press, is the same at all points.

Examples. 1. If the area of the pump plunger be 2 sq. in. and that of the press 1 sq. ft., what pressure will be exerted by the press when the load on the pump is 100 lb.?

Using equation 1 we have $a = 2$ sq. in., $A = 144$ sq. in., and

$$P = 100 \text{ lb., whence } W = 144 \times \frac{100}{2} = 7{,}200 \text{ lb. Ans.}$$

2. If a pressure of 10 tons be desired and the area of the press plunger be 200 sq. in., and the available pressure on the pump plunger be 150 lb., what area must be given to the pump plunger?

Here $W = 10 \times 2{,}000 = 20{,}000$ lb., $A = 200$ and $P = 150$. Using equation 1 and letting x = desired area, we have $20{,}000 = \frac{200 \times 150}{x}$. Solving for x we have $x = \frac{200 \times 150}{20{,}000}$

$= 1.5$ sq. in. Ans.

6. Pressure Due to the Weight of Water. Let Fig. 4 represent a vessel of water. Consider a vertical column of the water of height h and a cross-section of one square foot. Its volume will be h cubic feet and it will weigh $62.5 \times h$ pounds. As it is supported entirely by the water underneath, it therefore exerts a pressure upon that water of $62.5 \times h$ pounds. Likewise the pressure at any other point in the vessel at a distance h below the surface is $62.5 \times h$ pounds per square foot. Furthermore, since the water exerts equal pressures in all directions it follows that the pressure against the sides of the vessel at this depth, or against any object immersed in the water, will also be $62.5 \times h$ pounds per square foot.

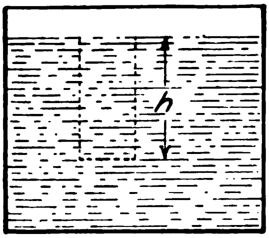


Fig. 4.

Since the weight of water is so nearly constant we may conveniently use the depth h as a measure of the pressure. When so used it is called the *pressure head* or simply the "head" acting on the given surface. For each foot of head the pressure will be 62.5 pounds per square foot, but in expressing pressure in pounds it is customary to use the square inch. A pressure of 62.5 pounds per square foot being equal to $\frac{62.5}{144}$ or .434 pounds per square inch, it follows that one foot of *head* gives a pressure of .434 pounds per square inch. Conversely, a pressure of one pound per square inch requires a head of $\frac{1}{.434}$ or 2.304 feet.

Rule. *To convert feet of head to pounds pressure multiply by* .434. *To convert pounds pressure to feet of head multiply by* 2.304. **(2)**

Examples. 1. What will be the pressure per square inch in the vessel of Fig. 5 at a point a 10 feet below the water surface? Assume the vessel to be round with a diameter of bottom = 6 feet and of upper part = 2 feet.

Here the head is 10 feet, and by the above rule the pressure per square inch = $10 \times .434 = 4.34$ pounds. It acts equally in all directions and is independent of the shape of the vessel.

2. What will be the total pressure on the bottom of the vessel?

The area of the bottom in sq. ft. $= \frac{3.14 \times 6^2}{4} = 28.26$ sq. ft.

The head is 14 feet and hence the pressure per sq. ft. $= 14 \times 62.5 = 875$ pounds. The total pressure on the bottom $= 875 \times 28.26 = 24,728$ pounds.

3. What will be the total upward pressure on the portion AB?

The area of this portion is the difference between the two circles respectively 6 feet and 2 feet in diameter. This is equal to $\frac{(6^2 - 2^2) \times 3.14}{4} = 25.12$ sq. ft. The head is 8 feet and the pressure, therefore, $8 \times 62.5 = 500$ pounds per sq. ft. Total upward pressure $= 500 \times 25.12 = 12,560$ pounds.

4. What is the entire weight of water in the vessel?

The volume of the lower part of the vessel $= \dfrac{6 \times 6^2 \times 3.14}{4}$ 169.56 cu. ft., and of the upper part $= \dfrac{8 \times 2^2 \times 3.14}{4} = 25.12$ cu. feet. Total volume $= 194.68$ cu. ft., and weight of water $= 194.68 \times 62.5 = 12{,}167$ pounds.

Note that the difference between the downward pressure on the bottom and the upward pressure on AB = 12,168 lbs., which is equal to the total weight of the water, or the net pressure of the vessel upon its support, if we neglect the weight of the vessel itself.

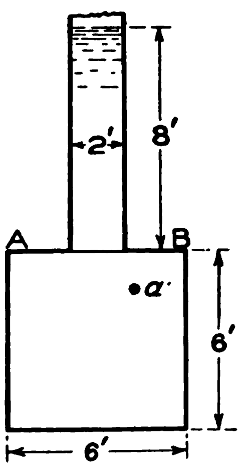


Fig. 5.

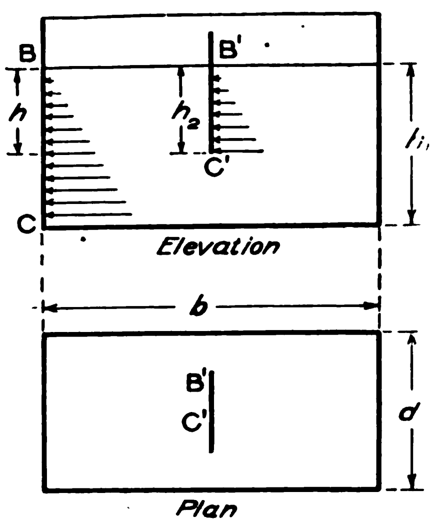


Fig. 6.

7. Pressure of Water upon Plane Areas in General. In the preceding articles it has been shown that the pressure per square inch upon any submerged body is equal to .434 h where h is the head in feet; furthermore, that this pressure is at right angles to the surface of the body. Let Fig. 6 represent a vessel of rectangular shape containing water of a depth h_1. The pressure on the bottom is then $h_1 \times .434$ pounds per square inch. If the area of the bottom be A $(= bd)$, then the total pressure on the bottom is $A \times h_1 \times .434$ pounds. In this case the pressure is the same per square inch at all points of the surface considered.

Consider now the pressure on one of the sides, as BC. In this case the pressure per square inch is not uniform, varying from nothing at B to a maximum at C where it is equal to $h_1 \times$ .434 pounds per square inch, the same as on the bottom. At any depth h the pressure is $h \times .434$ pounds per square inch. This variation in pressure is represented in Fig. 6 by the variation in length of the arrows acting against BC. From an inspection of the figure it is evident that the average length of these arrows is equal to one-half the length of the one at the bottom, or in other words, the *average* pressure per square inch against BC is equal to one-half the maximum, or $\frac{1}{2}\ h_1 \times .434$, which is the same as the pressure at the center of BC. The *total* pressure on the entire surface is then equal to this average pressure multiplied by the total area, or equal to $\frac{1}{2}\ h_1 \times .434 \times h_1\ d$.

If the area in question be a plate B′C′ immersed in the water to a depth h_2 the result is the same, except in this case there is an equal pressure on each side. As before, the pressure on either side of the plate is equal to $\frac{1}{2}\ h_2 \times .434 \times$ (area of submerged portion of plate).

If the plate be wholly submerged, as BC, Fig. 7, the pressure per square inch at B will be $h_1 \times .434$, and that at C will be $h_2 \times .434$, and the variation in pressure will be represented by a trapezoid of arrows instead of a triangle. The average pressure will now be $\frac{h_1 + h_2}{2}$ which is again the same as the pressure at the center of BC. The total pressure will be this average pressure multiplied by the area of the plate.

In all the above cases it will be seen that the average pressure found is the same as the pressure at the center of the plate. In a similar way it can be shown that for plates of *any shape* the average pressure is equal to the pressure at the *center of gravity* of the area, hence the following:

Rule. *The total pressure on a submerged vertical plane surface is equal to the pressure per unit area at its center of gravity multiplied by its area.* **(3)**

Suppose now the plate BC, Fig. 8, be an inclined plate immersed in water. From the principles already explained the pressure per square inch will be the same at any given depth as if the plate were vertical. Hence at B the pressure is $h_1 \times .434$ and that at C is $h_2 \times .434$. The average pressure is again $\frac{h_1 + h_2}{2} \times .434$, or the pressure at its center, and the total is equal to this pressure multiplied by the area of the plate. Whence the more general rule,—

Rule. *The total pressure on any submerged plane surface is equal to the pressure per unit area at its center of gravity multiplied by its area. Such pressure always acts at right angles to the surface.* (4)

8. Pressure in a Given Direction. In the above discussion we have considered only the total pressure of the water, which always acts perpendicular to the surface of the body. In Fig. 9 let P represent this total pressure on the surface BC, which has a

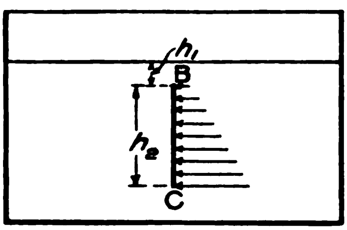


Fig. 7.

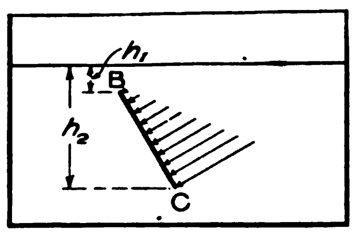


Fig. 8.

length l and a width d (its area equals ld). Suppose it is desired to find the horizontal and vertical components P_h and P_v of this pressure. Since P is perpendicular to BC the inclination of P from the horizontal is the same as that of BC from the vertical. Call this angle θ. From Mechanics we have at once, $P_h = P \cos \theta$ and $P_v = P \sin \theta$ From the foregoing articles we also have $P = h \times .434 \times ld$, in which h is the depth of the center of gravity of BC. Hence we have $P_h = P \cos \theta = .434\, h \times \cos \theta \times ld$, and $P_v = P \sin \theta = .434\, h \times \sin \theta \times ld$. From the figure we see that the area of the vertical projection of the plate BC $= m \times d = l \cos \theta \times d$, and the horizontal projection $= n \times d = l \sin \theta \times d$. Whence we have $P_h = .434\, h \times md$ and $P_v = .434$

$h \times nd$. That is, $P_h = .434\ h \times$ (vertical projection of plate), and $P_v = .434\ h \times$ (horizontal projection of plate). Whence the general

Rule. *The horizontal component of the pressure on a plate is equal to the pressure per square inch at its center of gravity multiplied by the area of its vertical projection, and the vertical component of the pressure is equal to the pressure at its center of gravity multiplied by its horizontal projection.* **(5)**

Examples. 1. What will be the horizontal and vertical components of the pressures on a plate, BC, as in Fig. 9, which is inclined at an angle of 10° to the vertical, the length l of the plate

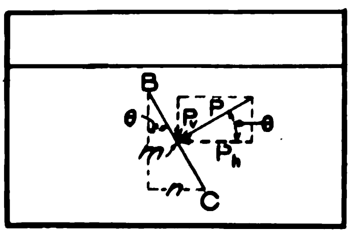


Fig. 9.

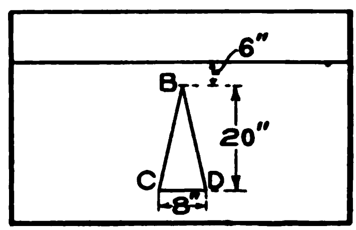


Fig. 10.

being 30 in. and the width 10 in., and the center being 2 feet below the water surface.

Here the pressure per sq. in. at the center is that due to a head of 2 ft., or is equal to $2 \times .434 = .868$ lb. per sq. in. The vertical projection of the plate is equal to $30 \times \cos 10°$ and its horizontal projection $= 30 \times \sin 10°$. Cos. $10° = .985$ and $\sin 10° = .174$, hence by rule 5 the required horizontal component $= .868 \times 30 \times .985 \times 10 = 256$ lb., and the vertical component $= .868 \times 30 \times .174 \times 10 = 45$ lb. Ans.

2. Required the horizontal and vertical components of the pressures on the three faces of the wedge shown in Fig. 10, the length of the wedge perpendicular to the paper being 12 in.

Face BC. The depth of the center of BC below the surface is evidently 16 in. The pressure per sq. in. at this depth $= \frac{16}{12} \times .434 = .579$ lb. The vertical projection of BC $= 20 \times 12$

= 240 sq. in., and its horizontal projection = $4 \times 12 = 48$ sq. in., whence the desired components are: Horizontal component $= 240 \times .579 = 13.89$ lb., vertical component $= 48 \times .579 = 27.8$ lb.

Face BD. The pressures are the same as on BC, the horizontal component acting towards the left and the vertical component acting downwards. The total downward pressure $= 2 \times 27.8 = 55.6$ lb.

Face CD. The pressure per sq. in. at this depth $= .434 \times \frac{26}{12} = .940$ lbs. Total upward pressure $= 8 \times 12 \times .940 = 90.2$ lb.

9. Pressure on Curved Surfaces. If we are dealing with a curved surface as BC, Fig. 11, the pressure is still at all points normal to the surface, but the varying direction of the pressures makes it difficult to determine readily the resultant pressure. The results of the preceding article will, however, enable us to solve the problem sufficiently accurate for all purposes. Suppose the pressure on each square inch be resolved into vertical and horizontal components. Each of these components will equal the normal pressure at the center of the square inch multiplied by the horizontal or vertical projection of the inch of area. Adding all together we find that the total horizontal pressure will equal a certain average horizontal pressure multiplied by the vertical projection of the entire area, and the vertical pressure will equal a certain average vertical pressure multiplied by the horizontal projection. It can be shown that the average value of the horizontal pressure is equal to the pressure at the center of gravity of the vertical projection, but the average value of the vertical pressure cannot be readily determined with accuracy. It may always be estimated by taking as near as may be a pressure corresponding to the average depth of the area below the water surface. Where the body is submerged a great distance, or is under a great pressure in a closed vessel, the error will be unimportant.

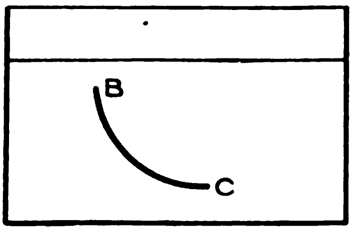


Fig. 11.

10. Bursting Pressure in Pipes and Cylinders. Let BECD, Fig. 12, be the cross-section of any pipe of diameter d and length l and containing water under a head h. The figure shows the pipe connected to an open vessel with water standing at a height h above the center. This free surface of water may represent a reservoir at a height h above the pipe, or the pipe may be entirely closed and the pressure head h exerted upon the water by means of a force pump or a pumping engine. The pressure per square inch at the center of the pipe will be $h \times .434$ pounds. The pressure against the pipe BEDC will be perpendicular to the surface at all points, and if the diameter is small compared to the height h, this pressure will be practically the same at all points and equal to $h \times .434$ pounds per square inch.

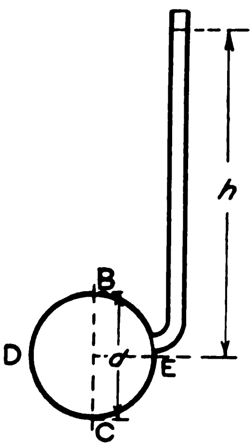


Fig. 12.

Suppose we wish to find the total horizontal force acting against the half BDC. By the foregoing article we may consider the pressure on its vertical projection BC. The center of gravity of this vertical projection will be at the center of the pipe and the pressure per square inch at that point will be $h \times .434$. The area of the projection BC is equal to $d \times l$. Hence the total horizontal pressure against BDC will equal $h \times .434 \times dl$. The pressure against the side BEC will be the same, but opposite in direction.

The action of the pressures on BDC and BEC tends to burst the pipe at points B and C. This is resisted by the stress in the pipe, the amount of which at each of these points is one-half the total horizontal pressure on BDC or BEC, or equal to $\frac{1}{2} h \times .434 \times dl$. If we consider a length of pipe of only one inch then $l = 1$ and we have the important formula for the bursting stress in a pipe:

$$S = \frac{1}{2} h \times .434 \times d \qquad (6)$$

in which S = stress per lineal inch of pipe

h = head of water in feet

and d = diameter of pipe in inches.

By expressing the pressure-head in pounds per square inch instead of feet head we have

$$S = \frac{pd}{2} \qquad (7)$$

in which p = pressure per square inch at center of pipe.

If t = thickness of pipe in inches and s = stress on the metal per square inch then

$$s = \frac{pd}{2t} \qquad (8)$$

For large pipes and low heads the stress at C will be a little larger than at B.

11. Longitudinal Stress in Closed Pipes and Cylinders. Let Fig. 13 represent a side view of a short pipe or cylinder, closed at

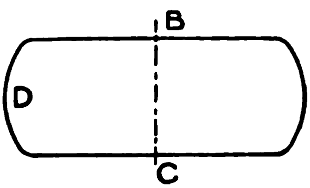


Fig. 13.

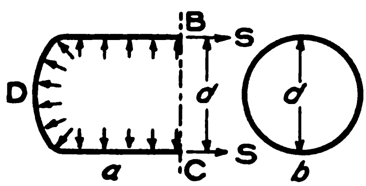


Fig. 14.

the ends like a steam boiler and containing water under a pressure p per square inch. Consider the portion to the left of a section BC. (See Fig. 14a.) The cross-section of the cylinder at BC will be a circle of diameter d on which there will be a stress S due to the horizontal water pressure on the end of the cylinder at D. This total horizontal pressure may be found as in the preceding article. It is equal to the average pressure p multiplied by the vertical projection of the area of the end. This projection, Fig. 14b, is equal to the area of the circle of diameter d, or to $\frac{1}{4}\pi d^2$. Hence the total horizontal force is $p \times \frac{1}{4}\pi d^2$, and hence

$$\text{Total stress} = p \times \tfrac{1}{4}\pi d^2.$$

This stress is distributed entirely around the circumference of the cylinder, or over a distance equal to πd. The stress per inch of circumference is then equal to

$$S' = \frac{p \times \frac{1}{4}\pi d^2}{\pi d} = \frac{pd}{4} \qquad (9)$$

This is seen to be just one-half of the stress in a circumferential direction, as given by formula 7.

If t equal thickness of cylinder, then the horizontal stress per square inch of metal is

$$s' = \frac{pd}{4t} \qquad (10)$$

Examples 1. What will be the stress per lineal inch in a pipe 30 in. in diameter under a water pressure of 40 feet?

The pressure per lineal inch is equal to, by equation 6, $\frac{1}{2} \times 40 \times .434 \times 30 = 260.4$ lb. Ans.

2. If the safe strength of the metal of a pipe in example 1 is 2,000 lb. per sq. in., what will be the necessary thickness of the pipe wall?

The stress per lineal inch is 260.4 lb., and if the safe stress is 2,000 lb. per sq. in., the necessary thickness will be equal to $260.4 \div 2{,}000 = .130$ inch. Ans.

EXAMPLES FOR PRACTICE.

1. What is the bursting stress per square inch in a pipe $\frac{1}{2}$ inch thick and 4 feet in diameter under a pressure head of 400 feet? 8,330 lb. Ans.

2. What is the stress per square inch in a boiler plate 1 inch thick, the boiler being 6 feet in diameter working under a pressure of 150 lb. per sq. in. (Use equation 8.) 5,400 lb. Ans.

3. What is the horizontal stress per sq. in. in the boiler of example 2? 2,700 lb. Ans.

12. Center of Pressure on Rectangular Areas. In the preceding discussion of Arts. 7 and 8 the *total* pressure was the quantity determined. In the case of the plate reaching to the surface, Fig, 6, the variation in the pressure was represented by a triangle of forces, and where the plate was wholly submerged, Fig. 7, it was represented by a trapezoid. In either case the "center of pressure," or the point where the resultant of the pressure forces would be applied, will be opposite the *center of gravity* of the pressure area. In the case of the triangle the center of gravity is two-thirds the distance from the apex to the base, hence,

The center of pressure against a rectangular plate which reaches to the surface, or projects above it, is two-thirds the distance from the surface to the lower edge of the plate. **(11)**

In the case of the wholly submerged plate, with trapezoidal pressure diagram BC, Fig. 15, the pressure head at B is equal to h_1 and that at C h_2, which heads will be equal to the heights BD and CE of the trapezoid representing the pressures.

By the method explained in the paper on Strength of Materials, Art. 48, we find the center of gravity of the trapezoid of length l to be at a distance from B equal to

$$x = \frac{2}{3} l \frac{\frac{h_1}{2} + h_2}{h_1 + h_2} \qquad (12)$$

Another form of expression can be obtained by noting that if the point A be the intersection of the plane BC, produced, with the surface of the water, the line DE, produced, will also pass through A, since a plate AC would have a triangle of pressures

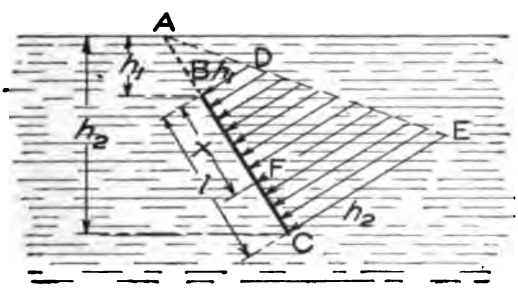


Fig. 15.

represented by AEC. Then by proportion we would have $\frac{h_1}{h_2} = \frac{AB}{AC}$, or $h_1 = h_2 \frac{AB}{AC}$. Substituting this value for h_1 in equation 12, and reducing, we have

$$x = \frac{2}{3} l \frac{\frac{AB}{2} + AC}{AB + AC} \qquad (13)$$

13. Center of Pressure on Plane Areas of Any Form. The center of pressure of irregular plane areas can be found by the following rule, the demonstration of which is here omitted. Let BC, Fig. 15, represent a plane area of any form, then

The distance AF from the surface to the center of pressure is equal to the moment of inertia of the given area about an axis at A divided by the product of the area times the distance from A to its center of gravity. **(14)**

Examples. 1. What force S will be required to lift a sluice gate BC, Fig. 16, placed on the sloping face of a dam and hinged at B? The gate is 3 feet wide, 4 feet long from B to C, and has such a slope that the vertical projection BD = 3.5 ft., and the

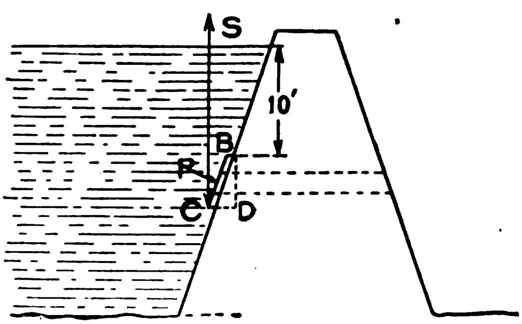
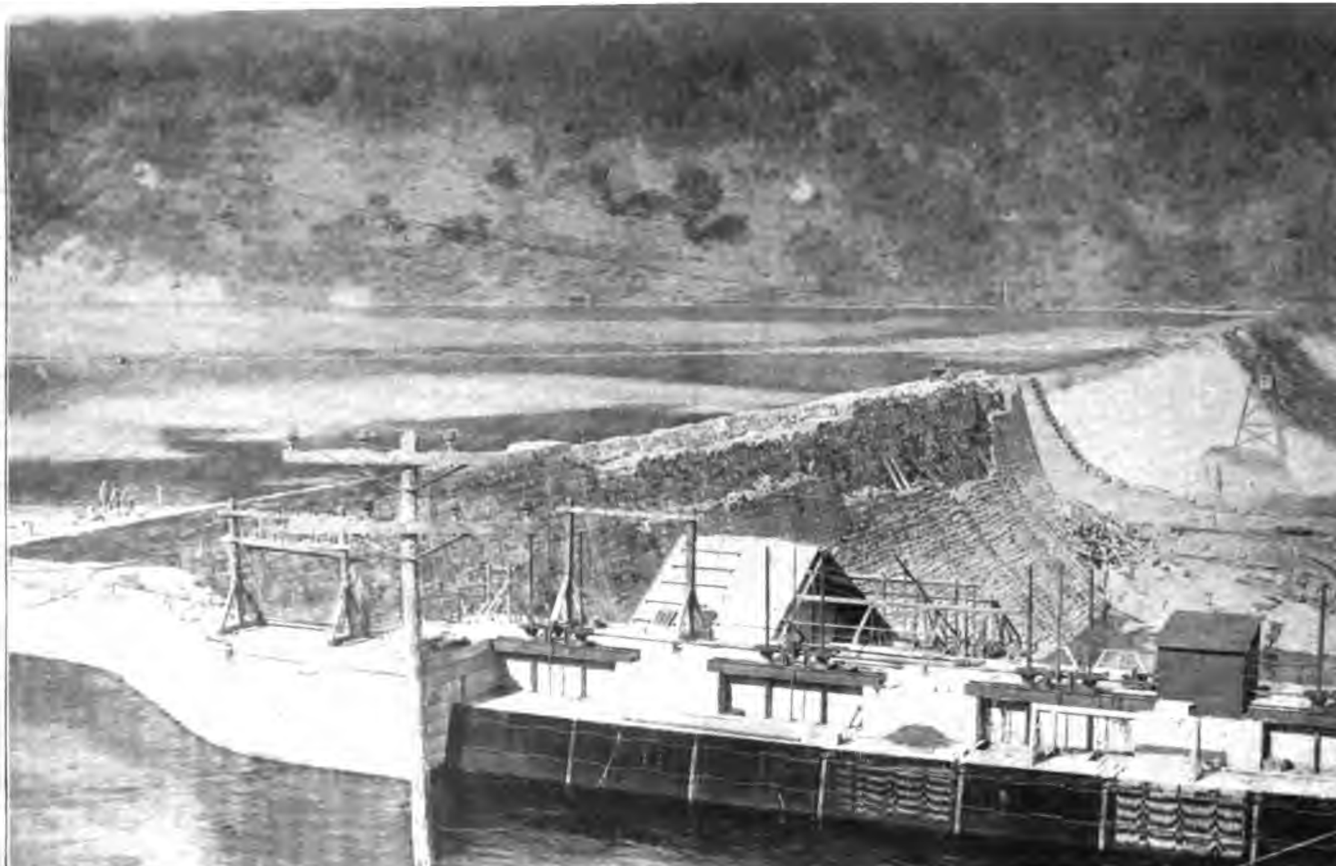


Fig. 16.

horizontal projection CD = 1.93 ft. The depth of B below the surface is 10 ft.

We will first find the total pressure P against the gate. By Art. 7 this will be the pressure per sq. in. at the center multiplied by the area. The depth of the center is $10 + \frac{1}{2}$ BD $= 10 + \frac{3.5}{2}$ $= 11.75$ feet. The pressure per sq. ft. $= 11.75 \times 62.5 = 734$ lb. The total pressure $= 734 \times 3 \times 4 = 8{,}808$ lb.



CANAL AND DAM AT SPIER FALLS, NEW YORK

At this plant, which is controlled by the Hudson River Power Company, about 80,000 horse-power is developed, which, combined with the output from other water-power plants in the vicinity, is supplied to the shops of the General Electric Company at Schenectady, and also for light and power purposes to Albany, Troy, and other neighboring towns.

The center of pressure will be found next. This is at a distance from B given by formula 12, in which $h_1 = 10$ and $h_2 = 13.5$. We have then $x = \frac{2}{3} \times 4 \times \frac{\frac{10}{2} + 13.5}{10 + 13.5} = 2.1$ feet.

Now taking moments about B we have $S \times 1.93 = P \times 2.1$ or $S = 8{,}808 \times \frac{2.1}{1.93} = 9{,}590$ lb. Ans.

2. Find the water pressure on a gate AB, Fig. 17, one foot long, when the heads on the two sides are different; also find the reactions R_1 and R_2 of the gate against sills at A and B.

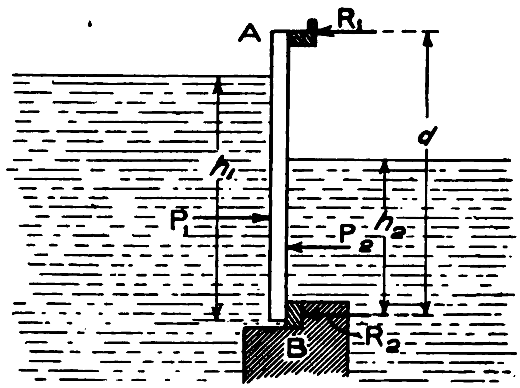


Fig. 17.

By Art. 7 the total pressure P_1 of the water on the left side of the gate is equal to the pressure at the half depth $\frac{h_1}{2}$ multiplied by its submerged area. Taking here the square foot as the unit and letting w = weight of a cubic foot of water, the pressure at a depth $\frac{h_1}{2}$ is $\frac{h_1}{2} \times w$ pounds per square foot, and as the exposed area is $h_1 \times 1$ the total pressure $P_1 = \frac{h_1}{2} \times w \times h_1 = \frac{1}{2} h_1^2 \times w$. The center of pressure, or point of application of P_1, is $\frac{2}{3} h_1$ below the surface (Art. 12).

In like manner the pressure $P_2 = \frac{1}{2} h_2^2 w$, and its point of application is $\frac{2}{3} h_2$ below the water surface on that side.

The forces P_1 and P_2 being known, the reaction R_1 may be found by taking moments about B as explained in Mechanics.

There results the equation

$$R_1 \times d - P_1 \frac{h_1}{3} + P_2 \frac{h_2}{3} = 0$$

whence

$$R_1 = \frac{1}{3} \frac{P_1 h_1 + P_2 h_2}{d}$$

Substituting the values of P_1 and P_2 above given, we have

$$R_1 = \frac{1}{3} \times w \times \frac{1}{2} \frac{h^3_1 + h^3_2}{d}$$

$$= \frac{1}{6} w \frac{h^3_1 + h^3_2}{d} \qquad (15)$$

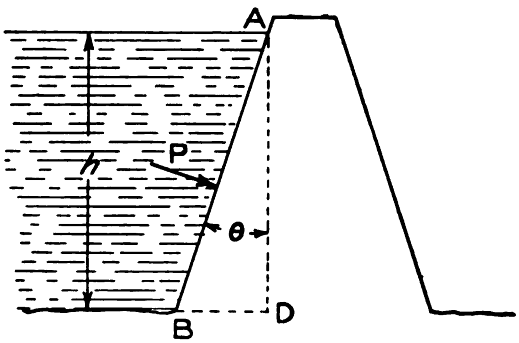


Fig. 18.

in which all dimensions are to be expressed in feet, and the result will be for a gate one foot long. For other lengths the value of R_1 will be proportional to the length.

3. Find the pressure P on a dam AB, Fig. 18. Let h = depth of water against the dam. Consider a length of dam of one foot. By Art. 7 the total pressure P is equal to the pressure at the half depth multiplied by the area of AB or

$$P = \frac{h}{2} \times w \times (\text{length of AB}) \times 1.$$

The center of pressure, by Art. 12, is two-thirds of the distance from A to B.

The horizontal component of the pressure is, by Art. 8, equal to the pressure per square foot at mid-depth multiplied by the vertical projection of the face AB, or

$$P_h = \frac{h}{2} \times w \times h \times 1 = \frac{1}{2} wh^2 \qquad \textbf{(16)}$$

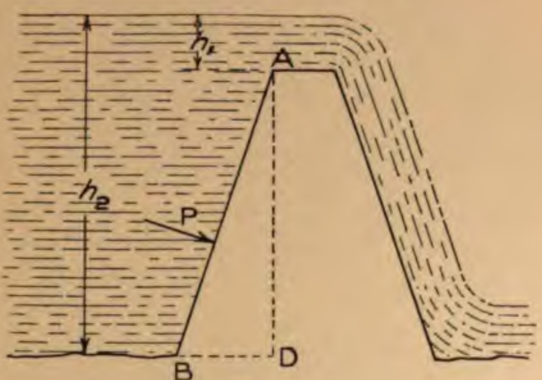


Fig. 19.

The vertical component is likewise

$$P_v = \frac{h}{2} \times w \times (\text{length BD}),$$

but we can write BD $= h \tan \theta$. Hence

$$P_v = \frac{1}{2} wh^2 \tan \theta. \qquad \textbf{(17)}$$

If the dam is submerged, as shown in Fig. 19, then the method employed in example 1 of this Article must be used.

EXAMPLES FOR PRACTICE.

1. What is the horizontal pressure on a dam one foot long on which the water has a depth of 80 feet; and where is the center of pressure? 200,000 lb., and 26 ft. 8 in. from the bottom. Ans.

2. What is the vertical component of the pressure in example 1 if the face of the dam slopes 1 inch horizontally to 1 foot vertically? (The horizontal projection $= \frac{1}{12} \times 80 = 6\frac{2}{3}$ ft.)

16,670 lb. Ans.

3. In Fig. 19 if $h_1 = 10$ ft., $h_2 = 40$ ft., AD $= 30$ ft., and BD $= 10$ ft., what will be the horizontal and vertical components

of the pressure P? The center of gravity of the area is 30 ft. deep. Use rule 5.

Hor. comp. = 46,875 lb.; Vert. comp. = 15,625 lb.
Ans.

4. How far from A is the center of pressure in example 3? The length of AB = 31.62 ft. Use equation (12). 18.97 ft. Ans.

14. Buoyant Effect of Water on Submerged Bodies. If a body AB, Fig 20, be submerged, the water exerts an uplift upon it owing to the fact that the pressure upwards on the bottom of the body is greater than the pressure downwards on the top. The net upward force, or buoyant effect, is exactly equal to the weight

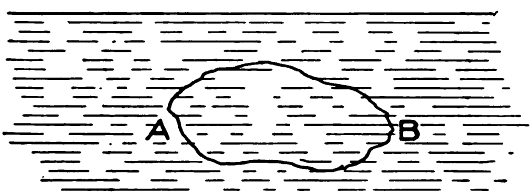


Fig. 20.

of a volume of water equal to that of the body AB. It is plain that this must be so, for if AB be replaced by water, the water would tend neither to rise nor fall, that is, it would be just supported by the surrounding pressures. Hence the following well-known law:

The weight of a body in water is less than its weight in air by an amount equal to the weight of an equal volume of water.

15. *The Specific Gravity* of a substance is the ratio of its weight to that of an equal volume of water. The specific gravity is found by weighing a body in air and then in water. The difference is the weight of an equal volume of water. Then if W equals weight in air; and W′ equals weight in water, then W − W′ = weight of water displaced, and

$$\text{Specific gravity} = \frac{W}{W - W'}. \qquad (18)$$

as explained in Elementary Mechanics.

EXAMPLES FOR PRACTICE.

1. If a body weighs 100 lb. in air and 40 lb. in water, what is its specific gravity? 1.67. Ans.

2. If a body of .6 cu. ft. in volume weighs 75 lb., what is its specific gravity, the weight of water being 62.5 lb. per cu. ft.? 2.0. Ans.

3. If a body of 3 cu. ft. in volume has a specific gravity of .75, what force is necessary to submerge it? Here the buoyant effect is greater than the weight of the body. 46.9 lb. Ans.

FLOW OF WATER THROUGH ORIFICES.

16. Velocity of Flow Through Orifices. If AB, Fig. 21, be a vessel containing water of depth h, and C and D are any open tubes connected therewith, the water will stand in these tubes at

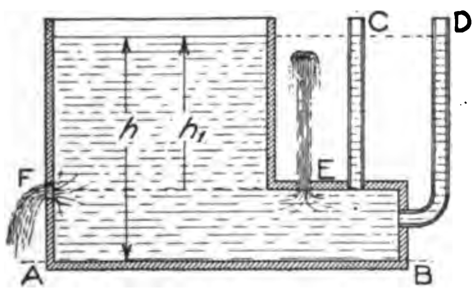


Fig. 21.

the same height h above the base level AB as in the large vessel, and the pressure in the tubes at any given depth is the same as in the large vessel at the same depth. If we now make an opening at E so that the water will issue in a vertical direction it has been experimentally demonstrated that the water will rise very nearly to the same level as it will in the tubes. The discrepancy is due to the air resistance and a slight friction at the opening. Neglecting this discrepancy the velocity of the water at E can be determined on the principle that it must be sufficient to cause the water to rise the distance h_1. In Mechanics it was shown that, neglecting air resistance, the velocity a body must have to cause it to rise against gravity a distance h_1 is the same as the velocity acquired by a body falling through the same distance. This velocity is given by formula

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

in which v = velocity in feet per second
g = acceleration of gravity
= 32.2 feet per second
and h = height of fall, or the height a body will rise when started with a velocity v.

Applying this to the jet issuing from E we find that the theoretical velocity of efflux is

$$v = \sqrt{2gh_1}$$

If the orifice be in the side of the vessel, as at F, on the same level as E, it is plain that the water will issue with the same force as at E, since the pressure is the same. Hence in general:

The theoretical velocity of efflux from an orifice in any direction is

$$v = \sqrt{2gh} \qquad \textbf{(19)}$$

where h is the pressure head in feet at the orifice.

In practice the velocity is a little less than that given by the formula, the actual velocity being from 97% to 99% of the theoretical. This ratio of .97 to .99 is called the *coefficient of velocity.*

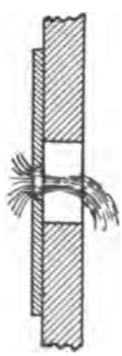


Fig. 22.

17. Use of Orifices for Measuring Water. In making use of an orifice for measuring water it is desirable, for the sake of accuracy, that the orifice be constructed in such a way that the water in passing out will touch the inner edge only. This may be done by making the orifice of a very thin plate, or cutting it on a bevel so that the water will not come in contact with the side, as shown in Fig. 22. To get accurate results an orifice should be made of metal, such as brass, and fastened to the inside of the tank as in Fig. 22, but in many cases sufficiently accurate results can be obtained by cutting a beveled hole in the side of a tank.

To give reliable results the orifice should be located a distance from the nearest side or the bottom of the tank not less than three times the width of the orifice. The tank or channel should also have a cross-section much larger than that of the orifice so that the velocity of the water as it approaches the orifice will be small, otherwise the discharge will be affected by this "velocity of approach". If the cross-section of the tank is as much as twenty times that of the orifice this effect is of no consequence.

18. Discharge Through Small Orifices. When water flows through an orifice, such as shown in Fig. 22, the direction of the flow at the edges is such as to cause the water vein to contract as it issues from the orifice. The area of the contracted vein at its smallest section is only 60 to 70 per cent of the full area of the orifice, the exact value depending upon the size of the orifice, and the pressure. Now the discharge through any orifice, pipe, or channel is equal to the area of the cross-section of the stream of water multiplied by its velocity at that point. If we measure the cross-section in square feet and the velocity in feet per second, then the discharge will be expressed in cubic feet per second. In the case of the orifice, then, to determine the discharge per second we would need to multiply the area of the cross-section of the vein of water by its velocity. In Art. 16 it was shown that the actual velocity of the jet was about 97 to 99 per cent of the theoretical velocity v, which refers to the velocity of the vein at the contracted section where the velocity is a maximum. The discharge will then be found by multiplying this actual velocity by the actual area at the point of contraction. Thus if we take the coefficient of velocity as .98 and the coefficient of contraction as .65, the discharge would be

$$Q = .98\, v \times .65\, A$$

where Q = discharge in cubic feet per second

v = theoretical velocity in feet per second by equation 19, and A = area of orifice in square feet.

If we substitute for v its value $\sqrt{2gh}$ we have

$$Q = .98 \times .65\, A \sqrt{2gh}$$
$$= .637\, A \sqrt{2gh},$$

The coefficient .637 in this case is called the *coefficient of discharge*, and as it varies with different conditions, it is desirable to use the more general formula

$$Q = c\, A \sqrt{2gh}. \qquad \textbf{(20)}$$

in which c = coefficient of discharge, which varies in value from about .66 to .60. This coefficient is the product of the "coefficient of velocity" and the "coefficient of contraction."

TABLE NO. 3.

Coefficients for Circular Vertical Orifices.

Head, h, in Feet.	Diameter of Orifice in Feet.						
	0.02	0.04	0.07	0.10	0.2	0.6	1.0
0.4	....	0.637	0.624	0.618			
0.6	0.655	.630	.618	.613	0.601	0.593	
0.8	.648	.626	.615	.610	.601	.594	0.590
1.0	.744	.623	.612	.608	.600	.595	.591
1.5	.637	.618	.608	.605	.600	.596	.593
2.0	.632	.614	.607	.604	.599	.597	.595
2.5	.629	.612	.605	.603	.599	.598	.596
3	.627	.611	.604	.603	.599	.598	.597
4	.623	.609	.603	.602	.599	.597	.596
6	.618	.607	.602	.600	.598	.597	.596
8	.614	.605	.601	.600	.598	.596	.596
10	.611	.603	.599	.598	.597	.596	.595
20	.601	.599	.597	.596	.596	.596	.594
50	.596	.595	.594	.594	.594	.594	.593
100	.593	.592	.592	.592	.592	.592	.592

TABLE NO. 4.

Coefficients for Square Vertical Orifices.

Head, h, in Feet.	Side of the Square in Feet.						
	0.02	0.04	0.07	0.1	0.2	0.6	1.0
0.4	....	0.643	0.628	0.621			
0.6	0.660	.636	.623	.617	0.605	0.598	
0.8	.652	.631	.620	.615	.605	.600	0.597
1.0	.648	.628	.618	.613	.605	.601	.599
1.5	.641	.622	.614	.610	.605	.602	.601
2.0	.637	.619	.612	.608	.605	.604	.602
2.5	.634	.617	.610	.607	.605	.604	.602
3	.632	616	.609	.607	.605	.604	.603
4	.628	.614	.608	.606	.605	.603	.602
6	.623	.612	.607	.605	.604	.603	.602
8	.619	.610	.606	.605	.604	.603	.602
10	.616	.608	.605	.604	.603	.602	.601
20	.606	.604	.602	.602	.602	.601	.600
50	.602	.601	.601	.600	.600	.599	.599
100	.599	.598	.598	.598	.598	.598	.598

TABLE NO. 5.

Coefficients for Rectangular Orifices 1 Foot Wide.

Head, h, in Feet.	Depth of Orifice in Feet.						
	.125	.25	.50	.75	1.0	1.5	2.0
.4	.634	.633	.622				
.6	.633	.633	.619	.614			
.8	.633	.633	.618	.612	.608		
1.	.632	.632	.618	.612	.606	.626	
1.5	.630	.631	.618	.611	.605	.626	.628
2.	.629	.630	.617	.611	.605	.624	.630
2.5	.628	.628	.616	.611	.605	.616	.627
3.	.627	.627	.615	.610	.605	.614	.619
4.	.624	.624	.614	.609	.605	.612	.616
6.	.615	.615	.609	.604	.602	.606	.610
8.	.609	.607	.603	.602	.601	.602	.604
10.	.606	.603	.601	.601	.601	.601	.602
20.				.601	.601	.601	.602

19. Experimental Coefficients of Discharge. Many experiments have been made on different kinds of orifices to determine the value of c, equation 20, so that by means of this formula and a table of coefficients, orifices could readily be used for measuring water. The accompanying tables give these coefficients for circular, square, and rectangular orifices in vertical planes, the rectangular orifices all being one foot wide.

Example 1. What is the discharge from a circular orifice 3 in. in diameter under a pressure head of 10 feet?

By Table No. 3 the coefficient of discharge for an orifice of a diameter of .25 ft. and under a head of 10 feet is .597. The area of the orifice $= \frac{1}{4} \times 3.14 \times .25^2 = .049$ sq. ft. Then by equation 20 the discharge will be $.597 \times .049 \times \sqrt{2 \times 32.2 \times 10} = .743$ cu. ft. per sec. Ans.

2. What will be the velocity of flow in example 1, the coefficient of velocity being taken equal to .97?

By equation 19 the velocity $= .97 \sqrt{2 \times 32.2 \times 10} = 24.6$ ft. per sec. Ans.

EXAMPLES FOR PRACTICE.

1. What will be the discharge from an orifice 4 in. square under a head of 16 feet? 2.14 cu. ft. per sec. Ans.

2. What must be the diameter of a circular orifice acting under a head of 25 feet to discharge 1 cu. ft. per sec.? (Assume $c = .6$ for a trial solution.) 2.76 in. Ans.

3. A pipe discharges 1.5 cu. ft. per sec. into a tank from which the water escapes through an orifice 6 in. square. How

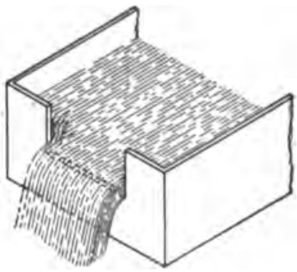


Fig. 23*a*.

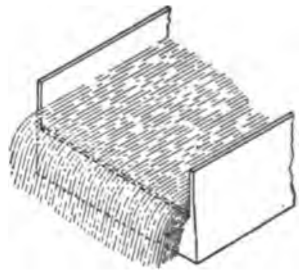


Fig. 23*b*.

deep will the tank be filled above the orifice when the outflow is just equal to the inflow? 1.53 ft. Ans.

FLOW OF WATER OVER WEIRS.

20. General Explanation. The term weir is usually given to a notch cut in the side of a tank or reservoir through which water may flow and be measured. The notch is usually rectangular and may have a width less than that of the tank, as shown in Fig. 23*a*, or equal to that of the tank, as in Fig. 23*b*. Such weirs are often used for measuring the flow of a small stream by building a small dam and leading all the water through a notched plank or timber wall. For accurate work weirs should be sharp-crested (the "crest" is the lower edge over which the water passes) so that the water will touch the inner corner only as in the case of the standard orifice described in Art. 18. The back side of the weir should be smooth and vertical for a considerable distance downwards from the crest.

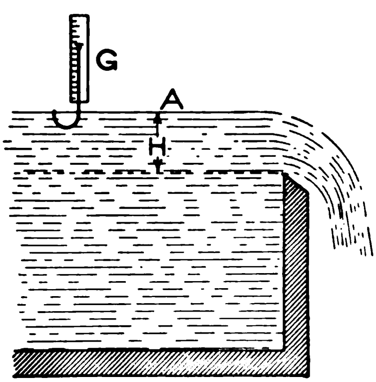


Fig. 24.

If the weir is made as in Fig. 23*a* the water in passing out will cause a contraction of the stream laterally, but if made as in Fig. 23*b* the water will pass out parallel to the sides of the tank and there will be no lateral, or, as it is called, "end contraction". In either case reliable results may be obtained by the use of the proper coefficients, but if the form of Fig. 23*a* be used, the distance of the notch from the side of the tank or channel should be at least three times the depth of the water on the weir in order that the contraction may be complete.

The measurement of water flowing over a weir is accomplished by merely measuring the depth of the water flowing over it. Then knowing this and the length of the weir the discharge can be calculated. In measuring this depth of water, or "height" of water on the weir, as it is commonly called, it is necessary to take the level of the water some distance back from the weir, as at A, Fig. 24, in order to avoid the effect of the curvature of the water surface. The difference between the level of the water and that of the weir is then the desired height H. The necessary distance back from the weir may be taken as 2 or 3 feet for small weirs to 8 or 10 feet for large ones.

A common and accurate way of determining the level of the water at A is by means of a submerged hook, shown at G, Fig. 24, called a *hook gauge*, arranged to be easily moved vertically along a scale. Fig. 25 shows such a gauge in detail. The gauge is set by moving it until the hook comes to the surface of the water. The scale is then read and the level of the water determined.

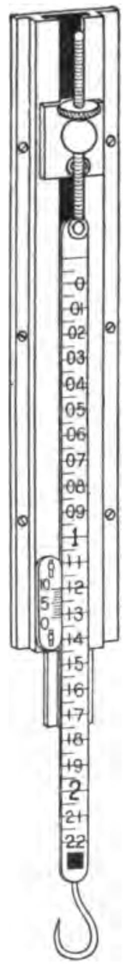


Fig. 25.

21. Formulas for Discharge. If the weir were a rectangular orifice at a considerable depth below the surface its discharge would be given by the formula

$$Q = c \times b \times d \sqrt{2gh} \qquad \textbf{(21)}$$

as in equation 20 of Art. 18. In this expression b = breadth and d = height of orifice, and h = average depth of orifice below the surface, or the average pressure head. In the case of the weir the depth d is the height H, and the average pressure head, h, is something less than H, varying from nothing for the water at the surface to the full value H for the water at the crest. For a case like this where the square root of a quantity, h, is taken, that varies from zero to a given value H, the *average* value of this square root is two-thirds the square root of the maximum limit H. That is, for h in equation 20 we may substitute $\frac{2}{3}\sqrt{H}$, giving for the discharge

$$Q = cb \times \frac{2}{3} H \sqrt{2gH}$$

$$= c \times \frac{2}{3} b \sqrt{2g} \times H^{\frac{3}{2}} \qquad (22)$$

in which c is the coefficient of discharge and equal to about .60 to .65 as for orifices.

If the channel is small the "velocity of approach" will have an appreciable effect upon the discharge, increasing it somewhat above what it otherwise would be. This is taken account of by calculating approximately the velocity of the water in the channel of approach at the place where the level of the water is measured, and determining the head h corresponding to this velocity by the formula

$$h = \frac{v^2}{2g}.$$

Then the discharge will be, for weirs with end contractions,

$$Q = c \times \frac{2}{3} b \sqrt{2g}\,(H + 1.4h)^{\frac{3}{2}} \qquad (23)$$

and for weirs without end contractions

$$Q = c \times \frac{2}{3} \sqrt{2g}\ \left(H + 1\frac{1}{3}h\right)^{\frac{3}{2}} \qquad (24)$$

The coefficient c should, in all cases, be selected according to the character of the weir.

In calculating "velocity of approach", it is necessary first to get an approximate value for the discharge Q by omitting the term h. The resulting discharge, divided by the cross-section of the tank or channel will be, with sufficient accuracy, the desired velocity of approach.

22. Coefficients of Discharge. Tables Nos. 6 and 7 give values of the coefficient c for the above formulas for rectangular sharp-crested weirs.

TABLE NO. 6.

Coefficients for Contracted Weirs.

Effective Head in Feet, h.	Length of Weir in Feet, b.						
	0.66	1	2	3	5	10	19
0.1	0.632	0.639	0.646	0.652	0.653	0.655	0.656
0.15	.619	.625	.634	.638	.640	.641	.642
0.2	.611	.618	.626	.630	.631	.633	.634
0.25	.605	.612	.621	.624	.626	.628	.629
0.3	.601	.608	.616	.619	.621	.624	.625
0.4	.595	.601	.609	.613	.615	.618	.620
0.5	.590	.596	.605	.608	.611	.615	.617
0.6	.587	.593	.601	.605	.608	.613	.615
0.7	....	.590	.598	.603	.606	.612	.614
0.8	....	....	.595	.600	.604	.611	.613
0.9	....	....	.592	.598	.603	.609	.612
1.0	....	....	.590	.595	.601	.608	.611
1.2	....	....	.585	.591	.597	.605	.610
1.4	....	....	.580	.587	.594	.602	.609
1.6	....	....	....	.582	.591	.600	.607

TABLE NO. 7.

Coefficients for Weirs without Contractions.

Effective Head in Feet, h.	Length of Weir in Feet, b.						
	19	10	7	5	4	3	2
0.1	0.657	0.658	0.658	0.659			
0.15	.643	.644	.645	.645	0.647	0.649	0.652
0.2	.635	.637	.637	.638	.641	.642	.645
0.25	.630	.632	.633	.634	.636	.638	.641
0.3	.626	.628	.629	.631	.633	.636	.639
0.4	.621	.623	.625	.628	.630	.633	.636
0.5	.619	.621	.624	.627	.630	.633	.637
0.6	.618	.620	.623	.627	.630	.634	.638
0.7	.618	.620	.624	.628	.631	.635	.640
0.8	.618	.621	.625	.629	.633	.637	.643
0.9	.619	.622	.627	.631	.635	.639	.645
1.0	.619	.624	.628	.633	.637	.641	.648
1.2	.620	.626	.632	.636	.641	.646	
1.4	.622	.629	.634	.640	.644		
1.6	.623	.631	.637	.642	.647		

23. The Francis Formula. The most widely used weir formula for large weirs without end contractions is that derived by Mr. James B. Francis from an extensive series of experiments on weirs 10 feet long. His formula is

$$Q = 3.33\ bH^{\frac{3}{2}}, \qquad (25)$$

in which the unit of length must be the foot. This is equivalent to the use of a constant value of the coefficient c of equation 22, equal to .623. It gives results sufficiently close for most purposes. With end contractions the length b is to be reduced by .1 H for

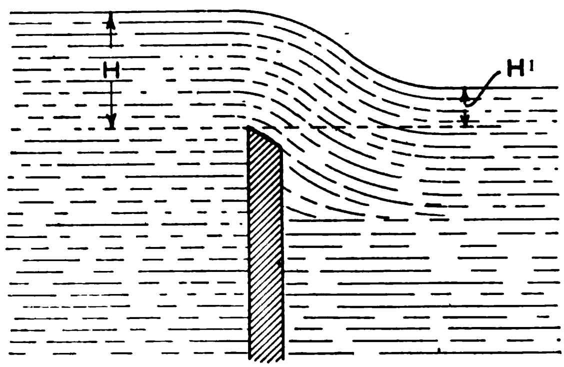


Fig. 26.

one end contracted and by .2 H for both ends contracted. The formula is further modified to allow for velocity of approach, but where this element enters, use may be made of the other formula.

24. Submerged Weirs. Where the water on the downstream side of a weir is higher than the crest, as in Fig. 26, the discharge is closely given by the formula

$$Q = 3.33\ b\ (nH)^{\frac{3}{2}}, \qquad (26)$$

where H is the height of the water on the upper side and n is a coefficient depending on the ratio of the head on the lower side, H^1, to the head H. The values of n are as follows:

TABLE NO. 8.

Values of n for Submerged Weirs.

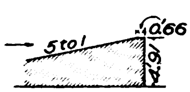
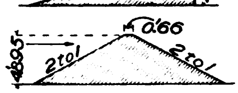
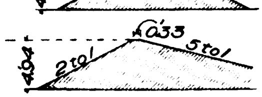
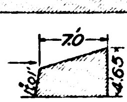
$\frac{H'}{H}$	n	$\frac{H'}{H}$	n	$\frac{H'}{H}$	n	$\frac{H'}{H}$	n
.00	1.000	.20	0.985	.45	0.912	.70	0.787
.02	1.006	.25	0.973	.50	0.892	.75	0.750
.05	1.007	.30	0.959	.55	0.871	.80	0.703
.10	1.005	.35	0.944	.60	0.846	.90	0.574
.15	0.996	.40	0.929	.65	0.819	1.00	0.000

25. Weirs of Irregular Section. In many cases it is desirable to determine the flow of a stream by measurements taken of the height of water flowing over some dam or weir; and, on the other hand, in the design of waste-weirs some method of estimating their capacity is essential. The law of flow over such weirs

TABLE NO. 9.

Values of the Coefficient C in the Formula

$$Q = C l H^{\frac{3}{2}} \text{ for irregular weirs.}$$

	Form of Weir.	Height on Weir in Feet.							
		1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0
1		3.51	3.37	3.33	3.31	3.29	3.23	3.16	3.14
2		.....	3.76	3.68	3.68	3.70	3.75	3.83	
3		.....	3.68	3.71	3.81	3.90	4.00	4.06	
4		3.81	3.61	3.68	3.65	3.72	3.80	3.93	
5		3.81	3.61	3.57	3.63	3.62	3.67	3.71	3.80

is very complicated, and the only accurate way of determining the constants for any particular case is by means of experiments on a section of the same form as the one in question. If this is impossible, the best substitute for it is to use constants which have been determined for a weir agreeing as closely in form as may be to the one under consideration.

In Table No. 9 are given several sets of coefficients for five forms of dams, as determined by experiment. This coefficient is to be used in place of the value 3.33 in equation 25.

It will be noted by comparing Nos. 1 and 3 that the discharge falls off considerably by using a flat slope for the back of the dam.

Examples. 1. What will be the discharge of a sharp-crested weir 4 ft. long with $H = 6$ inches, there being contraction at both ends?

By Table No. 6 the coefficient may be taken at .610. Then by equation 22, $Q = .61 \times \frac{2}{3} \times 4 \times \sqrt{64.4} \times \left(\frac{1}{2}\right)^{\frac{3}{2}} = 4.6$ cu. ft. per sec. Ans.

2. If the channel of approach in example 1 be 6 feet wide by $2\frac{1}{2}$ feet deep, what will be the effect of the "velocity of approach"?

Assuming the same discharge as above, the velocity of flow in this channel will be $\frac{4.6}{6 \times 2\frac{1}{2}} = .11$ ft. per sec. The head h corresponding to this velocity $= \frac{v^2}{2g} = \frac{.11^2}{64.4} = .0002$ ft. Introducing this value for h in equation 23, it is seen that the additional term $1.4\ h$ is too small to be of any practical consequence.

FLOW OF WATER THROUGH PIPES.

26. Discharge Through Pipes for Different Velocities. The rate of discharge through a pipe is equal to the average velocity of the flowing water multiplied by the cross-section of pipe. Velocities are usually expressed in feet per second and discharge in cubic feet per second or gallons per minute. The diameter of a pipe is always given in inches. These differences in units make it desirable to have a table at hand giving for a velocity of one foot per second the discharge of pipes of various diameters expressed both in cubic feet per second and in gallons per minute. Such a table is given below:

POWER INSTALLATIONS AT NIAGARA FALLS, NEW YORK, USING "SAMSON" TURBINES

Courtesy of James Leffel & Co., Springfield, Ohio

TABLE NO. 10.

Discharge of Pipes in Cubic Feet Per Second and in Gallons Per Minute for a Velocity of One Foot Per Second.

(For other velocities multiply the discharge here given by the velocity expressed in feet per second.)

Diameter of Pipe in Inches.	Discharge.	
	Cubic Feet Per Second.	Gallons Per Minute.
1	.0055	2.4
2	.0218	9.8
3	.0491	22.0
4	.0873	39.1
6	.1964	88.1
8	.3491	157
10	.5454	245
12	.7854	352
14	1.069	480
16	1.396	627
20	2.182	978
24	3.142	1410
30	4.909	2200
36	7.069	3155
42	9.621	4317
48	12.568	5639

EXAMPLES FOR PRACTICE.

1. What will be the discharge in gallons per minute of a 6-inch pipe for a velocity of 4.5 feet per sec.? 396 gallons per min. Ans.

2. What velocity will be required to discharge 1,000,000 gallons per day through an 8-inch pipe? 4.4 ft. per sec. Ans.

3. What diameter of pipe will be required to discharge 1,000 gals. per min. at a velocity of 5 feet per sec.?

An 8-inch pipe will discharge 785 gals. per min. and a 10-inch pipe will discharge 1,225 gals. per min. A 10-inch pipe would therefore be necessary if no intermediate size is available. Ans.

27. General Principles Governing the Flow of Water Through Pipes. Let ABCD, Fig. 27, be any pipe leading from a reservoir and having a stop valve at D. Also suppose B*b* and C*c* are tubes connected with the pipe at B and C and open at the top.

From the laws of pressure explained in Art. 6 we know that if the valve D be closed so that there will be no motion of the water the water will rise in the tubes B*b* and C*c* to the same level as that in the reservoir. The pressure at A will be represented by the head h_1 and that at B and C by the heads h_2 and h_3 respectively, which heads may be greater or less than the head at A according as the pipe slopes downwards or upwards from A.

Now let the valve at D be opened partly so as to permit the water to escape slowly. It will be found that the pressures at B and C will immediately decrease and that the water in the tubes will fall to some lower levels b' and c'. This decrease in pressure

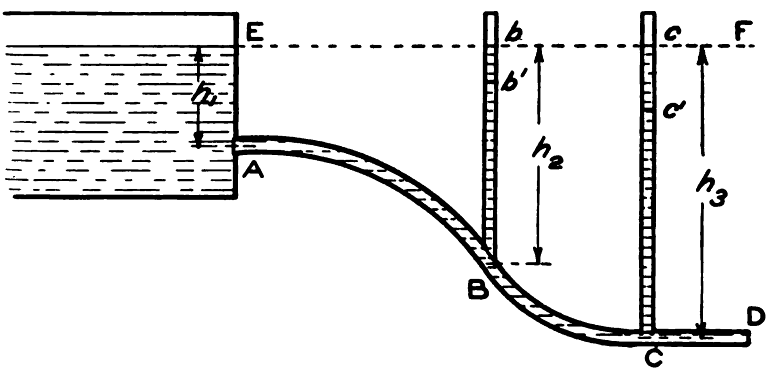


Fig. 27.

is due to two causes. First, a part of the pressure head has been used to give the water some velocity in the pipe, and second, a part has been consumed in the friction of the water in passing from A to B and C. The portion used in giving the water its velocity is the same as the head required to produce a given velocity of efflux, v, from an orifice, and is found from the formula $v = \sqrt{2gh}$. Solving for h we have

$$h = \frac{v^2}{2g} \qquad (27)$$

in which v is the actual velocity of flow in the pipe.

The pressure head lost in friction is usually much greater than that used in velocity and is the most important as well as the most difficult part of the problem of determining the flow in pipes.

If H represents the total loss of pressure head between the reservoir and any point B, h_v the head necessary to produce the given velocity at B ("velocity head") and h_f the pressure lost by friction between A and B we then have in general

$$H = h_v + h_f \qquad (28)$$

or from equation 27

$$H = \frac{v^2}{2g} + h_f \qquad (29)$$

In the figure, bb' represents the head H for point B and cc' that for point C. Between B and C the loss in head is the *difference* between bb' and cc' and is all due to friction, since the velocity is the same at the two points, the pipe being of uniform size.

If now we open the valve D farther so as to give the water a higher velocity the level of the water in the tubes bB and cC will fall still more, that is, there will be a greater loss of pressure head, H, than before. This increase in loss of pressure is due mainly to the increased friction loss h_f caused by the higher velocity, but to a small extent also to the increased energy transformed into velocity head.

In any case that part of the head H needed to produce the velocity v, which is equal to $\frac{v^2}{2g}$, can readily be calculated or can be obtained from the following table:

TABLE NO. 11.

Velocity Heads

$$h = \frac{v^2}{2g}$$

Corresponding to Various Values of v.

v feet per sec.	h ft.	v ft. per sec.	h ft.	v ft. per sec.	h ft.	v ft. per sec.	h ft.
2.0	0.06	4.0	0.25	6.0	0.56	8.0	0.99
2.2	0.08	4.2	0.28	6.2	0.60	8.2	1.04
2.4	0.09	4.4	0.30	6.4	0.64	8.4	1.10
2.6	0.10	4.6	0.33	6.6	0.68	8.6	1.15
2.8	0.12	4.8	0.36	6.8	0.72	8.8	1 20
3.0	0.14	5.0	0.39	7.0	0.76	9.0	1.26
3.2	0.16	5.2	0.42	7.2	0.80	9.2	1.31
3.4	0.18	5.4	0.45	7.4	0.85	9.4	1.37
3.6	0.20	5.6	0.49	7.6	0.90	9.6	1.43
3.8	0.22	5.8	0.52	7.8	0.94	9.8	1.49

The usual problem in practice consists in calculating the friction loss h_f between any two given points in a pipe for a given velocity v; or, conversely, to determine the velocity which will occur with a given loss of head h_f.

28. Formulas for Friction Loss in Pipes. A great number of experiments have been made to determine the friction loss in the flow of water through pipes. The results show great variations due to many causes, chief of which is the variation in the character of the pipe as to material, degree of roughness of the interior, diameter, etc. Consequently much less accuracy is possible in the estimation of the flow of water through pipes than through orifices or over weirs. Theory is of very little assistance here, and the only practicable method of calculation is to express by some formula the approximate law of variation in friction, and then use coefficients as determined from experiments.

Results of experiments show that the friction loss in a pipe is approximately proportional to the length of the pipe and to the square of the velocity of the water, and is inversely proportional to the cross-section of the pipe divided by its circumference. If we let

h_f = loss by friction between any two points;
l = length of pipe between same two points;
v = velocity of water in pipe;
r = ratio of cross-section to circumference of pipe, called the "hydraulic mean radius",

we then have, according to the above law,

$$h_f = \frac{v^2 l}{r} \times k$$

where k is some coefficient.

It is usual to write this formula so as to express directly the value of v. By solving for v we have

$$v = \sqrt{r\frac{h}{l}} \times \sqrt{k}$$

Putting C for $\sqrt{k}$ we may write

$$v = \mathrm{C}\sqrt{r\frac{h}{l}} \qquad (30)$$

which is known as the Chezy formula. The values of r, h, and l are to be expressed in feet, and the result will give v in feet per second.

The above formula may be used for all kinds of pipe by using a suitable value of C as determined by experiments on similar pipe. For ordinary cast iron pipe the value of C varies from about 100 for pipes 1 or 2 inches in diameter to 140 or 150 for pipes 4 or 5 feet in diameter. Various diagrams and formulas for C have been devised for cast iron pipe, all of which are more or less unsatisfactory. Mr. Hamilton Smith has constructed a diagram which is probably as satisfactory as any now in use. This diagram is not entirely convenient in form, and instead of it we give below an extended table giving the actual velocities of flow v for various diameters of pipe and various losses of head for a length of 100 feet for pipes from $\frac{3}{4}$ in. to 3 in. in diameter, and for a length of 1,000 feet for larger pipes. This table is very convenient to use in calculations, as the desired velocity or loss of head can be seen at a glance.

TABLE NO. 12.

Discharge, Friction Head, and Velocity of Flow Through Smooth Pipes such as Cast Iron.

Discharge, Gals. per Minute.	¾-inch Pipe.		1-inch Pipe.		1½-inch Pipe.	
	Loss of Head, Feet per 100 Feet.	Velocity, Feet per Second.	Loss of Head, Feet per 100 Feet.	Velocity, Feet per Second.	Loss of Head, Feet per 100 Feet.	Velocity, Feet per Second.
1	0.5	0.72	0.02	0.41		
2	2.0	1.4	0.6	0.82		
3	4.0	2.2	1.1	1.2		
4	7.2	2.9	1.8	1.6		
5	11.0	3.6	2.6	2.0		
6	15.0	4.3	3.6	2.4		
7	20.4	5.1	4.8	2.9		
8	25.5	5.8	6.2	3.3		
9	32.0	6.5	7.7	3.7		
10	39.0	7.2	9.4	4.1	1.1	1.8
12			13.0	4.9	1.6	2.2
14			17.1	5.7	2.2	2.5
16			21.8	6.5	2.8	2.9
18			27.1	7.3	3.5	3.3
20			33.0	8.2	4.3	3.6
30					9.5	5.4
40					16.0	7.2
50					24.0	9.1
60					34.0	10.9
70					45.0	12.7

TABLE NO. 12.—Continued.

Dicharge. Gals. per Minute.	2-inch Pipe.		2½-inch Pipe.		3-inch Pipe.	
	Loss of Head, Feet per 100 Feet.	Velocity, Feet per Second.	Loss of Head, Feet per 100 Feet.	Velocity, Feet per Second.	Loss of Head, Feet per 100 Feet.	Velocity, Feet per Second.
10	.4	1.0	0.1	0.65	0.05	0.45
20	1.2	2.0	0.4	1.3	0.2	0.90
30	2.4	3.1	0.8	1.9	0.4	1.4
40	4.0	4.1	1.4	2.6	0.7	1.8
50	6.1	5.1	2.1	3.3	1.0	2.3
60	8.6	6.1	2.9	3.9	1.4	2.7
70	11.5	7.1	3.9	4.6	1.8	3.2
80	14.8	8.2	5.0	5.2	2.3	3.6
90	18.4	9.2	6.3	5.9	2.8	4.1
100	22.2	10.2	7.7	6.5	3.4	4.5
120			10.8	7.8	4.8	5.4
140			14.3	9.1	6.3	6.3
160			18.3	10.4	8.0	7.2
180			22.7	11.8	9.9	8.1
200			27.5	13.1	12.0	9.0
250					18.0	11.3
300					25.0	13.6

TABLE NO. 12.—Continued.

Discharge, Gals. per Minute.	4-inch Pipe.		6-inch Pipe.		8-inch Pipe.	
	Loss of Head, Feet per 1,000 Feet.	Velocity, Feet per Second.	Loss of Head, Feet per 1,000 Feet.	Velocity, Feet per Second.	Loss of Head, Feet per 1,000 Feet.	Velocity, Feet per Second.
50	2.3	1.3				
75	5.2	1.9				
100	8.7	2.5	1.2	1.1		
125	13.1	3.2	1.8	1.4		
150	18.3	3.8	2.5	1.7		
175	24.3	4.5	3.3	2.0		
200	31.0	5.1	4.2	2.3	1.1	1.3
250	46.5	6.4	6.3	2.8	1.6	1.6
300	65.0	7.7	8.9	3.4	2.2	1.9
350			11.9	4.0	2.9	2.2
400			15.1	4.5	3.7	2.6
450			18.7	5.1	4.6	3.9
500			22.7	5.7	5.6	3.2
600			31.8	6.8	7.9	3.8
700			42.2	7.9	10.5	4.5
800			54.0	9.1	13.4	5.1
900					16.6	5.8
1000					20.2	6.4
1100					24.1	7.0

TABLE NO 12.—Continued.

Discharge, Gals. per Minute.	10-inch Pipe.		12-inch Pipe.		16-inch Pipe.	
	Loss of Head, Feet per 1,000 Feet.	Velocity, Feet per Second.	Loss of Head, Feet per 1,000 Feet.	Velocity, Feet per Second.	Loss of Head, Feet per 1,000 Feet.	Velocity, Feet per Second.
200	.35	.82	.14	.57		
300	.73	1.2	.30	.85		
400	1.24	1.6	.51	1.1		
500	1.87	2.0	.78	1.4	.18	.80
600	2.6	2.4	1.10	1.7	.26	.96
700	3.5	2.9	1.45	2.0	.34	1.1
800	4.4	3.3	1.82	2.3	.43	1.3
900	5.5	3.7	2.3	2.6	.54	1.4
1000	6.7	4.1	2.8	2.8	.66	1.6
1100	8.0	4.5	3.3	3.1	.78	1.8
1200	9.4	4.9	3.9	3.4	.92	1.9
1300	10.9	5.3	4.5	3.7	1.06	2.1
1400	12.6	5.7	5.1	4.0	1.22	2.2
1500			5.8	4.2	1.38	2.4
1600			6.5	4.5	1.55	2.6
1700			7.3	4.8	1.74	2.7
1800			8.1	5.1	1.93	2.9
1900			9.0	5.4	2.1	3.0
2000			9.9	5.7	2.3	3.2
2200			11.7	6.2	2.8	3.5
2400					3.3	3.8
2600					3.8	4.2
2800					4.4	4.5
3000					5.0	4.8
3500					6.6	5.6

Examples. 1. What is the head lost in friction due to the flow of 800 gallons per minute in a 6-inch pipe?

From Table No. 12 we see that the friction head in a 6-inch pipe for a flow of 800 gals. per min. is 54.0 ft. for each 1,000 ft. of pipe. Ans.

2. What size of pipe will be required to convey 700 gallons of water per minute a distance of 8,000 feet with a total loss of head of 40 feet?

The loss of head per 1,000 ft. is $40 \div 8 = 5$ ft. From the table we find that for a discharge of 700 gallons per min. the loss of head in an 8-in. pipe is 10.5 ft. per 1,000, and in a 10-in. pipe it is 3.5 ft. A 10-in. pipe would then be required if the assumed loss is not to be exceeded. Ans.

3. If a town is supplied with water from an elevated reservoir through a pipe line 15,000 feet long, how high must the

TABLE NO. 12.—Continued.

Discharge, Gals. per Minute.	20-inch Pipe.		24-inch Pipe.		30-inch Pipe.	
	Loss of Head, Feet per 1,000 Feet.	Velocity, Feet per Second.	Loss of Head, Feet per 1,000 Feet.	Velocity, Feet per Second.	Loss of Head, Feet per 1,000 Feet.	Velocity, Feet per Second.
1000	.23	1.0	.08	.71		
1200	.32	1.2	.12	.85		
1400	.42	1.4	.16	.99		
1600	.52	1.6	.20	1.1		
1800	.64	1.8	.25	1.3		
2000	.77	2.0	.31	1.4	.10	.91
2200	.92	2.2	.37	1.6	.12	1.00
2400	1.08	2.5	.43	1.7	.14	1.09
2600	1.25	2.7	.50	1.8	.17	1.18
2800	1.43	2.9	.58	2.0	.19	1.27
3000	1.62	3.1	.66	2.1	.22	1.36
3200	1.82	3.3	.74	2.3	.24	1.45
3400	2.04	3.5	.83	2.4	.27	1.55
3600	2.27	3.7	.92	2.5	.30	1.64
3800	2.51	3.9	1.02	2.7	.33	1.73
4000	2.76	4.1	1.12	2.8	.36	1.82
4500	3.43	4.6	1.39	3.2	.46	2.05
5000	4.16	5.1	1.68	3.5	.56	2.27
5500	4.96	5.6	2.00	3.9	.67	2.50
6000	5.80	6.1	2.35	4.3	.78	2.73
6500					.90	2.96
7000					1.03	3.18
7500					1.17	3.41
8000					1.32	3.64
9000					1.64	4.09
10000					2.00	4.55

reservoir be above the town and what must be the size of the pipe line so that the pressure of water in the distributing pipes be not less than 60 pounds per sq. in., equivalent to 60 × 2.3 = 138 ft. head. The amount of water required is 1,800 gals. per min.

This problem has several solutions since various sizes of pipe may be assumed and the reservoir placed at the elevation to furnish the necessary pressure. An examination of Table No. 12 shows that to deliver 1,800 gals. per min. a 12-in. pipe would consume in friction 8.1 ft. of head per 1,000 ft., a 16-in. pipe would consume only 1.93 ft. per 1,000 ft., and a 20-in. pipe only .64 ft. No value is given for a 10-in. pipe, but it would evidently be 20 feet or more per 1,000, which would give a total loss for 15,000 ft. of 300 feet, a loss which would ordinarily be impracticable

If we use a 12-in. pipe the total loss in friction will be 8.1 × 15 = 121.5 ft. The velocity of flow will be 5.1 ft. per sec. and the necessary velocity head, h_v, will be, by Table No. 11, .4 ft. The total head = 121.5 + .4 = 121.9 ft., and the necessary elevation of the reservoir = 138 + 121.9 = 259.9 ft. above the town.

If a 16-inch pipe be assumed, the friction loss = 1.93 × 15 = 28.9 ft., the velocity head = .1 ft., and the total head = 29 ft. Elevation of reservoir = 29.0 + 138 = 167 ft.

If a 20-inch pipe be used, the friction head = .64 × 15 = 9.6 ft., the velocity head is less than .1 ft. and may be neglected. The required height of reservoir = 147.6 ft.

Still larger sizes will give still lower elevations for the reservoir, but it is evident that the reservoir in any case must have an elevation somewhat greater than 138 ft.

From the above results we see that a 12-in. pipe requires the reservoir to be at an elevation of 259.9 ft., a 16-in. pipe requires an elevation of 167 ft., and a 20-in. pipe an elevation of 147.6 ft. The proper size to use would be that size which would give the cheaper construction for the pipe and reservoir combined.

29. The Hydraulic Grade Line. Referring again to Fig. 27, it will be seen that the drop in pressure between B and C will be proportional to the length of the pipe from B to C, and if we have a long pipe with several open tubes attached to it like *b*B and *c*C, the level of the water in them would be lower and lower as we proceed along the pipe, the drop being uniform so long as the pipe is of the same size and kind, the amount of the drop per 1,000 feet being given in Table No. 12. If now a line were drawn from E through the points *b*, *c*, etc., so that the height of this line above the pipe would represent the pressures in it, this line would be called the "hydraulic grade line" for the pipe under the given conditions. It is convenient in various problems to construct such a grade line. Its position will evidently vary with the velocity of the flow and will be a horizontal line when the water is still, and always a straight line for a pipe of uniform conditions.

30. Siphons. If in any case a pipe line rise above this hydraulic grade line, as shown in Fig. 28, the pressure in such portion of the pipe will be less than atmospheric, the pressure measured by the grade line as described above referring in all cases to the pressure in excess of the usual atmospheric pressure. That portion of the pipe BC lying above the grade line is called a *siphon*. The greatest height above the grade line which it is practicable to operate a siphon is considerably less than the height of the water barometer given in Art. 4. Evidently since the velocity of flow, and hence the hydraulic grade line, can be varied by varying the opening at D, a pipe which may act as a siphon at one time may not so act at another. Thus in the figure, if valve D be nearly closed so that the flow is reduced and hence also the frictional loss,

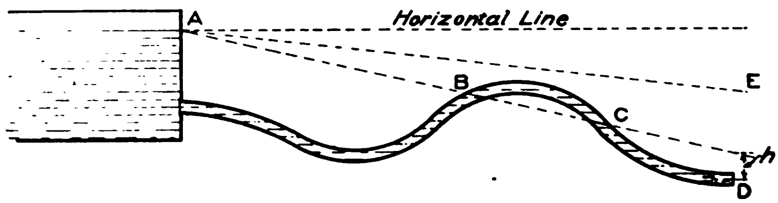


Fig. 28.

the grade line will rise to some position such as AE and there will be pressure in excess of atmospheric at all points.

31. Flow Through Special Forms of Pipes. *Riveted Pipe.* The friction loss in riveted pipes depends upon the thickness of the plates and the manner of making the joints. Experiments on this class of pipes are not sufficiently numerous to enable any general expression to be formulated, so that in the design of such pipes the selection of coefficients must be made by reference to the experimental data. In general it is found that the coefficient C, of equation 30, changes little with change in diameter or velocity, and in this respect exhibits considerable difference from its variation in cast-iron pipe. For ordinary velocities the value of C, for new pipe appears to range between 100 and 115. A value of 100 is as great as it is well to use.

32. Wood Stave Pipe. Few experiments have been made on this class of pipe although it has been used quite extensively in the West. The pipe is usually quite smooth and not subject to deterioration on the interior, so that its discharging capacity is high. For ordinary velocities the value of C, equation 30, may be taken at 110.

33. Fire Hose. In making provisions for fire protection it becomes necessary to estimate the effectiveness of a stream of water when led through a given length of hose for a given pressure at the hydrant, or to find what pressure is required to throw a stream a given height or a given distance. The usual size of fire hose is $2\frac{1}{2}$ inches. At the end of the hose is attached a nozzle of a diameter usually of 1 in., $1\frac{1}{8}$ in., or $1\frac{1}{4}$ in., which partly controls the amount and pressure of the water discharged. If there were no friction in the hose the water could be thrown nearly to a height corresponding to the pressure head at the hydrant, but the hose friction is very great, and two or three hundred feet of hose will cut down the effective pressure often more than one-half. Evidently the more rapid the flow through the hose the greater the friction loss, hence if the nozzle is small so that the discharge will be small, the effective pressure near the nozzle will be greater than with a large nozzle and large discharge. Hence a higher stream can be thrown through a small nozzle with a given hydrant pressure and length of hose than through a large nozzle, although the stream is not so effective in quenching a fire as the larger stream.

In Table No. 13 are given the necessary data for estimating the loss of head and effectiveness of fire streams for various pressures and for three sizes of nozzles

In the table, page 44, the pressure given is that at the nozzle instead of at the hydrant. To get the latter, it is necessary to add to the nozzle pressure the head lost in the hose. The result will be the hydrant pressure, providing nozzle and hydrant are at same level. If not, then a correction would need to be made for this difference in elevation. The vertical height and horizontal distances are to be measured from the nozzle. The heads are given in pounds per square inch, which is the customary unit in this class of work. To reduce to feet of head multiply pounds pressure by 2.3.

Examples. 1. What hydrant pressure will be required to throw a stream of water 75 feet vertically through a $1\frac{1}{8}$-in. nozzle and 300 feet of hose.

In the table for the $1\frac{1}{8}$-in. nozzle we see that for a height of 75 feet the loss of head per 100 feet of hose is 20 pounds, and the pressure at the nozzle is (in first column of table) 50 pounds. The

TABLE NO. 13.

Hose and Fire-Stream Data.

Pressure at Nozzle (Base of Play-pipe).	1-inch Smooth Nozzle.					1⅛-inch Smooth Nozzle.					1¼-inch Smooth Nozzle.				
	Discharge in Gallons per Minute.	Loss of Head per 100 Feet of Ordinary Hose.	Vertical Height of Jet for Good Fire-streams.	Maximum Horizontal Distance for Good Fire-streams.	Extreme Drops at Level of Nozzle.	Discharge in Gallons per Minute.	Loss of Head per 100 Feet of Ordinary Hose.	Vertical Height of Jet for Good Fire-streams.	Maximum Horizontal Distance for Good Fire-streams.	Extreme Drops at Level of Nozzle.	Discharge in Gallons per Minute.	Loss of Head per 100 Feet of Ordinary Hose.	Vertical Height of Jet for Good Fire-streams.	Maximum Horizontal Distance for Good Fire-streams.	Extreme Drops at Level of Nozzle.
lb.		lb.	ft.	ft.	ft.		lb.	ft.	ft.	ft.		lb.	ft.	ft.	ft.
20	132	5	35	37	77	168	8	36	38	80	209	12	37	40	83
30	161	7	51	47	109	206	12	52	50	115	256	19	53	54	119
40	186	10	64	55	133	238	16	65	59	142	296	25	67	63	148
50	208	12	73	61	152	266	20	75	66	162	331	31	77	70	169
60	228	15	79	67	167	291	24	83	72	178	363	37	85	76	186
70	246	17	85	72	179	314	28	88	77	191	392	43	91	81	200
80	263	20	89	76	189	336	32	92	81	203	419	49	95	85	213
90	279	22	92	80	197	356	36	96	85	214	444	55	99	90	225
100	295	25	96	83	205	376	40	99	89	224	468	62	101	93	236

hydrant pressure will then be $50 + (20 \times 3) = 110$ pounds per square inch. The discharge will be about 266 gallons per minute.

2. With a hydrant pressure of 100 pounds, what will be the discharge through 250 feet of hose with a 1¼-in. nozzle, and how high can such a stream be thrown with effectiveness?

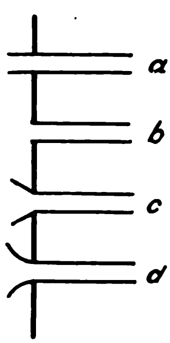


Fig. 29.

This problem must be solved by trial. In the table for 1¼-in. nozzles, we see that for a discharge of 269 gallons the nozzle pressure is 40 pounds, and the loss of head per 100 feet of hose is 25 pounds; for a discharge of 331 gallons the nozzle pressure is 50 pounds, and the loss of head per 100 feet is 31 pounds, etc. We have given the head of 100 pounds, which must equal the sum of the nozzle pressure and the loss in the hose. If we try the first value for discharge, we have a nozzle pressure of 40 pounds and a total loss in the hose of $25 \times 2.5 = 62.5$ pounds, or a total of $40 + 62.5 = 102.5$ pounds. This being a little more than the total available head, it is evident that we have assumed too high a discharge. The next lower value is 256 gallons, giving a nozzle pressure of 30 pounds and a total hose loss of 19 × 2.5 or 47.5 pounds, giving a total of 30 + 47.5 = 77.5 pounds. Evidently the correct value is somewhere between 296 and 256, and further that it is but very little below the former value. For a total change in discharge of 40 gallons we have a change in total head of 102.5 – 77.5 or 25 pounds. Hence for a change of 2.5 pounds the discharge will vary about $\frac{1}{10}$ of 40 gallons, or 4 gallons. The discharge may then be taken as 292 gallons per minute. The effective height will be between 67 feet and 53 feet, but only a little less than the former value, say 65 feet. This is as close an estimate as the conditions of the problem will warrant, since the hose friction is a factor that varies greatly according to the character of the hose.

34. Minor Losses of Head in Pipes. In most of the following formulas the quantity $\frac{v^2}{2g}$ occurs. For given values of v this quantity can readily be taken from Table No. 11.

Loss of Head at Entrance. This is expressed by the formula

$$h = \left(\frac{1}{c^2} - 1\right)\frac{v^2}{2g}, \qquad (31)$$

where v = velocity in the pipe, and c is the coefficient of discharge. For various forms at entrance, as shown in Fig. 29, we have the following values:

	c.	$\frac{1}{c^2} - 1$.
Pipe projecting into reservoir, Fig. (*a*)	.72	.93
End of pipe flush with reservoir, Fig. (*b*)	.82	.49
Conical or bell-shaped mouth, Fig. (*c*) or (*d*)	.93 to .98	.15 to .04

Loss of Head at Bends. For 90° bends this is equal to

$$h = n\frac{v^2}{2g} \qquad (32)$$

in which n has the following values according to the ratio of the radius of the pipe r to the radius of curvature R:

$\frac{r}{R}$	.1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
n	.13	.14	.16	.21	.29	.44	.66	.98	1.41	1.98

Loss of Head in Valves. Weisbach's experiments on small gate-valves gave values for n in the expression $h = n\frac{v^2}{2g}$ as follows:

Ratio of height of opening to diameter	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{8}$
Values of n	.07	.26	.81	2.1	5.5	17	98

In applying the above formula v is the velocity in the pipe.

For a throttle-valve placed at various angles θ with the axis of the pipe, Weisbach found the following values of n:

θ..	5°	10°	20°	30°	40°	50°	60°	65°	70°
n..	.24	.52	1.5	3.9	11	33	118	256	750

Experiments on large gate-valves have been made by Kuichling and by J. W. Smith. The following table gives values of the coefficient c in the expression $Q = cA\sqrt{2gh}$. In this expression A is the area of the opening, h is the head lost in the valve, Q is the rate of discharge.

TABLE NO. 14.

Coefficients for Large Gate-Valves.

Ratio of height of opening to diameter	.05	.1	.2	.3	.4	.5	.6	.7	.8
Ratio of area of opening to total area	.05	.10	.23	.36	.48	.60	.71	.81	.89
Coefficient c for 24-in. valve	1.7	1.0	.72	.70	.77	.92	1.2	1.6	
Coefficient c for 30-in. valve	1.2	.9	.83	.82	.84	.90	1.05	1.35	2.1

Example. If a pump draws water from a pipe projecting into a reservoir what will be the loss of head at entrance, the velocity of water in the pipe being 6 feet per second

Using equation 31 of Art. 34 the value of $(\frac{1}{c^2} - 1)$ is, for this case, about .93. The loss of head is then $.93 \times \frac{v^2}{2g}$ which by Table No. 11 $= .93 \times .56$ or .52 feet. Ans.

If the pipe is flush with the reservoir the loss of head will be only $.49 \times .56$, or .27 feet.

Finally, if the pipe is enlarged to a bell-mouth or conical form the loss of head will be very small, say $.10 \times .56$ or .056 feet.

FLOW OF WATER IN OPEN CHANNELS.

35. General Formula. Where water flows in an open channel like a ditch, or a concrete, brick or tile sewer flowing less than full, the inclination of such channel is what furnishes the necessary fall or head to the water for overcoming friction. In this case there is no pressure at any point, and the loss of head from point to point will be the difference in level of the water surface between the given points. This difference in level, or head, after the flow has become steady is equal to the loss of head due to friction in the same distance.

The frictional loss in open channels is expressed by the same general formula as that used for pipes in Art. 28. It is

$$v = c\sqrt{rs} \qquad (33)$$

in which as before

v = velocity in feet per second,
c = a coefficient,
r = hydraulic mean radius = the cross-section of the actual stream of water divided by that part of the perimeter that is under water ("wetted perimeter").
s = slope of channel, or ratio of fall to length $= \dfrac{h}{l}$

For open channels the value of c varies much more than for pipes, as the nature of the channel varies more. Thus the channel may be a smooth tile sewer where c may be 100 or more, which is about the same as for iron pipe; or the channel may be a rough natural water-course for which the value of c will be only 30 or 40. Estimates of flow in very rough channels are obviously subject to great uncertainties, but for sewers and open masonry drains or conduits, estimates may be quite closely made, as the values of c have been quite well determined.

For convenience the value of c has been expressed in a formula, called Kutter's formula, in which the condition of the channel is taken account of by a special coefficient n, called the coefficient of roughness. This formula for ordinary cases is

$$c = \frac{\frac{1.8}{n} + 45}{1 + \frac{45n}{\sqrt{r}}} \qquad (34)$$

in which r = hydraulic mean radius in feet, and n = coefficient of roughness, varying from a value of about .009 for smooth plank to .030 for natural channels full of stone, etc.

The following are the values of n usually assumed for the various surfaces mentioned:

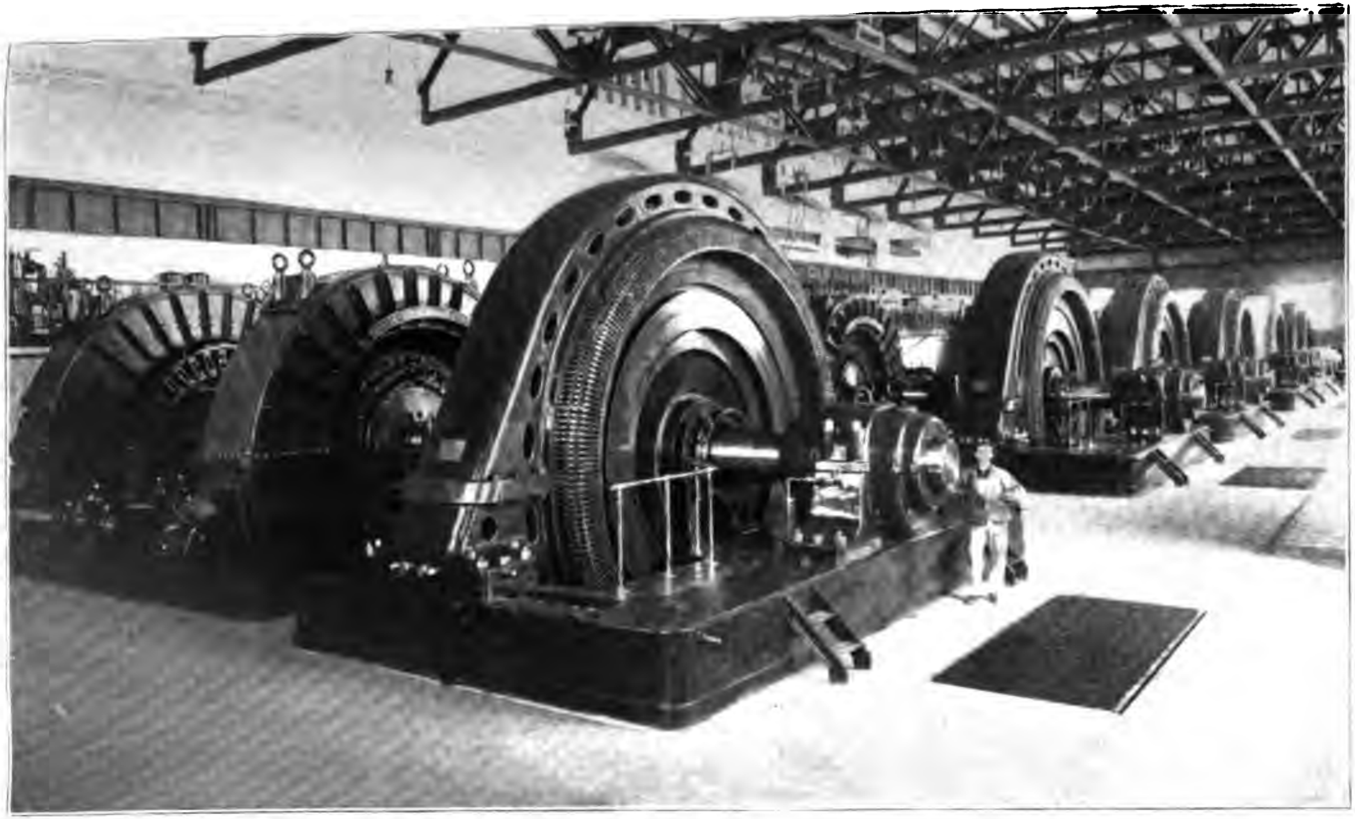
	n
Channels of well-planed timber	.009
" " neat cement or of very smooth pipe	.010
" " unplaned timber or ordinary pipe	.012
" " smooth ashlar masonry or brickwork	.013
" " ordinary brickwork	.015
" " rubble masonry	.017
" in earth free from obstructions	.020 to .025
" with detritus or aquatic plants	.030

After selecting the value of n, the value of c can readily be obtained from Table No. 15.

TABLE NO. 15.

Values of c in Kutter's Formula, for Various Values of n.

r in Feet.	Values of n.									
	.009	.010	.011	.012	.013	.015	.017	.020	.025	.030
.1	108	94	82	73	65	53	45	35	26	20
.2	129	113	100	89	80	66	56	45	34	26
.3	142	124	111	99	90	75	63	52	38	30
.4	150	132	118	106	96	80	69	56	42	34
.5	157	139	124	111	101	85	73	60	45	36
.6	162	143	128	116	105	89	76	63	48	38
.7	166	147	132	119	109	92	79	65	50	40
.8	170	151	135	122	112	95	82	68	52	42
.9	173	154	138	125	114	97	84	70	54	43
1.0	175	156	140	127	116	99	86	71	55	45
1.2	180	160	145	131	120	103	89	74	58	47
1.4	184	164	148	135	124	106	92	77	60	49
1.6	187	167	151	137	126	108	94	79	62	51
1.8	189	169	153	140	129	110	97	81	64	53
2.0	191	172	155	142	130	112	98	83	65	54
2.5	196	176	160	146	135	116	02	86	69	57
3.0	199	179	163	149	138	119	105	89	71	59
3.5	202	182	163	152	140	122	107	91	73	61
4.0	204	184	168	154	143	124	110	93	75	63
4.5	206	186	170	156	144	126	111	95	77	64
5.0	208	188	172	158	146	127	113	97	78	66



INTERIOR OF GENERATING STATION OF THE ONTARIO POWER COMPANY, NIAGARA FALLS, ONT.
The generators are direct-connected to horizontal double turbines.

36. The Hydraulic Mean Radius r. As before explained, this is a name given to the quotient found by dividing the actual cross-section of a stream of water by the "wetted perimeter," or that part of the perimeter of the cross-section of the channel that is under water. In the case of a pipe flowing full, of diameter d, the cross-section is $\frac{1}{4}\pi d^2$ and the perimeter is πd, hence the value of r is $\frac{1}{4}\pi d^2 \div \pi d = \frac{1}{4}d$. For a pipe flowing half full it is, similarly, $\frac{1}{8}\pi d^2 \div \frac{1}{2}\pi d$ or $\frac{1}{4}d$, the same as when flowing full. When less than half full the cross-section of the stream falls off more rapidly than the wetted perimeter, so that the value of r decreases. Hence we see from equation 33 that the velocity also falls off.

For any given form of channel filled to a given point the value of r can readily be found by plotting the cross-section to a large scale and measuring the area and the wetted perimeter.

Example. What will be the velocity and discharge of water flowing in a concrete channel 4 ft. wide and 3 ft. deep and having a slope of 1 ft. per 1,000 ft.?

Equation 33 must be used. We will first get the values of r and s. The value of r is equal to the cross-section of the stream of water divided by the wetted perimeter $= \dfrac{3 \times 4}{4 + 3 + 3} = 1.2$ ft. The slope $s = \dfrac{1}{1,000} = .001$. The value of c is to be obtained from Table No. 15, n being taken at .013, say, the same as for brickwork. For $n = .013$ and $r = 1.2$ Table No. 15 gives $c = 120$. Substituting then in equation 33 we have $v = 120 \times \sqrt{1.2 \times .001} = 4.16$ ft. per sec. The discharge will be $4.16 \times 4 \times 3 = 49.92$ cu. ft. per sec. Ans.

37. Flow Through Ordinary Sewers. Sewers are usually constructed of vitrified earthen pipe or of brick or concrete. For the former material the value of n in equation 34 is usually taken at .013, and for brick and concrete about .015. If the concrete is smoothly finished n may be taken at .013.

The following Table No. 16 gives the velocities and discharges for circular sewers flowing full. For sewers flowing half full the velocity will be the same and the discharge one-half of the given values.

TABLE NO. 16.

Velocity and Discharge for Pipe Sewers (n = .013);

Velocity in Feet per Second (V); Discharge in Cubic Feet Per Second (Q).

(For n = .011 add 20 per cent.)
(For n = .015 subtract 16 per cent.)

Fall of Sewer, in Feet per 100 ft.	4-inch.		6-inch.		8-inch.		10-inch.		12-inch.		15-inch.		18-inch.	
	V	Q	V	Q	V	Q	V	Q	V	Q	V	Q	V	Q
10.	5.75	.50	7.99	1.57	10.04	3.50	11.94	6.51	13.78	10.78	16.24	19.93	18.59	32.86
5.	4.06	.35	5.64	1.11	7.09	2.48	8.43	4.60	9.70	7.62	11.48	14.08	13.13	23.22
4.	3.63	.32	5.05	.99	6.34	2.21	7.54	4.11	8.65	6.80	10.26	12.59	11.74	20.73
3.	3.15	.27	4.25	.83	5.49	1.92	6.53	3.56	7.51	5.90	8.89	10.91	10.17	17.97
2.	2.57	.22	3.56	.70	4.48	1.56	5.33	2.91	6.13	4.82	7.25	8.90	8.30	14.67
1.	1.82	.16	2.52	.49	3.17	1.11	3.77	2.06	4.33	3.40	5.13	6.30	5.87	10.38
.8	1.61	.14	2.25	.44	2.83	.99	3.37	1.84	3.87	3.04	4.59	5.63	5.25	9.28
.6	1.38	.12	1.95	.38	2.45	.86	2.92	1.59	3.35	2.64	3.97	4.89	4.55	8.04
.4			1.59	.31	2.00	.69	2.38	1.30	2.74	2.15	3.24	3.97	3.70	6.55
.2					1.40	.49	1.67	.91	1.91	1.51	2.27	2.79	2.60	4.60
.1							1.17	.64	1.35	1.06	1.60	1.96	1.83	3.34
.09											1.51	1.86	1.73	3.06
.08													1.63	2.88
.07													1.53	2.69

TABLE NO. 16.—Continued.

Fall of Sewer, in Feet per 100 ft.	20-inch.		22-inch.		24-inch.		30-inch.		33-inch.		36-inch.	
	V	Q	V	Q	V	Q	V	O	V	Q	V	Q
10.	20.08	43.8	21.51	56.8	22.91	72.0	26.84	131.7	28.69	170.3	30.46	215.3
5.	14.18	30.9	15.20	40.1	16.19	50.9	18.97	93.1	20.27	120.4	21.54	152.3
4.	12.69	27.7	13.59	35.9	14.47	45.5	16.96	83.8	18.13	107.7	19.26	136.5
3.	10.98	24.0	11.77	31.1	12.53	39.4	14.69	72.1	15.70	93.6	16.68	118.0
2.	8.97	19.6	9.61	25.4	10.23	32.2	11.99	58.9	12.82	76.1	13.62	96.3
1.	6.34	13.8	6.79	17.9	7.24	23.3	8.48	41.6	9.06	53.8	9.63	68.1
.8	5.67	12.4	6.07	16.0	6.47	20.8	7.58	37.2	8.11	48.1	8.61	60.9
.6	4.91	10.7	5.26	13.9	5.60	19.6	6.57	32.2	7.02	41.7	7.46	52.7
.4	4.00	8.7	4.29	11.3	4.56	14.3	5.35	26.3	5.72	34.0	6.08	43.0
.2	2.81	6.1	3.01	7.9	3.21	10.1	3.76	18.5	4.02	23.9	4.28	30.2
.1	1.98	4.3	2.12	5.6	2.26	7.1	2.66	13.0	2.84	16.9	3.02	21.3
.09	1.87	4.1	2.01	5.3	2.14	6.7	2.51	12.3	2.69	16.0	2.86	20.3
.08	1.76	3.8	1.89	5.0	2.02	6.3	2.37	11.6	2.53	15.0	2.69	19.0
.07	1.64	3.6	1.76	4.6	1.88	5.9	2.20	10.8	2.36	14.0	2.51	17.7
.06	1.51	3.3	1.63	4.3	1.73	5.4	2.04	10.0	2.18	12.9	2.32	16.4
.05			1.48	3.9	1.58	5.0	1.86	9.1	1.99	11.8	2.11	14.9
.04			1.32	3.5	1.40	4.4	1.65	8.1	1.77	10.5	1.88	13.3
.03					1.20	3.8	1.40	6.9	1.52	9.0	1.62	11.4
.02					0.96	3.1	1.13	5.6	1.22	7.2	1.30	9.2

TABLE NO. 17.

Velocity and Discharge for Brick and Concrete Sewers ($n = .015$); Velocity in Feet per Second (V); Discharge in Cubic Feet Per Second (Q).

(For n = .013 add 19 per cent.)
(For n = .017 subtract 13 per cent.)

Fall of Sewer, in Feet per 100 ft.	33-inch.		36-inch.		42-inch.		4-foot.	
	V	Q	V	Q	V	Q	V	Q
.5	17.17	102.0	18.27	129.3	20.37	196.1	22.36	281.1
.4	15.36	91.2	16.34	115.5	18.21	175.3	20.00	251.3
.3	13.30	79.0	14.15	100.0	15.77	151.8	13.31	217.6
.2	10.85	64.5	11.55	81.7	12.88	123.9	14.13	177.6
.1	7.68	45.6	8.16	57.7	8.90	87.6	9.99	125.6
.8	6.86	40.7	7.30	51.6	8.14	78.3	8.93	112.3
.6	5.94	35.2	6.32	44.6	7.04	67.8	7.73	97.2
.4	4.84	28.8	5.15	36.4	5.75	54.0	6.31	79.3
.2	3.41	20.3	3.63	25.7	4.05	39.0	4.45	55.9
.1	2.40	14.3	2.52	18.1	2.85	27.5	3.13	39.4
.09	2.27	13.5	2.42	17.1	2.70	26.0	2.97	37.3
.08	2.14	12.7	2.28	16.1	2.55	24.5	2.80	35.2
.07	2.00	11.9	2.13	15.0	2.38	22.9	2.61	32.9
.06	1.85	11.0	1.97	13.9	2.20	21.1	2.42	30.4
.05	1.68	10.0	1.79	12.6	1.95	18.8	2.20	27.6
.04	1.49	8.9	1.59	11.3	1.78	17.1	1.96	24.6
.03	1.28	7.6	1.37	9.7	1.53	14.7	1.68	21.2
.02					1.23	11.9	1.36	17.1
.15							1.16	14.6

TABLE NO. 17.—Continued.

Fall of Sewer, in Feet per 100 ft.	5-foot.		6-foot.		8-foot.		10-foot.	
	V	Q	V	Q	V	Q	V	Q
5.	26.05	512						
4.	23.30	457	26.34	745				
3.	20.17	396	22.81	645				
2.	16.47	323	18.62	527	22.53	1133	26.03	2045
1.	11.64	228	13.17	372	15.93	801	18.41	1446
.6	10.41	204	11.78	333	14.25	716	16.46	1293
.8	9.01	177	10.19	288	12.33	620	14.25	1119
.4	7.36	144	8.32	235	10.07	506	11.63	914
.2	5.19	102	5.87	166	7.10	357	8.21	645
.1	3.66	72	4.14	117	5.02	252	5.81	456
.09	3.47	68	3.92	111	4.76	239	5.51	433
.08	3.27	61	3.70	105	4.49	226	5.19	408
.07	3.05	60	3.46	98	4.20	211	4.86	382
.06	2.82	55	3.20	90	3.88	195	4.50	353
.05	2.57	50	2.92	82	3.54	178	4.10	322
.04	2.29	45	2.60	74	3.16	159	3.66	288
.03	1.97	39	2.24	63	2.73	137	3.17	249
.02	1.60	31	1.82	51	2.22	112	2.58	203
.015	1.37	27	1.56	44	1.93	97	2.23	175
.012			1.39	39	1.70	86	1.99	156
.010					1.55	78	1.81	143
.0095					1.25	63	1.77	139
.0090							1.72	135

MEASUREMENT OF THE FLOW OF STREAMS.

38. General Methods. For measuring the flow of a small stream the best method is by the use of a weir constructed of plank and built into a temporary dam of earth. Such weirs can readily be used for streams up to 3 or 4 feet in depth and 40 or 50 feet wide, although streams normally of such size would have flood flows many times greater and which could not be so measured. Where a dam already exists in a stream, observations of the flow over such a dam will give fairly good results when the coefficient of discharge is carefully selected as noted in Art. 25.

Where a weir cannot be used, then the flow must be measured by actually determining the mean velocity of the flow at a given

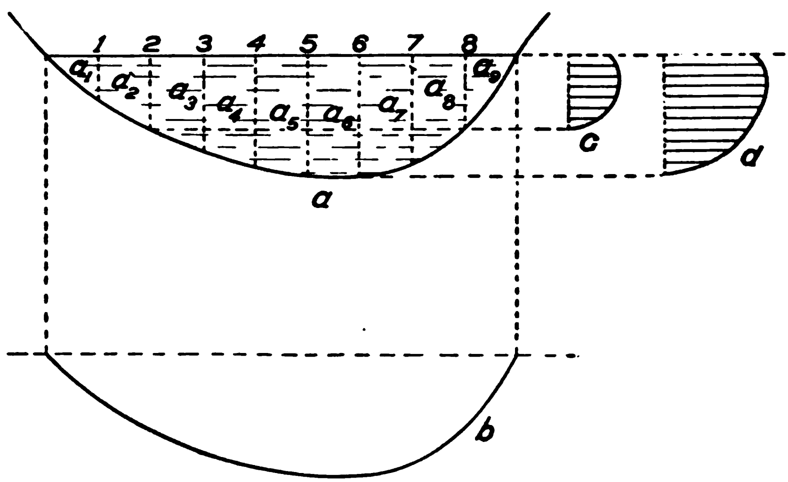


Fig. 30.

section and the area of such cross-section, then the discharge will be equal to the product of these quantities.

39. Variations in Velocity. Owing to the disturbing effect of the bottom and sides of a channel, the velocity of the water will not be the same at all points in a given cross-section. In general the velocity will be greater near the center of a stream than near the edges, and will be greater where the water is deep than where it is shallow. Thus if Fig. 30*a* represents the cross-section of a stream, the velocity of flow along the surface will vary in some such way as is represented in Fig. *b*, being greatest near the deepest parts and very small near the banks. Likewise if we consider the velocities along the vertical section 2 they will vary somewhat as shown in Fig. *c*, and at section 6 they will be as shown in Fig. *d*. In both Figs. *d* and *c* the maximum velocity is shown to be a little below the surface. This is usually the case, although it depends somewhat on the effect of the wind.

From these statements it will be seen that there are great variations in the velocity throughout the cross-section, and therefore the determination of the average velocity is not readily accomplished.

Instead of trying to get the average velocity through the entire cross-section, it is usual to divide the section of the stream into several vertical strips as shown in Fig. *a*. Then get the average velocity and discharge of each strip separately. In doing this a place should be selected where the flow is as uniform and the channel as regular as possible. In case floats are used to get velocities, as described later, it is necessary to establish two sections 100 feet apart or more, between which points the velocities are measured. In either case careful soundings must be taken and an accurate plot made of the cross-section, and the area of each division a_1, a_2, etc., determined. The divisions of the section may be marked by knots or tags on a rope stretched across the channel. The sections having been divided off, it remains to determine the average velocity in each.

40. Use of the Current Meter. The most accurate method of finding the velocity is by means of the current meter, one form of which is illustrated in Fig. 31.

The essential part of the current meter consists in the series of cups mounted on a wheel with vertical axis shown at the left of the vertical rod. This wheel being submerged, is rotated by the current, and the number of revolutions is recorded by an electrical device which may be held in a boat or on shore. The long vane attached to the wheel is to keep the meter always parallel with the current. A heavy weight is attached to the bottom of the rod to keep the meter steady, the whole apparatus being suspended by means of a rope from a boat or bridge. The number of revolutions per minute of the wheel being known, the velocity of the water at the wheel is calculated by multiplying by a coefficient determined by previous experiments with the meter.

The average velocity for any given strip is determined either by getting the velocity along the center of the strip at several different depths and taking the average, or by moving the meter slowly from top to bottom and then back to the top and taking a single reading. Whichever way determined the resulting velocity multiplied by the area of the strip in question equals the discharge of that strip. Then the total discharge equals the sum of the discharges of all the strips.

The coefficient to use in calculating actual water velocities

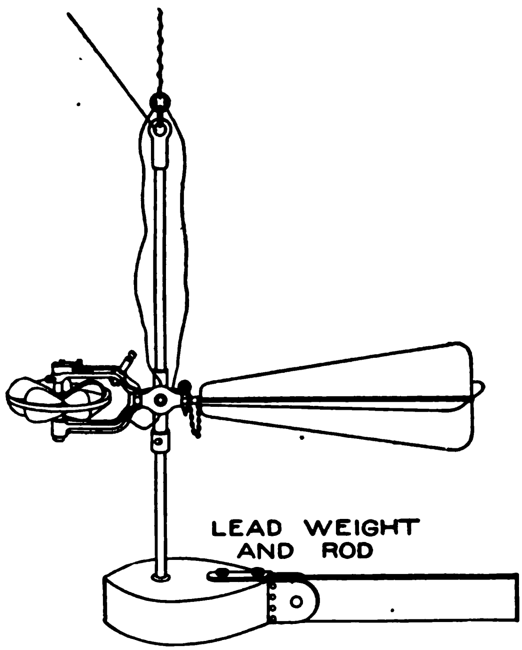


Fig. 31.

from meter readings is determined by a "rating" of the meter. This rating is done by moving the meter at various known velocities through still water in a reservoir, pond, or canal. Then knowing the velocity of the meter through the water and its readings, a rating curve or table of coefficients can be worked out.

41. Use of Floats. Very often a meter is not at hand, and a less accurate method must be employed. That most often used is by means of floats. These are of three kinds—*surface floats*, *subsurface floats*, and *rod floats*. The best form is the rod float.

The *rod float* is a rod of wood, or a tube of tin, which is weighted at one end so that it will float in an upright position and as near to the bottom of the stream as practicable. The float is then placed in the stream at the desired point, and far enough up stream from the upper of two measured cross-sections so that it will acquire the same velocity as the water by the time it reaches such section. The time of its passage from the upper to the lower section is then observed and its velocity deduced therefrom. In this way observations are made for each of the vertical strips in which the stream section is divided. The average velocity of each strip is taken equal to that of the rod itself.

The *surface float* may be made of any convenient form which will be readily seen from the point of observation. Its use will give only the surface velocities of the several strips and not the desired average velocities. To get the average velocity, we may use the approximate formula,

$$\text{Average velocity} = .9 \times \text{surface velocity} \qquad \textbf{(35)}$$

whence the discharge of the several strips can be calculated as before. This method is not so accurate as the use of rod floats and is not to be recommended except for very rough determinations. It is much influenced by the wind, and observations should, if possible, be made on still days.

Sometimes a very rough determination is desired from one or two measurements of velocity. If the surface velocity is measured at a point where it is a maximum (near the center of the stream), then the average velocity for the entire stream may be taken at about $\frac{8}{10}$ of the measured velocity, although the exact value of this coefficient will vary between quite wide limits. The discharge then equals the total cross-section multiplied by the average velocity.

The *sub-surface float* consists of a submerged body a little heavier than water that is attached by means of a fine cord to a surface float of much smaller size. The sub-surface float can be adjusted to float at any desired depth. By setting it at mean depth the observed velocity will be approximately the average velocity of the vertical strip. The use of such floats is not looked upon with much confidence. Rod floats are much better.

UNRIVALED NIAGARA

WATER-POWER DEVELOPMENT

PART I

1. **Introduction.** One of the fundamental teachings of science is that all energy in the solar system is derived from the sun. Through the agency of that luminary, water from the earth's oceans, seas, and lakes is transformed into vapor, and in this condition is diffused throughout the atmosphere, transported by the winds—themselves created by this same solar energy—over long distances and wide areas, and finally precipitated over land and water, hills and valleys, mainly in the form of rain and snow. Of the total precipitation on the continents, part is evaporated from land and water surfaces, vegetation, etc.; part runs off more or less rapidly as surface flow into the nearby drainage channels, and thence, more or less directly, to the ocean; and part sinks into the ground. Of this last, a portion is retained by capillary attraction within reach of vegetation, to be taken up slowly by the rootlets and transpired through the leaves; the balance percolates downward until it reaches the surface of the underground water flow, which it joins in its relatively slow motion to some nearby stream, lake, or other drainage course, or directly to the ocean. It is then again evaporated into the atmosphere, with a continuous repetition of the cycle described above.

Thus every elevated body of water, every running stream, is a source of power whose energy has been derived or borrowed from the sun; and under proper conditions, a large proportion of this energy may be transformed into useful work.

2. **Unit of Work.** For industrial purposes, the unit of work most generally adopted is the *foot-pound* (ft.-lb.), which represents the quantity of work done in lifting a mass of one pound through a height of one foot against the opposing force of gravity—or in raising a weight of one pound through a height of one foot. Since the force of gravity, and therefore the weight of a given mass, is not constant for

all points on the surface of the earth, it follows that the foot-pound, or *gravitation-measure* of work, is not a constant unit. Its variation, however, is so small as to be negligible for ordinary purposes; and, being much simpler than the theoretically accurate units which must occasionally be employed in scientific investigation, it has remained in very general use. Thus the work done in raising 20 pounds of water through a height of 1 foot, or 1 pound of water through a height of 20 feet, or 5 pounds of water through a height of 4 feet, is said to be 20 foot-pounds.

3. **Power.** In the preceding definition, the element of *time* was not considered; thus, in the above example, 20 foot-pounds of work were done, whether the indicated operation took one minute to perform or extended over a period of one hour, or longer. The term *power* is defined as the amount of energy that can be exerted, or work done, *in a given time.*

4. **Unit of Power.** For industrial purposes, the unit most commonly employed is the *horse-power* (h.p.), which represents the capacity to perform 33,000 foot-pounds of work in one minute, or 550 foot-pounds of work in one second; it thus indicates the *rate of work.*

Example 1. A pump raising 7,500,000 gallons of water* in 10 hours to an elevated tank 50 feet high, is performing:

$$\frac{7,500,000 \times 62.5 \times 50}{7.5} = 3,125,000,000 \text{ ft.-lbs. of useful work; or,}$$

$$\frac{3,125,000,000}{10 \times 60} = 5,208,333 \text{ ft.-lbs. per minute,}$$

which is equivalent to:

$$\frac{5,208,333}{33,000} = 157.8 \text{ h.p.}$$

This amount of horse-power is the rate of work which, in the example above, must be continued for 10 hours in order to raise the total quantity of water. The entire problem may be conveniently performed in one operation, thus:

$$\frac{7,500,000 \times 62.5 \times 50}{7.5 \times 10 \times 60 \times 33,000} = 157.8 \text{ h.p.}$$

5. **Energy.** The amount of energy existing in any agent is measured by the quantity of work it is able to do; *energy* and *work*

*One cubic foot of water weighs 62.5 lbs. and contains 7.5 gallons (approximately).

are therefore measured by the same unit. "When energy is exerted, work is done against resistance." As usually stated in Theoretical Mechanics, energy may exist as *potential energy*—energy of position; or *kinetic energy*—energy of motion; or partly in one form, and partly in the other. Thus (see Fig. 1) a cannon-ball weighing W pounds, located in an elevated position h feet above any plane of reference, possesses Wh foot-pounds of potential energy with respect to that plane, by virtue of its position. If it be allowed to fall to the plane, it will, at its lowest point, theoretically have acquired a velocity of $v(=2gh)$ feet per second, and will therefore, at that level, possess kinetic energy to the amount of $W\frac{v^2}{2g}$ $(=Wh)$ foot-pounds by reason of its motion. Further, if we analyze the conditions at some intermediate plane h_1 feet below its original position, and h_2 feet above the lower level, we shall find that the ball has acquired at this point a velocity of v_1 (= $\sqrt{2gh_1}$) feet per second, and therefore possesses kinetic energy to the amount of $W\frac{v_1^2}{2g}$ (= Wh_1) foot-pounds due to its motion; but, by reason of its position h_2 feet above the lower plane, it still possesses Wh_2 foot-pounds of potential energy; consequently, with respect to the lower plane, the ball possesses a total energy represented by $W\left(\frac{v_1^2}{2g}+h_2\right) = W(h_1+h_2) = Wh = W\frac{v^2}{2g}$ foot-pounds. Thus potential and kinetic energies are mutually convertible, theoretically without loss; practically, more or less energy will be transformed into heat during the conversion, and dissipated. But the great principle of the Conservation of Energy teaches that the *total quantity* of energy existing, or stored in the ball in any position, is theoretically a constant quantity.

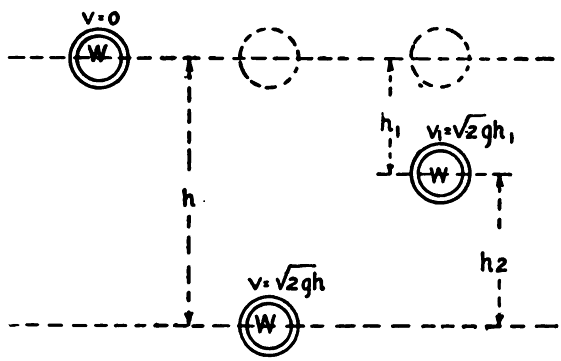


Fig. 1. Illustrating Relation between Potential and Kinetic Energy.

6. **Pressure-, Velocity-, and Gravity-Head.** In hydraulic work, because of the nature of the medium dealt with—water being considered in this connection a perfect fluid, and incompressible—and because of the character of the problems presented, it is customary and convenient to consider the energy of water as capable of existing in three forms—*Pressure*, *Velocity*, and *Gravity*. Thus, in Fig. 2, with the conditions as represented (see also "Hydraulics," page 34), if the valve at D be closed, the water will rise in tube CC (called a *piezometer tube*) to the same level EF as that existing in the reservoir, and the pressure in the pipe at C will be represented by the head h

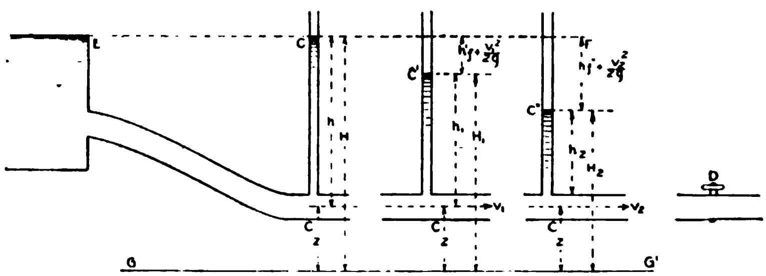


Fig. 2. Illustrating Relations of Pressure-, Velocity-, and Gravity-Head.

feet. Now, if the valve at D be partially opened, so that there is some velocity of flow v_1, in the pipe at section C, the column of water in the tube CC will sink to some lower level, as CC^1, and the pressure in the pipe at C will be that due to the head h_1 feet. Similarly, if the valve be now completely opened, so that the velocity of flow v_2, in the same section, becomes greater than v_1, the column of water in the tube will sink still lower, as CC'', indicating a pressure in the pipe at C represented by the head h_2 feet. If the loss of head in friction, etc., in the two cases of flow indicated above be respectively represented by h'_f and h''_f, the important relations existing are clearly shown in this diagram. It is evident that at the end of the pipe, where the water discharges freely into the air, no pressure-head exists, all the energy possessed by the issuing water being kinetic.

7. **Total Head.** Now let GG^1 represent any horizontal plane of reference—for example, the level of the tail-race water in a hydraulic power plant. With reference to this plane, the total effective head existing in the pipe at the section C, is:

(*a*) *For the case of no flow*—

$$z + h = H \text{ feet};$$

(*b*) *For the case of partial flow*—

$$z + h_1 + \frac{v_1^2}{2g} = H - h'_f \text{ feet};$$

(*c*) *For the case of full flow*—

$$z + h_2 + \frac{v_2^2}{2g} = H - h''_f \text{ feet.}$$

The distance z may be called the *gravity-head* (it corresponds to the head in potential energy already referred to); $\frac{v_1^2}{2g}$ and $\frac{v_2^2}{2g}$ are properly termed the *velocity-heads* (they correspond to the heads in kinetic energy already explained); h_1 and h_2 are known as the *pressure-heads* (see "Hydraulics," Article 6); h'_f and h''_f represent the heads lost in overcoming the various resistances to flow, principally friction in the pipe for the usual cases; but in the general case they include losses of head due to entrance, valves, curves, etc. (see "Hydraulics," Articles 28 and 34).

8. **Energy per Pound of Water.** The quantities stated above as number of feet in (*a*), (*b*), and (*c*) may be understood in another sense. Each may represent the total number of foot-pounds of energy existing *in every pound of water* in, or passing through, the pipe at section C; thus,

(*a*) $z + h = H$ foot-pounds per pound of water

(*b*) $z + h_1 + \frac{v_1^2}{2g} = H - h'_f$ " " " " " "

(*c*) $z + h_2 + \frac{v_2^2}{2g} = H - h''_f$ " " " " " "

9. **Total Energy.** Now suppose W_1 and W_2 pounds of water per second respectively to pass the section C in the two cases of flow considered; then, with respect to the plane GG^1, the total energy of the water as it passes this section is, for the one case:

(*b*) $W_1\left(z + h_1 + \frac{v_1^2}{2g}\right)$ foot-pounds;

and for the other:

(*c*) $W_2\left(z + h_2 + \frac{v_2^2}{2g}\right)$ foot-pounds;

and these expressions represent, for the two cases considered, the *total amount of energy* possessed by the water, *with respect to the plane* GG^1, and theoretically capable of being delivered to a machine or motor, by the descent of the water from the upper level EF to the lower level GG^1.

Where the water issues freely into the air from the extremity of the pipe, or through a nozzle at the end, no pressure exists; therefore, in the expression corresponding to (*b*) or (*c*), above, for such section, the term representing pressure-head disappears, leaving the two terms indicating gravity-head and velocity-head.

Further, if the plane of reference passes through the center of the end of the pipe or nozzle opening, the term representing gravity-head also disappears, leaving the velocity-head alone to indicate the energy of the stream at this point.

It is usually more convenient to express the sum of gravity-head and pressure-head in a single term: thus, $z + h_1 = H_1$; and $z + h_2 = H_2$; here H_1 and H_2 may be called the *piezometer heights*.

10. **Efficiency.** The efficiency of any apparatus for utilizing the kinetic energy of moving water, or the potential energy of still water, is *the ratio of the amount of work given out by the apparatus to the amount of work delivered to it*; or, as it is sometimes stated, it is *the ratio of the useful work to the theoretic energy.* This topic will be treated more fully in a later article; for the present, if e represent the efficiency of a motor, then,

$$e = \frac{\text{Foot-pounds or horse-power } \textit{given out by} \text{ motor}}{\text{Foot-pounds or horse-power } \textit{delivered to} \text{ motor}}.$$

As will be seen later, the denominator does not represent the full theoretic energy of the waterfall, since more or less of this energy must be utilized in overcoming the resistances encountered in conducting the water to the motor.

Example 2. A motor is operated by a stream of water discharged through a 2-foot pipe with a velocity of 10 feet per second. The motor gives out at its shaft 4.4 horse-power. What is the efficiency of the motor?

$$\frac{3.1416 \times 10 \times 62.5 \times 100}{550 \times 64.4} = 5.5 \text{ horse-power delivered to motor}$$

$$e = \frac{4.4}{5.5} = 80 \text{ per cent efficiency.}$$

Example 3. A small turbine wheel using 100 cubic feet of water per minute under a head of 45 feet, is found to give 6 horse-power. What is the efficiency of the wheel?

$$6 \text{ Horse-Power} = 6 \times 33{,}000 = 198{,}000 \text{ ft.-lbs. per min.}$$

$$e = \frac{198{,}000}{100 \times 62.5 \times 45} = 70.4 \text{ per cent efficiency.}$$

Theoretic Efficiency. If the efficiency of the motor actuated by the water were 100 per cent, it would give out at its shaft, as useful work, the same number of foot-pounds that were delivered to it. It is also interesting to note that if the efficiency of the hydraulic parts of the plant were 100 per cent—that is, if there were no hydraulic losses of head—the total energy of the water (see Fig. 2) represented by the total head H feet, or H foot-pounds per pound of water, would be available; and, if operating a motor of 100 per cent efficiency, the total energy of the water would be given out as useful work at the shaft of the motor. In practice these ideal conditions can never be fully realized, for there are certain hydraulic and mechanical losses of energy, which, while they may be reduced to the lowest limits by means of proper design, nevertheless, cannot be entirely eliminated.

Example 4. A pond containing 2,000,000 cubic feet of water is at an average elevation of 50 feet above the lower level. How much potential energy does this theoretically represent at the lower level?

$$2{,}000{,}000 \times 62.5 \times 50 = 6{,}250{,}000{,}000 \text{ ft.-lbs.}$$

If this water is fed to a small motor at the rate of 100 cubic feet per minute, what horse-power does this represent, and how long may the motor be operated?

$$\frac{100 \times 62.5 \times 50}{33{,}000} = 9.5 \text{ h.p.}$$

$$\frac{2{,}000{,}000}{100 \times 60 \times 24} = 13\tfrac{7}{8} \text{ days, or 13 days 21 hours.}$$

Assuming that the motor has an efficiency of 75 per cent, how much power may be taken off at its shaft?

$$9.5 \times .75 = 7.1 \text{ h.p.}$$

Example 5. The discharge of a stream is 1,000 cubic feet per second; its mean velocity is 3 feet per second. What horse-power does this represent?

$$\frac{1{,}000 \times 62.5 \times (3)^2}{550 \times 64.4} = 1{,}588.1 \text{ h. p.}$$

Example 6. Water issues from a nozzle at the rate of 50 feet per second; the area of the nozzle opening is 0.1 square foot How many foot-pounds of kinetic energy does this represent? How many horse-power? If this jet operates a motor of 80 per cent efficiency, what horse-power will the motor actually yield?

$$0.1 \times 50 \times 62.5 \times \frac{(50)^2}{64.4} = 21{,}125 \text{ ft.-lbs. per second.}$$

$$\frac{21{,}125}{550} = 22 \text{ h.p.}$$

$$22 \times .80 = 17.6 \text{ h.p.}$$

11. Pipe End with Nozzle. *Pressure at Base of Nozzle.* For many purposes—as in hydraulic mining, in the operation of certain types of water motor (described later), and at the extremity of fire-hose—water is delivered at considerable velocity through a nozzle attached to the end of a pipe. It is therefore desirable to develop a formula for velocity of flow, and quantity of discharge, for such cases.

If the pressure-head h_1 (Fig. 3) at the entrance or base of a *smooth* nozzle be observed, either by a piezometer tube or by a pressure

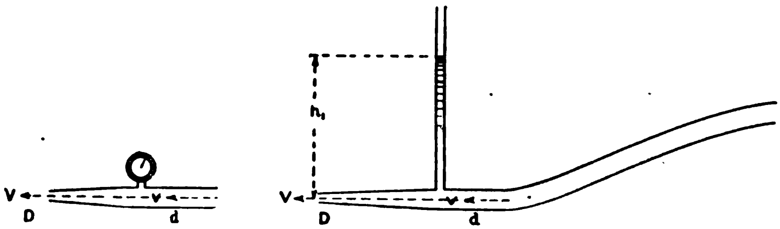


Fig. 3. Pipe with Nozzle Attachment.

gauge, then, since the nozzle velocity V is a consequence of the pressure-head h_1 and the velocity-head $\frac{v^2}{2g}$ of the water in the pipe approaching the nozzle with a velocity of v feet per second, the real or effective head on the nozzle is $h_1 + \frac{v^2}{2g}$; the theoretic velocity from the nozzle is:

$$V = \sqrt{2g\left(h_1 + \frac{v^2}{2g}\right)};$$

and the actual velocity is:

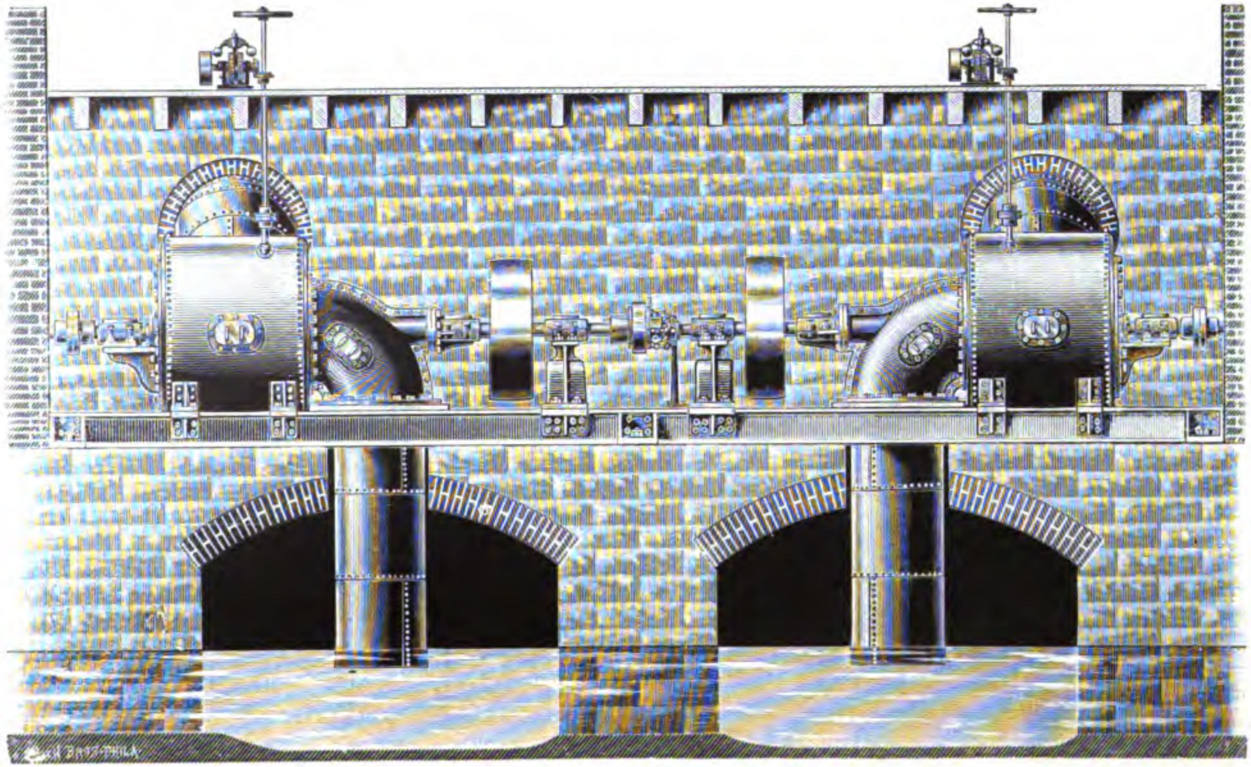
$$V = c_1 \sqrt{2g\left(h_1 + \frac{v^2}{2g}\right)},$$

in which c_1 denotes the coefficient of velocity, which, for smooth nozzles, is the same as the coefficient of discharge. In these equations, h_1 is expressed in feet; V and v in feet per second. Let D and d be the diameters, in feet, of the nozzle and pipe respectively.

Since the Discharge q = Area × Velocity,

$$q = \frac{\pi D^2}{4} V = \frac{\pi d^2}{4} v;$$

therefore,



TURBINE INSTALLATION FOR A PAPER MILL

Two single 27-inch horizontal-shaft turbines operating under 35 feet head and developing 548 horse-power. The wheels are coupled direct to line shaft in the mill, from which the engines and other machinery are driven. The wheels are located in the middle of the building, and are connected by a friction cut-off coupling, so that they can be run together or independently, as may be preferred.

Courtesy of S. Morgan Smith Co., York, Pa.

$$v = \left(\frac{D}{d}\right)^2 V.$$

Substituting this value of v in the equation above, and solving for V, there results:

$$V = \sqrt{\frac{2gh_1}{\left(\frac{1}{c_1}\right)^2 - \left(\frac{D}{d}\right)^4}} \quad \ldots\ldots\ldots (1)$$

in feet per second; and the discharge (area × V) is:

$$q = 0.7854\ D^2 \sqrt{\frac{2gh_1}{\left(\frac{1}{c_1}\right)^2 - \left(\frac{D}{d}\right)^4}} \quad \ldots\ldots\ldots (2)$$

in cubic feet per second; and the velocity-head of the issuing jet is:

$$\frac{V^2}{2g} = \frac{h_1{}^2}{\left(\frac{1}{c_1}\right)^2 - \left(\frac{D}{d}\right)^4} \quad \ldots\ldots\ldots (3)$$

In many cases it is common to read the pressure at the base of the nozzle in pounds per square inch; then h_1 (in feet) equals $2.304\ p_1$ (in pounds per square inch); and the discharge is frequently stated in gallons per minute; making these substitutions in Equation 2, above, we have:

$$q = 29.83\ D^2 \sqrt{\frac{p_1}{\left(\frac{1}{c_1}\right)^2 - \left(\frac{D}{d}\right)^4}} \quad \ldots\ldots\ldots (4)$$

in gallons per minute.

Example 7. The pressure-gauge at the base of a smooth 1¼-inch nozzle reads 80 pounds per square inch; compute the velocity and discharge from the nozzle, the velocity-head of the issuing stream, and the mean velocity in the pipe, if the latter be 2½ inches in diameter. Assume 0.97 as the value of the coefficient.

Substituting the given numerical values in Equation 1, we have:

$$V = \sqrt{\frac{64.4 \times (2.304 \times 80)}{\left(\frac{1}{.97}\right)^2 - \left(\frac{1}{2}\right)^4}} = 38.5 \text{ foot per second.}$$

$$q = \text{Area} \times V = \frac{0.7854 \times 1.25}{144} \times 38.5 = 0.33 \text{ cubic foot per second.}$$

$$\frac{V^2}{2g} = \frac{(38.5)^2}{64.4} = 22.7 \text{ feet.}$$

$$v = \left(\frac{D}{d}\right)^2 V = \tfrac{1}{4} \times 38.5 = 9.6 \text{ feet per second.}$$

What horse-power does this represent?

$$\frac{0.33 \times 62.5}{550} \times 22.7 = 0.85 \text{ h.p}$$

With a motor of 80 per cent efficiency, how much useful work will be obtained?

$$0.85 \times 0.80 = 0.68 \text{ h.p.}$$

12. **Pipe Line with Nozzle.** In Fig. 4, let h be the total head on the end of the nozzle, D its smaller diameter in feet, and V the velocity of the issuing stream in feet per second. Let d and v be the corre-

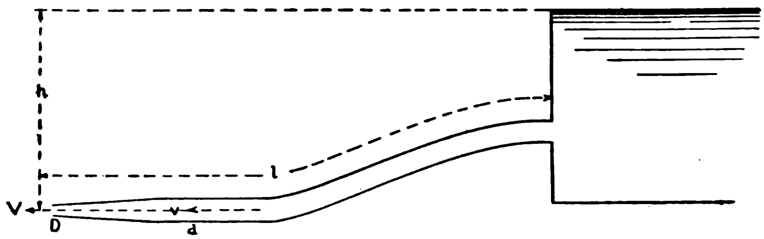


Fig. 4. Loss of Head in Pipe and Nozzle.

sponding quantities for the pipe or hose, and l its length in feet. Of the total available head h on the end of the nozzle, only $\frac{V^2}{2g}$ remains; so that $h - \frac{V^2}{2g}$ represents the head lost or dissipated in overcoming various resistances to flow, from the reservoir to the tip of the nozzle. This lost head consists of several parts (see "Hydraulics," Article 34); we may therefore write:

$$h - \frac{V^2}{2g} = \left\{ \left(\frac{1}{c}\right)^2 - 1 \right\} \frac{v^2}{2g} + \frac{8gl}{C^2 d}\,\frac{v^2}{2g} + m\,\frac{v^2}{2g} + n\,\frac{v^2}{2g} + m'\,\frac{V^2}{2g} \quad \ldots (7)$$

in which,

$\left\{ \left(\frac{1}{c}\right)^2 - 1 \right\} \frac{v^2}{2g}$ = Loss of head at entrance; $\frac{8\,gl}{C^2 d}\,\frac{v^2}{2g}$ = Head lost in friction in the pipe (see "Hydraulics," Articles 28 and 36); $m\,\frac{v^2}{2g}$ = Head lost in bends and curves; $n\,\frac{v^2}{2g}$ = Head lost by the passage of the water through valves and gates; and, lastly, $m'\,\frac{V^2}{2g}$ = Head lost in passing through the nozzle.

The equation for the value of m^1 assumes a form similar to that for entrance loss into a pipe:

$$m^1 = \left\{ \left(\frac{1}{c_1}\right)^2 - 1 \right\},$$

in which c_1 is the coefficient of velocity, which, for smooth nozzles, is the same as the coefficient of discharge; its value may be taken as 0.97 for such nozzles, with the small diameter between $\frac{3}{4}$ inch and $1\frac{1}{2}$ inches, under ordinary range of pressures.

Since, in steady flow, the velocities v and V are inversely proportional to the areas of the corresponding cross-sections,

$$V = v\left(\frac{d}{D}\right)^2$$

Inserting this value of V in Equation 5, and solving for v, there results:

$$v = \sqrt{\frac{2gh}{\left\{\left(\frac{1}{c}\right)^2 - 1\right\} + \frac{8gl}{C^2 d} + m + n + \left(\frac{1}{c_1}\right)^2 \left(\frac{d}{D}\right)^4}} \quad \ldots(6)$$

for the velocity of flow in the pipe, in feet per second.

The velocity and discharge from the nozzle are then:

$$V = \left(\frac{d}{D}\right)^2 v, \quad \ldots\ldots(7)$$

and,

$$q = \frac{1}{4}\pi D^2 V \quad \ldots\ldots(8)$$

In many cases the sum of the losses at entrance, through valves and gates, and around bends and curves, is sufficiently small, in comparison with the loss in pipe friction, to be negligible; in such cases, Equation 6 reduces to

$$v = \sqrt{\frac{2gh}{\frac{8gl}{C^2 d} + \left(\frac{1}{c_1}\right)^2 \left(\frac{d}{D}\right)^4}} \quad \ldots\ldots(9)$$

Example 8. A smooth nozzle with a small diameter of 1 inch is attached to a 3-inch pipe 1,500 feet long; the tip of the nozzle is 64 feet below the surface of the water in an elevated reservoir. Assume $C = 100$, and determine the velocity of flow in the pipe, and through the nozzle. Find also the discharge, and the efficiency of the pipe and nozzle.

Since in this case the entrance loss is relatively small, because the pipe is long in comparison with its diameter, and therefore pipe friction is relatively large, Equation 9 may be used:

$$v = \sqrt{\frac{64.4 \times 64}{\frac{8 \times 32.2 \times 1{,}500}{(100)^2 \times 0.25} + \left(\frac{1}{.97}\right)^2 \left(\frac{3}{1}\right)^4}} = 4.14 \text{ feet per second,}$$

for the velocity of flow in the pipe.

$$V = v\left(\frac{d}{D}\right)^2 = 4.14 \times 9 = 37.26 \text{ feet per second,}$$

for the velocity of the jet issuing from the nozzle.

$$q = \frac{\pi d^2}{4} v = \frac{3.1416 \times \left(\frac{1}{4}\right)^2}{4} \times 4.14 = 0.20 \text{ cu. ft. per second.}$$

The energy of the jet is:

$$W \frac{V^2}{2g} = \frac{.20 \times 62.5 \times (37.26)^2}{64.4} = 269.5 \text{ ft.-lbs. per second.}$$

The theoretic energy is:

$$Wh = .20 \times 62.5 \times 64 = 800 \text{ ft.-lbs. per second.}$$

The efficiency of pipe and nozzle, therefore, is:

$$\frac{269.5}{800} = 33.7 \text{ per cent.}$$

13. If, under the conditions just stated, we suppose the nozzle removed, the last term in the denominator of Equation 9 will disappear, and the equation will assume the form:

$$v = \frac{C}{2}\sqrt{\frac{hd}{l}} = C\sqrt{r\frac{h}{l}} = C\sqrt{rs} \quad \ldots\ldots (10)$$

which is Equation 30 in "Hydraulics," for the case of a pipe of uniform diameter; or Equation 33, for flow in open channels.

14. Equation 7, taken in connection with Equation 6 or its simpler form, Equation 9, shows that the smaller the nozzle diameter compared with that of the pipe, within ordinary practical limits, the greater will be the nozzle *velocity*; but the greatest *discharge* will occur (Equation 8) when the nozzle diameter is as large as possible; that is, when it is equal to the pipe diameter—in other words, when there is no nozzle attached.

15. **Relation of Pipe and Nozzle Diameters.** When the object of attaching a nozzle to a pipe is to utilize the velocity-head of the issuing jet $\left(=\frac{V^2}{2g}\right)$ without regard to the quantity of water discharged, a large pipe and a relatively small nozzle should be employed. When the object is to obtain as large a discharge as possible, no nozzle should be used, and the pipe should be as large as practical considerations will warrant. When the object is to utilize the energy of the jet in producing power by means of a water-motor, in which case both velocity-head and quantity of discharge are concerned, there is a definite relation existing between the diameters of nozzle and pipe that will render this a maximum.

16. **Maximum Power Derivable from Nozzle Jet.** From Equations 9 and 7, we derive:

$$V = \sqrt{\frac{2gh}{\frac{8gl}{C^2 d}\left(\frac{D}{d}\right)^4 + \left(\frac{1}{c_1}\right)^2}} \quad \ldots\ldots\ldots (11)$$

Then, if w be the weight in pounds of a cubic foot of water, we have, for the theoretical kinetic energy of the issuing jet in foot-pounds per second (weight of discharge in pounds per second $\times$ velocity-head):

$$K = w\tfrac{1}{4}\pi D^2 V \frac{V^2}{2g} = \frac{w\pi D^2 V^3}{8g} \quad \ldots\ldots (12)$$

Substituting in this equation the value of V above (Equation 11), and ascertaining, by the procedure usually adopted in such cases (differential calculus), the value of D to render K a maximum, we obtain:

$$D = \tfrac{1}{2}\left(\frac{C^2 d^5}{g c_1^2 l}\right)^{\frac{1}{4}}, \quad \ldots\ldots\ldots\ldots\ldots (13)$$

which is a formula for diameter of nozzle in terms of diameter and length of pipe (all in feet) to produce the maximum kinetic energy of the jet issuing from the nozzle.

With a nozzle of this diameter, the velocity of the issuing jet is obtained by placing the value of D from Equation 13 in Equation 11, with the result:

$$V = 2c_1\sqrt{\frac{gh}{3}} = c_1\sqrt{2g\left(\frac{2}{3}h\right)} = 0.816c_1\sqrt{2gh} \quad \ldots (14)$$

Since the value of c_1 for ordinary cases is about 0.97, it may be said that the nozzle velocity necessary to produce the *maximum power* is about 80 per cent of the theoretic velocity due to the actual static head on the nozzle tip.

17. **Relation between Total Head and Friction Head for Maximum Power.** The relation expressed by Equation 14 leads to some interesting conclusions. Since $V = .80\sqrt{2gh}$ for maximum power, $\frac{V^2}{2g} = .64\,h$; therefore, since the total head is h, $.36h$ must be used in overcoming pipe and nozzle resistance, to give the most advantageous velocity for power purposes. Again, omitting nozzle resistance (as represented by c_1), $\frac{V^2}{2g} = .667h$; therefore $.333h$ must be used in overcoming pipe friction alone. That is to say, with the conditions

arranged to furnish maximum power, ⅓ of the total static head on the nozzle tip is being used to overcome pipe friction, and the remaining $\frac{2}{3}h$ is transformed into the velocity-head of the issuing stream after due deduction or allowance for nozzle resistance. The second value of V (Equation 14) shows this directly. If no nozzle is attached, therefore, the conditions for maximum power obtain when ⅓ the total static head is used in overcoming pipe friction, the remaining ⅔ of the head being available as velocity-head, or as pressure-head, or partly in one form and partly in the other.

18. Usually the discharge in cubic feet per second (q) is known; then, by simple substitution (Equations 8 and 14), the values for maximum work are:

$$D = \left(\frac{12\, q^2}{\pi^2\, c_1{}^2\, g h} \right)^{\frac{1}{4}} \qquad \textbf{(15)}$$

and, from Equations 13 and 15:

$$d = 2 \left(\frac{6\, q^2\, l}{\pi^2\, C^2\, h} \right)^{\frac{1}{5}} \qquad \textbf{(16)}$$

in which D and d are the diameters in feet of nozzle tip and pipe to furnish maximum power. Being stated in terms of q, l, and h, these equations are occasionally the most convenient to use in solving problems.

Example 9. By damming a stream, an impounding reservoir was created, capable of supplying uniformly 5.92 cubic feet of water per second to a power-house below. The nozzle tip is to be 590 feet below the average water level in the reservoir; the length of pipe is 6,000 feet from reservoir to nozzle; the pipe being of riveted steel, and making due allowance for deterioration of surface with age, C was assumed to have the low value 83. What size pipe and nozzle should be used to give the maximum power? What will be the nozzle velocity? What horse-power will be developed at the nozzle? What efficiency does this represent for pipe and nozzle? What power may be derived from a wheel of 75 per cent efficiency, driven by the jet? What is the efficiency of the whole system?

From Equation 16:

$$d = 2 \left\{ \frac{6 \times (5.92)^2 \times 6{,}000}{(3.1416)^2 \times (83)^2 \times 590} \right\}^{\frac{1}{5}} = 1 \text{ foot, pipe diameter.}$$

From Equation 15:

$$D = \left\{ \frac{12 \times (5.92)^2}{(3.1416)^2 \times (.97)^2 \times 32.2 \times 590} \right\}^{\frac{1}{4}} = 2.67 \text{ inches, nozzle diameter,}$$

or Equation 13 may be used to determine D.

From Equation 14:

$$V = 0.816 \times 0.97 \sqrt{64.4 \times 590} = 152 \text{ feet per second, nozzle velocity.}$$

$$\text{Horse-power} = \frac{W V^2}{2g \times 550} = \frac{5.92 \times 62.5 \times (152)^2}{64.4 \times 550} = 241 \text{ h.p.}$$

$$\text{Theoretic horse-power} = \frac{Wh}{550} = \frac{5.92 \times 62.5 \times 590}{550} = 397 \text{ h.p.}$$

$$\text{Efficiency} = \frac{241}{397} = 61 \text{ per cent (nearly).}$$

$$\text{Useful work from wheel} = 241 \times .75 = 181 \text{ h.p.}$$

$$\text{Efficiency of whole system} = .75 \times .61 = 46 \text{ per cent (or } \tfrac{46}{100}\text{).}$$

19. **Multiple Nozzles.** Sometimes an impulse wheel is driven by means of jets issuing from two or more nozzles of the same or of different diameters. Then, for maximum power, the sum of the areas of the several nozzles must equal the area corresponding to D, as computed for a single nozzle, on the assumption that the nozzle tips are at substantially the same level, and that the coefficient c_1 has the same value for each. Thus, if there be two nozzles with diameters D_1 and D_2,

$$D_1^2 + D_2^2 = \frac{1}{4}\left(\frac{C^2 d^5}{g c_1^2 l}\right)^{\frac{1}{2}} = \frac{C d^2}{4 c_1}\sqrt{\frac{d}{g l}} \quad \ldots\ldots (17)$$

One diameter, as D_1, may be assumed, and the other computed from the above relation.

If the two nozzles are of equal diameter D_1,

$$D_1^2 = \frac{1}{8}\left(\frac{C^2 d^5}{g c_1^2 l}\right)^{\frac{1}{2}};$$

therefore,

$$D_1 = \frac{1}{2}\left(\frac{C^2 d^5}{4 g c_1^2 l}\right)^{\frac{1}{4}} \quad \ldots\ldots\ldots\ldots (18)$$

If the value of D for one nozzle has already been determined, then, for two nozzles of equal diameter D_1, from the relation stated above,

$$\frac{2\pi D_1^2}{4} = \frac{\pi D^2}{4};$$

therefore,

$$D_1 = \frac{D}{\sqrt{2}} \quad \ldots\ldots\ldots\ldots (18a)$$

With three or more nozzles, of the same or of different diameters, the relation of areas stated above will furnish a means of readily determining the diameters. Thus, for three nozzles of equal diameter D_1,

$$D_1 = \frac{D}{\sqrt{3}} \quad \ldots\ldots\ldots\ldots (18b)$$

If the discharge q is known, an analysis similar in all respects to that above will give, in place of Equation 17:

$$D_1^2 + D_2^2 = \frac{2q}{\pi c_1}\sqrt{\frac{3}{gh}}; \quad \ldots\ldots\ldots (19)$$

and, in place of Equation 18:

$$D_1 = \left(\frac{3q^2}{\pi^2 c_1^2 gh}\right)^{\frac{1}{4}}, \quad \ldots\ldots\ldots\ldots (20)$$

which will prove more convenient for use in some problems.

Example 10. If, in example 9, two nozzles of equal diameter were required, the diameter of each nozzle could be determined directly from Equation 18; or more simply, from Equation 18*a*, since the value of D has already been found:

$$D_1 = \frac{D}{\sqrt{2}} = \frac{2.67}{1.41} = 1.9 \text{ inches for each nozzle.}$$

If three equal nozzles were required, then, from Equation 18*b*:

$$D_1 = \frac{D}{\sqrt{3}} = \frac{2.67}{1.73} = 1.5 \text{ inches for each nozzle.}$$

IMPULSE, REACTION, AND DYNAMIC PRESSURE

20. **Impulse and Reaction of Water in Motion.** Let W be the number of pounds of water discharged per second from an orifice, pipe, or nozzle, or flowing in a stream, with a uniform velocity of v feet per second; then,

$$F = W\frac{v}{g} \text{ pounds} \ldots\ldots\ldots\ldots (21)$$

is called the *impulse* of the moving water. It may be regarded as a continuous pressure in the direction of motion; and it will be exerted as such upon a surface placed in the path of the jet or stream, with an intensity varying with the conditions, and ranging to the maximum value F, above. The *reaction*, or *back-pressure*, is equal in value to the impulse, but opposite in direction. For example, if a vessel containing water be freely suspended at A (Fig. 5), and water be allowed to flow out through an orifice at B, the pressure due to the head of water h causes W pounds of water per second to be discharged with the velocity v (= theoretically $\sqrt{2gh}$) feet per second. In the direction of the jet, the impulse produces motion; in the opposite direction, it produces an equal back-pressure (action and reaction being equal in amount and opposite in direction), causing the vessel to swing to the right. The first of these forces is the *impulse*, and the second is the *reaction* of the jet; and if a force R be applied as shown, of just sufficient intensity to prevent this motion of the vessel, its value is:

$$R = W\frac{v}{g} = F, \ldots\ldots\ldots\ldots (22)$$

which is the reaction of the jet.

21. The impulse or reaction of a jet issuing from an orifice is double the hydrostatic pressure on the area of the orifice. For, if a is the area of the orifice, and w the weight of a cubic unit of water, the normal hydrostatic pressure on the area of the orifice when closed (see "Hydraulics," Article 6) is:

Hydrostatic pressure $= wah$ pounds.

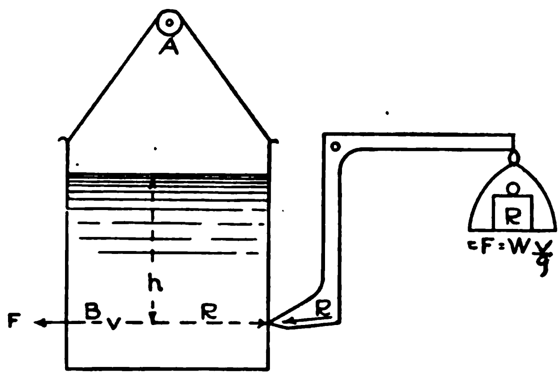


Fig. 5. Measuring the Reaction of a Jet by Weighing.

When the orifice is opened, the weight of the discharge per second (see "Hydraulics," Article 18) is theoretically $W = wav$; hence,

$$F = R = W\frac{v}{g} = wav\,\frac{v}{g} = \frac{2wav^2}{2g} = 2\,wah. \ldots\ldots (23)$$

This conclusion has been verified by many experiments (see Fig. 6).

Example 11. What must be the velocity of a jet of water 1 inch in diameter, issuing from a nozzle, in order that its impulse may be 100 pounds? What will be the discharge in cubic feet and in gallons per second?

$$F = \frac{Wv}{g} = \frac{wav^2}{g} = 100\;;$$

$$\therefore v = \sqrt{\frac{100 \times 32.2}{62.5 \times .0054}} = 97.7 \text{ foot per second.}$$

$$q = av = .0054 \times 97.7 = .53 \text{ cubic foot per second.}$$

$$.53 \times 7.5 = 4 \text{ gallons per second.}$$

22. **Dynamic Pressure of Water in Motion.** If a jet of water strike a stationary plane normally, it produces a dynamic pressure on that plane equal to the impulse of the jet; that is:

$$P = F = W\frac{v}{g}$$

If a jet moving with a velocity v_1 be retarded by a surface so that its velocity becomes v_2, without changing its direction, the impulse in the first case is:

$$F_1 = \frac{Wv_1}{g};$$

and in the second case:

$$F_2 = \frac{Wv_2}{g};$$

and the difference,

$$P = F_1 - F_2 = W\left(\frac{v_1 - v_2}{g}\right) \quad \ldots\ldots\ldots (24)$$

is a measure of the dynamic pressure which has been developed in

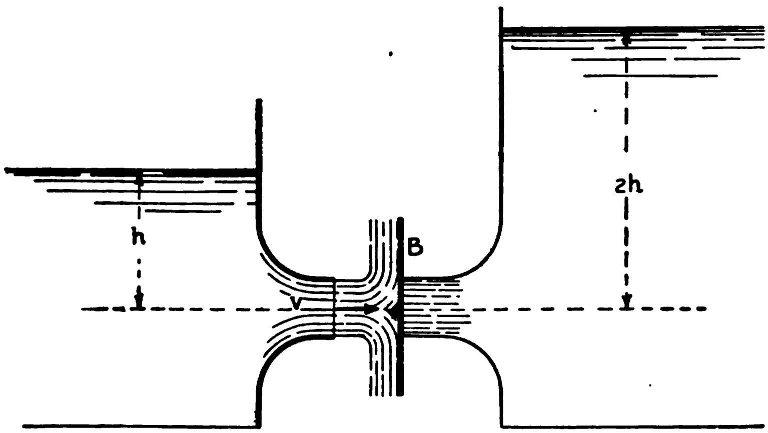


Fig. 6. Illustrating Relation between Impulse and Hydrostatic Pressure.

the direction of motion by the retardation of the velocity. If a jet of water impinge upon a stationary surface which changes its direction of motion without changing its velocity, a dynamic pressure is developed, its amount depending upon the velocity and the change in direction. In all cases this pressure is exerted upon the surface causing the retardation of velocity or change in direction of flow.

23. **Static and Dynamic Pressures.** *Dynamic pressure* must be clearly distinguished from *static pressure*, the laws governing in the two cases being entirely different. A static pressure due to a given head will cause a jet of water to be discharged from an orifice with a velocity proportional to the head; if this jet impinge upon a surface, a dynamic pressure will be exerted upon it, which may be equal to, greater than, or less than the static pressure due to the head,

depending upon the circumstances. Again, at any point below the surface of water, static pressure is exerted with equal intensity in all directions; dynamic pressure is exerted with different intensities in different directions.

24. **Definitions.** From a comparison of Equations 21 and 24, we may now define the *impulse* of a jet or stream of water as the dynamic pressure which it is capable of producing in the direction of its motion when its velocity in that direction is entirely destroyed. This may be accomplished by carefully deflecting the jet 90 degrees to its original path by means of a smooth surface, so that, no energy being dissipated in overcoming frictional or other resistances, the velocity of the water is not changed, but its component in the original direction is zero; and the *reaction* of a jet or stream of water may be defined as the backward dynamic pressure, in the line of motion, which is exerted against a vessel out of which it issues, or against a surface away from which it moves.

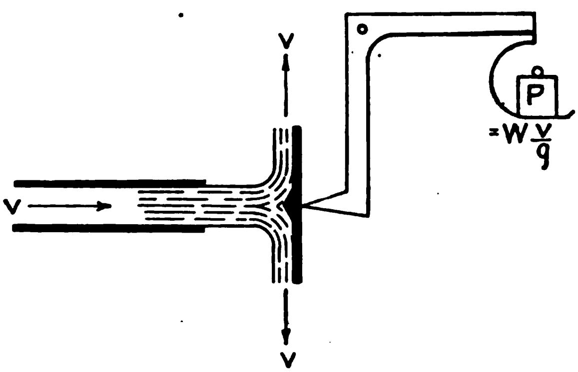


Fig. 7. Measuring Pressure of a Jet on a Plane Surface, by Weighing.

25. **Laboratory Experiments on Impulse, Reaction, and Dynamic Pressure.** Fig. 5 shows how the reaction of a jet may be measured; the necessary weight in the scale pan to prevent motion of the vessel has been found to be very nearly:

$$R = F = \frac{Wv}{g} = 2wa\frac{v^2}{2g}.$$

Fig. 6 shows how the pressure due to the impulse of a jet may be made to balance the hydrostatic pressure due to twice the head causing the flow. *B* is a loose plate with surface carefully finished to fit the mouthpiece so as to prevent leakage. Fig. 7 illustrates a simple device for measuring by weighing the dynamic pressure exerted upon a surface by the impulse of a jet impinging upon and gliding over it, when its motion in the original direction has been entirely destroyed by being deflected 90 degrees. The result of the experiment is found to show very nearly that:

$$P = W\frac{v}{g} = 2\,wa\frac{v^2}{2g},$$

as theory requires.

Fig. 8 illustrates a case of dynamic pressure exerted upon a curved surface, due to both impulse and reaction, the former being due to the direct impact of the jet, the latter to the circumstance that the deflected stream leaves the surface in a direction which has a component of velocity parallel to the original path, but opposite in direction. Here experiment shows:

$$P > W\frac{v}{g} < 2W\frac{v}{g},$$

as theory requires.

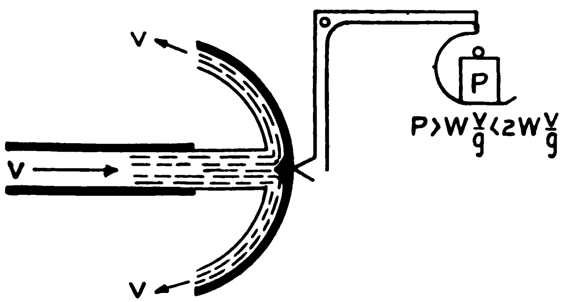


Fig. 8. Measuring Pressure from a Jet on a Curved Surface, by Weighing.

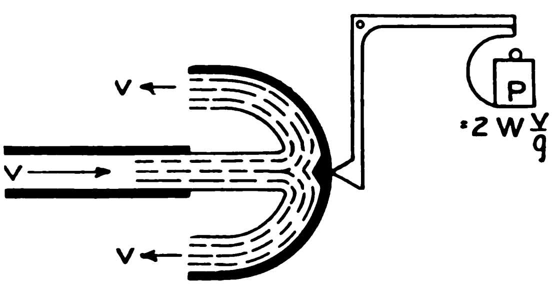


Fig. 9. Measuring Pressure from a Jet whose Direction is Completely Reversed.

Fig. 9 shows the case where the stream is deflected 180 degrees; that is, there is a complete reversal in the direction of motion; and we should expect the dynamic pressure exerted upon the surface to be equal to the sum of both impulse and reaction; namely,

$$P = F + R = 2F = 2W\frac{v}{g} = 4\,wa\frac{v^2}{2g},$$

which agrees quite closely with the results of laboratory experiments.

Example 12. In Fig. 7 the diameter of the tube is 1 inch; there is no contraction of the jet; and the discharge is .5 cubic foot per second. What is the velocity, and the dynamic pressure against the plane? What would be the dynamic pressure in the case represented by Fig. 9?

$$v = \frac{q}{a} = \frac{.5}{.0054} = 92.6 \text{ feet per second.}$$

$$P = W\frac{v}{g} = \frac{.5 \times 62.5 \times 92.6}{32.2} = 90 \text{ pounds.}$$

$$P = 2W\frac{v}{g} = 2 \times 90 = 180 \text{ pounds.}$$

FIXED SURFACES

26. **Dynamic Pressures on Fixed Surfaces.** When a stream of water impinges with a uniform velocity v on a smooth surface at rest, it glides over the surface and leaves it with the original velocity v, since there are supposed to be no frictional or other resistances, only its direction of motion being changed. The water, as it strikes the

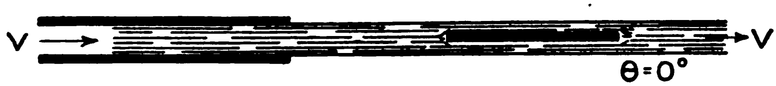


Fig. 10. Illustrating Case of No Dynamic Pressure.

surface, exerts upon it an impulse F in the direction of the path of entry; as it leaves the surface, it exerts on it an equal reaction F, in a direction opposite to its path of exit (see Figs. 11 to 14). The dynamic pressure thus developed depends upon velocity v, and change of direction of stream (angle θ). The stream is assumed to be moving horizontally while in contact with the surface, so that its velocity is not affected by gravity.

27. **Resultant Dynamic Pressure.** From the principle of Composition of Forces (Mechanics), the resultant dynamic pressure upon a fixed surface struck by a jet may be readily found by constructing the parallelogram of the forces of impulse and reaction, as shown in Fig. 15, in which $ab = bc = F = R$; from which we deduce (Trigonometry) that the value of this resultant pressure is:

$$P_R = F\sqrt{2(1-\cos\theta)} = 2 \text{ Sin } \frac{1}{2}\theta. \; W\frac{v}{g}. \quad . \quad . \quad \textbf{(25)}$$

and that it makes an angle of $(90° - \frac{1}{2}\theta)$ with the original direction of the jet. Its line of action passes through the intersection of F and R, and it bisects the angle between them.

28. **Dynamic Pressure Parallel to Initial Direction of Jet.** This is simply the component of the Resultant Dynamic Pressure in the

desired direction. From Fig. 16, this is found to be (Resolution of Forces) $ab = bc \cos(90 - \frac{1}{2}\theta)$; so that,

$$P_J = P_R \cos\left(90 - \frac{1}{2}\theta\right) = (1 - \cos\theta)\, W \frac{v}{g} \quad \ldots (26)$$

If, in this equation, $\theta = 0$, the stream glides over the surface without change of direction or retardation of velocity, and $P = 0$;

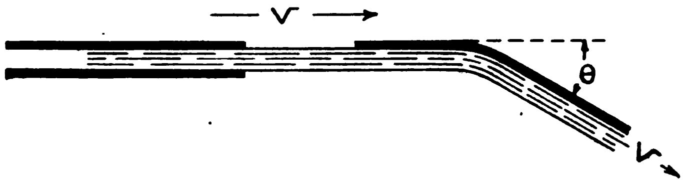


Fig. 11.

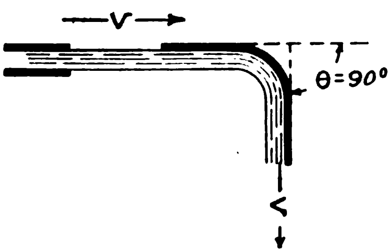


Fig. 12.

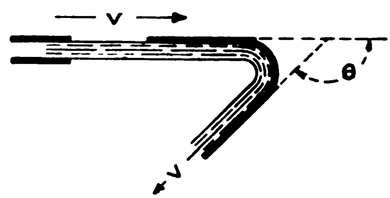


Fig. 13.

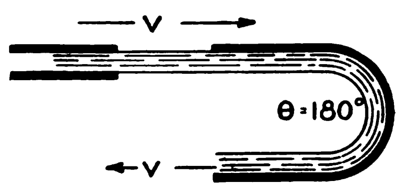


Fig. 14.

Illustrating Dynamic Pressure of Jet on Various Fixed Surfaces.

that is, no dynamic pressure is exerted upon the surface (see Fig. 10).

If $\theta = 90°$, $\cos\theta = 0$ (see Figs. 7 and 12), and therefore the dynamic pressure is:

$$P = F = W \frac{v}{g}$$

Here the escaping jet has no component of velocity normal to the surface; therefore the reaction has no influence on the pressure.

If $\theta = 180°$ (see Figs. 9 and 14), indicating a complete reversal in the direction of the stream, $\cos \theta = -1$; hence the dynamic pressure is:

$$P = 2F = 2W \frac{v}{g}.$$

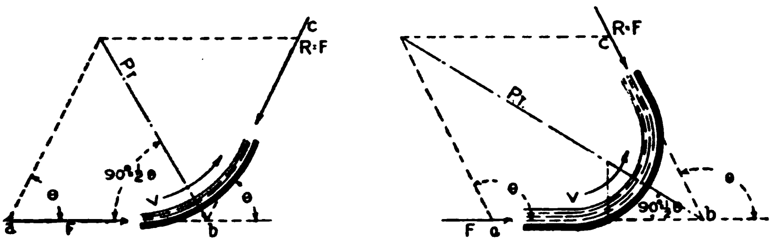


Fig. 15. Resultant Dynamic Pressure.

Here the pressure is a consequence of both impulse and reaction to their full amount.

29. **Dynamic Pressure in Any Given Direction.** It is frequently of importance to determine the dynamic pressure *in a given direction* exerted on a fixed surface by a stream of water. This may be ascertained by resolving the *resultant* dynamic pressure into its two components, parallel and at right angles to the required direction; the former represents the pressure in the required direction. Or the impulse and reaction may be separately resolved into their rectangular components, as above, and the algebraic sum taken of the two components parallel to the required direction. Thus, in Fig. 17, let it be required to find the dynamic pressure in a direction represented by the arrow x, which makes an angle a with the direction of the entering, and an angle θ with that of the departing stream. The components of the impulse and the reaction in the required direction, since $R = F$, are:

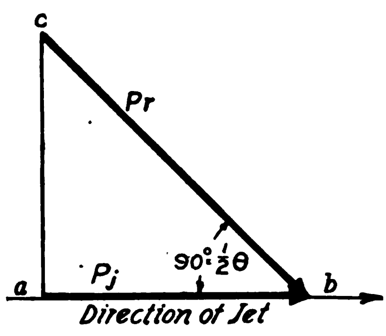


Fig. 16. Dynamic Pressure Parallel to Initial Direction of Jet.

$$P_1 = F \cos \alpha; \text{ and } P_2 = -F \cos \theta;$$

and therefore:

$$P = P_1 + P_2 = F(\cos \alpha - \cos \theta) = (\cos \alpha - \cos \theta) W \frac{v}{g}. \quad \textbf{(27)}$$

If, in this general equation (27), $a = 0°$,

$$P = (1 - \cos \theta)\, W \frac{v}{g},$$

as in Equation 26.

If $a = 0°$, and $\theta = 90°$,

$$P = F = W \frac{v}{g}, \text{ as in Figs. 7 and 12}$$

If $a = 0°$, and $\theta = 180°$,

$$P = 2F = 2W \frac{v}{g} \quad \text{as in Figs. 9 and 14.}$$

If $a = 0°$, and $\theta = 0°$, $P = 0$, as in Fig. 10.

Example 13. Let the jet of Problem 7 impinge tangentially upon the fixed curved vane of Fig. 15, with $\theta = 60°$. What is the resultant dynamic pressure upon the vane, in intensity and direction? What is the dynamic pressure in a direction parallel to the jet? What is the dynamic pressure in a direction making an angle of 30 degrees with the direction of the jet?

From Equation 25 and Problem 7:

$$P_R = 2 \sin \tfrac{1}{2} \theta\, W \frac{v}{g}$$

$$= 2 \times \tfrac{1}{2} \times \frac{.33 \times 62.5 \times 38.5}{32.2} = 24.7 \text{ pounds.}$$

From Equation 26:

$$P_1 = (1 - \cos \theta)\, W \frac{v}{g}$$

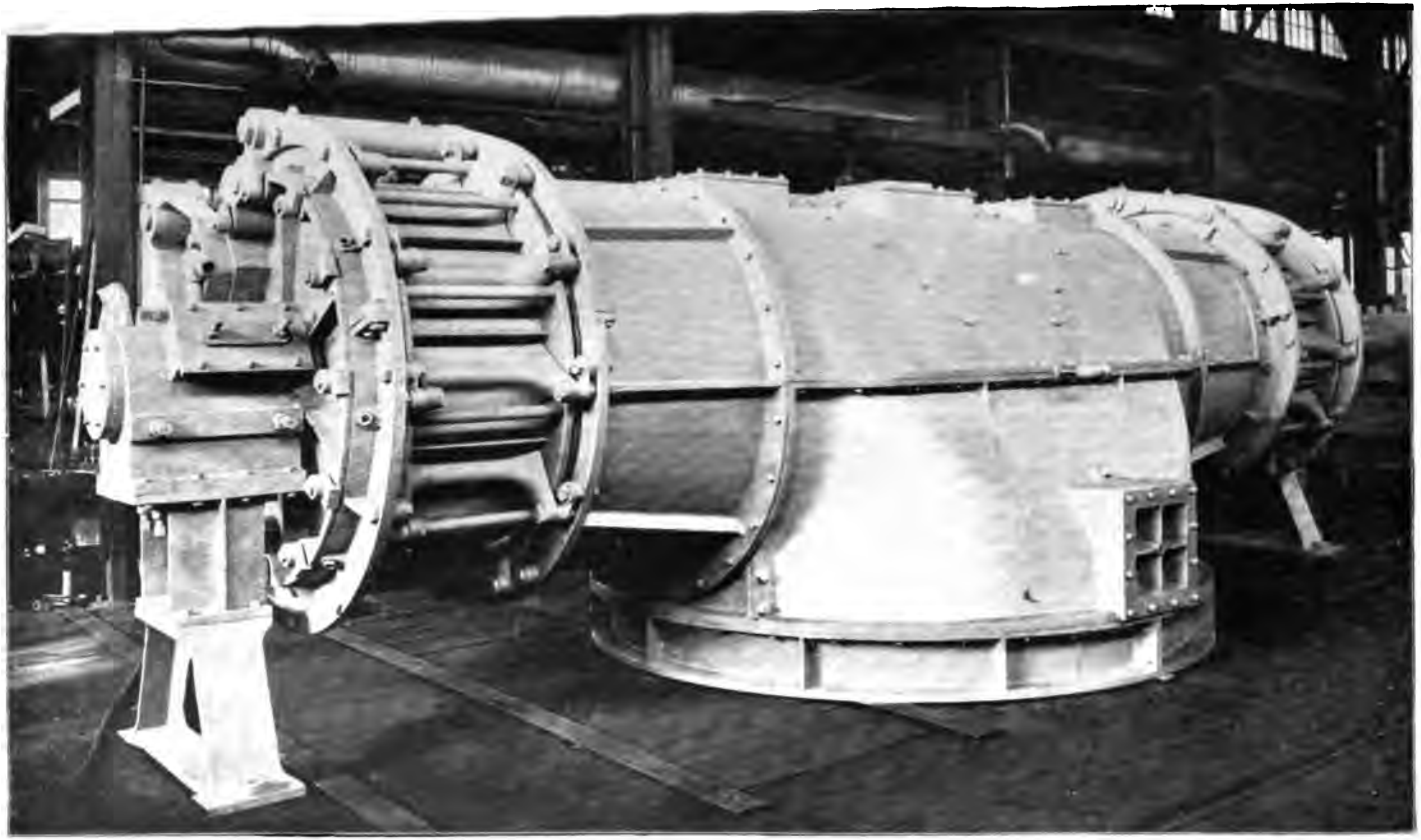
$$= (1 - \tfrac{1}{2}) \frac{.33 \times 62.5 \times 38.5}{32.2} = 12.4 \text{ pounds.}$$

From Equation 27 (Fig. 17):

$$P = (\cos a - \cos \theta)\, W \frac{v}{g}$$

$$= (.866 - .500) \frac{.33 \times 62.5 \times 38.5}{32.2} = 4.5 \text{ pounds.}$$

30. **Weight of Water Impinging.** In all the preceding equations, W represents the weight of water in pounds per second impinging upon the surface; and, since the surface has in each case been assumed to be stationary, W is also the weight of water in pounds per second issuing from the nozzle or orifice, or flowing in the stream. It is to be clearly kept in mind that this statement is not necessarily true if the surface is supposed to move; as, for example, in the case of a jet impinging upon the vanes or blades of a water wheel. Such cases will be considered later.



UNIT CONSISTING OF TWO 56-INCH, HORIZONTAL-SHAFT, 1,500-H. P. "SAMSON" WATER TURBINES

Courtesy of James Leffel & Co., Springfield, Ohio.

31. **Force and Work.** It must also be clearly realized that the dynamic pressures are *forces*; they are not expressed in terms of *energy* or *work*; just as a weight resting upon a table produces *pressure* thereon, but does not perform *work*. A force must be exerted against a resistance through a definite distance, in order that work may be done; the weight may be allowed to move, and thereby compress a spring, for example, thus doing work. Similarly, the above pressures must be exerted against resistances over some definite distances, in order that work may be done. In general, if P is the

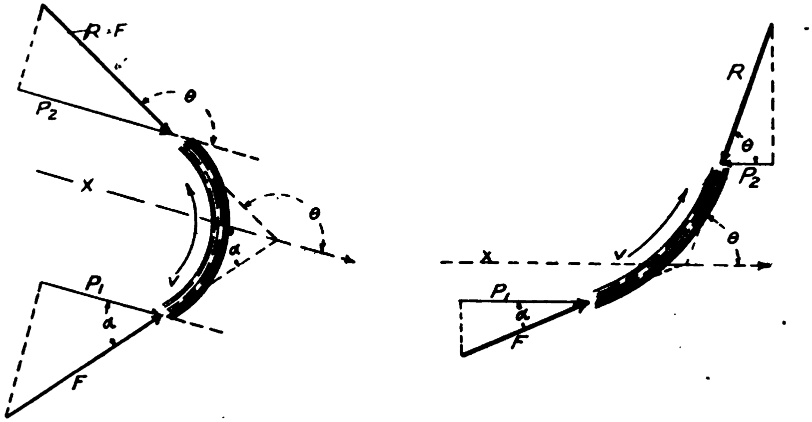


Fig. 17. Dynamic Pressure in Any Given Direction.

dynamic pressure on the surface in pounds, and if the surface is supposed to move a distance of u feet per second while overcoming some resistance, then,

$$\text{Work} = P \times u \text{ foot-pounds per second} \quad . \; . \; . \quad (28)$$

It is by reason of the dynamic pressures defined and explained above, produced by a retardation in velocity, or a change in direction of flow, that turbine wheels and other water-motors are able to transform the kinetic energy of moving water into useful work—such pressures being exerted over definite distances against resistances.

32. **Losses of Energy.** In the above discussion, no frictional or other losses of energy were considered. It is clear that if the surfaces are rough, or if the jet impinges on the surface in such a way as to produce "shock" or "eddies" or "foam," some of the original energy of the jet will be dissipated as heat, and the resulting pressures will be correspondingly reduced below the values indicated by the fore-

going formulæ. These losses may be largely eliminated by having the surfaces smooth and properly curved, and by so directing the jet as to strike the surface tangentially.

ABSOLUTE AND RELATIVE VELOCITIES

33. **Definitions.** While all velocities are in reality relative, it is convenient to define *absolute velocity* as the rate of speed of a moving object with respect to the surface of the earth; and *relative velocity* as the rate of speed of a moving object with respect to another moving body—or as the velocity the object would appear to have to a person standing upon, and viewing it from, the second moving body. In the one case, velocity is measured from, or referred to, the earth,

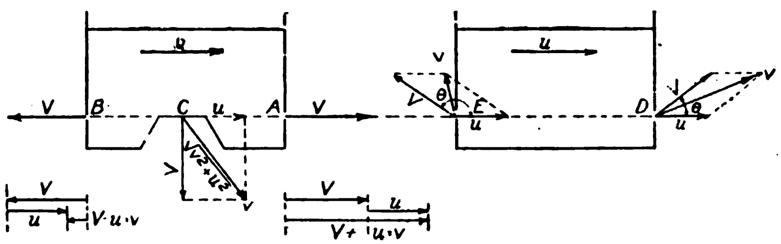


Fig. 18. Illustrating Absolute and Relative Velocities.

which is regarded as stationary; in the other case, the velocity is measured from, or referred to, the second moving body, regarded as stationary for this purpose. Thus, let Fig. 18 represent a tank so mounted that it may move horizontally to the right with a uniform absolute velocity of u feet per second; and let water issue from the various openings as indicated. Theoretically, the following absolute and relative velocities will result:

Orifice	Relative Velocity (to tank)	Absolute Velocity (referred to the earth)	θ	Cos θ
A	$V = \sqrt{2gh}$	$v = V + u$	0°	1
B	$V =$ "	$v = V - u$	180°	−1
C	$V =$ "	$v = \sqrt{V^2 + u^2}$	90°	0
D	$V =$ "	$v = \sqrt{V^2 + u^2 + 2Vu \cos \theta}$	θ	cos θ (positive)
E	$V =$ "	$v = \sqrt{V^2 + u^2 + 2Vu \cos \theta}$	θ	cos θ (negative)

The expression for absolute velocity from orifice D or E may be regarded as a general formula, and the formulæ for the other cases may be simply derived from it by assigning the proper values to θ. These considerations of absolute and relative velocities are of great importance in determining the dynamic pressures produced by a stream of water on the moving vanes or blades of water-motors. For example, consider Fig. 19, which represents a revolving wheel having an orifice from which water issues horizontally with the relative velocity V (velocity relative to wheel), while the orifice itself is moving horizontally with an absolute velocity u (velocity relative to the ground); then, from what has preceded.

$$v = \sqrt{V^2 + u^2 + 2\,Vu\cos\theta} \quad . \quad . \quad \text{(29)}$$

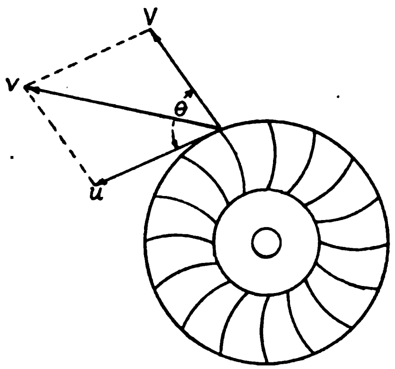


Fig. 19. Velocity of Stream Leaving or Striking Revolving Vane.

is the absolute velocity of the water as it leaves the wheel (velocity with respect to the ground). In all cases, then, *the absolute velocity of a stream of water striking or leaving a moving surface is represented in magnitude and direction by the diagonal of a parallelogram of which one side is the velocity of the stream relative to the moving surface, and the other side is the absolute velocity of that surface (with reference to the ground); i. e., it is the resultant of these two velocities.*

If the directions of the component velocities lie in the same straight line, $\theta = 0°$ or $180°$; and, applying Equation 29, we derive the special formulæ:

$$v = V + u;\ \text{or},\ v = V - u \quad . \ . \ . \ . \ . \ . \ . \quad \text{(29a)}$$

SURFACES MOVING IN A STRAIGHT LINE

34. **Dynamic Pressure on Moving Surfaces.** When a stream of water impinges upon a moving surface, the conditions are essentially different from those just discussed for surfaces at rest. Because the surface is continually moving away from the stream, two important results follow—the stream does not strike the surface with its full or absolute velocity, and the quantity of water reaching the surface per second is less than the stream discharge.

35. Case I. Jet Striking a Moving Flat Vane Normally. Let a jet (Fig. 20) whose absolute velocity is v, and cross-section a, impinge normally upon a smooth surface which is itself moving with a uniform absolute velocity u in the same direction as the jet. The *relative* velocity of the jet, or the velocity with which it strikes the surface, is $v-u$; the weight of water *leaving the orifice* per second is

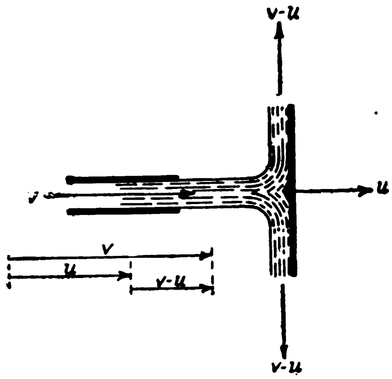


Fig. 20. Jet Striking a Moving Flat Vane Normally.

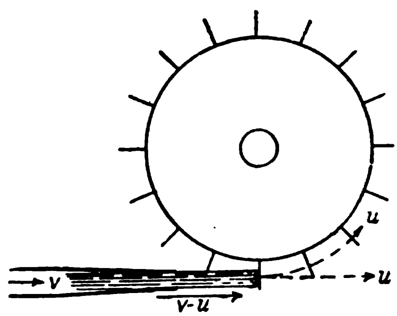


Fig. 21. Jet Striking Flat Radial Vanes of a Revolving Wheel.

$W = wav$; the weight of water *striking the surface* per second is $wa(v-u)$, if w represents the weight of a cubic unit of water; accordingly, the dynamic pressure exerted upon the surface, in the direction of motion, is:

$$P = wa\,(v-u)\,\frac{(v-u)}{g} = \frac{wa}{g}\,(v-u)^2, \quad \ldots\ldots \mathbf{(30)}$$

which is equivalent to considering the surface stationary, and the stream moving with an absolute velocity of $(v-u)$ feet per second.

36. *Work Done upon (or Given Up to) the Moving Body per Second.* The work done in one second by the force P (Force × Distance) is:

$$\text{Work} = Pu = \frac{wa\,(v-u)^2\,u}{g} \quad \ldots\ldots\ldots \mathbf{(31)}$$

The work is zero if $u = v$; or $u = 0$; and it is a maximum, and equal to:

$$\text{Work (Max.)} = \frac{4}{27}\,\frac{wav^3}{g} = \frac{8}{27}\,W\,\frac{v^2}{2g} \quad \ldots \mathbf{(32)}$$

when $u = \frac{1}{3}\,v$.

37. *Efficiency.* Since the theoretic energy of the impinging jet is $W\frac{v^2}{2g}$, the efficiency in the case just considered is $\frac{8}{27}$, or about

30 per cent. It is evident, however, that no practical motor could be constructed on such a plan.

Case II. This represents a wheel (Fig. 21) provided with many flat radial vanes against which, in rapid succession, a jet of water impinges. The resultant action of the jet in this case is not precisely the same as in the preceding example; but if we assume that the jet impinges normally on the vanes, and that, as the vanes come in rapid succession under the influence of the jet, and several vanes are more or less under action at the same time, the quantity of water impinging is the same as the nozzle discharge ($W = wav$); also, that the vanes move away from the jet in the direction of the latter while under impact, then we obtain for the *approximate* value of the dynamic pressure, if u represents the linear absolute velocity of the vanes at the center of impact:

$$P = W \frac{v - u}{g} = wav \frac{v - u}{g} \quad \ldots\ldots\ (33)$$

38. *Work Done upon (or Given Up to) the Wheel per Second.*

$$\text{Work} = Pu = \frac{wav}{g}(v - u)u \quad \ldots\ldots\ (34)$$

The work is zero if $u = v$, or $u = 0$; and it is a maximum and equal to:

$$\text{Work (Max.)} = \tfrac{1}{4}\frac{wav^3}{g} = \tfrac{1}{2}W\frac{v^2}{2g} \quad \ldots\ (35)$$

when $u = \frac{1}{2}v$.

39. *Efficiency.* Since the jet has a theoretic energy of $W\dfrac{v^2}{2g}$ foot-pounds, it is seen that the highest efficiency that can theoretically be obtained by means of a jet impinging upon rotating flat vanes is 50 per cent.

The preceding analysis applies more directly to the case of a series of flat vanes moving in a straight line, as indicated in Fig. 20, and coming in rapid succession under the influence of the jet. A motor constructed on this plan is, however, impracticable.

40. Case III. Jet Striking a Moving Curved Vane Tangentially. Fig. 22 represents a case in which the jet, with an absolute velocity v, impinges tangentially upon a vane which moves in the same direction with the uniform absolute velocity u. The velocity of the stream relative to the surface is $v-u$; and the dynamic pressure is the same as though the surface were at rest, and the stream

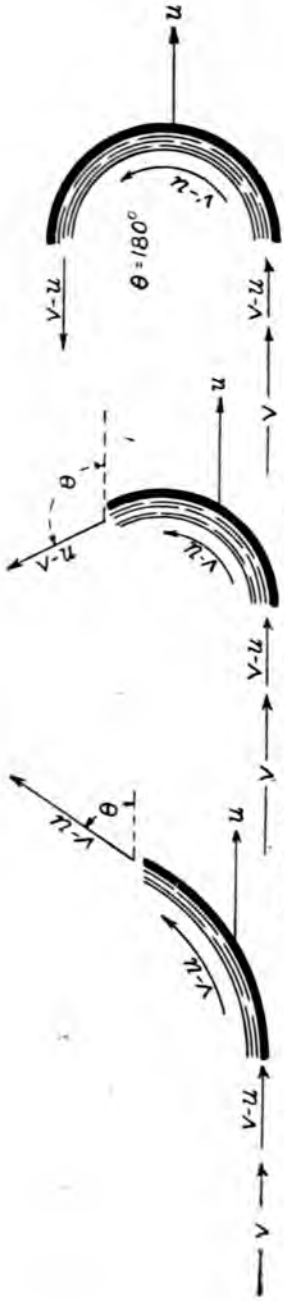


Fig. 23. Jet Striking Moving Curved Vane Tangentially.

moving and impinging with the absolute velocity $v-u$. Hence, for the dynamic pressure in the direction of the jet, we may use Equation 26, substituting $v-u$ for v; so that,

$$P = (1-\cos\theta)\,W\frac{v-u}{g} \quad \ldots\ldots \quad \textbf{(36)}$$

While the dynamic pressure may be exerted with different intensities upon different parts of the vane, the total value, in the direction of motion, is that indicated by Equation 36.

41. *Work Done.* If a is the area of the cross-section of the jet, the weight of water issuing from the nozzle per second is $W = wav$; the weight striking the vane is $wa\,(v-u)$; and therefore the work is:

$$\text{Work} = Pu = (1-\cos\theta)\frac{wa}{g}(v-u)^2\,u \quad \ldots\ldots \quad \textbf{(37)}$$

The work is zero when $v = u$, and when $u = 0$; also when $\theta = 0°$; and it is a maximum, and equal to:

$$\text{Work (Max.)} = \frac{4}{27}(1-\cos\theta)\,wa\,\frac{v^3}{g} = \frac{8}{27}(1-\cos\theta)\,W\frac{v^2}{2g} \quad \ldots\ldots \quad \textbf{(38)}$$

when $u = \frac{1}{3}v$.

42. *Efficiency.* Since the theoretic energy of the impinging jet is $W\frac{v^2}{2g}$, the efficiency is:

$$e = \tfrac{8}{27}(1-\cos\theta) \quad \ldots\ldots \quad \textbf{(39)}$$

If $\theta = 0°$, work $= 0$, and $e = 0$; in this case the vane is a flat surface whose plane is in the direction of the stream, which therefore glides over the surface without doing work.

If $\theta = 90°$, the water leaves the vane at right angles to the direction of motion, and the maximum work, from Equation 38, is:

$$\text{Work (Max.)} = \tfrac{8}{27} W \frac{v^2}{2g} \quad \ldots\ldots\ldots (40)$$

and the efficiency is $\frac{8}{27}$, or about 30 per cent. (Compare with Equation 32.)

If $\theta = 180°$, the stream is completely reversed. In this case, (since $\cos 180° = -1$),

$$\text{Work (Max.)} = \tfrac{16}{27} W \frac{v^2}{2g} \quad \ldots\ldots\ldots (41)$$

and the efficiency is $\frac{16}{27}$, or about 60 per cent.

43. Case IV. If, instead of a simple curved vane, as in the preceding case, we consider a wheel with a large number of such vanes, as in Fig. 23, and assume the jet to impinge tangentially, and the vanes to move in the direction of the jet while under its influence, and also the quantity of water impinging to be equal to the nozzle discharge, by an analysis similar to that which has preceded, we obtain:

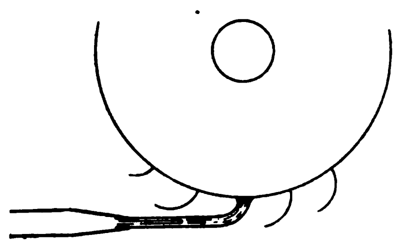


Fig. 23. Jet Striking Curved Vanes of a Revolving Wheel Tangentially.

44. The work is:

$$\text{Work} = (1 - \cos \theta)\, W \frac{(v - u)\, u}{g} \quad \ldots\ldots (41a)$$

This is zero when $u = 0$, or when $u = v$; also when $\theta = 0°$; and it is a maximum, and equal to:

$$\text{Work (Max.)} = \tfrac{1}{2}(1 - \cos \theta)\, W \frac{v^2}{2g} \quad \ldots\ldots (42)$$

when $u = \frac{1}{2}v$.

45. *Efficiency.* The efficiency is:

$$e = \tfrac{1}{2}(1 - \cos \theta) \quad \ldots\ldots\ldots\ldots\ldots (43)$$

When $\theta = 0°$, the stream merely glides along the surface without doing work, and $e = 0$.

When $\theta = 90°$, the jet is deflected normally to the direction of motion, and,

$$\text{Work (Max.)} = \tfrac{1}{2} W \frac{v^2}{2g} \quad \ldots\ldots\ldots\ldots (44)$$

and efficiency is $e = \frac{1}{2}$, or 50 per cent, as for radial flat vanes.

When $\theta = 180°$, the stream is completely reversed, and

$$\text{Work (Max.)} = W\frac{v^2}{2g} \quad \ldots\ldots\ldots\ldots (45)$$

in which case the efficiency is $e = 1$, or 100 per cent. The preceding analysis applies more directly to the case of a series of curved vanes moving in a straight line parallel to the jet, and coming in rapid succession under its influence. Such a motor is evidently impracticable.

46. In applying these considerations to water wheels, we must bear in mind that losses due to impact and friction have not been considered. The conclusions are therefore, to that extent, theoretic; but they represent limiting values which may be approached more and more closely, as the frictional and other resistances are reduced by means of correct design and construction. In the case of the conditions represented by Equation 45, since the efficiency is theoretically 100 per cent, it is clear that all the energy of the jet has been given up to the wheel, which would indicate that the absolute velocity of the water leaving the vanes must be zero; for if the water thus leaving has any absolute velocity, it still possesses some energy after passing clear of the wheel, which represents a portion of the original energy of the jet which has not been imparted to the wheel; the efficiency then could not be 100 per cent. This conclusion may be readily reached from the preceding analysis; for, since the best *absolute* velocity of the vane is $\frac{1}{2}v$, the water upon its surface has the *relative* velocity $v - \frac{1}{2}v = \frac{1}{2}v$, which is the same as the velocity of the vane, but in the *opposite direction;* then, if $\theta = 180°$, as in the case under discussion, the *absolute velocity of the water* as it leaves the vane, is $\frac{1}{2}v - \frac{1}{2}v = 0$.

While the above discussion shows that for maximum efficiency the velocity of the vanes should be one-half the velocity of the jet, the efficiency is not much lowered by slight variations of the vane velocity above or below the value indicated. It is also clear that to thus realize the full energy of the stream, we suppose the jet to both enter and leave the vanes in a direction tangential to the circumference, and a complete reversal is effected. It will be shown in a subsequent article that certain practical considerations render it impossible to fully realize these theoretic conditions.

47. If the vanes are plane radial surfaces, as in Fig. 21, the water passes from the wheel normally to the circumference, and

the highest obtainable efficiency is (theoretically) 50 per cent (Equation 35). In this case the water leaving the wheel still possesses absolute velocity to the extent of $\frac{v}{\sqrt{2}}$, the component of which, in the direction of motion of the vanes, is $\frac{1}{2}v$; this represents a dynamic pressure of $W\frac{\frac{1}{2}v}{g}$ pounds in that direction, or $W\frac{\frac{1}{2}v}{g} \times \frac{1}{2}v$ $(= P \times u)$ $= \frac{1}{2}W\frac{v^2}{2g}$ foot-pounds of work; that is, one-half of the original energy of the jet is carried away by the escaping water, and is thus lost to the wheel. Or, an absolute velocity of $\frac{v}{\sqrt{2}}$ represents kinetic energy to the amount of $\frac{W\left(\frac{v}{\sqrt{2}}\right)^2}{2g} = \frac{1}{2}W\frac{v^2}{2g}$. Equation 58 shows even more clearly that in order to realize the full theoretic energy of the stream, the absolute velocity of the departing water ($v_1 = \frac{v}{\sqrt{2}}$ for this case) must be zero.

48. CASE V. GENERAL. In the usual case the direction of motion of the vane is not the same as that of the jet. In Fig. 24, let the arrow marked v represent the direction of the jet as it impinges on the vane with an absolute velocity v; and let the arrow marked u represent the direction of motion of the vane, as well as its absolute velocity. While this case can be analyzed and solved in a manner similar to that employed in the preceding cases, it will be well here to adopt another procedure illustrating an important and useful principle:

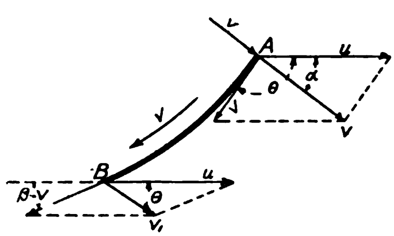


Fig. 24. General Case of a Jet Impinging on a Moving Vane.

The difference between the components of the absolute impulses of the entering and departing streams, in the direction of motion, is the resultant dynamic pressure in that direction.

49. *Dynamic Pressure in a Given Direction.* The absolute velocity of entry v being known, it remains to determine the absolute velocity of exit, v_1. By means of the principle enunciated in Article 33, we first find the relative velocity V with which the jet strikes the surface at A, by drawing to scale the lines v and u (both known) and

completing the parallelogram. V then represents, in intensity and direction, the relative velocity of the stream at A. The stream passes over the surface, and leaves it at B with this same relative velocity, if not retarded by friction or shock. Now, by the principle just referred to and used for the point A, the absolute velocity of the stream as it leaves the vane at B may be determined. Draw u and V, and complete the parallelogram; v_1 then represents the absolute velocity of the escaping water at B.

The absolute impulse of the stream before striking the vane at A is $W\frac{v}{g}$; its component in the direction of motion is $W\frac{v}{g}\cos a$. The absolute impulse of the stream as it leaves the vane at B is $W\frac{v_1}{g}$; its component in the direction of motion is $W\frac{v_1}{g}\cos\theta$. Hence the dynamic pressure in the direction of motion is:

$$P = W\frac{v\cos a - v_1\cos\theta}{g} \quad \ldots\ldots\ldots (46)$$

This is a general formula for the dynamic pressure in any given direction exerted by a jet of water upon a vane moving in a direction parallel to a straight line, if a and θ be the angles between that direction and the directions of v and v_1.

If the surface is at rest, $v = v_1$, and Equation 46 becomes $P = (\cos a - \cos\theta)\, W\frac{v}{g}$, which is Equation 27.

50. Usually, in the case represented by Fig. 24, the angles a and β are known, or assumed, and θ is unknown; it therefore becomes desirable to express the angle θ in other and known terms. By taking the components of the velocities at B in the direction of motion, it is evident that $v_1\cos\theta = u - V\cos\beta$; if this value be substituted in Equation 46, there will result:

$$P = W\frac{v\cos a - u + V\cos\beta}{g}, \quad \ldots\ldots (47)$$

in which,

$$V^2 = u^2 + v^2 - 2uv\cos a \text{ (Trigonometry, from the triangle } A\,u\,v) \quad \ldots (47a)$$

51. *Curvature of Vane at Entrance.* In order that the stream may strike the vane without shock, the curve of the vane at A should be tangent to the direction of V. It therefore becomes important to express the angle ϕ in known terms. From either triangle at A,

making use of the trigonometric principle that the sides of any plane triangle are proportional to the sines of their opposite angles, we obtain:

$$\frac{\text{Sin}(\phi -)\alpha}{\text{Sin } \phi} = \frac{u}{v} \quad \ldots\ldots\ldots\ldots (\mathbf{48})$$

which may be reduced, by known trigonometric relations, to:

$$\cot \phi = \cot \alpha - \frac{u}{v \sin \alpha} \quad \ldots\ldots\ldots (\mathbf{48a})$$

Equation 48*a* determines the angle ϕ, when u, v, and the angle α are known; and this fixes the proper curvature of the vane at the point A.

Example 14. In Fig. 24, let $u = 70.71$, $v = 100$, $\alpha = 45°$, and $\beta = 30°$. What is the dynamic pressure on the vane in the direction of motion, when 1 cubic foot of water strikes the vane per second? What should be the value of the angle ϕ in order that no loss by impact may occur?

From Equation 47*a*:

$$V = \sqrt{70.71^2 + 100^2 - 2 \times 70.71 \times 100 \times .707} = 70.71 \text{ feet per second.}$$

From Equation 47:

$$P = 62.5 \frac{100 \times .707 - 70.71 + 70.71 \times .866}{32.2} = 1{,}356 \text{ pounds.}$$

From Equation 48*a*:

$$\text{Cot } \phi = 1 - \frac{70.71}{100 \times .707} = 0 ; \quad \therefore \phi = 90°$$

REVOLVING SURFACES

52. Case VI. In the case of water motors, the vanes upon which the jet impinges revolve about an axis. The motion of every point on the vane is therefore circular; hence, at any instant, the direction of motion of any point is tangent to the circumference drawn through, or it is normal to the radius drawn to, that point. At any point, therefore, that portion of the dynamic pressure which is effective in producing motion is its component in the direction of motion of that point. Fig. 25 illustrates two cases of wheels with vertical axes, the vanes revolving in horizontal planes. In the one case (B), the water, after impinging, passes outward, or away from the axis; in the other (a), the stream passes inward, or toward the axis. The following analysis, however, is general, and therefore applies to both types. As heretofore, v and v represent the absolute,

and V and V_1 the relative velocities of the entering and departing streams; u and u_1 (drawn normal to the radii r and r_1) represent the absolute velocities and directions of motion of the points A and B on the vane; the angles to be used in the analysis are sufficiently clear from the diagram, in view of what has preceded. Constructing the two parallelograms in the usual manner, there is obtained, at

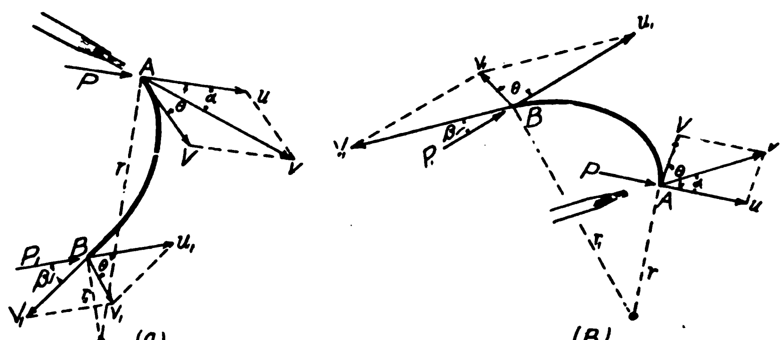


Fig. 25. Wheels with Vertical Axes, the Vanes Revolving in Horizontal Planes.

the point A, V as the relative velocity of the entering stream; and at the point B, v_1 as the absolute velocity of the departing stream. For the parallelogram at B, however, the value of V_1 must first be computed by means of Equation 54.

53. *Components of Pressures in Direction of Motion.* The total dynamic pressure exerted in the direction of motion will depend upon the impulses of the entering and the departing streams. The absolute impulse of the water on entering is $W\frac{v}{g}$; and that of the water on leaving is $W\frac{v_1}{g}$. The components of these in the directions of the motion of the vane at entrance and departure, are respectively:

$$P = W\frac{v \cos \alpha}{g}\ ; \text{ and } P_1 = W\frac{v_1 \cos \theta}{g} \quad \ldots\ldots (49)$$

Since their directions are not parallel, and the velocities of the points A and B are not equal, their difference cannot be taken to give the resultant dynamic pressure, as was done in Case V, which represented motion in a straight line; but this resultant pressure is not important. The two expressions in Equation 49, however, are useful in an analysis of the work that can be delivered by the vane.

54. **Useful Formulæ.** Since in any rotating body the linear velocities of points are directly proportional to their distances from the axis of rotation,

$$\frac{r}{r_1} = \frac{u}{u_1} \quad \ldots\ldots\ldots\ldots\ldots \quad \textbf{(50)}$$

The relative velocities V and V_1 are connected with the velocities of rotation by the following simple relation:

$$V_1^2 - V^2 = u_1^2 - u^2 \quad \ldots\ldots\ldots\ldots \quad \textbf{(51)}$$

Ordinarily, for a revolving vane, the data given or assumed will be the angles α, ϕ, and β; the radii r and r_1; the absolute velocity of the jet, v; the number of revolutions per second, n; and the weight of water delivered to the vane per second, W. Then,

$$u = 2\pi r n\ ; \quad \text{and } u_1 = 2\pi r_1 n, \quad \ldots\ldots \quad \textbf{(52)}$$

from which u and u_1 may be determined.

In the triangle Auv (sides are proportional to sines of opposite angles),

$$V = \frac{v \sin \alpha}{\sin \phi}, \quad \ldots\ldots\ldots\ldots\ldots \quad \textbf{(53)}$$

which determines the relative velocity of entrance, V.

From Equation 51:

$$V_1 = \sqrt{u_1^2 - u^2 + V^2}, \quad \ldots\ldots\ldots\ldots \quad \textbf{(54)}$$

which gives the value of the relative velocity of exit, V_1. Finally, taking the components of the velocities at B in the direction of motion of that point, there results:

$$v_1 \cos\theta = u_1 - V_1 \cos\beta \quad \ldots\ldots\ldots\ldots \quad \textbf{(55)}$$

From the above equations, the numerical values of P and P_1 of Equation 49 can be fully determined.

Example 15. In Fig. 25; suppose $r = 2$ ft.; $r_1 = 3$ ft.; $\alpha = 45°$; $\phi = 90°$; $v = 100$ ft. per second; $n = 6$ revolutions per second. Compute the velocities u, u_1, V, and V_1.

From Equation 52:

$$u = 2 \times 3.1416 \times 2 \times 6 = 75.4 \text{ feet per second.}$$

$$u_1 = \frac{3}{2}u \qquad = 113.1 \quad \text{"} \quad \text{"} \quad \text{"}$$

From Equation 53:

$$V = \frac{100 \times .707}{1} = 70.71 \text{ feet per second.}$$

From Equation 54:

$$V_1 = \sqrt{\overline{113.1^2 - 75.4^2} + 70.71^2} = 110 \text{ feet per second.}$$

55. **Work Derived from Revolving Vanes.** In the discussion of "Work" and "Efficiency" under Cases IV and V, it was assumed that all points of the vane move with the same velocity; and in Case IV, that the stream enters upon it in the same direction as that of motion, or that $a = 0$. Considering the general case just discussed, it may be said that the work of a series of vanes arranged around a wheel may be regarded as that due to the absolute impulse of the entering stream in the direction of motion of the point of entrance, minus that due to the absolute impulse of the departing stream in the direction of motion of the point of exit; or,

$$\text{Work} = Pu - P_1u_1 , \quad \ldots\ldots\ldots\ldots \quad \mathbf{(56)}$$

in which P and P_1 are the components of the dynamic pressures due to the absolute impulses at A and B, in the directions of motion of the points A and B, respectively, as shown in Fig. 25 and Equation 49. Using the values of Equation 49, in Equation 56, there results:

$$\text{Work} = W\frac{u\,v\cos\,\alpha - u_1v_1\cos\theta}{g} \quad \ldots\ldots \quad \mathbf{(57)}$$

This is a perfectly general formula, applicable to the work of all wheels with outward or inward flow. It shows that the useful work consists of two parts—one due to the entering, and the other to the departing stream.

Another very simple general expression for the work of a series of revolving vanes may be deduced as follows: The total absolute energy of the entering stream is $W\frac{v^2}{2g}$; the total absolute energy of the departing stream is $W\frac{v_1^2}{2g}$; hence, neglecting friction and other resistances, the difference represents the energy imparted to, or taken up by, the wheel from the stream; that is:

$$\text{Work} = W\frac{v^2 - v_1^2}{2g} \quad \ldots\ldots\ldots\ldots \quad \mathbf{(58)}$$

which is a useful formula of wide applicability. From Equation 58, the efficiency is:

$$e = \frac{v^2 - v_1^2}{v^2} = 1 - \left(\frac{v_1}{v}\right)^2 \quad \ldots\ldots\ldots \quad \mathbf{(59)}$$

Example 16. As a numerical example, consider the case of the outward-flow horizontal wheel driven by a jet from a fixed nozzle, shown in Fig. 26.

Let $r = 2$ feet;
$r_1 = 3$ feet;
$\alpha = 45°$ (approach angle);
$\phi = 90°$ (entrance angle);
$\beta = 15°$ (exit angle);
$v = 100$ feet per second;
$q = 2.2$ cubic feet per second;
$n = 337.5$ revolutions per minute.

It is required to find the useful work of the wheel, and its efficiency.

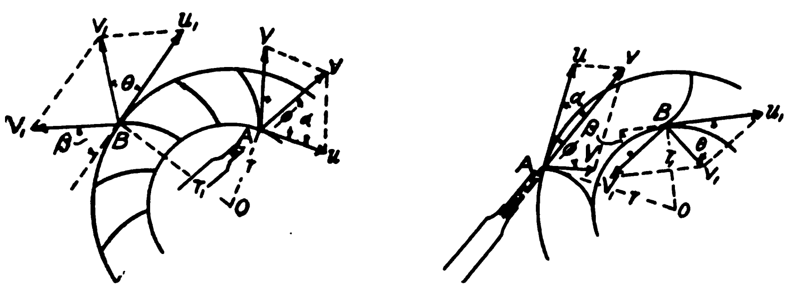


Fig. 26. Horizontal Wheels Driven by Jets from Fixed Nozzles.

From Equation 52:

$$u = 2\pi r n = 2 \times 3.1416 \times 2 \times \frac{337.5}{60} = 70.71 \text{ feet per second;}$$

and, from Equation 50:

$$u_1 = \frac{r_1}{r} u = \frac{3}{2} \times 70.71 = 106.06 \text{ feet per second.}$$

From Equation 53:

$$V = \frac{v \sin \alpha}{\sin \phi} = \frac{100 \times \sin 45°}{\sin 90°} = 100 \times 0.7071 = 70.71 \text{ feet per second.}$$

From Equation 54:

$$V_1 = \sqrt{u_1^2 - u^2 + V^2} = \sqrt{(106.06)^2 - (70.71)^2 + (70.71)^2} = 106.06 \text{ feet per second.}$$

From Equation 55:

$$v_1 \cos \theta = u_1 - V_1 \cos \beta = 106.06 - 106.06 \times \cos 15° = 3.61$$

Then, from Equation 57:

$$\text{Work} = 2.2 \times 62.5 \frac{70.71 \times 100 \times 0.707 - 106.06 \times 3.61}{32.2} = 19{,}712 \text{ ft.-lbs. per second.}$$

$$\frac{19{,}712}{550} = 35.8 \text{ horse-power.}$$

The theoretic energy of the jet is:

$$W\frac{v^2}{2g} = 2.2 \times 62.5\,\frac{(100)^2}{64.4} = 21{,}380 \text{ ft.-lbs. per second.}$$

$$\frac{21{,}380}{550} = 38.9 \text{ horse-power.}$$

Therefore the efficiency of the wheel is:

$$e = \frac{19{,}712}{21{,}380}, \text{ or } \frac{35.8}{38.9} = 92.2 \text{ per cent.}$$

This would seem to indicate a very high efficiency; but it must be borne in mind that losses in friction, shock, etc., have not been considered in the preceding analyses. The effect of such resistances will be to reduce the computed efficiency.

Example 17. In the above example, assume the same data, except that $\beta = 30°$.

The values of u, u_1, V, and V_1 are not altered.

$$v_1 \cos\theta = 106.06 - 91.85 = 14.21$$

and,

$$\begin{aligned} \text{Work} &= 14{,}910 \text{ ft.-lbs. per second,} \\ &= 27\ 2 \text{ horse-power.} \\ \text{Efficiency} &= 70 \text{ per cent.} \end{aligned}$$

In both of the above examples the work and efficiency may be simply computed from Equations 58 and 59, after the value of v_1 has been determined. From Fig. 24, parallelogram at B, since u_1 and V_1 are equal in the above examples, it follows that $\theta = \frac{1}{2}(180 - \beta)$; therefore, from Equation 55:

$$v_1 = \frac{u_1 - V_1 \cos\beta}{\cos\theta} = \frac{u_1 - V_1 \cos\beta}{\sin\frac{1}{2}\beta}$$

$$= \frac{106.06\,(1 - .966)}{.131} = 27.52 \text{ (for example 16);}$$

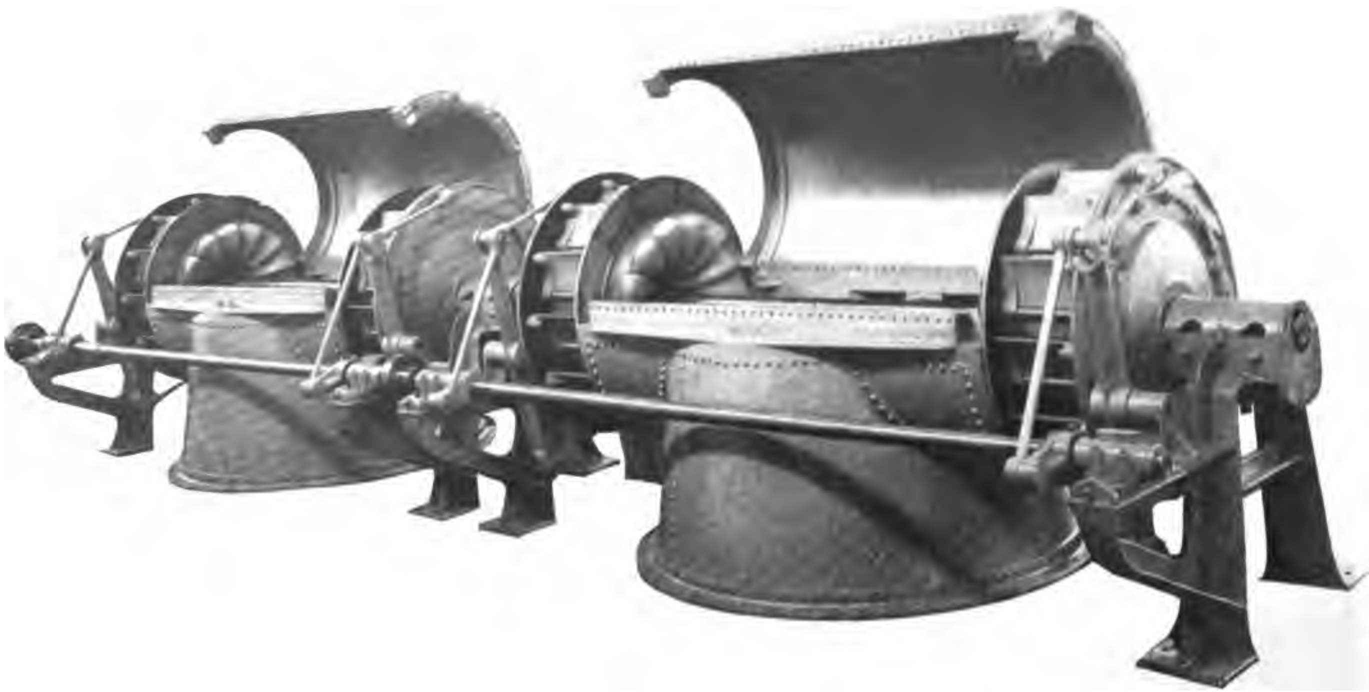
and,

$$v_1 = 106.06\,\frac{(1 - .866)}{.259} = 54.87 \text{ (for example 17).}$$

Substituting numerical values in Equations 58 and 59, the same results for the work and efficiency will be found as computed before.

HYDRAULIC MOTORS

56. **Definition.** A *hydraulic motor* may be defined as a machine in which the energy stored in water is utilized to produce motion and thus perform work. The energy of water, as was explained in Article 6,



UNIT CONSISTING OF TWO PAIRS OF 34-INCH "IMPROVED NEW AMERICAN" TURBINES ON HORIZONTAL SHAFT, FOR USE IN AN OPEN FLUME

View showing construction of central discharge cases, which admits of removal of upper half, thus giving easy access to the interior.

Courtesy of Dayton Globe Iron Works Company, Dayton, Ohio.

may exist in the form of gravity, of pressure, or of velocity; of these, gravity and pressure are not essentially or fundamentally separate and distinct phenomena, but rather the result of considering the weight of the water from different points of view. In general, then, it may be said that a hydraulic motor is an apparatus (usually a wheel) which is caused to move (usually rotate) by reason of a weight of water falling from a higher to a lower level, or because of the dynamic pressure induced by a change of direction, or of velocity, or both, in a moving stream. The dynamic pressure may be due to impulse, or reaction, or both. Many wheels are actuated by a combination in varying proportion of the above agencies, which are but manifestations of the energy existing in the water.

57. **General Requirements for High Efficiency.** The efficiency of any motor should, if possible, be independent of the quantity of water supplied to it; or, if the efficiency does vary with the supply, it should, when possible, be greatest in time of low water. It has already been shown that when W pounds of water fall through a height of h feet, or are delivered with a velocity of v feet per second, the theoretic energy in foot-pounds per second is:

$$K = Wh\ ; \quad \text{or } K = W\frac{v^2}{2g}.$$

58. The actual work performed, or that may be performed, per second is equal to the theoretic energy, *minus* all the losses of energy. It is convenient to subdivide these losses into four general classes:

(*a*) Losses incidental to the conduction of the water from the supply to the motor, occasioned by friction and the various other resistances usually encountered, such as bends, changes of section, passages through orifices or other controlling devices which are not essentially parts of the apparatus itself, etc.;

(*b*) Losses in passage through the motor, which include friction, losses in eddies resulting from abrupt change in cross-section and improper entrance angle, and losses in passage through controlling devices which form part of the apparatus, etc.;

(*c*) The residual energy still possessed by the departing water flowing away with an absolute velocity v_1;

(*d*) Shaft and journal friction.

Sometimes the friction of the moving parts in the air or water is included, but will not here be considered.

59. **Efficiency.** Let Wh' represent the energy lost in conduction; Wh'', that lost in passage through the wheel; $W\frac{v_1^2}{2g}$, the energy

still remaining in the departing water; and Wh''', the energy lost in shaft and journal friction; then,

$$k = W\left(h - h' - h'' - \frac{v_1^2}{2g} - h'''\right)$$

represents the actual useful work per second that the wheel is capable of performing. Accordingly, if v is the velocity due to the head h, the efficiency is:

$$e = \frac{k}{K} = 1 - \frac{h'}{h} - \frac{h''}{h} - \left(\frac{v_1}{v}\right)^2 - \frac{h'''}{h}.$$

This formula, being very general, leads to the four following broad statements of the conditions requisite for high efficiency:

(1) The water must be conducted to the motor, and

(2) The water must pass through the motor, with the minimum loss of energy.

(3) The water must reach the tail-race level with the minimum absolute velocity consistent with practical considerations, such as the necessity for quick and proper clearance of water from the buckets, etc.

(4) The friction and other mechanical resistances of the moving parts must be reduced to a minimum.

60. This analysis, with the corresponding formulæ, compares the energy of the entire waterfall with the ultimate output of the machine. In estimating the power and efficiency yielded by the motor itself, *regarded as a user of water delivered to it with a definite amount of energy*, certain of the above losses should be omitted. Thus, losses in the conduction of the water to the motor cannot properly be charged against the motor; nor should losses in journal and shaft friction, which are outside and independent of the wheel regarded as a water user; in fact, the overcoming of journal and shaft friction is part of the work performed by the wheel, though it is not *useful* work. The energy in the departing water is properly chargeable to the wheel, since it is directly dependent upon the design or construction of the wheel. Therefore the hydraulic efficiency of the wheel may be stated thus:

$$e = 1 - \frac{h''}{h} - \left(\frac{v_1}{v}\right)^2; \quad \dots\dots\dots\dots \quad \textbf{(60)}$$

or, as popularly stated, for high efficiency "the water should enter the wheel without shock, and leave without velocity." When the actual power and efficiency of a water motor are practically measured as described in Articles 115 *et seq.*, the shaft and journal friction and air or water resistance are automatically included in the result. This explains why the results of actual tests of power and efficiency are always lower than the corresponding values computed from formulæ derived without consideration of such losses. It is therefore well to employ two terms, *hydraulic efficiency* and *actual efficiency*, in order to distinguish clearly between the two sets of conditions involved.

61. **Classification.** In the absence of a uniform or generally accepted classification, hydraulic motors may be divided into two general classes:

(*a*) *Water-wheels*, in which the water does not enter and actuate the wheel around the entire circumference.

(*b*) *Turbines*, in which the water enters and actuates the wheel around the entire circumference.

Each of these main divisions has several subdivisions.

WATER-WHEELS

62. **Overshot Wheel.** In this form of wheel, the water enters at the top and acts mainly by its weight; nevertheless, in most forms, an appreciable amount of kinetic energy is likewise imparted to the wheel. Fig. 27 shows a vertical section of such a wheel. The *buckets* are formed by vanes or partitions made in two parts—one part *a* in line with the radius of the wheel, the other part *b* inclined in a direction definitely determined by the design. The bottom of the bucket is formed by the rim or *sole-plate F;* the side pieces are made by two *cheeks* or *shrouds E.* The whole is bolted to arms assembled on the hub, and supported by the axle.

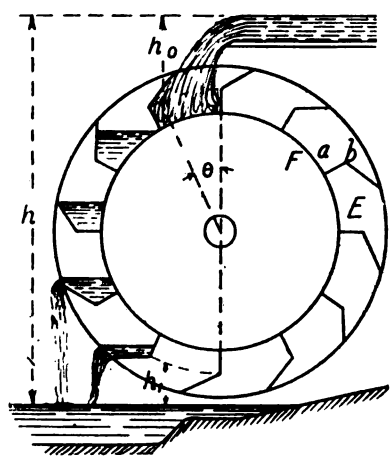


Fig. 27. Vertical Section of Overshot Wheel.

Let h be the total fall from the surface of the water in the head-race or flume to the surface of the water in the tail-race; and let W be the weight of water delivered to the wheel per second. The theoretic energy of the waterfall per second is Wh foot-pounds. The total fall h may be conveniently divided into three parts—namely,

h_0, the average head in filling the buckets; $h-h_0-h_1$, the average head of descent of the filled buckets; and h_1, that part of the head which remains between the empty buckets and the tail-race. The water strikes the buckets with a velocity v_0, approximately equal to $\sqrt{2gh_0}$; the buckets themselves are moving with a tangential velocity u approximately in the same direction as v_0; this occasions a loss of head in impact, h'' (Mechanics):

$$h'' = \frac{(v_0 - u)^2}{2g}.$$

The water then descends through the average distance $h-h_0-h_1$, acting by its weight alone; finally it drops out of the buckets, and reaches the level of the tail-race with the absolute velocity v_1, which represents part of the original energy wasted. Accordingly, the efficiency of the wheel is:

$$e = 1 - \frac{h''}{h} - \frac{v_1^2}{2gh}.$$

Since the water leaving the buckets has a velocity u when commencing the descent through height h_1, its velocity at the level of the tail-race is:

$$v_1 = \sqrt{u^2 + 2gh_1}.$$

Substituting the values h'' and v_1 in the equation of efficiency above,

$$e = 1 - \frac{v_0^2 - 2\,v_0\,u + 2\,u^2 + 2gh_1}{2gh},$$

and ascertaining by the usual procedure in such cases what value of u will render the efficiency e a maximum, it is readily found that:

$$u = \tfrac{1}{2}v_0; \qquad \textbf{(61)}$$

that is, theoretically, the velocity of the wheel should be one-half that of the entering water for maximum efficiency. With this value of u, the hydraulic efficiency is:

$$e\ (\text{Max.}) = 1 - \frac{1}{2}\frac{h_0}{h} - \frac{h_1}{h} \qquad \textbf{(62)}$$

and

$$\text{Work (Max.)} = Wh \times e = W\left(h - \frac{h_0}{2} - h_1\right) \qquad \textbf{(63)}$$

for the maximum efficiency and work of the overshot wheel. This equation teaches that one-half of the entrance drop h_0, and the whole of the exit drop h_1, are lost. Therefore, in order that the efficiency should be as high as possible, both h_0 and h_1 should be as small as practicable. The former requirement may be met by making the wheel of large diameter; but h_0 can never be zero, for in that case no water would enter the wheel; practically the size of wheel is usually such that θ equals 10 to 15 degrees. The fall h_1 is made small by giving to the buckets such a form that the water will be retained as long as possible, and by having as little clearance as practically advisable between the lowest point of the wheel and the tail-race level. In the design illustrated in Fig. 28, the buckets are deep in order to hold the water as long as possible; and moreover, they are shaped to conform to the direction of the entering water, thereby avoiding shock. Wheels of this description have been constructed 50 feet in diameter. In this case the power is taken from the axle of the small pinion, which is driven by a toothed ring attached to the circumference of the wheel. In other cases the power may be taken directly from the shaft of the water-wheel, through intermediate gearing, or by a crank-shaft.

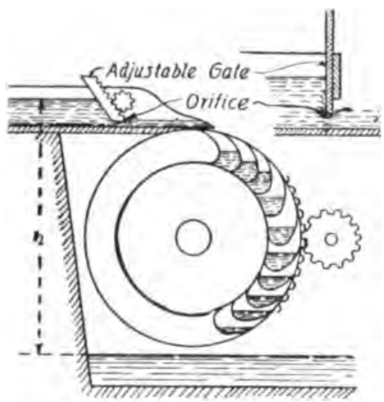


Fig. 28. Overshot Wheel with Deep Buckets to Hold Water as Long as Possible.

The method of regulating the supply of water to the wheel is also shown in the diagram. The theoretic advantageous velocity of the overshot wheel was shown to be $u = \frac{1}{2}v_0$; practically, this advantageous velocity is found to be about $u = 0.4\,v_0$; and the efficiency of the wheel is high, ranging from 70 to 85 per cent, or over. One great advantage of the overshot wheel is that its efficiency is highest in times of drought, when the supply is low, for then the buckets are but partly filled, they do not begin to empty at as high a point above tail-water as when they are full; hence h_1 becomes small, with corresponding increase in efficiency. The main disadvantage of the overshot wheel lies in its size and its cost of construction. Moreover, its speed being slow (commonly from 3 to 6 feet peripheral velocity), it often requires the installation of somewhat complicated and expensive transmission gearing in order to drive machinery at a suitable speed; it is therefore best adapted to drive slow-moving machinery, usually with heads from 10 to 40 feet (though much larger heads have been used), and with a supply of from 100 to 350 gallons per second. A peripheral speed much greater than that commonly employed would result in a waste of water from the buckets due to centrifugal force.

The number of buckets and their depth are sometimes determined by formulæ, but they are largely matters of experience. If r is the radius of the wheel in feet, the number of buckets is usually $5r$ or $6r$, and their radial depth 10 to 15 inches. The width of the wheel parallel to the shaft is governed by the quantity of water actuating the wheel; it should preferably be so great that the buckets will not be quite full, thus reducing the fall h_1. If the tail-water level is constant, the lowest part of the wheel should be set just clear of that level; if it is variable, just sufficient clearance should be allowed to prevent interference and resistance in times of high water.

These precautions are necessary, for it is clear that the direction of motion of the buckets in the lowest portion of the wheel is opposite to the stream flow in the tail-race; and even slight submergence, therefore, will offer great additional resistance to its motion. This difficulty is sometimes obviated, when for any reason the wheel is to be submerged 4 or 5 inches (as by reason of variable tail-race level), by adopting a reverse-feed arrangement at the end of the supply channel, by which means the water is introduced on the back instead of on the front of the wheel, causing it to revolve in the opposite direction, so that the lower buckets move in the same direction as the tail-water. Such a wheel is often called a *back-pitch* or *back-shot* wheel.

For shallow streams of water with fairly constant depth, the supply channel is usually open-ended, as in Fig. 27; for deeper streams, or greater falls, the supply channel is provided with a sluice-gate or other regulating device, as in Fig. 28. Such a supply-regulating device is especially necessary in case of variable stream-flow.

Perhaps the largest overshot wheel in existence is that at Laxey, Isle of Man (Fig. 29), off the west coast of England. It is 72 feet 6 inches in diameter, and is said to yield 150 to 200 horse-power useful work, which consists in draining a mine 1,200 to 1,380 feet deep. The water for operating is conveyed to the wheel in an underground conduit, and is carried up the masonry tower by pressure, flowing over the top into the buckets of the wheel. Probably the

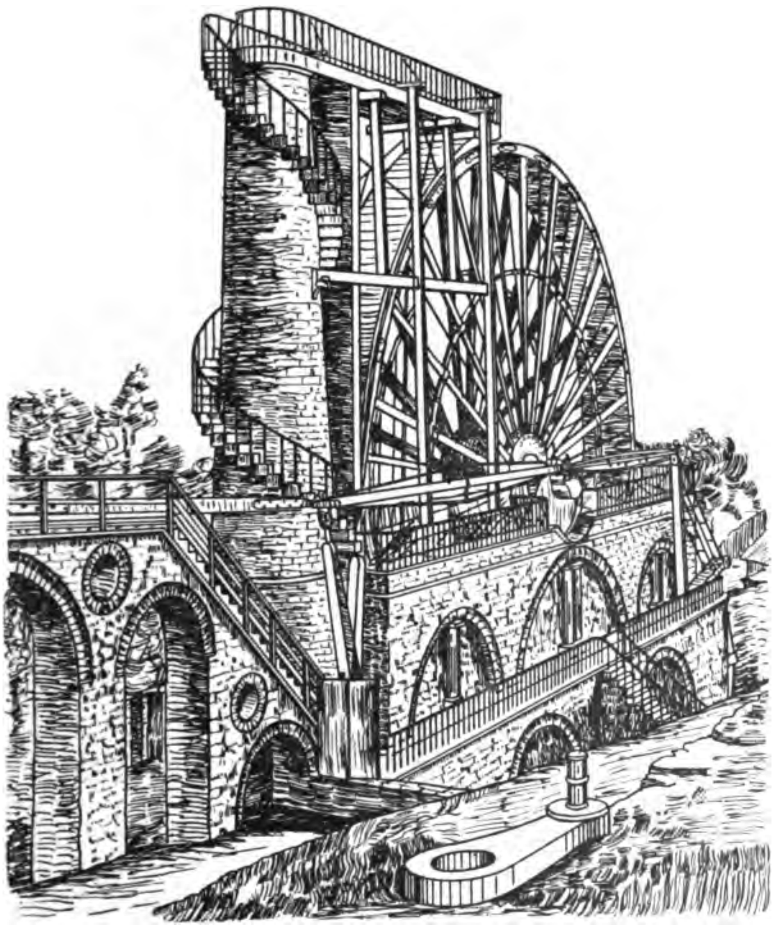


Fig. 29. Overshot Wheel at Laxey, Isle of Man.
Diameter of wheel, 72 ft. 6 in. Water carried up masonry tower by pressure, then flowing into buckets of the wheel.

largest wheel of this type in the United States was erected at Troy, N. Y., with a diameter of 62 feet and a width of 22 feet, developing 550 horse-power.

63. **Breast Wheel.** This type of wheel is designed to receive the water on one side, about or a little above the level of the hori-

zontal diameter; its lower portion, therefore, moves in the direction of the tail-water stream; for this reason the wheel may be *drowned*, or submerged, to a depth of 4 to 6 inches, which makes it suitable for use when both head-race and tail-race levels and supply are subject to variation. It is also evident from the manner of arranging the supply water, that this type is applicable only to small falls, from about 8 to 15 feet; for larger falls, the size of wheel would become impracticable. It is clear that the water acts both by impulse and by weight; therefore, to prevent the escape of the water before the buckets reach their lowest position, the lower quarter of the wheel is encased in a circular *breast* which encloses the buckets, thus practically compelling the water, or most of it, to remain therein until the lowest point is reached. In Fig. 30, water is conducted from the source in a channel or trough to and through an orifice *A*, which controls the supply to the wheel through regulation of the size of the orifice.

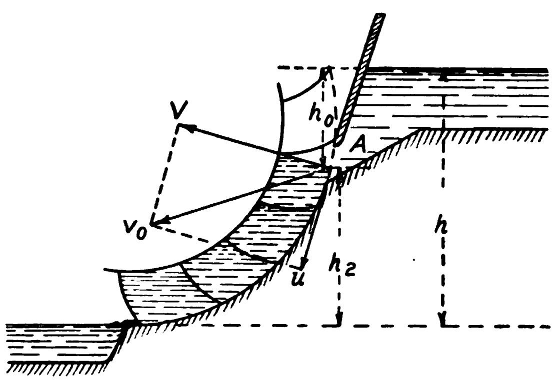


Fig. 30. Breast-Wheel with Supply Controlled through Size of Orifice.

In Fig. 31 the control of the supply is accomplished by means of a shuttle-gate arrangement which consists of a number of openings *J J* in the inclined end of the trough, one or more of which may be closed by shifting the sliding gate *B*. The guide-pieces are for the purpose of causing the water to enter the buckets in a direction most favorable for good efficiency. With the arrangement indicated in Fig. 31, considering the way in which the water enters the buckets, and observing that the mouths of the buckets are practically covered by the extension of the guide-pieces, it is evident that vents or air-holes *F F* in the sole-plate are necessary. Or the sole-plate may be dispensed with entirely, and the buckets formed of polygonal pockets, as *b a c*, in which the vents are naturally formed by the spaces left between the inner sides of consecutive buckets; these being at the top, the buckets may be completely filled with water.

Work and Efficiency. In Fig. 30, the water is admitted through the orifice A, under a head h_0; it therefore strikes the wheel with a velocity v_0, which is approximately equal to $\sqrt{2gh_0}$, and actually equal to $c_1\sqrt{2gh_0}$, where c_1 is the coefficient of velocity for the orifice at A. The water, being then confined between the vanes and the curved breast, acts by its weight alone through the distance h_2, which is approximately equal to $h - h_0$; finally it escapes at the level of the tail-race with the velocity u, or the velocity of the circumference of the wheel. The reasoning in the article on overshot wheels may be applied to this case, by making the fall h_1 equal to zero, and the resulting conclusions may be considered to apply approximately to the case of breast wheels. Accordingly, the following relations are approximately true:

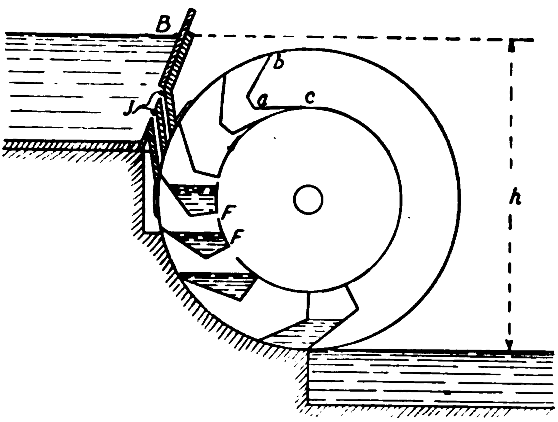


Fig. 31. Breast Wheel with Supply Controlled by Shuttle Gate.

The most advantageous theoretic velocity is

$$u = \tfrac{1}{2}\,v_0 = \tfrac{1}{2}\sqrt{2gh_0} \quad \ldots\ldots\ldots\ldots (64)$$

The maximum efficiency is theoretically:

$$e\,(\text{Max.}) = 1 - \tfrac{1}{2}\,\frac{h_0}{h} \quad \ldots\ldots\ldots\ldots (65)$$

The maximum work is theoretically:

$$\text{Work (Max.)} = W\,(h - \tfrac{1}{2}\,h_0) \quad \ldots\ldots\ldots (66)$$

Practically, the coefficient of velocity of the entrance orifice should be considered, as well as loss due to the clearance between wheel and breast, which will always exist; for any attempt to prevent this entirely by making the clearance less than about $\frac{3}{16}$ inch would result in a considerable increase in circumferential friction, and also, if the wheel is slightly off center, in repeated shocks. For these reasons the efficiency of the breast wheel is materially less than that of the overshot wheel, the usual values ranging from about 50 per cent for small wheels to about 75 per cent for large, well-designed wheels.

When the fall is not great, the wheel is sometimes designed to receive the supply water at a point appreciably below the horizontal diameter; in this case it is frequently termed a *side wheel.* Its efficiency is lower than that of the regular breast wheel. The best wheels of this type have been constructed with diameters ranging between 12 and 24 feet, running with circumferential velocities between 6 and 10 feet per second. They may be regarded as a type intermediate between the regular breast wheel and the undershot wheel. Breast wheels are sometimes provided with some simple automatic governing device controlled by the speed of the wheel, whereby the feed-water orifice is partially throttled when the speed of rotation exceeds a definite predetermined amount.

64. **Undershot Wheel.** The common undershot wheel is provided with plane radial vanes, and the wheel is so set that the water impinges on the lower vanes only, in an almost horizontal direction. In one sense, then, the undershot wheel may be regarded as a special kind of breast wheel, which is operated entirely by the impulse of the moving water. The formulæ developed for the case of breast wheels may therefore be applied approximately to the case of undershot wheels by changing h_0 to h, and v_0 to v; thus, for the most advantageous velocity of the wheel:

$$u = \tfrac{1}{2}v = \tfrac{1}{2}\sqrt{2gh}\,; \quad \ldots\ldots\ldots\ldots \quad (67)$$

the maximum efficiency is:

$$e = (\text{Max.}) = \tfrac{1}{2}, \text{ or } 50 \text{ per cent}; \quad \ldots\ldots \quad (68)$$

and the maximum work of the wheel is:

$$\text{Work (Max.)} = \tfrac{1}{2}Wh \quad \ldots\ldots\ldots\ldots \quad (69)$$

Here, also, the coefficient of velocity of the water in passing through the orifice should properly be considered. In this type, as well as in the last, for reasons set forth in a preceding article, the maximum efficiency and maximum work are practically less than indicated in the foregoing formulæ; also, the most advantageous speed of the wheel is more nearly $u = .40\sqrt{2gh}$ than $.50\sqrt{2gh}$. In practice the efficiencies of such wheels are found to lie between 20 and 40 per cent. The lowest efficiencies are obtained from wheels placed in an unconfined current of water, such as a wheel attached to a barge anchored in a stream; and the higher efficiencies may be expected from well-constructed wheels, in which the actuating stream of water is properly confined, so that it cannot spread laterally.

Fig. 32 shows a simple type of radial-vane undershot wheel operating under a head of water. Here it is seen that the wheel is set in a circular channel constructed with a radius a trifle larger than that of the periphery of the wheel. The sliding gate for regulating the supply from the penstock is arranged at an angle of about 45°, which enables its lower edge to be set close to the wheel rim. By this means the vanes are kept from contact with the moving water until they are almost vertical. The slight drop in the channel below the wheel compensates to some extent for the friction loss in passing the orifice of entry. The circular channel is succeeded by a gently inclined bed, so that the water maintains its uniform velocity after leaving the wheel, until, at a point well away from the wheel, the channel bed is given a sudden, steep inclination.

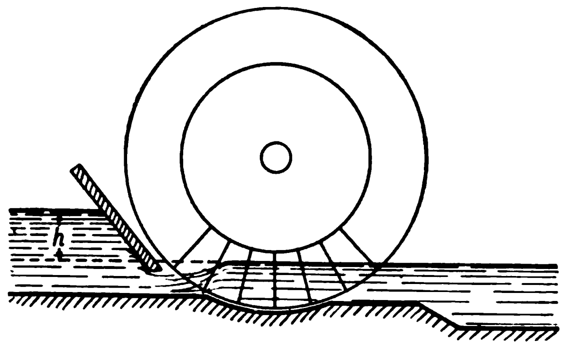


Fig. 32. Simple Type of Radial-Vane Undershot Wheel.

The depth of opening at the orifice usually varies from about 8 inches as a minimum, to about 20 inches in flood. The number of blades, N, is sometimes calculated from the empirical formula:

$$N = 4R,$$

in which R is the wheel radius. Then N and R will determine the spacing between the blades. In practice, this spacing may vary between 18 and 24 inches.

The undershot wheel is a relatively high-speed wheel; hence it may be made more compact than the types described before; its construction and installation are extremely simple, and, from these points of view, it is economical. But its efficiency is lower than that of the other types; it is suitable only for very simple installations, to drive machinery at relatively high speed, where an ample supply of water is available, under a low head.

65. **Poncelet Wheel.** In this wheel (Fig. 33), the vanes are curved in such a way that the water enters through the regulating orifice or opening without shock. Let v be the absolute velocity of the entering stream, and u the peripheral speed of the wheel. The stream, entering with the absolute velocity v, impinges tangentially on the smooth vanes, which are themselves moving in the same general direction with an absolute velocity u. The relative velocity of the water is therefore $v-u$; and it glides smoothly up the curved vane in the general direction of motion of the stream to a height corresponding to this velocity; when at its uppermost point, it is at rest relatively to the vane; it then falls, exerting pressure as it falls, gliding along the vane in the general direction opposite to the motion of the stream, attaining the velocity $v - u$ at the lowest point or extremity of the vane, and passing from the vane tangentially. Its dynamic pressure is therefore due to both impulse and reaction:

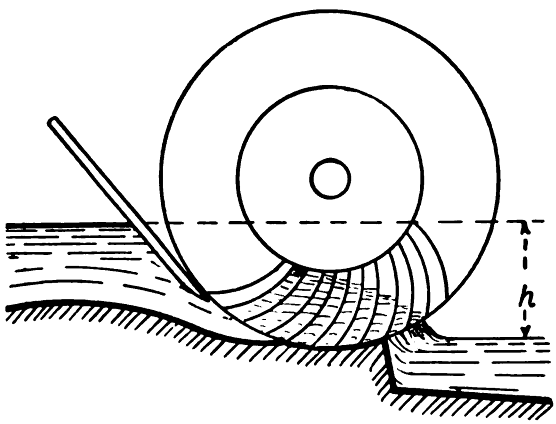


Fig. 33. Poncelet Wheel, Water Entering without Shock.

$$P = F + R = 2F = 2W\frac{v-u}{g};$$

the work of the wheel is $(P \times u)$;

$$\text{Work} = 2W\left(\frac{v-u}{g}\right)u;$$

and this is a maximum, and equal to:

$$\text{Work (Max.)} = W\frac{v^2}{2g} = Wh \cdot \cdot \cdot \cdot \cdot \cdot \cdot \cdot \quad \textbf{(70)}$$

when $u = \frac{1}{2}v$.

Since the theoretic energy of the stream is $W\frac{v^2}{2g} = Wh$,

$$e\ (\text{Max.}) = 1, \text{ or } 100 \text{ per cent} \quad \ldots\ldots\ldots (71)$$

This follows frcm the fact that with the advantageous velocity $u = \frac{1}{2}v$, the absolute velocity of exit is zero; hence the stream "enters without impact, and departs without velocity."

The preceding analysis and the conclusions are theoretic, since they do not consider the various losses of head or energy which must take place. Practically, the efficiency lies between 65 and 75 per cent.

The curved form is given to the bed of the channel of approach, in order to direct the entering stream of water so as to avoid shock. The depth of the vane should be such that the entering water may run up its length (due to its relative velocity) without interference. The spacing of the blades usually ranges between 10 and 18 inches.

The Poncelet wheel, like other undershot wheels, has a relatively high speed; its efficiency is almost independent of the flow, and also of the speed, when a curved channel of approach is used. Moreover, this speed does not vary much, in spite of considerable variations of head. This form of wheel may be used to advantage with a head not exceeding about 6 feet, when the application of power does not call for a high velocity, as for pumping, grinding, etc.

66. In the foregoing cases, the analytical relations have been deduced largely by comparison and analogy, resulting in conclusions more or less approximately true. In each case, however, these relations may be developed quite independently, giving theoretically accurate results. For example, take the case of the breast wheel represented in Fig. 30. In the figure, let Av_0 and Au represent in intensity and direction the velocities, respectively, of the entering water and of the vanes, inclined to each other at an angle α. The dynamic pressure exerted by the water on the vanes, in the direction of motion, is:

$$P = W \cdot \frac{v_0 \cos \alpha - u}{g};$$

and the work per second is:

$$K = W \cdot \frac{(v_0 \cos \alpha - u)}{g} u. \quad \ldots\ldots\ldots (72)$$

The work K, of the dynamic pressure alone, is a maximum, and equal to:

$$\text{Work (Max.)} = W\frac{v_0^2 \cos^2 \alpha}{4g} \quad \ldots\ldots\ldots (73)$$

when $u = \frac{1}{2}v_0 \cos \alpha$.

To this value of K must be added the term Wh_2, representing the work done by the weight of water in the buckets falling the distance h_2; this term is theoretically independent of the speed; accordingly,

$$\text{Total work (Max.)} = W\left(\frac{v_0^2 \cos^2 \alpha}{4g} + h_2\right); \quad \ldots\ldots (74)$$

but $v_0 = c_1\sqrt{2gh_0}$, where c_1 is the coefficient of velocity for the orifice at A. Therefore,

$$\text{Total work (Max.)} = W\left(\frac{1}{2}c_1^2 \cos^2 \alpha \,.\, h_0 + h_2\right); \quad (75)$$

and the maximum hydraulic efficiency is:

$$e\ (\text{Max.}) = \frac{1}{2}c_1^2 \cos^2 \alpha \,\frac{h_0}{h} + \frac{h_2}{h} \quad \ldots\ldots\ldots (76)$$

If, in these equations (73, 75, and 76), h_2 be replaced by its equal $h - h_0$, and if c_1 equals unity, and the angle a equals zero, there will result the approximate equations 64, 65, and 66, deduced in Article 63.

The angle a, however, cannot be zero; in fact it cannot practically be made less than about 10 degrees, for then little or no water would enter the wheel; it should, nevertheless, be as small as practicable, and is usually found between 10 and 25 degrees. The value of the coefficient c_1 is rendered large by well rounding the edges of the orifice; in this way c_1 may be made equal to .95 or even .98. In a manner similar to the above, formulæ for the other cases discussed may also be developed, with a greater degree of accuracy, theoretically considered. It is evident, however, that the approximate formulæ are sufficiently exact for most purposes, since the losses due to improper entry, foam, and leakage, cannot be algebraically expressed.

SPECIAL FORMS OF WHEELS

Water wheels in great variety have been in use from very early times, some of them operating with a fair degree of efficiency. A few of these forms will be very briefly described.

67. **Sagebien Side Wheel.** The buckets of this wheel (Fig. 34) are formed by flat vanes which are tangent to the horizontal cylinder *O*, whose axis is concentric with the shaft of the wheel. The depth of the bucket-ring is relatively large, and there is no sole-plate, each bucket forming a sort of vessel open on top and bottom. The wheel turns in a circular channel, prolonged upstream by a suitable iron casing, sometimes called a *swan's neck*. The side cheeks of the

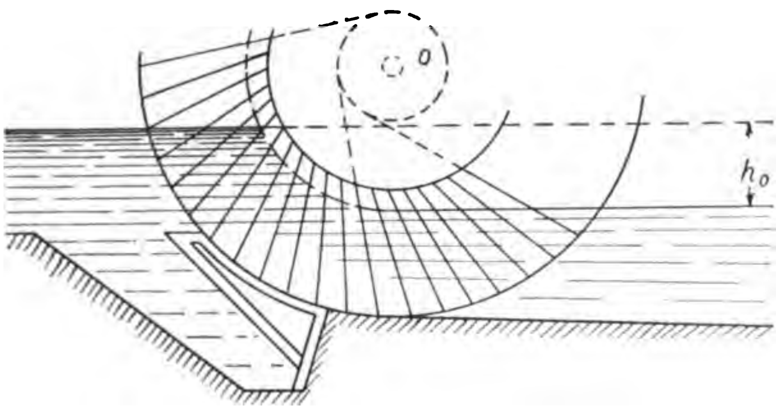


Fig. 34. Sagebien Side Wheel.

channel are continued downstream to the point where the wheel leaves the tail-race. There is very little work done by the water on the vanes beyond a point vertically below the center of the wheel. The inclination of the blades is not favorable for their easy emergence on the downstream side; but, as the speed of the wheel is rarely so great as 3 feet per second, being more usually between 1 and 2 feet, this resistance is small. The efficiency of this wheel, on account of its low speed (since resistances increase more or less rapidly with the speed), is very high, ranging from 80 to 90 per cent according to the height of the fall and the diameter, which varies between 20 and 40 feet, depending upon the fall available, the variability of the supply, and the fluctuation in the tail-race level. The number of revolutions per minute is often less than 1, and rarely exceeds 2½. The penstock speed is usually 1 to 2 feet per second, and this is about the velocity with which the water enters the wheel. The spacing of the blades, measured on the outside of the wheel is about 15 inches. This type of wheel is used for small falls, from 2 to 9 feet, and is suitable for large flows. On account of its slow speed, it is adaptable only for installations where the machinery runs slowly and opposes uniform resistance to driving.

68. **Millot Wheel.** This is a form of breast wheel (Fig. 35) in which the breast is not needed. The supply channel divides into two branches, which pass around to the inner side of the wheel, so that the water enters at the inner circumference. This wheel is difficult to construct, and can be used only for small powers, since, by reason of the feed-water arrangement, the arms must be placed in the middle section of the wheel, instead of being fixed to the flanges; for this reason the breadth is limited to about 5 feet.

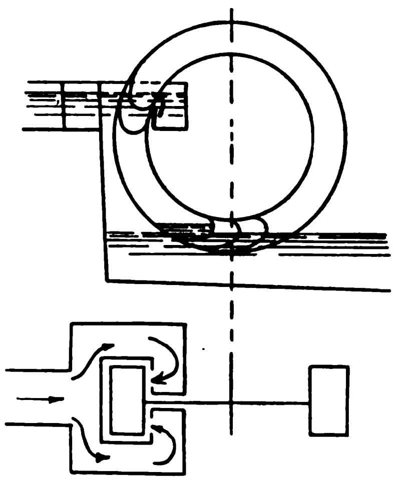
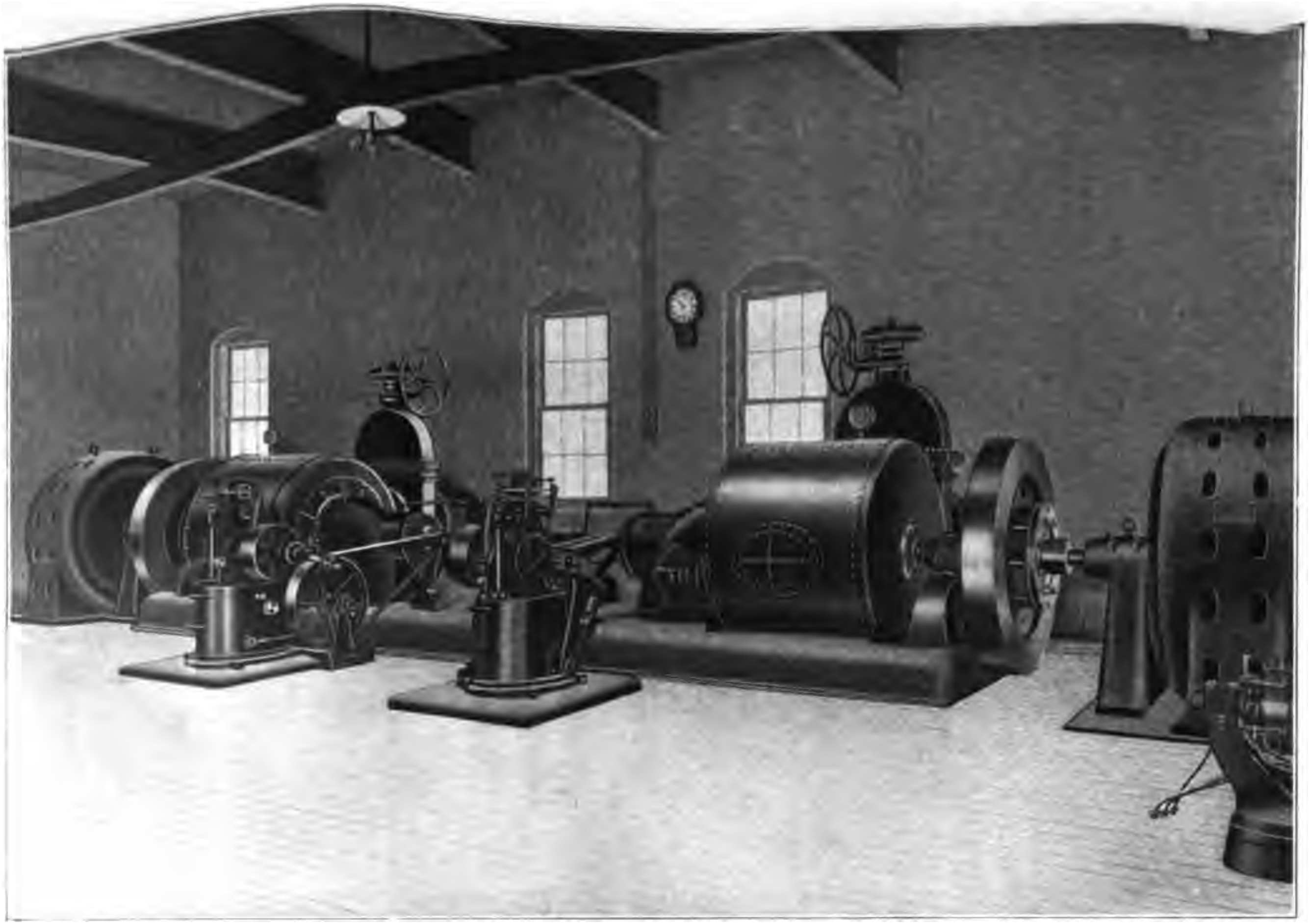


Fig. 35. Millot Wheel.

69. **Floating Wheel, or Current Wheel.** This type is simply an undershot wheel with flat, radial vanes, erected on a scow or barge intended to be anchored in a stream, or mounted on some suitable framework built up from the stream bed. The flat blades are attached to an inner circle, but are not enclosed in shrouds, so that the water has very free entry. As the barge rises and falls with the changes of the stream level, the depth of blade immersion is constant. The efficiency is theoretically a maximum, and equal to 50 per cent, when the peripheral speed of the wheel is one-half the velocity of the current; actually, it rarely reaches 40 per cent. When such a wheel is required to drive stationary machinery—that is, machinery so mounted that it does not follow the fluctuations in the surface level—some special device must be employed to insure the required condition of constant depth of paddle immersion. These wheels are extremely simple, but require to be of large size in order to develop even a moderate amount of power.

Wheels of this type have been used for operating dredges on the river Rhine, Germany; they have also been used to a limited extent, principally for irrigation purposes, in the western part of the



INTERIOR VIEW OF POWER PLANT OF THE DALTON POWER COMPANY, DALTON, MASSACHUSETTS
Two units of 21-inch "New American" turbines, operating under 152 feet head.

United States. One at Fayette Valley, Idaho, was said to be 28 feet in diameter, with 28 paddles, each 16 feet long and 2½ feet wide.

70. **Tympanium.** This is an ancient form of circular open-frame wheel (Fig. 36), fitted with radial partitions so directed as to point upward on the rising side of the wheel, and downward on the descending side. The wheel is mounted in such a way that its lowest parts are submerged to a convenient depth, and it may be turned by the impulse of the current impinging on radial vanes arranged around its circumference. The partitions scoop up a quantity of water, which, as the wheel revolves, runs back toward the axis, where it is discharged into a trough that conveys it away. A very evident disadvantage of this form of wheel is the fact that the water has to be

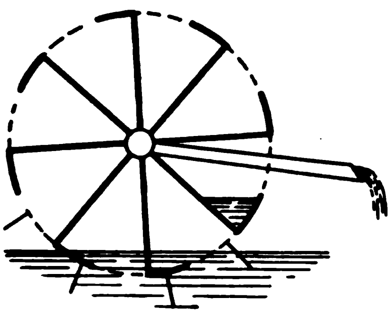


Fig. 36. Tympanium.

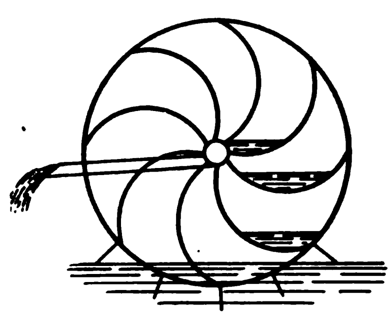


Fig. 37. Scoop Wheel.

raised at the extremity of each radius, so that its lever arm, and therefore its resistance, increases as it is raised to a horizontal plane. This defect does not exist in the next type.

71. **Scoop Wheel.** As this wheel (Fig. 37) revolves, the partitions dip into, and scoop up the water; and as they ascend, the water is discharged into a trough placed under one end of the shaft, which is arranged in as many compartments as there are partitions or scoops.

An improved form of scoop wheel is shown in Fig. 38, which consists of four curved scrolls or channels suitably mounted on the wheel body. The water is conveyed to the central chambers by the scrolls, and it then flows away in a channel or trough.

Many other forms of water motor might be shown, most of them ancient and obsolete, which were mainly used for the purpose of raising water; but the above examples serve to indicate some of the principal devices employed for the purpose.

72. **Ocean Waves.** Many attempts have been made to develop useful power from the almost ceaseless motion of the ocean waves. The essential mechanism usually consists of some form of float which is constrained by a fixed shaft, or a series of such shafts, fastened to a suitable foundation, to move in a vertical direction under the influence of the motion of the waves. The float, by its motion, operates a system of levers and wheels, or ropes and pulleys, which may be made by suitable connections to compress air, or to raise water from a lower to a higher level. In some such way, the irregular or intermittent character of the wave motion may be made to store up power, which, in turn may be released uniformly. Fig. 39 is a diagrammatic representation of such a device.

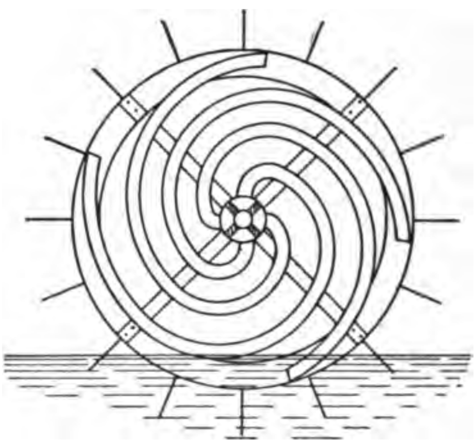


Fig. 38. Scoop Wheel, Improved Type.

73. **Tides.** The ocean tide furnishes a more reliable means of developing power under suitable conditions. Particularly in the vicinity of tidal rivers, and more rarely along shore, the physical configuration of the land may afford opportunity for impounding large volumes of water during the rising of the tide, which may be made to develop power at ebb by flowing out through a suitable channel and operating one or more wheels. Since the wheels must necessarily remain idle during the rising of the tide, some suitable means must be provided for storing power, so that the machinery dependent upon this power may be in continuous operation, or may operate at any time, irrespective of the tidal conditions. Where power is used intermittently—as in some pumping plants which operate only a certain number of hours each day of 24 hours— a system of power storage, while convenient and advisable to provide against the contingency of a breakdown or other mishap, is not so necessary.

74. **Water-Pressure Engine.** This is a hydraulic motor which

performs work by reason of the static pressure of water acting upon a piston or a revolving disc. The cylinder and piston type of motor has a reciprocating motion identical with that of the steam engine; and the operation is very similar, the water entering and leaving through ports which are opened and closed by valves properly connected with the piston-rod. The useful work is due to the difference in the pressure of admission and discharge. As in the case of the steam engine, the reciprocating motion is generally changed by suitable mechanism into rotary motion before being applied to drive machinery. In the other type, the rotary motion is obtained directly from the shaft of the rotating discs or vanes. This latter type has not been widely used, as in practice there are many inherent difficulties in this mode of transmitting high power.

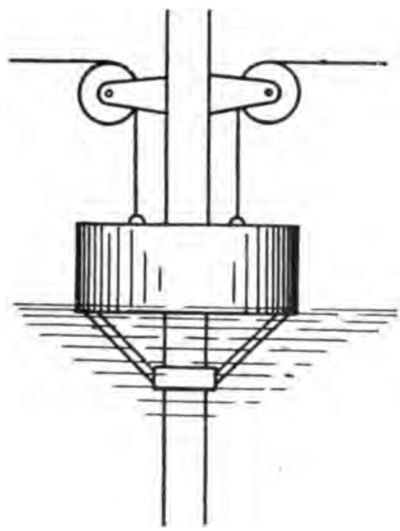


Fig. 39. Device for Utilizing Power of Wave Motion.

IMPULSE WHEELS

75. The term *impulse wheel* is sometimes used to include only those special forms of hydraulic motor which are driven by a jet of water issuing from a nozzle and impinging upon vanes or buckets of special shape attached to the circumference of the wheel. This definition would improperly exclude such motors as the undershot wheel, which is nevertheless a true impulse wheel actuated by a broad stream of water; and also several other types of true impulse wheels.

76. **Horizontal Impulse Wheels.** When a wheel operated by a stream of water issuing from a nozzle and impinging on its vanes is so placed that its plane of rotation is horizontal (the axis being vertical), it is called a *horizontal impulse wheel.*

There are two general classes of such wheels, the *outward-flow*, and the *inward-flow*, as described in Article 52 and illustrated diagrammatically in Figs. 25 and 26. In order to deduce the conditions or relations for maximum efficiency, consider Fig. 26, in which both types are represented, so that the following analysis and the resulting conclusions will be generally applicable to such wheels. The construction of the parallelograms, and the notation, being the same as heretofore, further explanation will be unnecessary.

In order that the water may enter the wheel without shock or foam, the relative velocity V should be tangent to the vane at A as explained before. This condition of tangency will obtain when u and v are proportional to the sines of their opposite angles, in the triangle Auv (as in Article 51, Equations 48 and 48a); that is:

$$\frac{u}{v} = \frac{\sin(\phi - \alpha)}{\sin \phi}; \text{ or, } \cot \phi = \cot \alpha - \frac{u}{v \sin \alpha}$$

The absolute velocity of exit v_1 should be very small (Equations 58 and 59), for the energy represented by this velocity is not given to the wheel, but wasted. Theoretically it should be zero for maximum efficiency, as has already been shown; but practically, if this were the case, the vanes would be unable to clear themselves of the contained water. This absolute velocity v_1 will be small when

$$u_1 = V_1 \qquad (77)$$

These two equations are usually given as representing the conditions of maximum hydraulic efficiency. Equation 77, however, is only approximately true, the real minimum value of v_1 is found when $V_1 = u_1 \cos \beta$, in which case $v_1 = u_1 \sin \beta$; but this equation leads to very complex formulæ. Hence the simpler relation of Equation 77, which is sufficiently accurate, will be used.

Referring to Equation 51, it is clear that if u_1 equals V_1, u must equal V. Then, from the parallelogram at A, Fig. 26, it is seen that when $u = V$, the diagonal bisects the angle ϕ; or,

$$\phi = 2a \qquad (78)$$

Using this value of ϕ in Equation 48, there results:

$$u = \frac{v}{2 \cos \alpha} \qquad (79)$$

Equations 78 and 79 state the conditions involved in Equations 48 and 77, for maximum hydraulic efficiency, in terms sometimes more convenient for use. When a wheel constructed according to this condition (Equation 78) is running with the advantageous velocity u of Equation 79, the absolute velocity of exit is:

$$v_1 = v\frac{r_1}{r}\frac{\sin\frac{1}{2}\beta}{\cos\alpha}\ ; \quad \dots\dots\dots\dots (80)$$

and the corresponding hydraulic efficiency (Equation 59) is:

$$e = 1 - \left(\frac{r_1}{r}\frac{\sin\frac{1}{2}\beta}{\cos\alpha}\right)^2 \quad \dots\dots\dots\dots (81)$$

77. An analysis of this formula teaches that, for high efficiency, both the approach angle α and the exit angle β should be small; but they cannot be zero, otherwise water would not pass into and out of the wheel. Values of 15 to 30 degrees are common. Since, for small angles, the sine varies much more rapidly than the cosine, the equation of efficiency also shows that β is more important than α; so that if β be very small, α may be as large as 40 or 45 degrees, with high efficiency. The equation further shows that for given values of α and β, the inward-flow wheel, in which r_1 is less than r, has a higher efficiency than the outward-flow wheel.

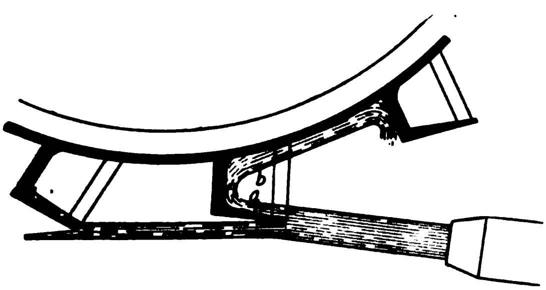


Fig. 40. Faulty Design of Vane.

The actual curve between the entrance and exit points of a vane is not of importance, provided it be smooth and gradual, as abrupt changes of direction lead to shock and to consequent loss of energy

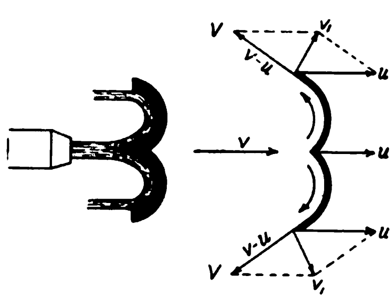


Fig. 41. Good Type of Vane, with Double Cups and Dividing Central Rib.

78. **Vertical Impulse Wheels.** Of this type of wheel (frequently called a ***hurdy-gurdy*** when the vanes are flat planes, and sometimes a ***tangent*** or ***tangential*** impulse wheel), there are several forms in the

market, differing merely in details, and known by various trade names, such as *Pelton, Doble, Cascade,* etc. Essentially this type consists of a wheel mounted on a horizontal shaft, which transmits the power received from a jet or several jets of water acting upon a series of cup-shaped vanes attached to its periphery. The simplest type would be a wheel with flat radial vanes, as in Fig. 21; but, as has already been shown, the efficiency in such a case would be low, so that in practice curved vanes are invariably used.

Fig. 42. A 5,000-H.P. Pelton Water-Wheel Runner.
This wheel, 9 ft. 10 in. in diameter, is capable of developing 5,000 h.p. at 325 r.p.m. under 865 feet of effective head.

In Fig. 40 is shown a faulty design of vane, for the water, after striking the outer lip, is abruptly changed in direction at the corners *a* and *b*, with consequent shock and loss of energy; also, after leaving a vane, the stream strikes the back of the one adjoining, thus producing back-pressure, with further loss of energy. For these reasons the cups or vanes must be very carefully designed.

In the best forms, the vanes are double cups or buckets with a central rib designed to divide and turn the stream sidewise, while at

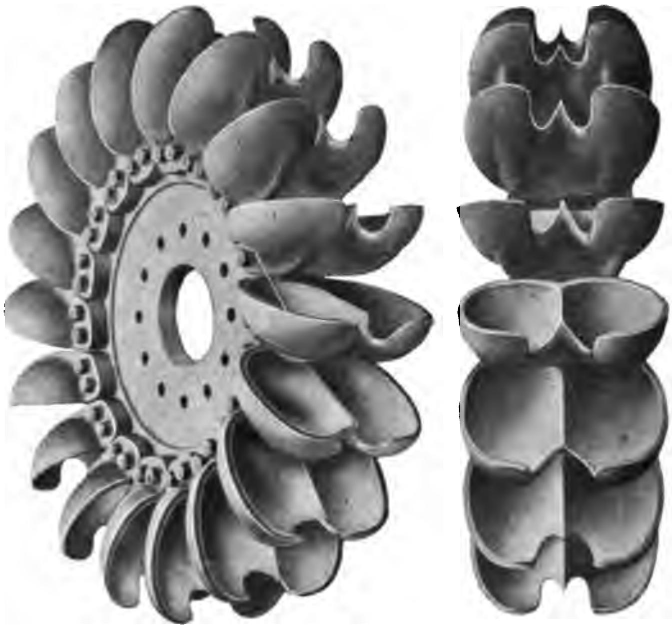


Fig. 43. Runner of 8,000-H.P. Doble Water-Wheel in DeSabla Power Plant. Velocity of jet, 20,000 ft. per minute.

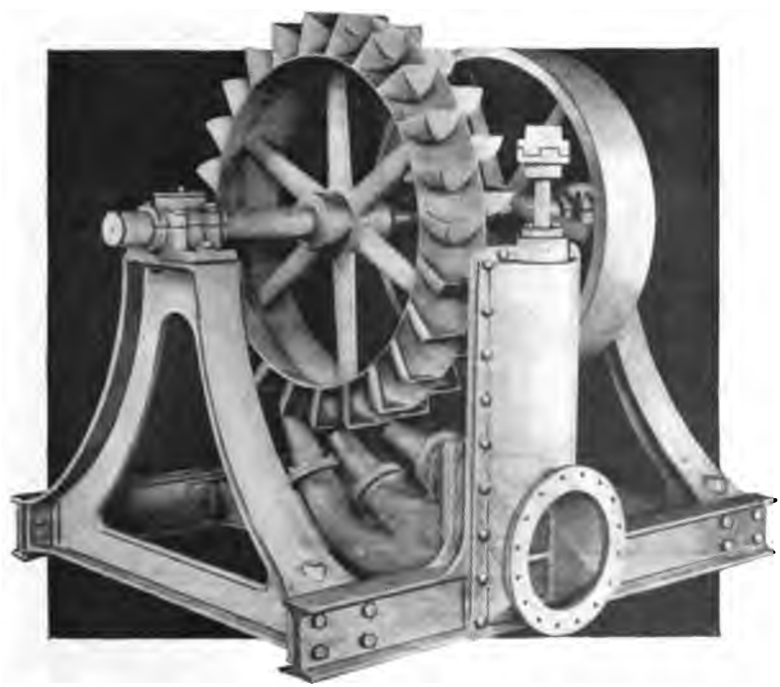


Fig. 44. The "Cascade" (Leffel) Impulse Wheel. Three-nozzle system.

the same time deflecting it backwards, opposite the direction of motion, as in Fig. 41.

Figs. 42 and 43 show the usual method of attaching the buckets to the wheel; it is clear that in these designs one or more buckets may be very easily and quickly removed and replaced when this is

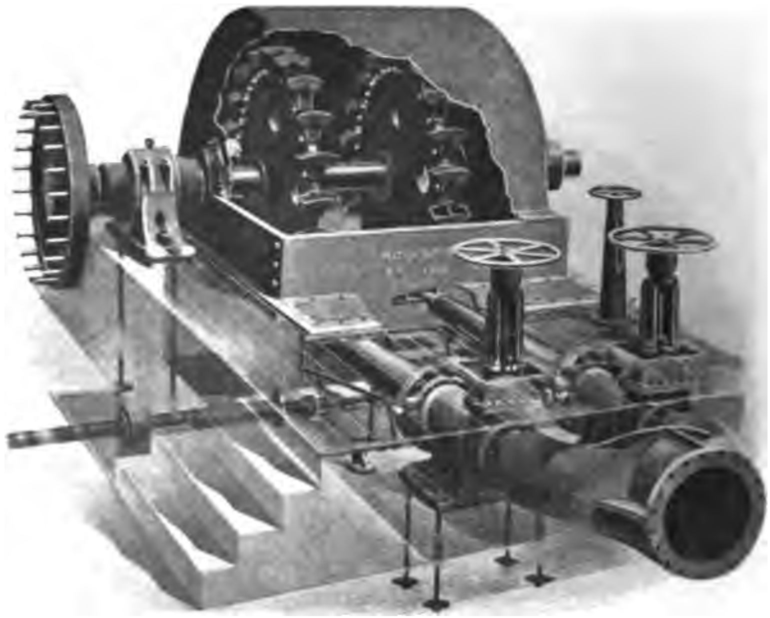


Fig. 45. A 2,000-Horse-Power Double Unit, 500 Feet Head, for Direct Connection to Generator.

rendered necessary by reason of wear or breakage. The buckets in Fig. 42 are of the *Pelton* type.

In the *Doble* vane, Fig. 43, the outer portion of the lip is dispensed with for the purpose of preventing interference between the jet and the approaching vane, though the central rib is retained for parting the stream sidewise.

In the *Cascade* (*Leffel*) wheel (Fig. 44), the *lobes* or half-buckets are set *staggering*, or *breaking joint*, on opposite sides of a thin circular disc, the sharp edge of which serves the same purpose as the central rib of the other forms in dividing the stream.

79. The analysis and conclusions of Articles 43 and 44, Fig. 23, apply in the case of these wheels; namely, the most advantageous velocity, theoretically, is:

$$u = \tfrac{1}{2} v ;$$

and at this velocity, the efficiency is a maximum, and equal to;

$$e \text{ (Max.)} = 1, \text{ or } 100 \text{ per cent,}$$

when $\theta = 180°$—that is, when the stream is completely reversed. However, θ cannot be made equal to 180°, so as to completely reverse the direction of the stream, without interference between the de-

Fig. 46. Interior of Power House of Puget Sound Power Company, Electron, Wash. Four wheel units aggregating 30,000 h.p. in this station; of the "double-overhung" type, coupled to 3,500 k.w. 225 r.p.m. generators. Each unit has an overload capacity of 7,000 h.p.

parting water and the adjoining vane, as shown in Fig. 40, where the water is deflected vertically; and this is equally true when the stream is deflected sideways. The vane is therefore so shaped as to throw the divided stream just clear of the next vane, which condition makes it necessary that θ shall be less than 180 degrees, and consequently the efficiency will be less than 100 per cent, even theoretically. Nevertheless this form of wheel probably comes as near as any to realizing the theoretic condition for maximum efficiency.

As in all the other cases discussed, the theoretic conclusions derived from analyses are not quite true practically. Thus the most advantageous velocity of the wheel is somewhat less than .5 of the jet velocity (though it is probably always considerably greater than .4 that velocity), while the maximum efficiency may be 90 per cent or somewhat higher.

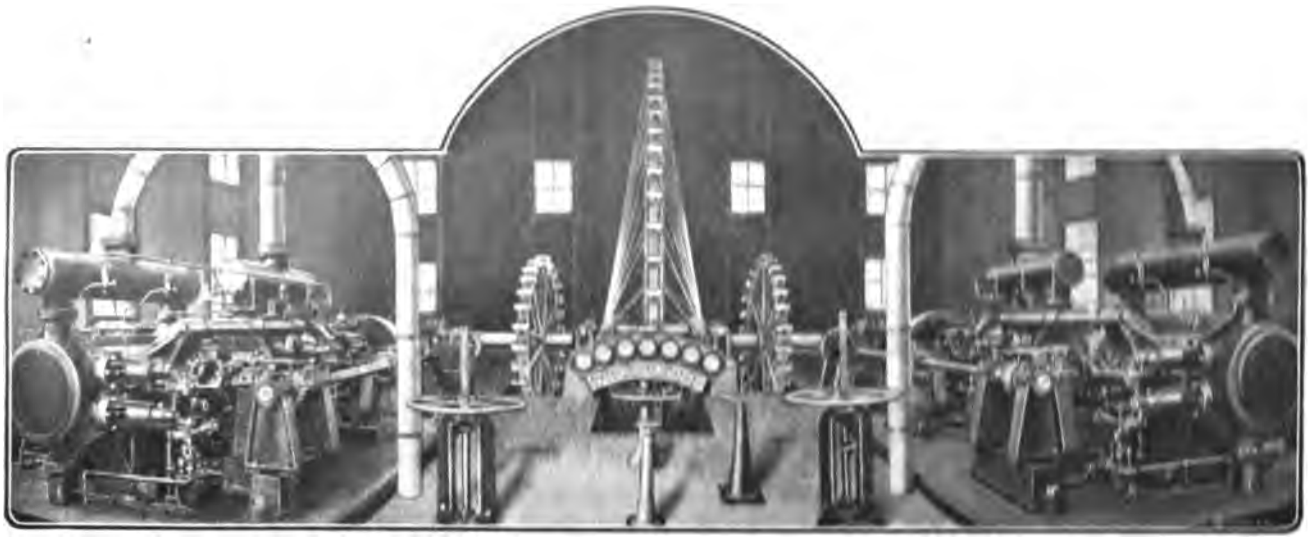


Fig. 47. Hydraulic Air-Compressor Plant. A 1,000-H.P. Duplex Air-Compressor Driven by 2 Pelton Water-Wheels Mounted Direct on Compressor Shaft. Operating under Three Separate Water Heads. Installed at the Morning Mine, Mullan, Idaho.

80. The simplicity, cheapness, and high efficiency of this type of water motor commend it for use when the head of water is not less than about 50 feet—though many are in operation with heads of about 25 feet—especially when the supply of water is not abundant. It has the further advantage, due to its simplicity and cheapness, of allowing of almost indefinite extension of the existing installation, and of division of the power into groups or units, by placing a number of wheels on the same shaft, as in Figs. 45 and 46, or providing a wheel for each machine or group of machines.

Further, several wheels mounted on the same shaft may be operated by jets of water issuing from nozzles under different heads, by properly proportioning the diameters of the wheels and nozzles, as shown in Fig. 47. Here the center wheel is 33 feet in diameter, which is unusually large for this type of motor, and therefore special care was necessary in the design. The two side wheels are each 12 feet in diameter. The variation in heads in this case is about 10 to 1.

For heads much lower than 50 feet, while this type of motor will, with proper regulation, still give a high efficiency, the construction is such that it cannot utilize a large quantity of water, and therefore the power output will not be great. This disadvantage may be obviated to some extent by mounting several wheels on the same shaft; but in the case of low heads, some form of turbine motor is to be preferred.

In setting up, this wheel must of necessity be placed above the tail-race level, and so high above it that there shall be no danger of interference from back-water. This means that a certain proportion of the total available head must be sacrificed to this condition; and unless the total head is sufficiently great to make the loss thus incurred relatively insignificant, this will not be the best type of motor for obtaining the greatest efficiency from the waterfall (see, however, article on "Draft-Tube"). These wheels are well adapted for running high-speed machinery, such as electric generators, air-compressors, etc., by direct connection, thus doing away with much belting or gearing with the attendant loss of power and expense of maintenance. These wheels have been used successfully with heads greater than 2,000 feet. They are manufactured in sizes from 6 inches in diameter to more than 30 feet for special cases, and two or more sizes of nozzle tips are usually provided for adjustment or regulation.

81. *Regulation.* In connection with the practical working of a water-wheel, an important matter is the quick and efficient control of the discharge from the nozzle in order to vary the power output of the wheel as the load varies, or to conform to fluctuations in the supply of water, so as to maintain a constant speed. Interchangeable nozzles of varying sizes have already been referred to; but this method requires hand manipulation, takes time, and requires attention. When the supply of water is adequate, and the power required sufficiently large, or the load variable, from two to five nozzles may be arranged to play simultaneously around the periphery of the wheel, as shown in Fig. 48. By this means, not only may much greater power be derived from one wheel; but, by shutting off one or more jets, the supply and power may be regulated to correspond to the load fluctuations with very little speed variation. Several wheels may be mounted upon the same shaft, each operated by its own jet or jets; and the regulation or control may be effected by shutting off the supply of one or more wheels, which would then run *dead.* In cases where the supply of water is abundant, so that waste is immaterial, good results can be obtained, especially with the smaller wheels, by mounting the two halves of the vanes on separate wheels (practically dividing the ordinary wheel, with its vanes, into two equal portions by a vertical plane at right angles to the axis). When the wheel is working at full power, the two halves are kept together, and thus form an ordinary wheel of this type; when, however, the speed increases, a governor

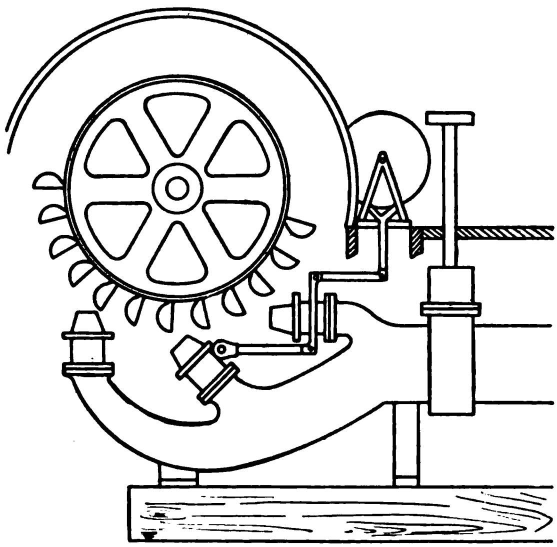


Fig. 48. Wheel Operated by Several Nozzles.

causes the two wheels to separate more or less, and thus some of the water is allowed to escape between. Several other ingenious devices have been developed for the purpose of accomplishing the same end; a description of some of them, taken mainly from manufacturers' catalogues, follows:

82. Under average conditions of operation, a governor is not necessary, as, with a constant load, the speed of the wheel is absolutely uniform. When slight and infrequent changes occur—such as are caused by hanging up stamps of a battery, for example—the wheel can be regulated by hand, by means of the main stop-gate, as shown in Fig. 45; but this would occasion considerable loss of energy, on account of the sudden change of section of the stream. It sometimes happens, however, especially when operating electric plants, that the fluctuations in speed are sudden and severe; and in these cases an automatic regulator is essential. In such cases the speed of the wheel may be controlled by means of various devices, among which may be described the following:

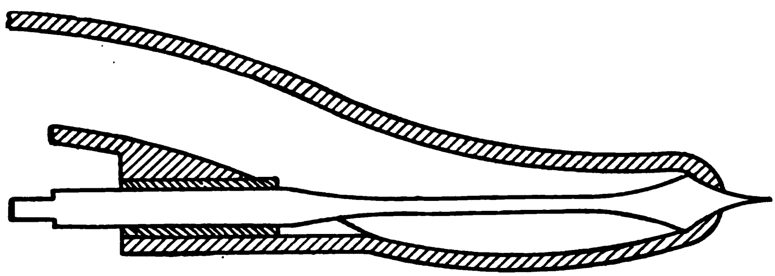


Fig. 49. Section of a Needle Nozzle.

The *deflecting nozzle* is a cast-iron nozzle provided with a ball and socket joint, which permits it to be raised or lowered, thus throwing the stream on or off the buckets; the power of the wheel is consequently increased or diminished to correspond to the change of load, and a constant speed is maintained. A steel deflecting plate, which deflects the stream itself—the nozzle remaining stationary—is sometimes used to accomplish the same results when the design will not admit of a deflecting nozzle. Both these devices are wasteful of water; but they effectually prevent *water-hammer*, which would result from a sudden decrease of velocity in the pipe.

The *stream cut-off* is a spherical plate fitting tightly over the end of the nozzle tip, which, by varying its position, changes the discharge area of the nozzle, and thus influences the power of the wheel.

The *needle nozzle* (Figs. 49 and 50) consists of a nozzle body in which is inserted a concentric tapered needle. A change of position of this needle produces a corresponding change of discharge area of the nozzle; the amount of water used is thus varied, and the power of the wheel influenced proportionally.

Fig. 50. Stream of Water from Pelton Needle Nozzle Operating under 390-Foot Head and Developing 1,500 H.P.

Note the shadow of needle showing through stream, and the perfect form of jet.

The *needle regulating and deflecting nozzle* (Figs. 51 and 52) is a most valuable combination, consisting of a deflecting nozzle swinging on a pair of trunnions, with which is incorporated a needle nozzle, with means for operating either the needle or deflecting nozzle simultaneously or separately. This accomplishes a twofold object—accurate regulation, and water economy without water ram. The deflecting nozzle is a most sensitive means of regulation when actuated by an automatic governor, but does not save water. On the other hand,

Fig. 51. Doble Needle Regulating and Deflecting Nozzle for 8,000-H.P. Wheel.

the needle nozzle, while it is extremely economical in the use of water, is difficult to control quickly by means of the governor. The operation of the combination is as follows:

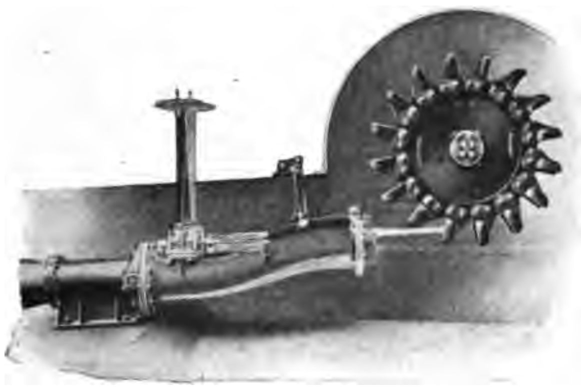


Fig. 52. Pelton Needle Regulating and Deflecting Nozzle in Operation.

Assuming the full load to be on the water-wheel, and the nozzle in position of greatest efficiency, a decrease in load, tending to cause increase in speed, will cause the nozzle to be suddenly deflected by the automatic governor. Simultaneously, the needle portion of the nozzle will be actuated by hand, or by another automatic device, tending to close the needle *gradually*

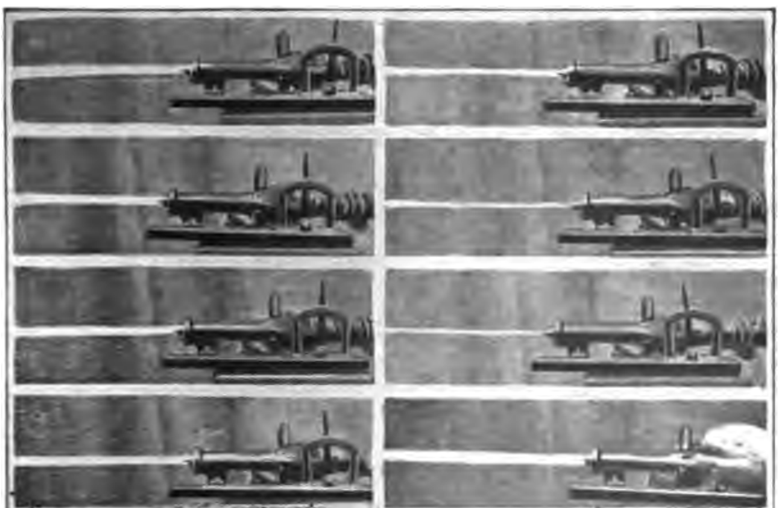


Fig. 53. Various Sized Jets from Small Doble Needle Regulating Nozzle.

and decrease the flow. The governor then raises the nozzle to accommodate the decreased flow of water (and consequent decrease of power), and the nozzle is then brought back to the position of greatest efficiency, having, at the same time, controlled the speed within the required limits.

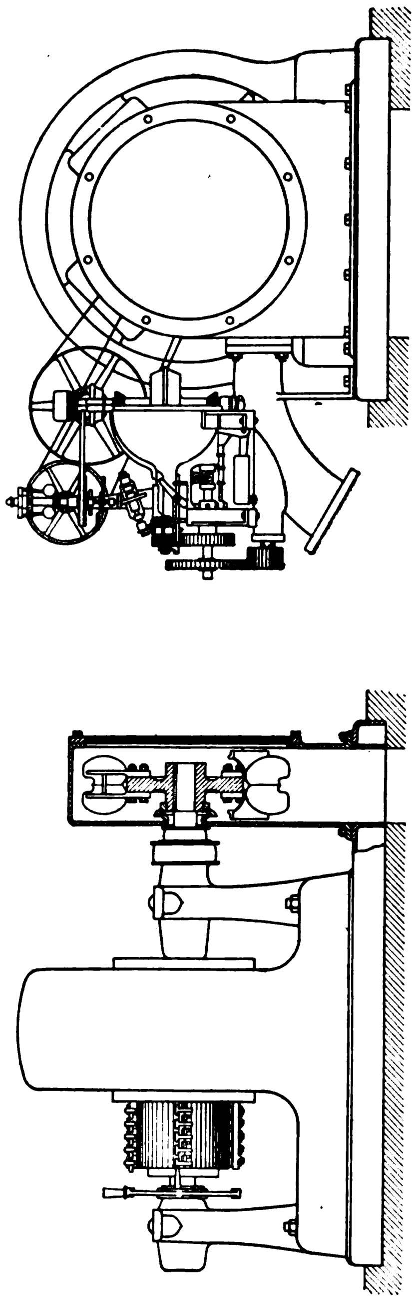
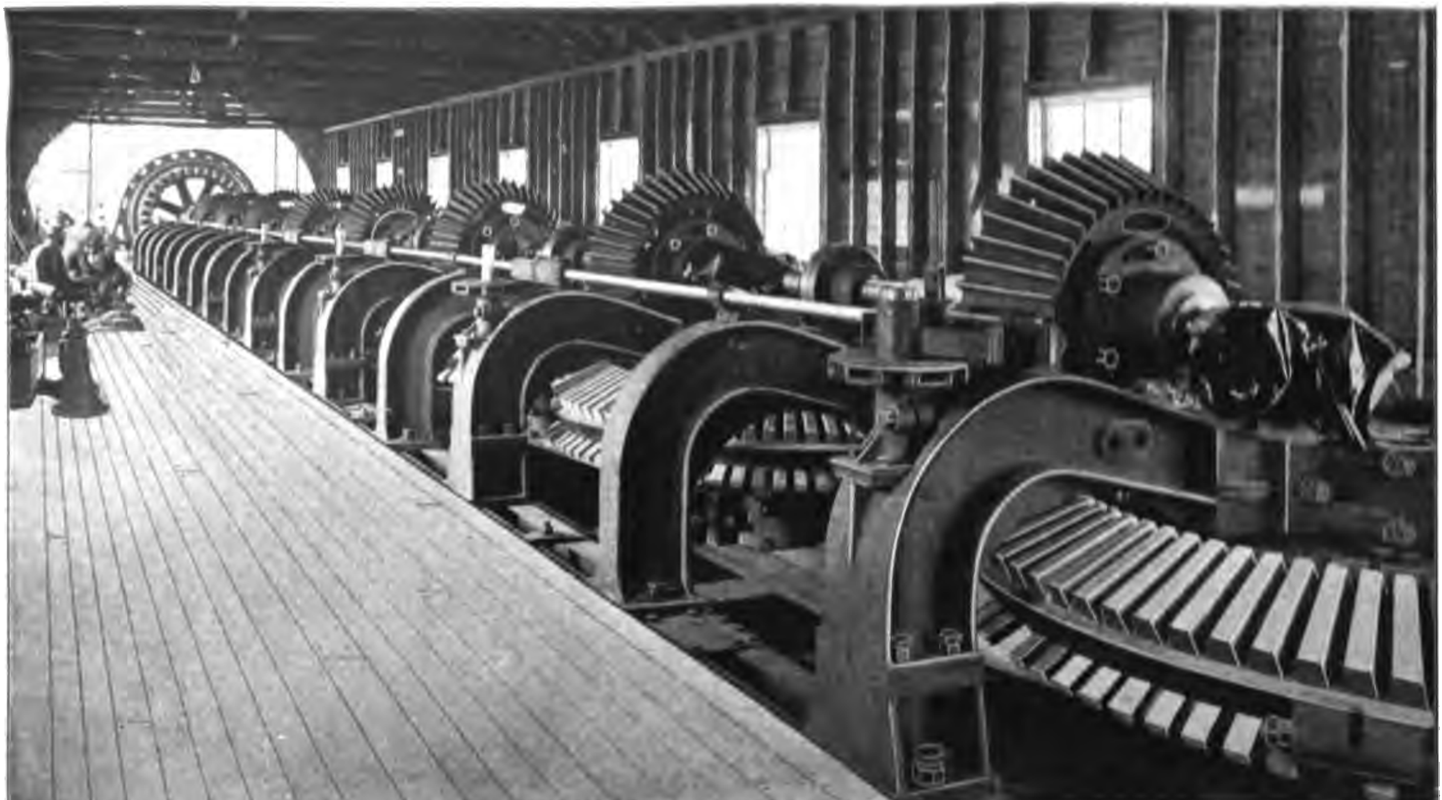


Fig. 54. Hydro-Electric Unit with Needle Nozzle Operated by "Woodward" Governor.

Such a device is essential where water is valuable, and where economy is necessary to carry over the peak load. The needle portion need not necessarily be operated by an automatic device, but may be controlled by hand, and the same results obtained, although necessarily in a longer period of time. In Fig. 52, the upper and lower lines indicate the limits of deflection. Fig. 53 shows how the size of the jet may be varied by means of the needle nozzle.

83. The conditions as to head, power, and character of load determine which device or combination is best suited to any individual case. These various mechanisms are actuated, through a proper system of rock-shafts and levers, by an automatic governor (Figs. 48 and 54), which, for ordinary machinery, may be a *mechanical* governor of the plain, centrifugal-ball type, the power to move the regulating device being furnished directly by the wheel itself; but where close regulation is required, as in driving electrical machinery, a



HYDRO-ELECTRIC POWER PLANT OF THE KALAMAZOO VALLEY ELECTRIC COMPANY AT OTSEGO, MICHIGAN

Consists of eight 56-inch "Samson" turbines, all driving one powerful generator.

Courtesy of James Leffel & Co., Springfield, Ohio.

more sensitive device is necessary. Fig. 55 represents a Lombard automatic governor of the *hydraulic* type, using direct water-pressure to actuate the pistons, which are controlled by balanced valves. Fig. 54 represents a hydro-electric unit in which the needle nozzle, instead of being arranged for hand control, is directly operated by a Woodward compensating governor mounted upon the nozzle body and geared to the needle shaft, which is threaded, and moves in a nut which forms part of the nozzle body, so that the action of the governor regulates directly the position of the needle. It is readily seen that the ball governor is the ultimate device, which actuates or sets in motion the controlling and regulating apparatus. This topic will be further considered under "Turbines" (see Articles 179 *et seq.*).

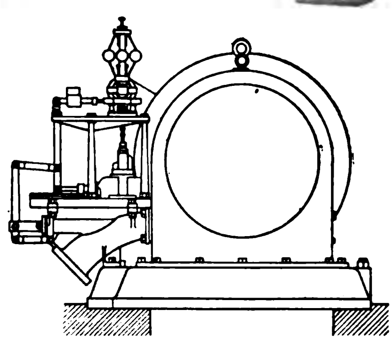


Fig. 55. "Lombard" Automatic Hydraulic Governor.

84. In the case of long pressure-pipes, especially when under high pressure, it is difficult and dangerous suddenly to vary the quantity of water delivered by the nozzle in such a manner as is necessary to regulate the speed of a hydro-electric generating unit subject to

sudden violent variations of load. Consequently it has become customary to regulate the speed of such units by deflecting the jet of water, so that all, or part of it, misses the water-wheel buckets, and is for the moment necessarily wasted. The water which is thus prevented from giving its energy to the water-wheel, is projected through the tail-race at a very high velocity—in some cases exceeding 300 feet per second (18,000 feet per minute)—and becomes destruc-

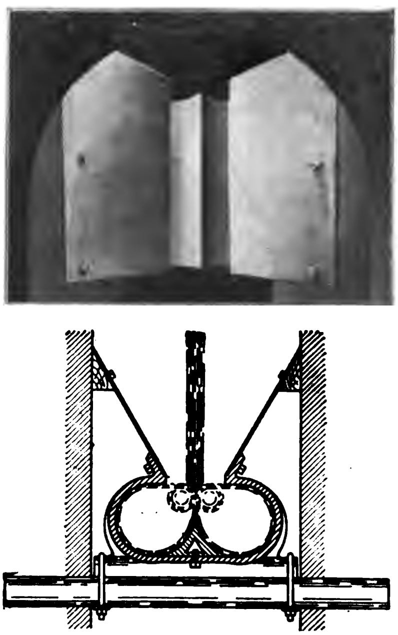


Fig. 56. "Ensign" Vortex Baffle-Plate as Installed in a Tail-Race.

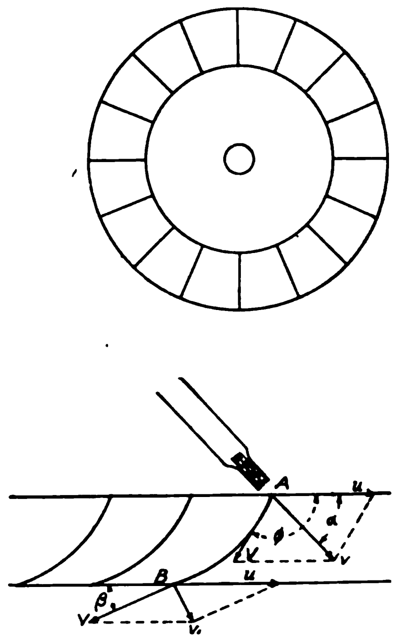


Fig. 57. Downward-Flow Impulse Wheel.

tive, particularly when the water unavoidably carries infinitesimal particles of sand. No masonry can long withstand the action of such a jet, and even iron and steel are rapidly worn away, as if by a terrific sand-blast.

The *Ensign Vortex Baffle-Plate* (patented), illustrated in Fig. 56, is designed to divide such a jet in halves, and deflect the halves until they impinge upon each other, and harmlessly spend their force. The device is a trough-like structure with a sharp central vertical dividing wedge, made to be replaceable in case of wear. The device splits the impinging jet, and guides each half around the curved surfaces, spreading it out into two thin sheets which meet and harmlessly spend their force against each other. The water then falls by gravity into the tail-race with very little disturbance.

85. **Downward-Flow Impulse Wheels.** In this type of motor, the horizontal impulse wheel is driven by the jet from a nozzle inclined downward at a convenient angle, as in Fig. 57, which represents in outline the plan and the development of part of a cylindrical section of such a wheel. The water, in passing through the wheel, neither approaches nor recedes from the axis of rotation; it is therefore sometimes called a *parallel-flow* or *axial-flow* wheel.

The stream enters at A, as shown, with the relative velocity V; passes downward over the vane, always maintaining the same distance from the axis; and, *neglecting the effect of friction and gravity*, issues from the vane at B, with the same relative velocity V.

As before, to prevent impact losses at A, the direction of the relative velocity V must be tangent to the vane at that point; and in order that the efficiency should be high, the absolute velocity of departure v_1 must be small, which later condition will be fulfilled if $u = V$ at B. Therefore, as explained in the preceding analyses, ϕ should be made equal to $2a$, and the best speed of the wheel is $u = \frac{v}{2 \cos a}$. The efficiency under these conditions is:

$$e = 1 - \left(\frac{\sin \frac{1}{2} \beta}{\cos \alpha}\right)^2,$$

which again shows that both a and β, particularly the latter, should be small for high efficiency.

In the above analysis, no account was taken of the force of gravity acting as the water descends through the vertical distance between A and B; this would increase the efficiency and the advantageous velocity above the values as found from the equations above.

It is evident that several nozzles might be employed also with this type of wheel, instead of one, where the supply of water is adequate.

(Articles 11 and 12 develop the hydraulic formulæ to be used in problems of nozzle discharge. Article 16 shows the proper relation between the diameters of nozzle and pipe to furnish maximum power; and Article 19 considers the case of multiple nozzles to fulfil the same condition.)

86. Girard Impulse Wheels. This type of wheel (Fig. 58) consists essentially of two flat, parallel, and concentric rings or *crowns*, between which are inserted the curved vanes or blades, the whole attached rigidly to the axle and forming the wheel proper, or *runner*. The feed or operating water issues from a nozzle placed inside the wheel as shown, in which case it is an *outward-flow* impulse wheel; or the nozzle may be placed outside, making it an *inward-flow* wheel; or several nozzles or groups of nozzles may be employed, located symmetrically around the circumference. The analyses and conclusions contained in the preceding articles apply to these cases.

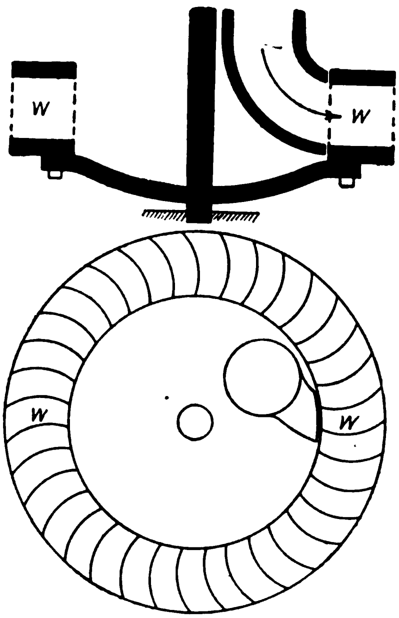


Fig. 58. Girard Impulse Wheel.

Axial or parallel flow may be applied to this type of wheel, as explained in Article 85, under the heading, "Downward-Flow Impulse Wheels."

Occasionally, with the outward-flow type, the two crowns are made to diverge (*i. e.*, their distance apart becomes greater) toward the outer circumference, constituting the so-called *bell-mouthed* profile. In this way, *choking* of the passageways, due to excessive narrowness, is avoided. Openings in the crowns to facilitate the escape of air are frequently made with the same object in view. This type of wheel is widely used in Europe, a number of single motors of this kind developing 1,000 horse-power. Among the several wheels installed at the Terni Steel Works, Italy, ranging from 50 to 1,000 horse-power each, under a head of about 600 feet, is a large 800-horse-power wheel which drives the rolling-mill machinery; its outer diameter is 9 feet 5 inches; its inner diameter, 8 feet 2.4 inches; distance between crowns at entrance, 4.91 inches; at exit, 16.14 inches. The quantity of water used is 16 cubic feet per second; and the normal speed, 200 revolutions per minute.

In the electric power station at Vernayaz, Switzerland, are six 1,000-horse-power Girard wheels, working under a head of 1,640 feet; the outer diameter of each wheel is about 6.5 feet; and the normal speed, about 540 revolutions per minute. These wheels work with but one *guide* (the nozzle tube) each.

A turbine built for the Ouiatchouan Pulp Company (Quebec) has two sets of such guides spaced 180 degrees apart. This wheel develops 1,000 horse-power under a head of 240 feet, running at 225 revolutions per minute; it is enclosed in a cast-iron case and provided with a draft-tube and air-admission valve, both of which contrivances will be described in a later article.

Example 18. Let us assume a Girard outward-flow impulse wheel, with $\alpha = 25$ degrees; $\beta = 20$ degrees; ratio $\frac{r_1}{r} = \frac{4}{3}$; supplied with 2 cubic feet per second through 12-inch pipe 2,000 feet long, with nozzle attached, having a coefficient of velocity of 0.95. Total head over nozzle tip, 152.00 feet, of which 8.3 feet are consumed in pipe friction and entrance losses. Wheel to make 240 r.p.m.

Velocity in the pipe is:

$$v_p = \frac{q}{a} = \frac{2}{\pi \frac{1}{4}} = 2.55 \text{ feet per second.}$$

Velocity of jet is:

$$v = 0.95 \sqrt{2g\,(152.00 - 8.3) + \overline{2.55}^2} = 91.4 \text{ feet per second.}$$

The best speed for the inner rim is

$$u = \frac{v}{2 \cos \alpha} = \frac{91.4}{2 \times 0.906} = 50.5 \text{ feet per second}$$

Since $2\pi rn = 50.5$.

$$r = \frac{50.5}{2\pi \frac{240}{60}} = 2.01 \text{ feet.}$$

$$r_1 = \frac{4}{3} \times 2.01 = 2.68 \text{ feet.}$$

The theoretic efficiency is:

$$e = 1 - \left(\frac{r_1}{r} \frac{\sin \frac{1}{2} \beta}{\cos \alpha}\right)^2 = 1 - \left(\frac{4}{3} \frac{0.174}{0.906}\right)^2 = 0.93.$$

The actual efficiency would probably have a value between 75 and 80 per cent.

The work imparted to the wheel is, theoretically:

Work $= W\frac{v^2}{2g} = 2 \times 62.5 \times \frac{\overline{91.4}^2}{64.4} = 16{,}213$ ft.-lbs. per second, if the nozzle be not considered a part of the motor, and if losses in the wheel itself be disregarded.

If the nozzle be considered part of the motor, the work imparted to it, disregarding wheel losses, is:

$$\text{Work} = wq\left(152.00 - 8.3 + \frac{\overline{2.55}^2}{2g}\right)$$

$$= 62.5 \times 2\left(143.7 + \frac{6.5}{64.4}\right) = 17{,}975 \text{ ft.-lbs. per second.}$$

If the wheel, under the second assumption, have an efficiency of 75 per cent, the useful work of the wheel is:

Useful work $= 17{,}975 \times 0.75 = 13{,}475$ ft.-lbs. per second;

$\frac{13{,}475}{550} = 24.5$ horse-power.

87. **Rotating Vessels.** As a preliminary to the study of the theory and operation of turbines, it will be necessary to consider very briefly some of the essential features of the action of rotating fluids.

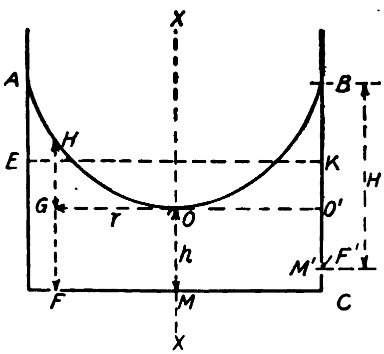


Fig. 59. Revolving Open Vessel.

Flow from a Revolving Vessel with Free Surface. Let AC (Fig. 59) be any open vessel, revolving about a vertical axis XX. It is shown in treatises on Hydraulics, that the water surface EK, which is horizontal before rotation, becomes, under the action of centrifugal force and gravity, a curved surface AOB, due to the rotation; that this surface is a paraboloid of revolution, so that any vertical section through the axis is a parabola with the vertex at O, and axis vertical and coincident with the axis of rotation; that the head of water over an orifice F in the base or side, at any distance r from the axis of rotation, is $h + \frac{u^2}{2g}$, if u be linear velocity of the center of the orifice, and h its distance below the lowest point O; and, therefore, that the relative velocity of efflux from F is:

$$V = \sqrt{2g\left(h + \frac{u^2}{2g}\right)} = \sqrt{2gh + u^2} \quad \ldots\ldots (82)$$

Let n be the number of revolutions per second; then $u = 2\pi rn$; and

$$V = \sqrt{2gh + 4\pi^2 r^2 n^2} \quad \ldots\ldots\ldots\ldots (83)$$

This result is independent of the shape of the containing vessel; and the axis of rotation may lie within or without it, the axis of the paraboloid in any case coinciding with the axis of rotation.

88. **Closed Vessel.** The above formulæ apply equally well to the case of a closed rotating vessel in which the curved surface is wholly or partially prevented from forming, as in Fig. 60. Here also h is the depth MO in the axis of rotation; and the parabola AOB represents the vertical section of the paraboloid of *pressures*. In both cases, then,

$$\frac{u^2}{2g} = \frac{2\pi^2 r^2 n^2}{g}$$

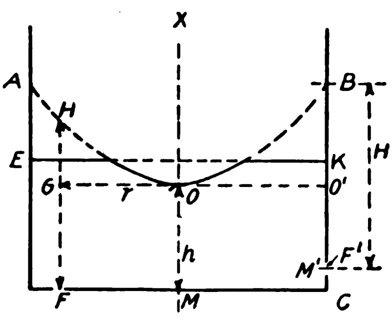


Fig. 60. Revolving Closed Vessel.

is the head GH, to be added to the minimum static head MO at the axis, to obtain the total pressure head over the orifice. If the orifice is in the vertical bounding wall of the vessel, as at F^1, the pressure head is $M^1O^1 + O^1B = H$.

89. **Revolving Tubes.** Fig. 61 represents the simple case of one or more hollow arms attached to a vessel, and rotating with it about a vertical axis. From what has preceded, it is clear that the static pressures at the points A and B in the tube, when rotation has been established, but *when no flow occurs*, are, respectively:

$$OM + GH = h + \frac{u^2}{2g}, \text{ for the point } A\text{; and,}$$

$$OM + G_1H_1 = h + \frac{u_1^2}{2g} \text{ for the point } B,$$

if u and u_1 be the linear velocities of the points A and B respectively.

When the orifices are opened and *flow takes place,* the pressure-head in each case falls by an amount equal to the velocity head *plus* the head lost in frictional resistances, as explained in Articles 6 and 7; and the line of pressure now assumes some other form, such as *LB*. Neglecting for the present the frictional losses, it is evident that the following relations must obtain, by reason of the principle of the conservation of energy:

$$h + \frac{u^2}{2g} = h' + \frac{V^2}{2g},$$

which becomes, for the point *B*, since no pressure head exists at the end of the tube when it discharges freely into the air:

$$h + \frac{u_1^2}{2g} = 0 + \frac{V_1^2}{2g}, \dots\dots\dots\dots \quad \textbf{(83a)}$$

so that

$$h = h' + \frac{V^2}{2g} - \frac{u^2}{2g} = \frac{V_1^2}{2g} - \frac{u_1^2}{2g}. \dots \quad \textbf{(83b)}$$

if V and V_1 represent the relative velocities in the tube at the points *A* and *B* respectively. If the tube is submerged as in Fig. 76, there is static pressure at the end; so that, if h'' is the static pressure on the end (the depth of submergence), then,

$$h + \frac{u_1^2}{2g} = h'' + \frac{V_1^2}{2g}, \dots\dots\dots\dots \quad \textbf{(84)}$$

and therefore,

$$h = h' + \frac{V^2}{2g} - \frac{u^2}{2g} = h'' + \frac{V_1^2}{2g} - \frac{u_1^2}{2g}. \quad \textbf{(84a)}$$

The above equation (84*a*) expresses the relation between the pressure-head, velocity-head, and rotation-head at any point of a revolving tube. In case the tube is only partly full, as when a stream impinges and glides along a vane (or one side of a tube or bucket of a water-motor), there can be no static pressure, and the above becomes:

$$V_1^2 - V^2 = u_1^2 - u^2, \dots\dots \quad \textbf{(84b)}$$

which is Equation 51, for the case of a jet impinging on a vane.

Fig. 61 represents essentially a reaction wheel, since the dynamic pressure causing rotation is caused entirely by the reaction of the issuing jets.

90. In order to discuss the work and energy of such an apparatus, we may use Equation 57, which expresses the work of the impulse of the entering stream and the reaction of the departing stream, by simply omitting the term representing the former Accordingly, for the work of a reaction wheel:

$$\text{Work} = W\frac{-u_1 V_1 \cos\theta}{g} \qquad (85)$$

$$= W\frac{u_1 V_1 \cos\beta - u_1^2}{gh} \text{(from Equation 55)} \qquad (86)$$

$$= W\frac{u_1 \cos\beta \sqrt{2gh + u_1^2} - u_1^2}{g} \text{(from Equation 83a)} \qquad (87)$$

Dividing the expression for Work by the theoretic energy Wh, we have:

$$\text{Efficiency} = \frac{u_1 \cos\beta \sqrt{2gh + u_1^2} - u_1^2}{gh} \qquad (88)$$

The work is zero when $u_1 = 0$—that is, when there is no rotation; also when $u_1^2 = 2gh \cot^2 \beta$; and it is a maximum, and equal to

$$\text{Work (Max.)} = Wh\,(1 - \sin\beta) \qquad (89)$$

when,

$$u_1^2 = \frac{gh}{\sin\beta} - gh, \qquad (90)$$

the efficiency, in this case, being:

$$e\ \text{(Max.)} = 1 - \sin\beta \qquad (91)$$

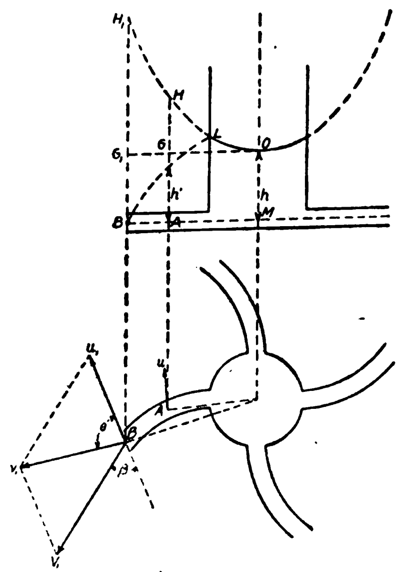


Fig. 61. Revolving Vessel with Hollow Arms Attached.

The work and efficiency, therefore, increase as the angle β decreases. When $\beta = 90$ degrees, the work and the efficiency both become zero, for the jet in such case issues radially; when $\beta = 0$ degrees, the work is Wh, and the efficiency is unity, or 100 per cent; but the velocity u_1 (*and therefore also* V_1) becomes infinitely great. It must be remembered that frictional and air resistances have not been considered in the above analysis; both increase rapidly with increased speed of rotation. In general, however, it may be stated that within certain limits the efficiency of a reaction wheel increases with the speed and with the smallness of the angle β; and it is greatest in any given case, when the angle β is zero—that is, when the water issues in a direction exactly opposite to that of rotation.

91. **Reaction Wheel.** Fig. 62 represents an apparatus com-

monly known as *Barker's Mill.* It is the reaction wheel described in the preceding article, with the direction of the issuing streams of water directly opposite to that of revolution, or $\beta = 0$. Making $\beta = 0$ in the preceding equations, we have:

$$\text{Work} = W \frac{u_1 \sqrt{2gh + u_1^2} - u_1^2}{g}; \quad \ldots\ldots (92)$$

$$\text{Efficiency} = \frac{u_1 \sqrt{2gh + u_1^2} - u_1^2}{gh}; \quad \ldots\ldots (93)$$

$$\text{Work (Max.)} = Wh; \quad \ldots\ldots (94)$$

$$\text{Efficiency (Max.)} = \text{unity, or 100 per cent,} \quad \ldots\ldots (95)$$

when u_1 = infinity; in which case also v_1 = infinity.

If a_1 be the area of the exit orifices, and w the weight of a cubic unit of water, the weight of water discharged in one second is $wa\,v$, which becomes infinite when $u_1 = V_1$ = infinity. As stated before, frictional and air resistances increase rapidly with the speed, so that the above relations, in deriving which these resistances have not been considered, are theoretic. It is evident, however, that the efficiency of a reaction wheel of this type increases with the speed within certain limits; and that the discharge varies with the speed.

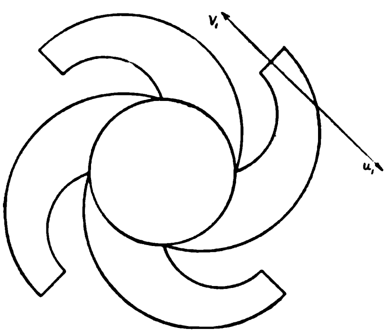


Fig. 63. Barker's Mill.

92. **Effect of Friction.** If c_v be the coefficient of velocity representing the effect of friction in the arms and orifice, then,

$$V_1 = c_v \sqrt{2gh + u_1^2}, \quad \ldots\ldots (96)$$

instead of the theoretical expression,

$$V_1 = \sqrt{2gh + u_1^2}$$

The expressions for the effective work of the wheel and the efficiency then become:

$$\text{Work} = W \frac{c_v u_1 \sqrt{2gh + u_1^2} - u_1^2}{g} \quad \ldots (97)$$

$$\text{Efficiency} = \frac{c_v u_1 \sqrt{2gh + u_1^2} - u_1^2}{gh}, \quad \ldots (98)$$

$$\text{Efficiency (Max.)} = 1 - \sqrt{1 - c_v^2} \quad \ldots\ldots (99)$$

when,

$$u_1^2 = \frac{gh}{\sqrt{1 - c_v^2}} - gh \cdots\cdots\cdots (100)$$

If $c_v = 1$—that is, when frictional loss is not considered—$e = 1$; and $u_1 = V_1$ = infinity, as before. When $c_v = .94$, the advantageous velocity $u_1 = \sqrt{2gh}$, and the efficiency is 65 per cent. Thus the effect of friction is greatly to decrease the theoretic efficiency. To render c_v large, the tubes should be smooth and well rounded by means of easy curves. In addition to the above considerations, the air resistance, which has not been included in the above analysis, increases very rapidly with the speed of rotation, and its effect is to reduce still further the computed efficiency. Because of the low actual efficiency resulting from the above factors, the reaction wheel is not used as a practical hydraulic motor.

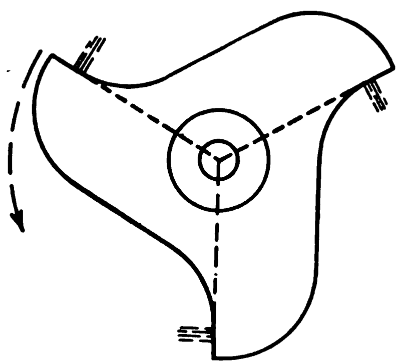


Fig. 63. Scotch Wheel.

93. The *Scotch wheel* (Fig. 63) is an improvement on the Barker's Mill; the three orifices are made adjustable in size by means of movable flaps, for the purpose of regulating the quantity of water and the power.

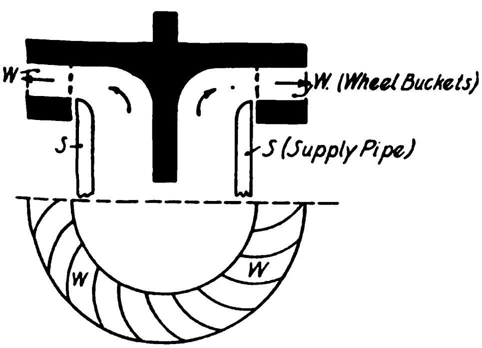
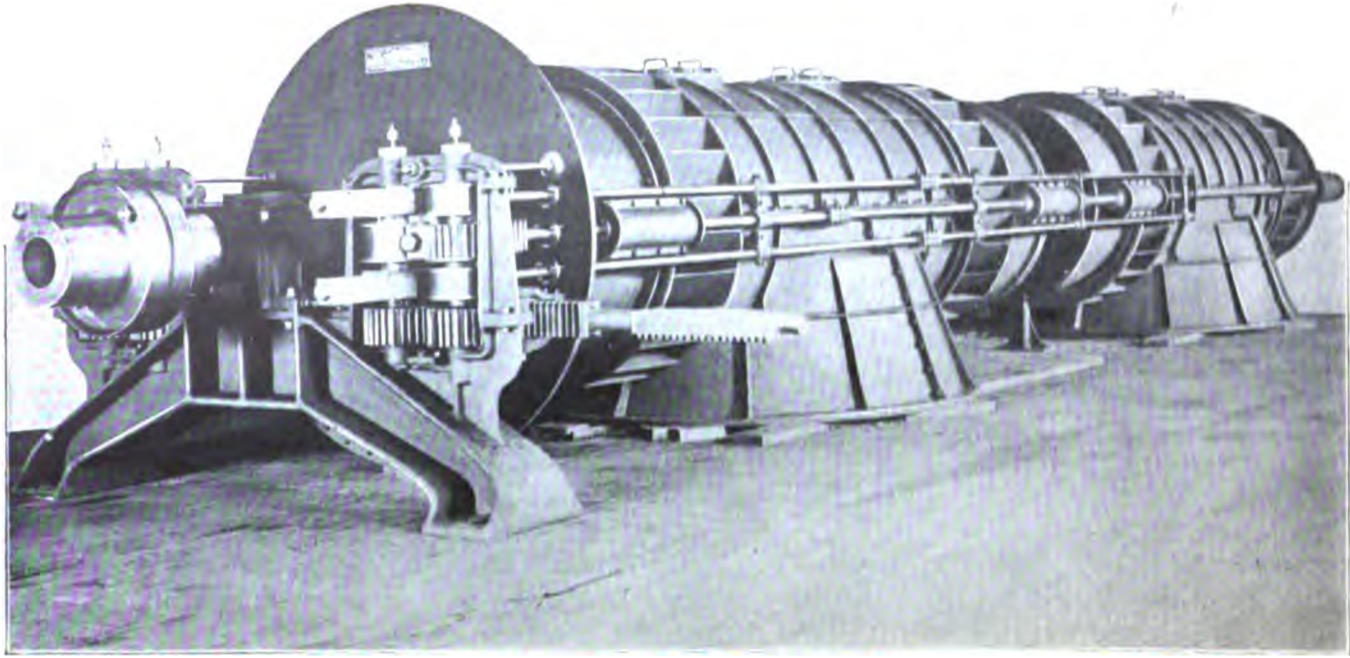


Fig. 64. Combe's Turbine.

94. The next advance in hydraulic motor design consisted in the employment of a large number of issuing streams, as in the *Combe's turbine* (Fig. 64). Here the supply pipe furnishes water directly to the buckets, without the directing intermediary of guides; the water completely fills the passageways, and discharges into the atmosphere with a somewhat low absolute velocity. A modification of this type, with supply pipe above, was called the *Cadiat turbine.*

95. "In 1826 the French engineer Fourneyron improved the Cadiat turbine by placing fixed guide-blades just inside the wheel-ring, around the entire circumference, by means of which the water received a forward direction of motion before entering the channels of the moving turbine. This rendered attainable a very low value of the absolute velocity of the water at exit from the outer rim of the wheel-ring. Also, the wheel being operated under water, the complete filling of the wheel-channels was assured when properly designed. This was the first modern turbine—a motor which, as varied and improved by Fontaine, Henschel, Jonval, and others in Europe, and by Boyden and Francis, and their successors in America, has grown in popular favor, and, together with the impulse wheels already described, has almost entirely supplanted the old forms of vertical water-wheels so long considered as giving the highest efficiency."*

*Church, "Hydraulic Motors."



UNIT CONSISTING OF TWO PAIRS OF 51-INCH "CYLINDER GATE NEW AMERICAN" TURBINES ON HORIZONTAL SHAFT

Designed for direct connection to 1,500-K. W. generator, operating under a variable head of 25 to 35 feet.

Courtesy of Dayton Globe Iron Works Company, Dayton, Ohio.

WATER-POWER DEVELOPMENT

PART II

TURBINES

96. A turbine consists essentially of a series of short, curved passageways or buckets divided from one another by vanes or blades, the whole forming a single rigid body attached to the axle, and called the *runner*. The water for operating the runner passes into the passageways through a set of fixed or stationary channels called *guides*, the feed-water being admitted around the entire circumference. For convenience and efficiency of operation, the turbine is provided with various controlling, regulating, and governing devices (Fig. 65).

97. **Classification.** Turbines are classified in several different ways, depending on the criterion used as the basis. Thus, with respect to the direction of flow of the water through the wheel, there are three classes—*Radial-Flow*, *Axial-Flow*, and *Mixed-Flow* turbines.

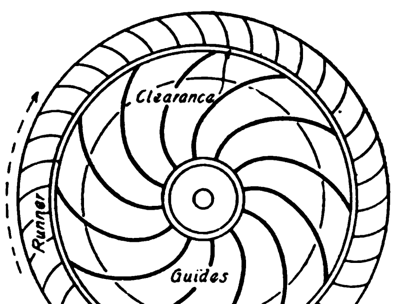


Fig. 65. Diagram of Typical Turbine.

A *radial-flow* turbine is one in which the path of a particle of water within the wheel lies in a plane perpendicular to the axis of rotation. The direction of flow may be either *outward* or *inward*; that is, the turbine may have *internal* or *external feed* (Figs. 66 and 67). An *axial-* or *parallel-flow* turbine is one in which the distance of a particle from the axis of rotation remains constant during its passage through the wheel (Fig. 68). *Mixed flow* is a combination of radial and axial flow; it is usually inward and axial.

98. Another classification divides these motors into *impulse*

and *reaction* turbines. If the wheel-passages are not completely filled with water, and if air enters freely so that the entire stream within each wheel-passage is under atmospheric pressure, the motor is called an *impulse* turbine.

If the wheel-passages are completely filled by the water flowing through them under pressure, the motor is called a *reaction* turbine.

The *limit* turbine is a type intermediate between the impulse and the reaction turbine; but, though it combines many of the advantages of both types, it has not as yet received much attention.

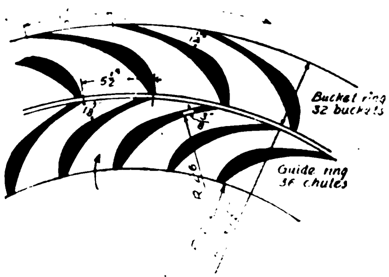


Fig. 66. Section of Guides and Buckets, Fourneyron Turbine, Niagara Falls.

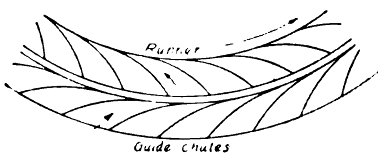


Fig. 67. Section of Runner of Francis Center-Vent Turbine.

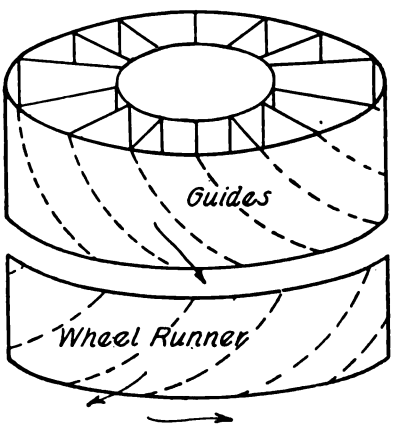


Fig. 68. Axial-Flow or Parallel-Flow Turbine.

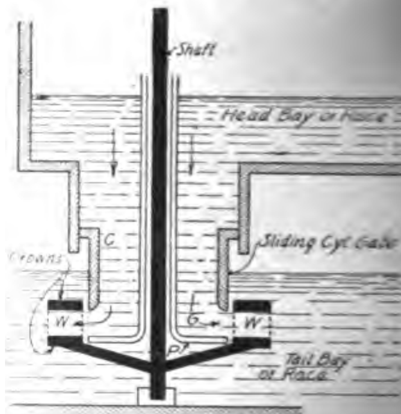
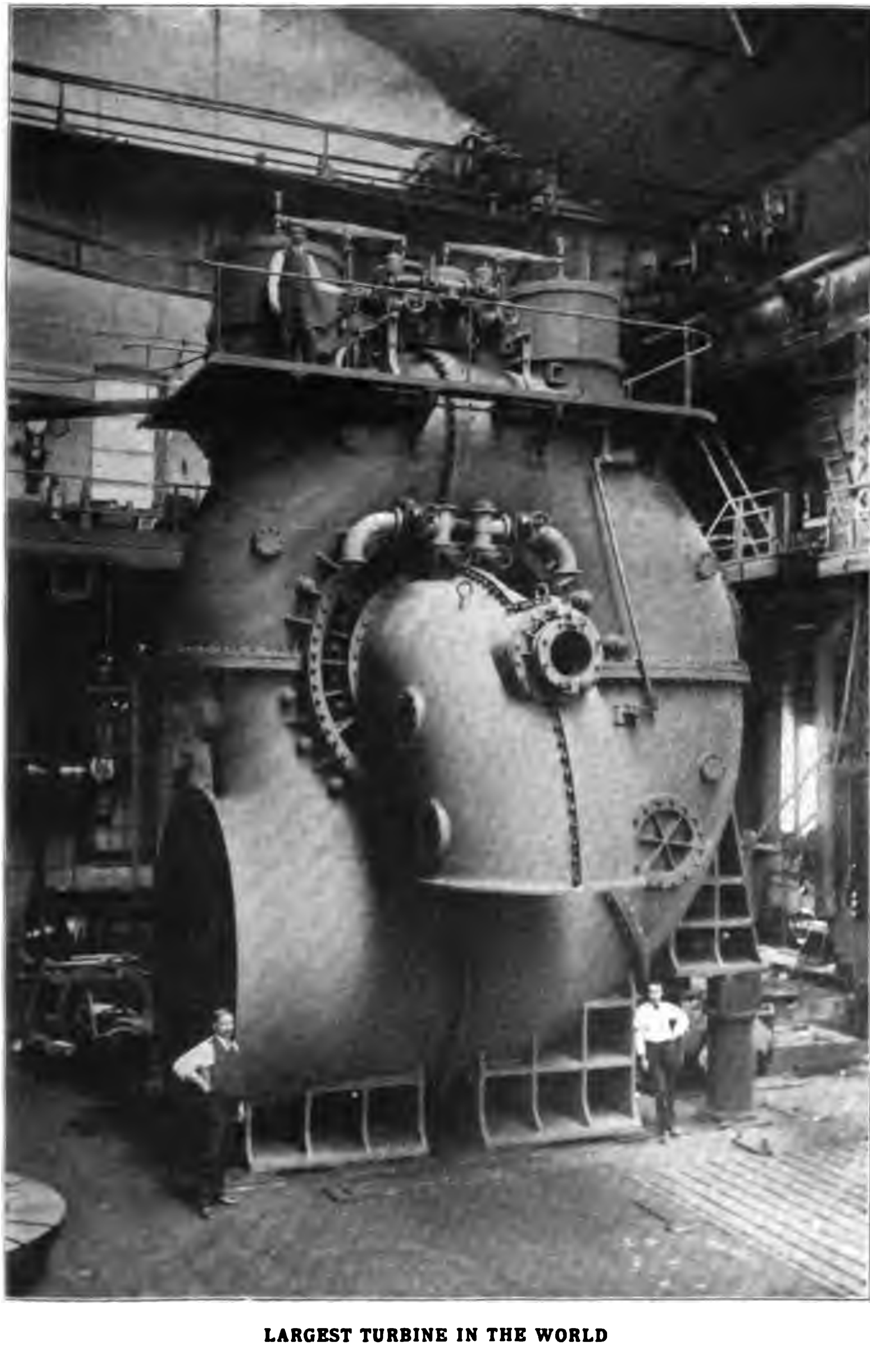


Fig. 69. Fourneyron Turbine. Radial outward flow.

99. The three typical classes of turbines above described are often called by the names of the eminent hydraulicians who invented or perfected them; thus the reaction turbine with radial outward flow



LARGEST TURBINE IN THE WORLD

Installed at the Shawinigan Falls Power Station on the St. Maurice river, about 84 miles northeast of Montreal, Que. Weight, 364,000 pounds; height, 30 feet; weight of wheel, 5 tons; weight of 22-in. shaft, 10 tons; intake, 10 feet in diameter; amount of water passing through per minute, 395,000 gallons.

is frequently called the *Fourneyron* turbine (Fig. 69). The guides *G* are rigidly attached to the fixed plate *P*, which is connected with the hollow pipe enclosing the shaft. In such a wheel the discharge may be either into the air or into a body of water; a suction or draft tube cannot very conveniently be used with this type of motor.

A reaction wheel similar to the above, but with radial inward flow, is often called a *Francis* turbine (Fig. 70).

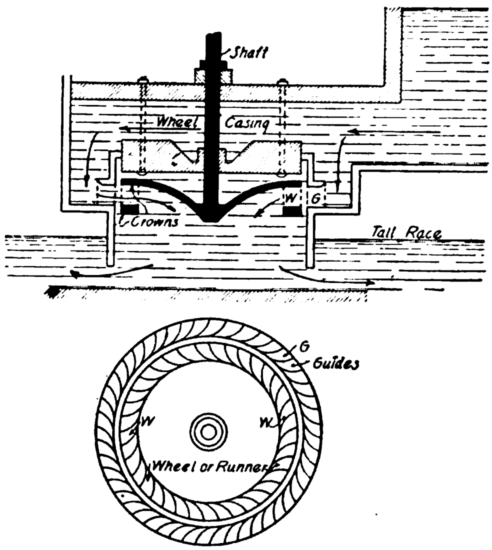


Fig. 70. Francis Turbine. Radial inward flow.

A reaction turbine with axial flow is generally termed a *Jonval* turbine (Fig. 71). The discharge from the two latter types of motor may take place into the air, directly into the tail-water, or into a suction (draft) tube. Limit turbines are sometimes called *Haenel* turbines; they may be considered impulse turbines without free deviation.

100. The most common forms of reaction turbines used in America, particularly for the smaller sizes, are of the mixed-flow type, having radial inward admission and axial downward discharge, as the *Swain* turbine; or of the plain inward-flow type.

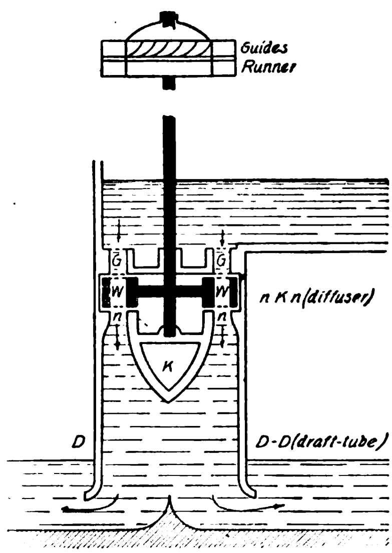


Fig. 71. Jonval Turbine. Axial flow.

Turbines of the *American* pattern (inward, downward, and outward), in which the water passes radially inward, then axially, and finally leaves the buckets at a slant between the axial and the radial outflow direction, are called *American vortex* turbines, types of which will appear in subsequent articles.

101. Any turbine may be made to act as either an impulse or a reaction turbine. If the conditions be such that the water, in passing through the vanes, fails to fill them completely, it is an impulse turbine; if the wheel be placed under water (or *drowned*), or if by any other means the water is compelled to completely fill all the passages under pressure, it acts as a reaction turbine.

102. In an impulse turbine, the energy of the water is wholly converted into kinetic energy at the inlet surface. Thus the water enters the wheel with a velocity due to the available head, and therefore without pressure; is received upon the curved vanes; and imparts to the wheel the whole of its energy through the agency of the dynamic pressure due to its impulse. Since special care must be exercised to insure that the water will be freely deviated on the curved vanes, such motors are sometimes called *turbines with free deviation*. For the above stated reasons, the water passages in such wheels should never be completely filled; and in order to insure an unbroken flow through the wheel-passages, and to prevent the formation of eddies at the backs of the vanes, ventilating holes are arranged in the wheel sides.

In a reaction turbine, only a portion of the available energy is converted into kinetic energy at the inlet surface of the wheel. Thus, such a wheel is driven by the dynamic pressure of flowing water which is at the same time under a certain degree of static pressure.

103. **Turbine Development in America.** "In 1834, M. Fourneyron, a French engineer, had brought out the radial outward-flow turbine known under his name; and in 1840, Mr. U. A. Boyden, of Massachusetts, commenced to study and to improve upon this type. M. Fourneyron's diffuser (described later) was also introduced in America by Mr. Boyden, and is therefore usually known as the *Boyden diffuser*. It should here be stated that the now obsolete diffuser was the forerunner of the conical draft-tube (described later) of the present day; and the same principles underlie the action of both.

"Mr. Boyden was soon followed in this work by Mr. James B. Francis. In 1849, however, Mr. Francis built a radial inward-flow or vortex turbine for the Booth Cotton Mills, Lowell, Mass. This turbine, which worked under a head of 19 feet, and when tested showed an efficiency of 79.7 per cent, or practically 80 per cent, may be regarded as the prototype of all American turbines.

"The number of revolutions varies as the square root of the head employed; and for the same head, the number of revolutions of different turbines is inversely proportional to their diameters. As all the early turbines were used with low heads (about 20 feet or less), and as even then (as now) the tendency was to increase the speed of shafting, machinery builders naturally reduced the turbine diameter.

"Mr. Francis's turbine was of the plain inward-flow type, with sufficient room in its interior for the water to turn and escape axially. With the continued reduction of the turbine diameter, this interior space became more and more reduced, so that it soon became necessary to turn the water in an axial direction while still in the runner-bucket—or, in other words, to curve the bucket from a radial to a more or less axial direction. This has been going on gradually, as can be seen by comparing the early forms of the *Humphrey* and the *Swain* turbines with the present form of the *Hercules*, *New American*, *Leffel*, and other turbines, which have scarcely more interior space than is required to pass the shaft through, and have a much greater part of the runner-buckets in the axial- or parallel-flow direction than in the radial-flow direction; while the runner-buckets have assumed such an intricate shape that it is very difficult to analyze the action of the water while flowing through these buckets, or to predetermine mathematically their shape for given conditions."

Clemens Herschel writes:

> "*American* turbines are mostly of a complex nature, as regards the action of the water on the buckets of the wheels, and have been perfected in efficiency by test, or, as it is irreverently called, by the 'cut and try' method of procedure. A wheel would be built on the inspiration of the inventor, then tested in a testing flume, changed in a certain part, and retested, until no further change in that particular could effect an improvement. Another part would then undergo the same process of reaching perfection; and thus, in course of time, the whole wheel would be brought up to the desired high standard of efficiency."

"Another consequence of the reduction of the diameter is that the inner ends of the buckets, which closely approach the center of the turbine, are located on a very small circle, which limits their number and gives them a very close spacing, while the spacing on the outer circumference becomes so large—6 inches and even 12 inches being not uncommon—that the buckets are unable properly to guide the water as the best efficiency would demand. The area through which the water enters the runner—being the outer circumference of the runner, multiplied by the axial dimension of the bucket entrance—decreases, of course, with the diameter of the runner; and with it, and in the same ratio, decreases the quantity of water passed through, and the power developed by the turbine.

"To prevent this decrease in entrance area, and in power, builders have gradually increased the axial dimension of the bucket entrance. Thus the efforts made towards greater speed and power have transformed the plain inward-flow turbine-runner of fifty years ago into the shape now generally employed."*

104. "The *American* type of turbine is thus distinguished by the great depth of its buckets, its great capacity in proportion to its diameter, and its high speed. It is also distinguished by the form of its buckets, which consist of a ring of curved vanes arranged parallel to the axis and inclosed within the guide-ring; while below the guide-ring, the buckets expand downward and outward, forming large cup-shaped outlets. The shape of the guide-buckets has not changed to the same extent.

105. "The type to be employed in each individual case should be in accordance with the height of the head to be utilized, as follows:

1. Low heads, say up to 40 feet: *American* type of turbine (*i. e.*, of the "inward and downward" variety), with horizontal or vertical shaft in open flume or case, nearly always with draft-tube. For heads up to about this limit, the *American* type of turbine has the great advantage over all other turbine types in common use, that it gives the greatest number of revolutions for a given head and power developed, or the greatest power for a given head and diameter of runner; while the *American* system of manufacturing only one line of turbines from stock patterns has the great advantage of enabling the builders to fill orders cheaply and quickly.

2. Medium heads, say from 40 to 300 or 400 feet: Radial inward-flow reaction or *Francis* turbine, with horizontal shaft and concentric or spiral cast-iron case with draft-tube.

3. High heads, say above 300 or 400 feet: Impulse wheel of the usual type (*Pelton*); or radial outward-flow segmental-feed, free deviation (a *Girard* impulse wheel); or a combination of both, with horizontal shaft and cast- or wrought-iron case, often with draft-tube.

*Thurso, "Modern Turbine Practice."

"Extremes in speed or power or both will, of course, often demand the use of a turbine type for a head outside of the range for which the type is here proposed.

"Turbines with horizontal shafts should be employed in all cases except where the use of turbines with vertical shafts is either imperative or gives a decided advantage over turbines on horizontal shafts. This advantage may often be gained by using dynamos with vertical shafts, direct-connected to the turbines, which gives an excellent, compact, and neat arrangement, as in the plant of the Niagara Falls Power Company and many others. Horizontal turbines are not only more convenient in attendance and easier of access for adjustment or repairs; but most of the transmission of power is done by horizontal shafts, and nearly all standard patterns of direct-driven dynamos or other machinery are arranged to connect to a horizontal driving-shaft. With a very low total head or pressure-head above the turbine, it will often be necessary to use vertical turbines to be able to utilize such a head at all. In many locations, horizontal turbines, on account of the great rise of the tail-water during times of flood, would have to be set at so great a height above low tail-water that the head below the turbine would be utterly beyond the practical working limit of draft-tube during the low-water season, and part of the head would thus have to be sacrificed just at the time of least water. In such a case, vertical turbines are of great advantage, as they may be set at any elevation, because their being submerged during times of flood does not interfere with their operation."*

ESTIMATES FOR WATER POWER†

106. The methods of estimating the water power that can be derived by damming a stream, are similar to those for water supply. In the absence of gaugings, the records of rainfall and evaporation are to be collected and discussed; but a few gaugings will give much more definite information, if records of water stages during several years can be had. Here, also, the minimum flow of the stream must receive careful attention, particularly when the plant is to generate electric power for trolley and light service, for the interruption of such serv-

*Thurso, "Modern Turbine Practice."

†Articles 106 to 121 inclusive have been taken, with slight changes, from Prof. Mansfield Merriman's "Treatise on Hydraulics."

ice is a serious public inconvenience. It has frequently happened, indeed, that a water-power plant built without sufficient investigation has proved unable to furnish sufficient power during dry seasons, and it has been necessary to install an auxiliary steam plant to make good the deficiency.

Let W be the weight of water delivered per second to a hydraulic motor, and h be its effective head as it enters the motor, h being due either to pressure or to velocity, or to pressure and velocity combined. The theoretic energy per second of this water is:

$$K = Wh;$$

and if W be in pounds, and h in feet, the theoretic horse-power of the water as it enters the motor is:

$$HP = \frac{Wh}{550};$$

and this is the power that can be developed by a motor of efficiency unity. The work k delivered by the motor is, however, always less than K, owing to losses in impact and friction, and the horse-power hp of the motor is less than HP. The efficiency of the motor is:

$$e = \frac{k}{K} = \frac{k}{Wh}; \text{ or, } e = \frac{hp}{HP};$$

and the value of this for turbine wheels is usually about 0.75; that is, the wheel transforms into useful work about 75 per cent of the energy of the water that enters it.

107. In designing a water-power plant, it should be the aim so to arrange the forebays and penstocks which lead the water to the wheel that the losses in these approaches may be as small as possible. The entrance from the head-race into the forebay, from the forebay into the penstock, and from the penstock to the motor, should be smooth and well-rounded; sudden changes in cross-section should be avoided; and all velocities should be low, except that at the motor. If these precautions be carefully observed, the loss of head outside the motor can be made very small.

Let H be the total head from the water level in the head-race to that in the tail-race below the motor. The total available energy per second is WH; and it should be the aim of the designer to render the losses of head in the approaches as small as possible, so that the effective head h may be as nearly equal to H as possible. Neglect of these precautions may render the effective power less than that estimated.

108. The efficiency e_1 of the approaches is the ratio of the energy K of the water as it enters the wheel, to the maximum available energy WH; or $e_1 = \frac{K}{WH}$. The efficiency E of the entire plant, consisting of both approaches and wheel, is the ratio of the work k delivered by the wheel to the energy WH; or,

$$E = \frac{k}{WH} = \frac{eK}{WH} = ee_1;$$

or, the final efficiency is the product of the separate efficiencies. If the efficiency of the wheel be 0.75, and that of the approaches 0.96, the efficiency of the plant as a whole is 0.72; or only 72 per cent of the theoretic energy is utilized. Usually the efficiency of the approaches can be made higher than 96 per cent.

In making estimates for a proposed plant, the efficiency of turbine wheels may be taken at 75 per cent; the effective work is then 0.75 Wh, where h is the actual *effective* head on the motor; and accordingly, if the wheels are required to deliver the work k per second, the approaches are to be arranged so that Wh shall not be less than 1.33 k. Especially when the water supply is limited is it important to make all efficiencies as high as possible.

109. **Water Delivered to a Motor.** To determine the efficiency of a hydraulic motor by formula, k is to be measured by the methods of Article 118, and h found by Articles 112 to 114. In order to find the weight W that passes through the wheel in one second, there must be known the discharge per second q, and the weight w of a cubic unit of water; then,

$$W = wq.$$

Here w may be found by weighing one cubic foot of the water; or, in approximate computations, w may be taken at 62.5 pounds per cubic foot. In precise tests of motors, however, its actual value should be ascertained as closely as possible.

110. The measurement of the flow of water through orifices, weirs, tubes, pipes, and channels is so fully discussed under "Hydraulics," that it only remains here to mention one or two simple methods applicable to small quantities, and to make a few remarks regarding the subject of leakage. In any particular case, that method of deter-

mining q is to be selected which will furnish the required degree of precision with the least expense.

For a small discharge, the water may be allowed to fall into a tank of known capacity. The tank should be of uniform horizontal cross-section, whose area can be accurately determined; and then the heights alone need be observed in order to find the volume. These, in precise work, will be read by hook gauges; and in cases of less accuracy, by measurements with a graduated rod. At the beginning of the experiment, a sufficient quantity of water must be in the tank so that a reading of the gauge can be taken; the water is then allowed to flow in, the time between the beginning and end of the experiment being determined by a stop-watch, duly tested and rated. This time must not be short, in order that the slight errors in reading the watch may not affect the result. The gauge is read at the close of the test after the surface of the water becomes quiet; and the difference of the gauge-readings gives the depth which has flowed in during the observed time. The depth, multiplied by the area of the cross-section, gives the volume; and this, divided by the number of seconds during which the flow occurred, furnishes the discharge per second q.

If the discharge be very small, it may be advisable to weigh the water rather than to measure the depths and cross-sections. The total weight divided by the time of flow then gives directly the weight W. This has the advantage of requiring no temperature observation, and is probably the most accurate of all methods; but unfortunately it is not possible to weigh a considerable volume of water, except at great expense.

When water is furnished to a motor through a small pipe, a common water meter may often be advantageously used to determine the discharge. No water meter, however, can be regarded as accurate until it has been tested by comparing the discharge as recorded by it with the actual discharge as determined by measurement or weighing in a tank. Such a test furnishes the constants for correcting the results found by its readings, which otherwise are liable to be 5 or 10 per cent in error.

111. The leakage which occurs in the flume or penstock before the water reaches the wheel, should not be included in the value of W, which is used in computing its efficiency, although it is needed in order to ascertain the efficiency of the entire plant. The manner of deter-

mining the amount of leakage will vary with the particular circumstances of the case in hand. If it be very small, it may be caught in pails and directly weighed. If large in quantity, the gates which admit water to the wheel may be closed, and the leakage being then led into the tail-race, it may be there measured by a weir, or by allowing it to collect in a tank. The leakage from a vertical penstock whose cross-section is known, may be ascertained by filling it with

Fig. 72. Weir for Modern Power Plant.

water, the wheel being still, and then observing the fall of the water level at regular intervals of time. In designing constructions to bring water to a motor, it is best, of course, so to arrange them that all leakage will be avoided; but this cannot often be fully attained, except at great expense.

The most common method of measuring q is by means of a weir placed in the tail-race below the wheel, as in Fig. 72. This has the disadvantage that it sometimes lessens the fall which would be otherwise available, and that often the velocity of approach is high. It has, however, the advantage of cheapness in construction and operation, and for any considerable discharge appears to be almost the only method which is both economical and precise. If the weir is placed above the wheel, the leakage of the penstock must be carefully ascertained.

112. **Effective Head on a Motor.** The total available head H between the surface of the water in the reservoir or head-race and that in the lower pool or tail-race, is determined by running a line of levels from one to the other. Permanent bench-marks being established, gauges can then be set in the head-race and tail-race, and graduated so that their zero points will be at some datum below the tail-race level. During the test of a wheel, each gauge is read by an observer at stated intervals; and the difference of the readings gives the head H. In some cases it is possible to have a floating gauge on the lower level, the graduated rod of which is placed alongside a glass tube that communicates with the upper level; the head H is then directly read by noting the point of the graduation which coincides with the water surface in the tube. This device requires but one observer, while the former requires two; but it is usually not the cheapest arrangement, unless a large number of observations are to be taken.

From this total head H, are to be subtracted the losses of head in entering the forebay and penstock, and the loss of head in friction in the penstock itself, and these losses may be ascertained by the methods discussed in "Hydraulics." Then,

$$h = H - h' - h''$$

is the effective head acting upon and chargeable to the wheel. In properly designed approaches, the lost heads h' and h'' are very small.

113. When water enters upon a wheel through an orifice which is controlled by a gate, losses of head will result, which can be estimated by the appropriate hydraulic formulæ. If this orifice is in the head-race, the loss of head should be subtracted, together with the other losses, from the total head H. But if the regulating gates are a part of the wheel itself, as is the case in a turbine, the loss of head should not be subtracted, because it is properly chargeable to the construction of the wheel, and not to the arrangements which furnish the supply of water. In any event that head should be determined which is to be used in the subsequent discussions: if the efficiency of the fall

is desired, the total available head is required; if the efficiency of the motor, that effective head is to be found which acts directly upon it.

114. When water is delivered through a nozzle or pipe to an impulse wheel, the head h is not the total fall, since a large part of this may be lost in friction in the pipe, but is merely the velocity-head $\frac{v^2}{2g}$ of the issuing jet. The value of v is known when the discharge q and the area of the cross-section a of the stream have been determined; and,

$$h = \frac{v^2}{2g} = \frac{q^2}{2ga^2}.$$

It is here assumed that the center of the nozzle is substantially at tail-water level. In the same manner, when a stream flows in a channel against the vanes of an undershot wheel, the effective head is the velocity-head; and the theoretic energy is, in either case:

$$K = Wh = W\frac{v^2}{2g} = \frac{wq^3}{2ga^2}.$$

If, however, the nozzle be above the elevation of tail-water, and the water, *in passing through the wheel*, falls a distance h_0' below the mouth of the nozzle, then the head which *actually acts* upon the wheel considered as a water motor merely, is given by:

$$h_1 - \frac{v^2}{2g} + h_0';$$

but the effective head chargeable to the wheel as part of the installation is:

$$h = \frac{v^2}{2g} + h_0,$$

in which h_0 is the distance of the nozzle center above the tail-water level. In order to utilize the fall h_0 efficiently, it is plain that the wheel should be placed as near the level of the tail-race as possible.

Lastly, when water enters a turbine wheel through a pipe, a piezometer or a pressure-gauge may be placed near the wheel entrance, to register the pressure-head during the flow; if this pressure-head, measured from the water level in the tail-race, be called h_0'', and if the velocity in the pipe be v, then,

$$h - \frac{v^2}{2g} + h_0''$$

is the effective head chargeable to the wheel as part of the installation.

The head chargeable to the wheel itself, regarded merely as a water-motor, without reference to its installation, is:

$$h_1 = \frac{v^2}{2g} + h_0',$$

in which h'_0 is the pressure-head measured upward from the lowest part of the exit orifices.

From the above discussion, it will be seen that a distinction is sometimes made between the efficiency of the motor itself, and the efficiency of the motor installed as a hydraulic machine. In the former case, only that part of the available head which is actually utilized by the motor should be used in the calculations; in the latter case, the entire available head. The case of discharge into a draft-tube is considered in a subsequent article.

115. **Measurement of Effective Power.** The effective work and horse-power delivered by a water-wheel or hydraulic motor are often required to be measured. Water-power may be sold by means of the weight W, or quantity q, furnished under a certain head, leaving the consumer to provide his own motor; or it may be sold directly by the number of horse-power. In either case, tests must be made from time to time, in order to insure that the quantity contracted for is actually delivered, and is not exceeded. It is also frequently required to measure the effective work, in order to ascertain the power and efficiency of the motor, either because the party who buys it has contracted for a certain power and efficiency, or because it is desirable to know exactly what the motor is doing, in order to improve if possible its performance.

116. The test of a hydraulic motor has for its object: *First,* the determination of the effective energy and power; *second,* the determination of its efficiency; and *third,* the determination of that speed which gives the greatest power and efficiency. If the wheel be still, there is no power; if it be revolving very fast, the water is flowing through it so as to change but little of its energy into work; and in all cases there is found a certain speed which gives the maximum power and efficiency. To execute these tests, it is not at all necessary to know how the motor is constructed, or the principle of its action, although such knowledge is very valuable, and is in fact indispensable, in order to enable the engineer to suggest methods by which its operation may be improved.

117. A method in which the effective work of a small motor may be measured, is to compel it to exert all its power in lifting a weight. For this purpose, the weight may be attached to a cord which is fastened to the horizontal axis of the motor, around which it winds as the shaft revolves. The wheel then expends all its power in lifting this weight W_1 through the height h_1 in t_1 seconds; and the work performed per second, then, is:

$$k = \frac{W_1 h_1}{t_1}.$$

This method is rarely used in practice, on account of the difficulty of measuring t_1 with precision.

118. The usual method of measuring the effective work of a hydraulic motor is by means of the *friction brake* or *power dynamometer* invented by Prony about 1780. In Fig. 73 is illustrated a simple method of applying the apparatus to a vertical shaft, the upper diagram being a plan, and the lower an elevation. Upon the vertical shaft is a fixed pulley; and placed against this, are seen two rectangular pieces of wood hollowed so as to fit it, and connected by two bolts. By turning the nuts on these bolts while the pulley is revolving, the friction can be increased at pleasure, even to the extent of stopping the motion; around these bolts, between the blocks, are two spiral springs (not shown in the diagram) which press the blocks outward when the nuts are loosened. To one of these blocks is attached a cord, which runs horizontally to a small movable pulley over which it passes, and supports a scale pan in which weights are placed. This cord runs in a direction opposite to the motion of the shaft, so that when the brake is tightened it is prevented from revolving by the tension caused by the weights. The direction of the cord in the horizontal plane must be such that the perpendicular let fall upon it from the center of the

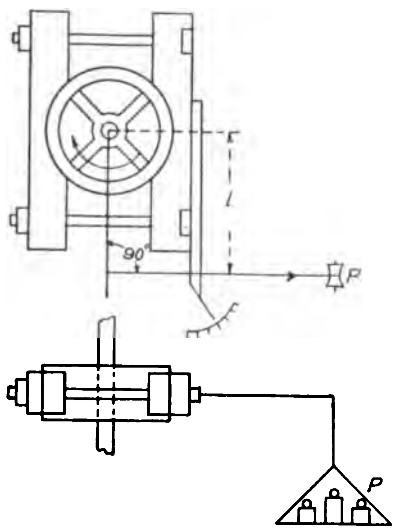


Fig. 73. Measuring Work of Motor by Prony Brake.

shaft, or its lever-arm, is constant; this can be effected by keeping the small pointer on the brake at a fixed mark established for that purpose.

119. To measure the work done by the wheel, the shaft is disconnected from the machinery which it usually runs, and is allowed to revolve, transforming all its work into heat by the friction between the revolving pulley and the brake, which is kept stationary by tightening the nuts and at the same time placing sufficient weight in the scale-pan to hold the pointer at the fixed mark. Let n be the number of revolutions per second, as determined by a counter attached to the shaft; P, the tension in the cord, which is equal to the weight of the scale-pan and its loads; l, the lever-arm of this tension with respect to the center of the shaft; r, the radius of the pulley; and F, the total force of friction between the pulley and the brake. Now, in one revolution, the force F is overcome through the distance $2\pi r$; and in n revolutions through the distance $2\pi rn$. Hence the effctive work done by the wheel in one second is:

$$K = F.2\pi rn = 2\pi nFr.$$

The force F, acting with the lever-arm r, is exactly balanced by the force P acting with the lever-arm l; accordingly the moments Fr and Pl are equal; and hence the work done by the wheel in one second is:

$$K = 2\pi n\ Pl \quad \ldots\ldots\ldots\ldots \quad \mathbf{(101)}$$

If P be in pounds, and l in feet, the effective horse-power of the wheel is given by:

$$hp = \frac{2\pi n\ Pl}{550} \quad \ldots\ldots\ldots\ldots \quad \mathbf{(101a)}$$

As the number of revolutions in one second cannot be accurately read, it is usual to record the counter readings every minute or half-minute. If N be the number of revolutions per minute,

$$hp = \frac{2\pi\ NPl}{33{,}000} \quad \ldots\ldots\ldots\ldots \quad \mathbf{(101b)}$$

It is seen that this method is independent of the radius of the pulley, which may be of any convenient size. For a small motor, the brake may be clamped directly upon the shaft; but for a large one a pulley of considerable size is needed, and a special arrangement of levers is used, instead of a cord.

120. The efficiency of the motor is now found by dividing the effective work per second by the theoretic work per second. Let K be this theoretic work, which is expressed by Wh; then,

$$e = \frac{k}{K}; \text{ or, } e = \frac{hp}{HP}. \quad \ldots\ldots (101c)$$

The work measured by the friction brake is that delivered at the circumference of the pulley, and does not include that power which is required to overcome the friction of the shaft upon its bearings. The shaft or axis of every water-wheel must have at least two bearings, the friction of which consumes probably about 2 or 3 per cent of the power. The hydraulic efficiency of the wheel, regarded as a user of water, is hence 2 or 3 per cent greater than the computed value of e.

121. There are in use various forms and varieties of the friction brake; but they all act upon the principle and in the manner above described. For large wheels, they are made of iron, and almost completely encircle the pulley; while a special arrangement of levers is used to lift the large weight P. If the work transformed into friction be large, both the brake and the pulley may become hot, to prevent which a stream of cool water is allowed to flow upon them. To insure steadiness of motion, it is well that the surface of the pulley should be lubricated, which, for a wooden brake, is well done by the use of soap. It is important that the connection of the cord to the brake should be so made that the lever-arm l increases when the brake moves slightly with the wheel; if this is not done, the wheel will be apt to cause the brake to revolve with it.

TURBINE TESTING*

122. **Holyoke Testing-Flume.** At Holyoke, Mass., where the Connecticut River furnishes a large water power, falling some 60 feet, the Holyoke Water-Power Company controls the water rights, and leases power to the many mill operators of that city. The mill-owners pay a certain price per annum per *mill-power,* which, in that locality, is the right to use 38 cubic feet of water per second under a head of 20 feet, either for continuous use (a 24-hour day) or for a definite fraction of each day.

*Articles 123 to 130 inclusive have been taken, with slight changes, from Professor Church's "Hydraulic Motors."

In order that the rate at which any mill turbine uses water at any stage or position of its gate or regulating apparatus may become known by simply observing the position of the gate, each turbine, before being installed in the mill where it is to work, is tested at the *testing-flume* of the company, and thus becomes a water-meter, whose indications, when the motor is in final place, are noted from day to day by an inspector to the company. In the same test, its power, best speed, and efficiency are also determined.

The testing-flume occupies the lower part of a substantial building, and its main features are shown in vertical section in Fig. 74. The

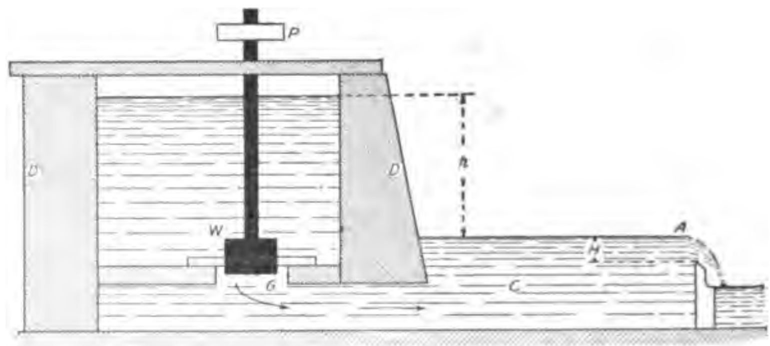


Fig. 74. Vertical Section of Holyoke Testing-Flume.

walls of the wheel-pit *DD*, which is 20 feet square, are built of stone masonry, and lined with brick laid in cement. The water is admitted to it from the head canal through a trunk or penstock, and vestibule, which are not shown in the figure. Over an opening in the floor of the wheel-pit, the wheel *W* to be tested is set in place, the water discharged from it finding its way through a large opening into the tail-race *C*, 35 feet long and 20 feet wide; and finally over a sharp-crested weir at *A*, into the lower canal. The whole head *h* available for testing may be from 4 to 18 feet for the smaller wheels, and from 11 to 14 feet for large wheels, up to 300 horse-power. The measuring capacity of the weir, which may be used to its full length, 20 feet (and then would have no end-contractions), is about 230 cubic feet per second. The head *h* becomes known in any test by observations of the water level in two glass tubes communicating with the respective bodies of water *W* and *C*. The water in channel *C*, which is a ***channel of approach***

DIVERTING DAM, INTAKE, AND HEAD-GATE FOR FLUME

Courtesy of Pelton Water Wheel Co., San Francisco, Cal for the weir A, communicates (at a point some distance back of the weir) by a lateral pipe with the interior of a vessel open to the air, in a side chamber. Water rises in this vessel, and finally remains stationary at the same level as that of the surface in the channel of approach. A hook gauge being used in connection with this vessel, observations and readings are taken, from which the value of H, or *head on the weir*, may be computed, for use in the proper weir formula for the discharge q.

Fig. 74 shows a turbine in position for testing, with a vertical shaft—the ordinary case. Upon the upper end of the shaft is secured a cast-iron pulley P, to the rim of which the Prony brake is fitted for purposes of test.

123. The procedure of testing was about as follows: The brake being carefully balanced and adjusted beforehand, a light weight was placed on the scale-pan, and the wheel started at full gate; sufficient friction was then produced to balance the weight, and the speed of wheel noted. The load was then increased at intervals of two or three minutes, by 25 lbs. at a time, until the speed of the wheel had fallen below that of maximum efficiency for the head; the weights were then reduced again, and the velocity of the wheel allowed to increase until the maximum was again passed. The same process was then repeated within a smaller range of speed and with smaller variations of load, until the speed of best work had been more exactly ascertained, and the performance of the turbine at maximum efficiency, under full head and at full gate, had been very precisely determined. This was repeated at each of the part gates, usually down to one-half maximum discharge.

124. Test of the Tremont Turbine. The test of the *Tremont Turbine,* a 160-horse-power turbine of the radial outward-flow type (Fourneyron), made at Lowell, Mass., in 1855 by Mr. J. B. Francis, was an event of special interest in the history of hydraulic science, and has become classic. Though the test is by no means recent, it was carried out so thoroughly as to make its details highly instructive to the student of hydraulics. The main features of this test will now be presented and commented on.

The inner and outer radii of the turbine were 3.37 and 4.14 feet respectively; height between crowns, 0.937 foot at entrance, and 0.931 foot at exit. There were 33 guide-blades and 44 turbine-vanes.

As to angles, $\alpha = 28°$, $\phi = 90°$, and $\beta = 22°$; and the head h on the wheel varied from about 12.5 to about 13.5 feet. The gate was a thin cylinder, movable vertically between the guides and the wheel. There were no horizontal partitions dividing up the wheel-channels—in fact, no special device for preventing the loss of head usually arising at part gate with this kind of regulating apparatus.

125. The annexed table (page 105) gives the principal data and results of Mr. Francis's test of the Tremont turbine, arranged in the order of the speed of wheel. In Experiments Nos. 1 to 15 (see column 1), the cylindrical gate was fully open ("full gate"); while in Experiments 16 to 20, it was in a single fixed position, leaving open, at the wheel-entrance, about one-quarter of the vertical height between crowns; in other words, the gate was drawn up about one-quarter of its full range of height. In this special "part-gate" position, however, the quantity of water passing per second was much greater than one-quarter of that passing at "full gate," as is seen from the values of q in column 4. For example, in Experiment 18, in which (for this position of the gate) the efficiency was a maximum, the value of q is about one-half of the q used in Experiment 6, which gives the maximum efficiency at full gate. It would be said, therefore, that in Experiment 18 the wheel was working at about "half gate." The heading of each column of the table shows clearly the nature of the quantity given in that column, and the units of measurement involved in its numerical value.

126. The rate of flow, or discharge in cubic feet per second, was measured by two weirs at the end of the tail-race, using the Francis weir formula; and the useful power was measured by means of the Prony brake, which in this case consisted of a large and strong friction brake with arcs of wood rubbing on the cast-iron pulley which was keyed to the turbine shaft, and arranged with a bell-crank lever and "dashpot" to "cushion" the motion of the lever. In this brake, $r = 2.75$ feet, and $l = 10.83$ feet.

127. To explain the computations connected with these investigations, Experiment No. 6 will be selected. In this test, 1,524 lbs. was placed in the scale pan, and the nuts tightened up, until the wheel raised this weight and held it just balanced. When the speed of the wheel had adjusted itself to the load, the speed counter indicated $n = 0.851$ revolution per second. Hence,

TEST OF THE TREMONT TURBINE

Selected Experiments

1	2	3	4	5	6	7
No of Exper	h (Feet)	n (Revs. per sec.)	q (Cub. ft. per sec.)	$2\pi\ nPl$ (Ft.-lbs. per sec.)	e (Effic.)	$h.\ p.$
FULL GATE						
1	12.80	0.00	135.6	0	0.00	
2	12.95	0.45	133.4	73,160	.68	
3	12.97	0.53	133.7	78,490	.72	
4	12.97	0.60	134.8	82,110	.75	
5	12.94	0.64	135.1	83,960	.77	
6	**12.90**	**0.85**	**138.2**	**88,210**	**.794**	**160.3**
7	12.90	0.88	139.0	88,190	.788	
8	12.90	0.90	139.6	88,076	.784	
9	12.85	1.00	141.9	86,310	.75	
10	12.85	1.06	142.5	83,970	.73	
11	12.80	1.18	144.8	77,150	.67	
12	12.70	1.31	147.3	66,840	.57	
13	12.65	1.46	152.3	51,680	.43	
14	12.55	1.60	156.6	33,350	.27	
15	12.54	1.79	162.3	0	0.00	
PART GATE						
16	13.51	0.00	60.3	0	0.00	
17	13.55	0.46	67.8	24,460	.43	
18	**13.48**	**0.67**	**71.8**	**27,980**	**.46**	**50.9**
19	13.39	0.96	76.6	21,250	.33	
20	13.34	1.25	80.4	0	.00	

$$
\begin{aligned}
K &= 2\pi\ nPl \\
&= 2 \times 3.1416 \times 0.851 \times 1{,}524 \times 10.83 \\
&= 88{,}214 \text{ foot-pounds per second} = 160.3 \text{ horse-power.}
\end{aligned}
$$

For computing the value of the discharge q, it is to be observed that the water passed over two contracted weirs with a combined length $b = 16.98$ feet; the number of end contractions was therefore $n = 4$; the head over the weir crest was $H = 1.87$ feet; velocity of approach was not considered. Therefore,

$$
\begin{aligned}
q &= 3.33\,(b - 0.1\,nH)\,H^{\frac{3}{2}} \\
&= 3.33\,(16.98 - .75)\,(1.87)^{\frac{3}{2}} \\
&= 138.2 \text{ cubic feet per second.}
\end{aligned}
$$

The difference in elevation between head and tail-water levels in this experiment was 12.90 feet; consequently the total available energy was:

$$K = 138.2 \times 62.5 \times 12.90 = 111{,}400 \text{ ft.-lbs. per second.}$$

Therefore the efficiency was:

$$e = \frac{88,214}{111,400} = 79.4 \text{ per cent.}$$

128. *Discussion of Test of Tremont Turbine.* In the experiments with full gate, Nos. 1 to 14 inclusive (see table, page 105), on account of the progressive lessening of the weight P in the scale pan (the brake friction being regulated each time to correspond), the uniform speed to which the wheel adjusts itself in successive experiments increases progressively from the zero value, or state of rest, of Experiment 1 (when the friction was so great as to prevent any motion), up to a maximum rate of 1.79 revolutions per second, attained when no brake friction whatever (*no load*) was present. In this last experiment, there being no useful work done, all the energy of the mill-site is wasted, partly in axle friction, but chiefly in fluid friction (eddying of the water, and finally, heat), both in the wheel-passages and also in the tail-race, where the water which has left the wheel with high velocity soon has its velocity extinguished. The same statement is true also for Experiment No. 1, except that axle friction is wanting. In both experiments the efficiency is, of course, zero.

The quantity of water discharged per second, q, is seen to increase slowly (after Experiment 2) from 133.4 to 162.3 cubic feet per second, though not differing from the average by more than ten per cent. This may be accounted for, in a rude way, as an effect of *centrifugal action* (as in a centrifugal pump), since the Tremont turbine is an outward-flow wheel. The reverse is found to be true for inward-flow turbines, notably the Thompson vortex wheel, which is therefore to some extent self-regulating in the matter of speed, since a less discharge at a speed higher than the normal diminishes the power, and hence the tendency to further increase of speed.

In the succession of experiments Nos. 1 to 15 (all at full gate and under practically the same head h), the efficiency is seen to have a zero value both at beginning and end of this series, and to reach its maximum at about the sixth experiment, in which the speed is noted as being about one-half that at which the turbine runs when entirely "unloaded" (Experiment 15). This is roughly true in nearly all turbine tests; but a notable feature of considerable practical advantage is that a fairly wide deviation from the best speed affects the efficiency but slightly. For instance, a variation of speed by 25 per cent either way from the best value (of 0.85 revolution per second) causes a diminution in the efficiency of only about four per cent.

It should be remembered, also, in this connection, that since the water used per second (q) is somewhat different at different speeds (at full gate), the speed of maximum power differs slightly from that of maximum efficiency.

129. In the five "part-gate" experiments, Nos. 16 to 20, the gate remains fixed in a definite position (about one-quarter raised, although the discharge is about one-half that of full gate) through all these five runs. The head is practically constant. At first the wheel is prevented from turning. The power and efficiency are then, of course,

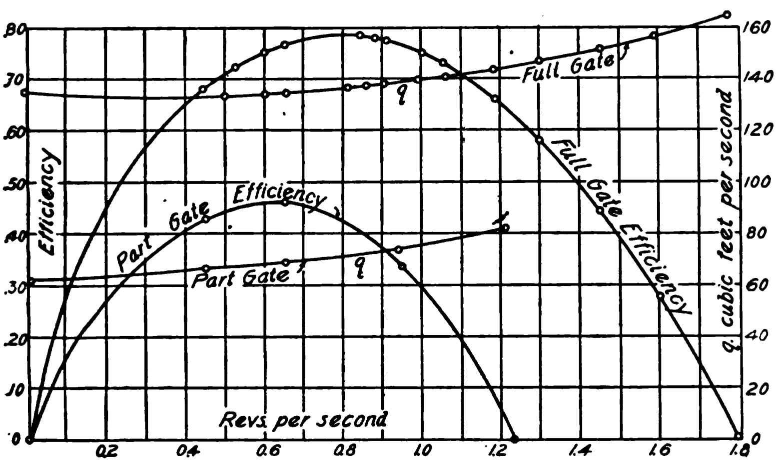


Fig. 75. Curves Showing Results of Test of Tremont Turbine.

zero; but $q = 60.3$ cubic feet per second. As the turbine is permitted to revolve under progressively diminishing friction, the speed of steady motion becomes greater, reaching its maximum (1.25 revolutions per second) when the wheel runs "unloaded," in Experiment 20; but the power reaches a maximum and then diminishes. The same is true of the efficiency, whose maximum (in Experiment 18) is seen to be about 46 per cent only. This forms a striking instance of the disadvantage and wastefulness of a cylindrical gate unaccompanied by other mitigating features, when in use at part gate. This defect, however, may be largely remedied by the use of horizontal partitions in the wheel-channels, as in Fig. 81, or by employing curved upper crowns, as in the *American* "inward and downward" turbines.

130. Fig. 75 is a graphic representation of the results of the test as set forth in the table (page 105).

REACTION TURBINES

131. **Formula for Inward or Outward Flow.** *Discharge.* The following analysis applies equally well with either direction of flow. The discharge from a reaction turbine, unlike that from an impulse turbine, depends on the speed of revolution, as well as on the orifice areas, as in the case of the reaction wheel already discussed. Let Fig. 76 represent diagrammatically an outward-flow turbine with the customary notation as shown in Fig. 77 and explained in preceding analyses. In addition, let a_1, a, and a_0 be the areas of the respective orifices or water-passages, measured normal to the directions of V_1, V, and v_0; and let H represent the pressure head on the guide-orifices at the gate openings, as would be indicated by piezometer tubes or pressure gauges if they were inserted at such points.

From Article 6, neglecting frictional losses,

$$H + \frac{v_0^2}{2g} = h + H_1;$$

also, from Article 89, neglecting frictional losses,

$$H_1 + \frac{V_1^2}{2g} - \frac{u_1^2}{2g} = H + \frac{V^2}{2g} - \frac{u^2}{2g}.$$

The addition of these two equations results in the following formula:

$$V_1^2 - V^2 + v_0^2 = 2gh + u_1^2 - u^2 \quad \ldots\ldots\ldots \mathbf{(102)}$$

Being a reaction turbine, the buckets are completely filled; therefore the same quantity of water must pass per second through each of the areas a_1, a, and a_0; from which condition the following relations are obtained:

$$V_1 = \frac{q}{a_1}; \quad V = \frac{q}{a}; \quad v_0 = \frac{q}{a_0}.$$

Substituting these values in the last formula above, and solving for q, there results:

$$q = c_d \sqrt{\frac{2gh + u_1^2 - u^2}{\frac{1}{a_1^2} - \frac{1}{a^2} + \frac{1}{a_0^2}}} = m\sqrt{2gh + u_1^2 - u^2} \quad \ldots . \mathbf{(102a)}$$

which is a formula for discharge through a reaction turbine, for either inward or outward flow. The coefficient c_d is introduced to take account of losses through leakage and friction, which factors were not considered in the above analysis. For an outward-flow turbine, u_1 is greater than u, consequently the discharge increases with the speed; for an inward-flow turbine u_1 is less than u, and therefore the discharge varies inversely with the speed.

The value of the coefficient c_d varies with the head of water and with the details of the wheel design. In one case of an outward-flow turbine, in which $r = 2.67$ ft.; $r_1 = 3.32$ ft.; with total head of water varying between 17.16 and 17.34 ft.; number of revolutions per minute, between 63.5 and 100; and the discharge with full gate, from 117 to 127.7 cubic feet per second, the value of c_d was found to range between .941 and .950.

A formula for discharge from a turbine operating at part gate is difficult to formulate theoretically, because of the losses of head resulting from the partial closure, analytical expressions for which are not definitely known for turbines. The values of q for any turbine operating at part or full gate may be obtained by measuring the quantity of water actually discharged, by any of the various methods described in works on hydraulics, and substituting the resulting values of q in Equation 102a.

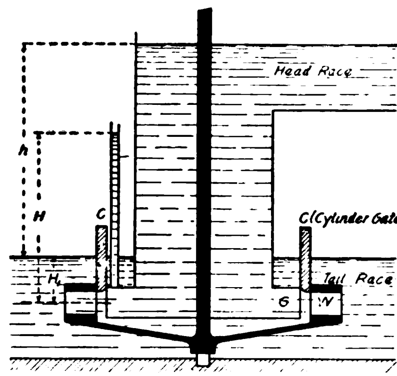


Fig. 76. Diagrammatic Representation of Outward-Flow Turbine.

132. **Work and Efficiency.** The following analysis is also valid for both directions of flow. In addition to the notation adopted and employed in the preceding article, and indicated on the corresponding diagrams, let d_1, d, and d_0 be the respective depths of the exit, the entrance, and the guide-orifices or water-passages. With gates fully open (in which case d_0 becomes equal to d), and neglecting the thickness of the vanes or passage walls,

$$a_0 = 2\pi rd \sin\alpha; \quad a = 2\pi rd \sin\phi; \quad a_1 = 2\pi r_1 d_1 \sin\beta.$$

Since, in a reaction turbine, the passageways are always completely full of water,

$$q = v_0 . 2\pi rd \sin\alpha = V . 2\pi rd \sin\phi = V_1 2\pi r_1 d_1 \sin\beta. \quad \text{(102b)}$$

The general conditions previously established and discussed—that for maximum efficiency the water must enter tangentially to the vanes, and the absolute velocity of the water at discharge must be as low as possible—will be fulfilled for the first case, when u and v_0 are proportional to the sines of their opposite angles—that is, when

$$\frac{u}{v_0} = \frac{\sin(\phi - \alpha)}{\sin \phi}; \quad \dots\dots\dots\dots \quad \mathbf{(103)}$$

and for the second case, approximately and simply, when

$$u_1 = V_1.$$

(The theoretic condition is $V_1 = u_1 \cos \beta$, as explained in a preceding article.)

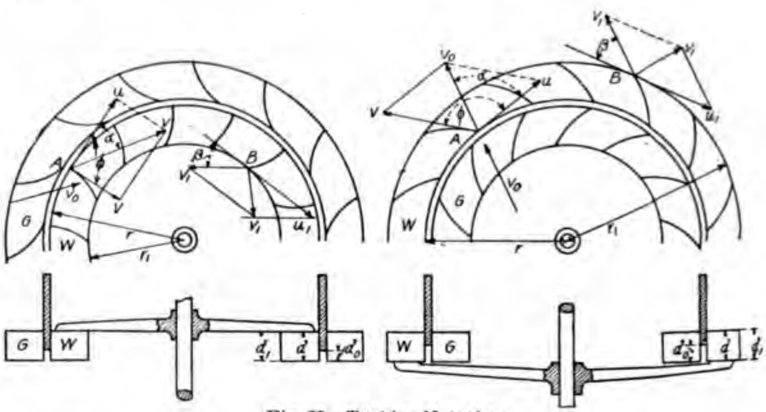


Fig. 77. Turbine Notation.

Substituting u_1 for V_1 in the third member of Equation 102*b*, and equating it to the first, there results:

$$\frac{u_1}{v_0} = \frac{rd \sin \alpha}{r_1 d_1 \sin \beta}.$$

Then, since $\frac{u}{u_1} = \frac{r}{r_1}$, the above equation may be reduced to:

$$\frac{u}{v_0} = \frac{r^2 d \sin \alpha}{r_1^2 d_1^2 \sin \beta} \quad \dots\dots\dots\dots \quad \mathbf{(103a)}$$

From the trigonometric relations at the point A, in Fig. 77,

$$V^2 = u^2 + v_0^2 - 2uv_0 \cos \alpha.$$

Substituting this value of V^2 in Equation 102, and making $V_1 = u_1$, as established above for one of the conditions of maximum efficiency, the following additional important relation immediately results:

$$uv_0 = \frac{gh}{\cos \alpha} \quad \dots\dots\dots\dots \quad \mathbf{(104)}$$

From the above necessary relations, the following practical formulæ may be developed by combining Equations 103 and 104:

$$u = \sqrt{\frac{gh \sin(\phi - \alpha)}{\cos \alpha \sin \phi}} \ldots\ldots\ldots (105)$$

and,

$$v_0 = \sqrt{\frac{gh \sin \phi}{\cos \alpha \sin(\phi - \alpha)}} \ldots\ldots\ldots (106)$$

Equation 105 gives the advantageous velocity of the circumference at the point of wheel entrance, from which the advantageous velocity of the circumference at exit may be obtained from the relation $\frac{u}{u_1} = \frac{r}{r_1}$; and Equation 106 gives the value of the absolute velocity of entrance of the water into the wheel.

By combining Equations 103 and 103*a*, there is obtained:

$$\frac{\sin(\phi - \alpha)}{\sin \phi} = \frac{r^2 d \sin \alpha}{r_1^2 d_1 \sin \beta},\ldots\ldots\ldots (107)$$

which establishes the necessary relations between the dimensions and angles of the wheel which must obtain in order that the above conclusions may be valid.

The work imparted to the wheel is, theoretically:

$$\text{Work} = W \frac{h - v_1^2}{2g};\ldots\ldots\ldots (108)$$

and the efficiency is theoretically:

$$e = 1 - \frac{v_1^2}{2gh}\ldots\ldots\ldots (108a)$$

in which h indicates the available head properly chargeable to the machine as installed.

By means of Equations 105 and 107, and the relations at the point B, the above value of the efficiency may be reduced to:

$$e = 1 - \frac{d}{d_1} \tan \alpha \tan \tfrac{1}{2} \beta \ldots\ldots (109)$$

in which d and d_1 are the entrance and exit depths, respectively.

The discharge is:

$$q = a_0 v_0;$$

and the useful work of the wheel is:

$$\text{Work} = e \times wqh.$$

It is to be observed that the foregoing formulæ for work and efficiency do not take into account losses of energy incurred during the passage of the water to and through the guide- and wheel-buckets and those due to clearance.

If losses due to impact and friction in the runner buckets be neglected, then the work imparted to the wheel is:

$$\text{Work} = W\frac{h^1 - v_1^2}{2g}; \dots\dots\dots\dots (109a)$$

and the efficiency is:

$$e = \frac{h^1 - v_1^2}{2gh} \dots\dots\dots\dots\dots (110)$$

in which h^1 is the head representing the total energy of the water as it enters the runner buckets.

133. **Downward-Flow or Parallel-Flow Reaction Turbine.** Let r be the mean radius, and u the mean velocity, of the entrance and exit orifices of the wheel, and let d and d_1 be the widths of the entrance and exit orifices respectively. The formulæ developed in the preceding articles for inward- and outward-flow reaction turbines may be adapted to this case by making $u_1 = u$, and $r_1 = r$.

Thus the advantageous velocity of entrance is:

$$v_0 = \sqrt{\frac{gh \sin \phi}{\cos \alpha \sin (\phi - \alpha)}} \dots\dots\dots (111)$$

The advantageous speed is:

$$u = \sqrt{\frac{gh \sin (\phi - \alpha)}{\cos \alpha \sin \phi}} \dots\dots\dots (112)$$

The necessary relation between the vane angles and the wheel dimensions is:

$$\frac{\sin (\phi - \alpha)}{\sin \phi} = \frac{d \sin \alpha}{d_1 \sin \beta} \dots\dots\dots\dots\dots (113)$$

and the hydraulic efficiency is:

$$e = 1 - \frac{d}{d_1} \tan \alpha \tan \tfrac{1}{2} \beta \dots\dots (114)$$

IMPULSE TURBINES

134. **Formulæ.** The velocity v_0 with which the water leaves the guide-orifices and enters the runner buckets, is, in the case of impulse turbines, theoretically equal to $\sqrt{2gh_0}$, where h_0 stands for the *effective* head on such orifices. The analysis and conclusions of Articles 76 and 77 apply directly to such motors. Accordingly, for a properly designed impulse turbine, the entrance angle should be double the approach angle; that is:

$$\phi = 2\alpha.$$

The advantageous speed is:

$$u = \sqrt{\frac{gh_0}{2 \cos^2 \alpha}}.$$

When the motor is running at this best speed, the absolute velocity of exit is:

$$v_1 = v_0 \frac{r_1 \sin \frac{1}{2}\beta}{r \cos \alpha}.$$

The work imparted to the wheel is:

$$\text{Work (Max.)} = W \frac{v_0^2 - v_1^2}{2g} = Wh_0 \left\{ 1 - \left(\frac{r_1 \sin \frac{1}{2}\beta}{r \cos \alpha} \right)^2 \right\};$$

and the efficiency is:

$$e\,(\text{Max.}) = \frac{h_0}{h} \left\{ 1 - \left(\frac{r_1 \sin \frac{1}{2}\beta}{r \cos \alpha} \right)^2 \right\},$$

in which h is the available head properly chargeable to the motor

It is clearly to be seen that both the approach angle α and the exit angle β should be small for high efficiency, and that the angle β exercises a greater influence on the efficiency than the angle α, both of which conclusions have already been discussed in the articles referred to above.

The discharge is:

$$q = a_0 v_0 = a_0 \sqrt{2gh_0};$$

and the work of the turbine per second is:

$$\text{Work} = e \times wqh.$$

With both reaction and impulse turbines, when the guide-buckets are considered part of the turbine, h is the head representing the total available energy existing in the water as it enters the guides. When the turbine runner alone is considered to constitute the motor proper, and the guides part of the approach, h is the head representing the total available energy existing in the water as it emerges from the guide-buckets. In the latter case, in the above analysis, h may be put equal to h_0. In both cases, however, there is a certain amount of loss in the clearance.

135. **General Definitions.** *Complete and Partial Admission.* These terms signify that water may be admitted to all the wheel passages at once, or to a limited number of them.

Gate-Opening; Part Gate. The area left open for the clear passage of water by the regulating gate or gates, is called the *gate-opening.* It should not be confounded with the amount of water flowing through the gate-opening at any particular position of the gate. The term *five-eighths gate opening*, etc., is often employed to mean that position of the regulating apparatus which allows five-eighths, etc., of the full discharge (*i. e.,* discharge at full gate) to take place. *Five-eighths discharge* would better express the meaning in such a case.

Clearance. The clear space between the guide-ring and the runner is called the *clearance.*

136. **Impulse and Reaction Turbines.** *Comparisons.* Reaction wheels necessarily have complete admission; and partial closure of the gates (to all the wheel passages simultaneously) results in material loss of energy, due to expansion of section after the contraction; while, with impulse wheels, the admission may be complete, or the supply may be decreased by partial admission, with little, if any sacrifice of efficiency, Moreover, the regulating gates for impulse turbines are of much simpler construction than those for reaction turbines.

The speed of an impulse turbine for a given head is less than that of a reaction turbine; but the relative velocity of entrance is greater; hence there is greater liability to shock and eddies.

The dimensions of an impulse wheel may vary between wide limits, so that for high falls with a small supply of water, a comparatively large wheel may be employed, with a low speed. The speed of a reaction turbine under such conditions would be inconveniently great; and any considerable increase in diameter to reduce the speed would increase the fluid friction, and render more troublesome the proper proportioning of the vanes. When there is an ample supply of water, and the fall is not too great, the reaction turbine is usually to be preferred; but on high falls, on account of the resulting speed becoming inconveniently great, a turbine of the impulse type should be adopted, which permits without disadvantage an increase in diameter with corresponding decrease in speed. On very high falls, an impulse wheel would be preferable.

The advantageous speed of an impulse turbine remains the same for all positions of the gate; but with reaction turbines it is less at part gate than at full gate; and it has already been stated that the partial closing of the gates results in a material loss of energy in the case of reaction wheels; and since, for many industrial purposes, it is essential to maintain a constant speed in spite of variations in power or useful work, it follows that to maintain this constant speed with a reaction turbine involves considerable loss of efficiency. It is also evident that a turbine of the impulse type has a marked advantage in point of efficiency when the supply is low.

137. To partially prevent the loss of head, with consequent loss of power, incidental to the operation of a reaction turbine at "part gate," the turbine runner may be divided into several parts or *stories* by means of horizontal partitions, as described later (see Figs. 81, 82). During partial operation, one or more of these divisions would be entirely closed off by the gate, the remaining divisions only being in action; the expansion of the stream due to partial throttling being thus avoided, the efficiency at "part gate", when using less than the usual quantity of water, would not be materially altered.

138. The *Duplex* motor, a double turbine of the parallel-flow type, consists of a pair of concentric runners made in one piece, supplied with water by a similar pair of concentric annular supply guide-passages. The supply for either division may be cut off independently, leaving the other runner in action without sacrifice of efficiency. Such motors must be distinguished on the one hand from double motors of the *two-story* type, as just described, and on the other hand from double turbines, in which two essentially different wheels are combined and mounted on the same shaft for the purpose of increasing the capacity of the turbine without increasing its diameter, as in the *Leffel* and other types.

139. The principal disadvantages of the impulse type are that the turbine must always work in air, for, under water, the runner buckets cannot be ventilated; or, if a draft-tube is used, an air-admission valve must be provided, both to supply air for ventilating the buckets, and to keep the surface of the water in the draft-tube below the exit orifices and moving parts of the motor, as explained in a subsequent article.

140. In impulse turbines, the entrance angle ϕ should be double the approach angle a; but in reaction turbines it is often greater than $3a$, and its value depends on the exit angle β; hence the vanes in impulse turbines are of sharper curvature for the same values of a and β. β is usually greater for inward-flow than for outward-flow reaction wheels, in order that the exit orifices may be sufficiently large. If the entrance angle ϕ is 90° (a good value), Equation 105 shows that the velocity u is that due to one-half the head. Equation 109 shows that the efficiency is increased by making the exit depth d_1 greater than the entrance depth d (*Bell-mouthed* profiles; *Diffuser*); but usually they do not differ very much, and frequently they are made equal.

141. **Discharge.** Impulse turbines always discharge into the air, at some distance above tail-water; consequently that part of the available head between the center of discharge and the tail-water level is lost, unless the motor is set to operate in the space above the "hanging column" of a draft-tube, in which case only part of the head is lost, as described later. A reaction turbine may discharge into the free air, in which case the same loss occurs; or it may be "drowned" —that is, set below the water level in the tail-race; or it may discharge into a suction or draft-tube; in these two latter cases the above-mentioned loss will not take place.

TURBINE ACCESSORIES

142. **Diffuser.** An apparatus for the purpose of providing a gradual enlargement of section for the passage of the discharge water

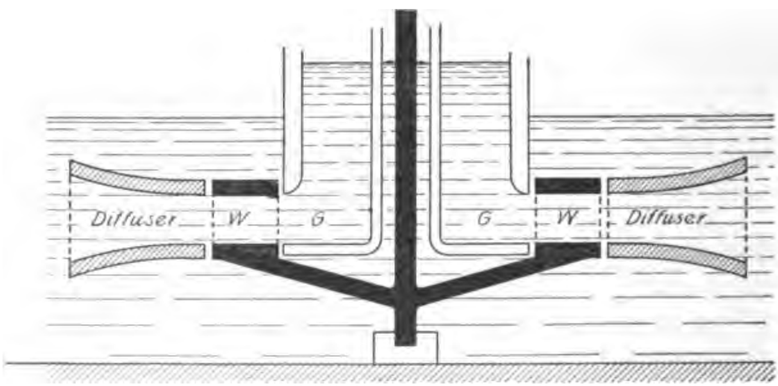


Fig. 78. Illustrating Operation of Diffuser.

as it leaves the runner-buckets of a radial outward-flow turbine, is called a *diffuser*. It usually consists of two fixed conical zones flaring out opposite the water edges of the turbine crowns, giving a *bell-mouthed* or divergent profile to the walls of the passageway at that point of the flow. The object of the diffuser is to prevent part of the loss of energy incurred in the general case, due to the absolute velocity of the escaping water. In this case the absolute velocity of discharge at the extremities of the runner buckets is slowly decreased as the cross-section of the passageway through the diffuser increases; and the discharge water finally passes out at a much lower absolute velocity than that at the runner, with a consequent gain in efficiency (see Fig.78, and nKn of Fig. 71). The efficiency of a reaction turbine is increased by making the exit depth d_1 greater than the entrance depth d, as shown by Equation 109; the stationary diffuser produces the same result.

143. **Draft-Tube.** A wheel set above the level of the tail-race and discharging at an elevation h feet above that level, loses h feet of available head. If the discharge takes place through a (substantially) vertical pipe which is always full of water, and the lower extremity of which is below the level of the tail-water, most of this head becomes effective, since the draft-tube virtually adds this additional head to the static head. In order that the tube may always remain full of water, the internal fluid pressure must be greater than zero; and therefore the draft-tube must not be placed more than about 25 feet (the practical suction limit) above the tail-water level. In practice, these tubes are rarely made longer than about 18 feet, their principal use being to render the turbine easily accessible for examination and repairs, without the necessity of draining the wheel-pit, and without loss of head in setting the turbine above tail-water.

144. The draft-tube is a suction-tube or *water-barometer;* therefore, if v be the velocity of the water in the tube, the *balancing* height of water in feet cannot be greater than $34 - \frac{v^2}{2g}$; and if the vertical length is greater than this, that portion of the tube above this height will contain a vacuum, which involves a loss of head equal to the length of this empty space. The above considerations do not take into account several additional losses, such as that due to entrance, friction, etc., as well as that carried away by the discharge water, the effect of which is to reduce the effective head; it is evident, therefore, that the total draft-head cannot be made available in the turbine. The water discharged at the lower end of the turbine should have a velocity of not less than 2 or 3 feet per second, in order that air bubbles may not rise in the tube and displace the water; and also to carry off any air that may be in the tube on starting the turbine.

145. In order to seal the draft-tube against an inrush of air, it should dip below the level of the tail-water from 6 to 12 inches for short and small draft-tubes, increasing to 20 to 24 inches for long and large tubes. The dip should be sufficient to insure an unbroken seal at all stages of tail-water level. To facilitate the escape of the water from a vertical draft-tube, the lower end should be flared outward,

trumpet-shape. These tubes are usually built-up of lap-riveted steel plates, of circular cross-section, as this form offers great resistance to collapse. They must be thoroughly air-tight, as any leakage of air will destroy the vacuum, with consequent loss of draft-head. When the power-house is built with a concrete substructure, it will often be found advisable to mould the draft-tube directly in the concrete, and so dispense with a metal-tube.

Example 19. A turbine (Fig. 79) receives a uniform supply of 20 cubic feet of water per second from a steel penstock 2 feet in diameter and 2,000

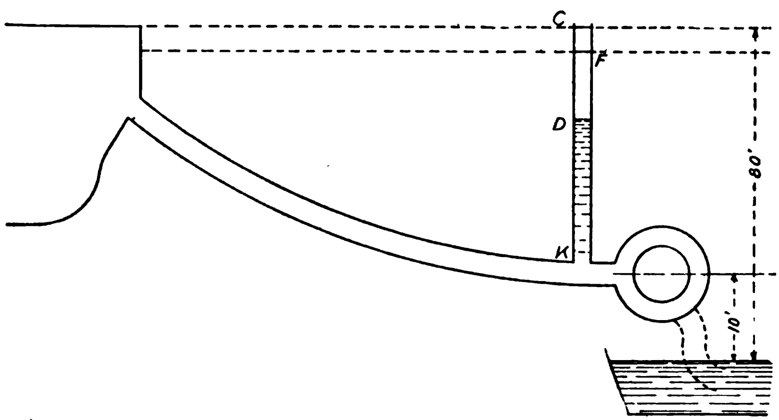


Fig. 79. Diagrammatic Representation of Turbine Installed, Showing Penstock, Head-Race, and Tail-Race.

feet long. The total drop from head-race to tail-race is 80 feet. The turbine is installed 10 feet above tail-water. Discuss its operation.

The mean velocity of flow in the pipe is:

$$v = \frac{q}{\pi r^2} = \frac{20}{3.1416 \times (1)^2} = 6.4 \text{ feet per second.}$$

The head lost in friction (taking the value of c as 100, from $v = c\sqrt{rs} = c\sqrt{\frac{d}{4}\frac{h_f}{l}}$) is:

$$h_f = 15.8 \text{ feet } (FD \text{ in the figure}).$$

The head lost at entrance (taking $m = 0.5$, from $h_e = m\frac{v^2}{2g}$), is:

$$h_e = 0.31 \text{ foot.}$$

The velocity-head of the moving water $(\frac{v^2}{2g})$, is:

$$\frac{v^2}{2g} = 0.68 \text{ foot.}$$

VIEWS OF THE DAM AND WORKS OF THE COMMONWEALTH POWER COMPANY OF JACKSON, MICH., AT PLAINWELL, MICH.

Plant comprises six 50-inch, upright-shaft "Samson" turbines operating under 14-foot head. Upper view, looking downstream; lower view, looking upstream.

Courtesy of James Leffel & Co., Springfield, Ohio.

In the figure, $CF = 0.34 \times 0.68 = 1.02$ feet; and $KD = 70 - (15.8 + 1.02) = 53.18$ feet. Consequently the total effective energy delivered to the turbine is represented by 53.18 feet of pressure and 0.68 foot of kinetic energy = 53.86 feet; but it has an additional gravity-head of 10 feet chargeable to it, which is wasted if the motor discharges into the atmosphere, and practically all utilized if it discharges into a draft-tube. In either case the head h chargeable to the installed motor, and to be used in computations of efficiency, is $53.86 + 10 = 63.86$ feet. At first sight it might appear that the head chargeable to the motor should be 80 feet, the total fall; but from what has preceded, it is evident that $15.8 + 0.34 = 16.14$ feet head is utilized in overcoming penstock resistances, and is not properly chargeable against the motor, either as a separate machine, or as installed. In the former case, the head h should be 53.86 feet; in the latter, 63.86 feet.

Thus, with a diameter of 2 feet, the loss of head in the penstock is $15.8 + 0.34 = 16.14$ feet; and the power lost is $16.14 \times 20 \times 62.5 = 20{,}175$ foot-pounds per second = 36.7 horse-power.

If the turbine discharge into the atmosphere, the head of 10 feet is also lost; this would occasion a further loss of $10 \times 20 \times 62.5 = 12{,}500$ foot-pounds per second, or 22.7 horse-power. With a draft-tube, this last loss would be avoided.

Suppose a 3-foot penstock to be substituted for the 2-foot, the discharge remaining as before.

The mean velocity in this case is 2.84 feet per second.

The head lost in friction (supposing c to be 110 in this case) is:

$$h_f = 1.8 \text{ feet.}$$

The head lost at entrance is:

$$h_e = 0.06 \text{ foot}$$

The velocity-head is:

$$\frac{v^2}{2g} = 0.12 \text{ foot.}$$

Thus the loss of head in the penstock is $1.8 + 0.06 = 1.86$ feet, which represents a loss of $1.86 \times 20 \times 62.5 = 2{,}325$ foot-pounds per second, or 4.2 horse-power.

146. **Flaring or Conical Draft-Tubes.** When draft-tubes are used, they should be of the flaring type, so as to change the speed of the water gradually; for when the tube is of the same cross-sectional area

for its entire length, much of the energy which should be made available owing to the lower velocity of discharge, is lost in shock when the water issuing from the runner at a relatively high speed strikes the water in the draft-tube moving with a lower speed. It has already been shown that the head corresponding to the absolute velocity with which the water leaves a wheel is in the general case entirely lost; but when the discharge takes place into a draft-tube of conical or flaring shape (the cross-sectional area increasing gradually from the turbine to the tail-water), the velocity of the water is gradually reduced below that of discharge at the extremities of the runner-buckets, so that the amount lost to the turbine is less, as in the case of the diffuser. Since the reduction of velocity should be gradual, the change in section must be gradual; and it is therefore sometimes advantageous to increase the total length of the draft-tube without increasing the draft-head, by curving or inclining the tube; this procedure may sometimes be adopted to save tail-race excavation.

147. Pulsation or oscillation of the water is frequently noticed, especially in connection with high draft-head, particularly if the turbine is subject to sudden changes of load, and is controlled by a quick-acting governor. This is extremely detrimental to good speed regulation; and in extreme cases, may even seriously injure the motor. Conical draft-tubes are not subject to these pulsations to the same extent as cylindrical tubes; they are also more efficient in expelling air when the turbines are started, and are better able to retain the draft-head when the motors are running with light loads. A draft-tube with diffuser is shown in Fig. 71.

148. The use of draft-tubes in recent years has marked a notable advance in turbine practice; it has made practicable the employment of turbines on horizontal shafts, the turbine being connected to the tube by means of a draft-tee or elbow.

Draft-tubes may be employed with greater or less convenience in connection with any class of turbine, though the Jonval and Francis types, with their modifications, are best adapted to their use. Even impulse wheels of the Pelton and other types have been fitted with draft-tubes; but in such cases the upper surface of the water in the tube must be maintained automatically at an elevation just below the lowest point reached by the revolving buckets, which thus move in rarified air within a strong casing forming the top of the draft-tube.

This gives the advantage, not only of added draft-head, as just described, but also of decreased air friction for the running parts. In such cases, that portion of the draft-head is necessarily lost which is represented by the distance between the water surface in the draft-tube and the center of the nozzle discharge opening; or, in case two or more nozzles are used, it is the mean vertical distance. In the case of an impulse turbine fitted with a draft-tube in the manner described above, the head lost is the vertical distance, or the mean vertical distance, from the water level in the draft-tube to the center of the guide-bucket discharge openings.

149. The principle of the air-admission valve for automatically maintaining a constant height of "hanging column" in the draft-tube, is very simple. A vertical pipe is placed at the side of the turbine, its upper end connected to the turbine case, and its lower end to the draft-tube, just as a water-gauge is connected to a boiler. In this pipe floats a copper ball connected with an air-admission valve in the turbine case; when the water level rises, the float rises with it and opens the valve, admitting air, which causes the water level and the copper ball to fall, thus closing the valve.

150. In European practice, gates are occasionally employed to close the lower end of draft-tubes for the purpose of filling them with water before starting the turbines (as in *priming* pumps), or for decreasing the feed when running under light load, thus reducing the tendency, under the above conditions, of the water to crowd to one side, or to drop through the draft-tube without expelling the air and producing suction, with consequent loss of draft-head. Curved or inclined draft-tubes are more likely to suffer in this way than vertical tubes; and large and long tubes more than small and short. But such gates are costly and cumbersome for large turbines, and their employment in American practice to any great extent is improbable.

151. **Fall-Increaser.** Under the name of *Fall-Increaser*, Mr. Clemens Herschel has patented an apparatus designed to increase the fall acting on hydraulic turbines by the use of freshet water otherwise going to waste. In this way the normal output of power of a hydraulic power-plant may be maintained at a constant normal quantity in spite of *back-water*, and for all those days in the year when there is water enough flowing in the river to produce so much as the normal output. In some cases of low fall, when there is an abundance of

water to be had, also for certain cases of tide-mills, the fall acting on the turbines may be increased *throughout the year*, over and above the natural fall, so as to produce a *greater speed*, and thus render the location more fit for generating electricity; while at the same time it will diminish the cost of the plant—generators, turbines, and building —*per horse-power* produced.

The fall-increaser, shown underneath the turbines, and operating at a time when the *direct* discharge of the turbines has been shut off,

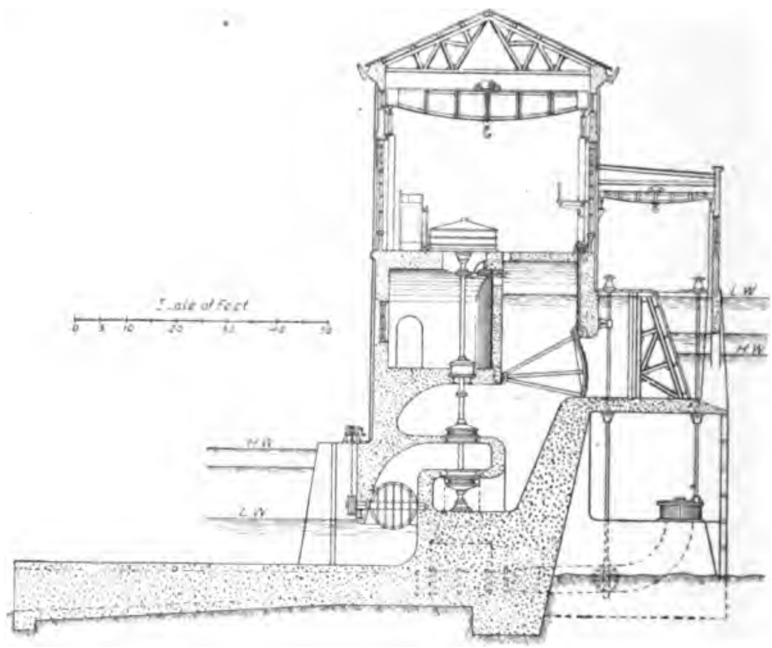


Fig. 80. Application of a Fall-Increaser to a Power Plant.

exhausts the turbine discharge from the vacuum-box, and also produces a partial vacuum in this vacuum box, thus increasing the fall that would otherwise act upon the turbines. Fig. 80 shows diagrammatically the application of a fall-increaser to a power plant.

152. **Regulating Gates.** For many industrial purposes the power required of a turbine is variable, as when the number of machines operated in a factory is not constant, or when running dynamos to supply electric current to meet the fluctuating demands of lighting or of transportation service; and since the speed of the tur-

bine should be fairly constant, the variation in power must be provided by varying the quantity of water supplied to the motor. Under these conditions, it is evident that the average position of the turbine gate is not that of full gate, and the problem of design to secure high efficiency at part gate, and also at full gate, is not a simple one. To regulate the quantity of water supplied to the motor without serious loss of efficiency, is more difficult in the case of reaction turbines than of impulse turbines, for, with the latter type, it is only necessary to vary the cross-section of the guide-passages at the place of discharge, which can be done with relatively little loss of energy. With a reaction turbine, however, since the wheel passages are always filled, a throttling of the stream at any point causes not only a contraction at that point, but a subsequent expansion, with a resulting loss of energy. Perhaps the most wasteful methods for regulating the discharge are by *throttling* the flow by means of the gate in the penstock or supply-pipe, or in the draft-tube; or by the use of a cylindrical gate encircling the lower end of the draft-tube; in such cases, losses invariably result from subsequent enlargement of section, or from impact. The plain cylindrical gate moving axially is open to the same objection, unless the turbine channels are provided with partitions, as already described, or have an upper crown which curves downward.

153. Theoretically the most perfect regulator for a radial-flow turbine is the device due to Nagel and Kaemp, in which the *roofs* of the guide-passages and the crown of the turbine are together movable, so that, in consequence of the crown and roof being always even and opposite, sudden enlargement at entrance is avoided for all positions of regulation. The design is, however, expensive, and involves difficulties of a practical nature.

154. The regulation of the Jonval or parallel-flow (axial-flow) turbine is usually accomplished by sliding (*register*) plates or swinging flaps for closing the guide-passages. It is found that the entire closure of a number of the guide-passages, instead of the partial closure of all, is conducive to higher efficiency, since in the former case the absolute velocity of entrance has the same value as when all guide-passages are open.

155. Of the great variety of gate arrangements that have been tried, only the following three have come into general use:

1. *Cylinder Gate.* The cylinder gate, moving in an axial direc-

tion, as used by Mr. Francis with his early turbines, is now by far the most extensively employed gate. It may be placed on the inlet or on the outlet circumference of the runner, and it regulates by cutting off the supply of water from the upper sections of the bucket orifices. To partially prevent the loss of energy incidental to the use of this regulator when operated at "part-gate," the width of the guide and runner-buckets (the distance between the crowns) may be divided into two or more spaces by additional crowns or partitions, thus virtually forming two or more turbines which are regulated by one common cylinder gate. Thus, for example, a triple or three-story turbine, when working at one-third or two-thirds gate-opening, has two or one turbine working at full gate with full-gate efficiency; while the

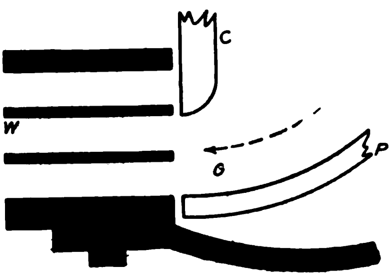


Fig. 81. Turbine Divided into Three Stories.

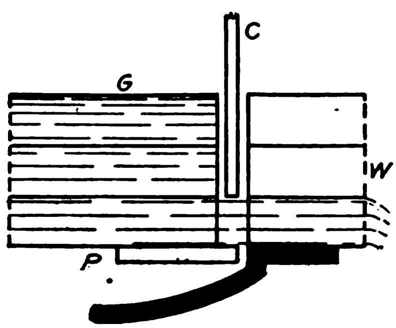


Fig. 82. Subdivision of Guide-Passage.

one turbine or two turbines remaining are entirely shut off. The cylinder gate in nearly all cases works between the guides and runner. Thus Fourneyron divided his turbine (Fig. 81) into three *stories* by means of horizontal partitions; and the guide-passages may also be divided to correspond, as in Fig. 82.

As examples of recent design illustrating this principle, may be mentioned a 700-horse-power horizontal-inflow turbine of Swiss design, with a width of bucket of about 30 inches, divided into five stories, all regulated by a single gate. The 5,000-horse-power double (or twin) turbines of the Niagara Falls Power Company in Power-House No. 1, are examples of three-story outflow turbines, with cylinder gates on the discharge (outer) side of the runners. The guide-passages are also divided into three stories.

2. *Register Gate.* The register gate (so named from its sim-

ilarity to a common hot-air register) may be of the plate or the cylindrical type, according to the kind of turbine to which it is applied. The latter type consists of a rotating cylinder having slots which correspond with the outlet openings of the guide-buckets. The axis of the cylinder is coincident with the axis of the turbine shaft; its motion is circumferential, so that it cuts off the supply from the sides of the guide-passages. When applied to parallel-flow turbines, the plate type of register gate must be used. Register gates are sometimes placed outside, and sometimes inside the guide-ring. This form of gate is no longer widely used. Many ingenious forms of this type of gate have been designed; in one form, a part of each of the vanes which form the guide-buckets is separate from the rest of the vane, being attached at each crown to a movable ring, so that by rotating these rings by a suitable device, the size of each of the clear openings between the vanes can be altered simultaneously in accordance with fluctuations in the load on the turbine. The movable part of the vanes may be either at the entrance or at the discharge side of the guide-buckets, though the former plan is now rarely used, as the shape of the bucket is thereby too much distorted when the gate is partly closed; whereas, with the latter arrangement, the shape of the bucket is much better maintained.

3. *Wicket- or Pivot-Gate.* In this type of gate, the whole guide-vane swings on pivots so located as to balance the vane as nearly as possible in every position. Here also the vanes move, and thus alter the size of the discharge openings of the buckets, simultaneously. Such gates maintain the correct shape of the guide-buckets at part gate, better than any other gate.

156. **The Case.** This term is usually applied to the fixed parts sustaining the guides and gate or gates, which the maker furnishes with the wheel, being regarded as part of the same. It includes the plate or disc which supports the guides, and a plate which relieves the wheel of the pressure of the water; in some types these two functions are performed by the same plate. The case usually carries the mechanism for operating the gate, and often the step on which the shaft runs. The term is sometimes used to signify the iron casing or vessel inclosing the entire mechanism, into which the water passes by connection with the penstock. *Closed turbine-chambers* would perhaps better express the meaning in the latter case.

157. **Turbine-Chamber, or Flume.** In order that the water may act upon a motor, the latter is placed in a chamber communicating with the upper level by a penstock or head-race, and with the lower level by a tail-race (with or without the intervention of a draft-tube), into which it discharges after passing through the motor. This chamber is frequently called a *flume*; but it is preferable to call such a contrivance an *open* or *closed turbine-chamber* (or the latter, a *turbine casing*), as the case may be, and to restrict the use of the term *flume* to mean a water conductor carrying water not under pressure.

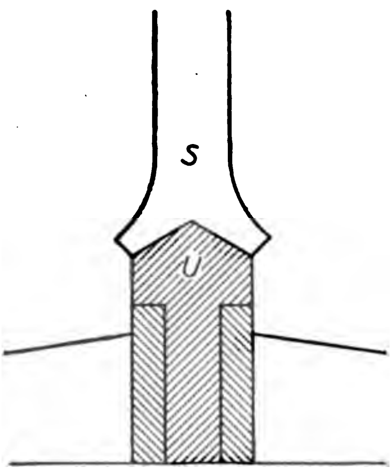


Fig. 83. Vertical Section of Lower Part of a Risdon Wheel-Shaft and Wooden Step-Bearing.

The turbine-chamber may be built of wood, iron, stone, or concrete, or a combination of these, the choice depending principally upon the head.

The best and simplest arrangement for a single or multiple horizontal turbine with draft-tube or tubes, is an open turbine-chamber, built of wood (for temporary purposes), masonry, concrete, or concrete and steel, forming a direct continuation or branch of the head-race or forebay; and this plan has been adopted in recent years for many important power-plants.

Open turbine-chambers have three advantages over closed turbine casings or chambers—namely, the friction of the water flowing to the motor is reduced to a minimum; the mechanism is readily accessible; and the arrangement is very convenient for speed regulation. The minimum depth of water above the highest point of the entrance rim of the guide-bucket for turbines operating with low draft-head should be 3.5 to 4.5 feet for open chambers, to avoid the formation of funnels, and the suction of air into the turbine.

158. **Step, Suspension, and Thrust Bearings.** Most vertical shafts of turbines run on wooden step-bearings, the block forming this bearing being sometimes arranged so as to be free to revolve also in its socket, so that if the upper surface becomes heated to such an

extent as to greatly increase its frictional resistance, the block will turn on its lower surface until the upper surface has cooled. Lignum vitæ, maple, and oak are the woods commonly employed for step-bearings; they are thoroughly dried, and then boiled to impregnation in linseed oil; they become, therefore, in a measure, self-lubricating. Fig 83 shows a vertical section of the lower part of a Risdon wheel-shaft (S) and wooden step-bearing (U). In some cases the shaft and attachments are suspended from a collar-bearing above, as in Fig. 87.

159. **Thrust or Balancing Piston.** This arrangement is principally used for horizontal inward- and outward-flow turbines, and takes the thrust both of the action of the water and the weight of the rotating parts. Fig. 84 (Niagara Falls Power Company, Power-House No. 1) represents an example of twin-flow wheels on the same shaft. The weight of the dynamo, shaft, and turbine (about 70 tons) is balanced, when the wheels are in motion, by the upward pressure of the water in the wheel-case on a *balancing* piston or disc B, placed above the upper wheel and rigidly attached to the shaft. The upper disc containing the guides is perforated so that the water pressure existing in the penstock can be transmitted directly through to the lower side of the balancing piston, while the upper side is open to atmospheric pressure. The lower disc is not perforated; and the weight of the water upon it is carried by inclined rods upward

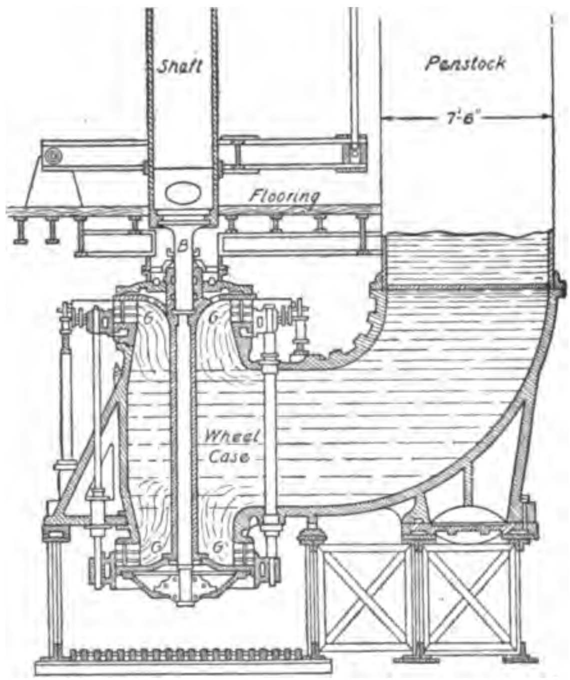


Fig. 84. Thrust Piston with Twin-Flow Wheels Mounted on Same Shaft. Niagara Falls Power Co., Power-House No. 1.

to the wheel-case, which, together with the penstock, is supported upon several girders. At the upper end of the shaft is a special thrust-bearing designed to receive the excess of vertical pressure, which may act either upward or downward, under different conditions of power and speed.

Fig. 85 shows diagrammatically the device applied to their single-inflow turbines by the Niagara Falls Power Company, in Power-House No. 2. The water from the penstock fills the annular chamber A under nearly hydrostatic pressure; passes through the guide-passages at G; and enters the wheel channels at C under reduced pressure and at high velocity. The revolving turbines, shaft, and attachments are shown in black. The water, leaving the turbine-channels W, enters the space D with low absolute velocity and low pressure. At the lower end of the shaft, while lateral support is provided by the step-bearing, a great lifting force is furnished by the admission of water directly from the head-race by an independent pipe under the full head-race pressure to the space UU on the under side of the conical shell or balancing disc VV, which is keyed upon the shaft and revolves with it. The pressure on the upper surface of the piston is small, being that of the water in the upper end of the draft-tube. In this way the larger part of the weight of the wheel, shaft, and armature of the electric generator is supported by fluid friction. The diameter of the balancing disc is 4.9 feet; the weight of the revolving mass is 71 tons, of which 66 tons are supported by the upward pressure of the disc, leaving five tons to be sustained by a *suspension* or collar bearing at the upper end of the shaft.

160. **Thrust-Chamber.** This arrangement is principally used to take the end thrust of vertical inflow turbines, and consists of an annular chamber, formed by the cast-iron turbine case, and open towards the runner, which revolves in front of it. The water pressure in the chamber is supplied by a pipe connected with the penstock, and provided with a valve for regulating the pressure. By means of this valve, the end thrust can be so regulated that the shaft will press against the step-bearing with just enough force to prevent any end motion.

According to Thurso, the use of wood with water lubrication for bearings and steps located under water, has been practically abandoned by European manufacturers; and metal bearings and steps, with forced oil lubrication, are employed instead, using a pressure and return pipe for circulating the oil. Frizell shows a type of such oil step-bearing (Fig. 86). The shaft passes through a stuffing-box, and rests on the revolving plate *a*; the surfaces of contact between *a* and *b* are dressed to an exact fit, and wear keeps them in that condition. Oil for lubrication passes through the pipe *e*, being forced in under pressure by a pump; it fills the space *f*, and exerts a lifting

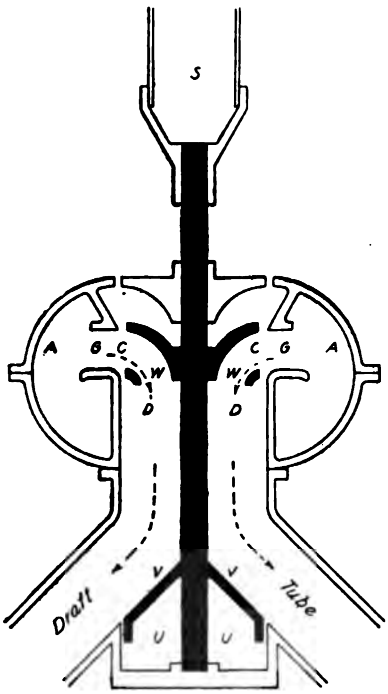


Fig. 85. Thrust Piston Device for Single-Inflow Turbine. Niagara Falls Power Co., Power-House No. 2.

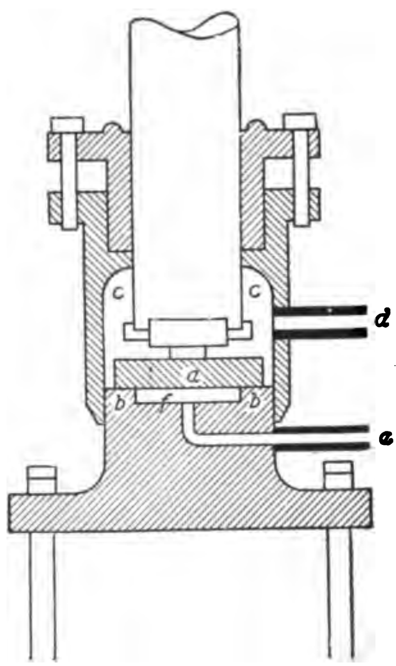


Fig. 86. Oil Step-Bearing.

pressure on the plate *a*, nearly equal to the weight of the shaft and its attachments, so that the contact surfaces sustain very moderate friction. The oil passes between these surfaces into the space *c*, and therefore the upward pressure on plate *a* would be neutralized, were it not that another pipe *d*, communicating with this space, conveys the oil back to the tank from which the pump draws its supply.

161. A metal collar thrust-bearing is shown in Fig. 87. Such metal bearings should never be located in the water, but are usually placed on the end of the shaft opposite to that from which the power

of the turbine is taken off. Both the straight and the collar bearings of the main turbine shaft should be adjustable, and should be lined with bronze as a base, and the bronze in turn lined with an anti-friction metal, or babbitt, well hammered and bored.

162. In double turbines working on the same shaft, the end thrusts balance and neutralize each other, provided both work at the same gate opening. Thurso recommends the employment of thrust-chambers for single turbines working under a high head, say several hundred feet; and in case the runners are of such size or shape as to preclude the use of such chambers, the thrust-piston should be used instead, placing the runner at one end of the case, and the piston at the other.

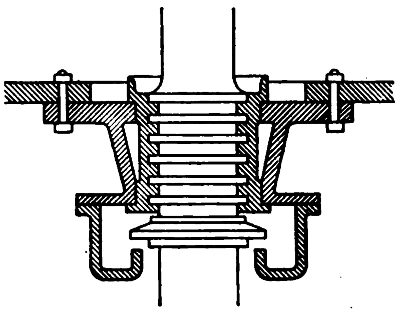


Fig. 87. Metal Collar Thrust-Bearing.

Turbines having thrust-chambers or pistons, or double turbines so mounted that the two end thrusts balance, should nevertheless be provided with a small collar-bearing, to take care of unavoidable variations in the end thrusts.

163. **Stop-Valves.** It is a matter of practical convenience to have a separate stop-valve in the supply-pipe near the turbine, in order that the water may be quickly shut off and the turbine stopped. They should also be placed in convenient locations, dividing the pipe-line into sections, so that single sections may be shut off for inspection, repairs, renewals, etc.

The turbine may be shut down by closing the regulating gates; but this method will not permit the turbine to be taken apart or repaired; and under high head the regulating gates are seldom so tight that their closure will bring the wheel to a standstill. Where several units are supplied by the same penstock, stop-valves are a necessity; for, otherwise, the stoppage of one unit for repairs by means of closing down the head-gate would involve the stoppage of all the turbines connected with that penstock. In American practice, gate-valves are most commonly used for this purpose. The small sizes are fitted with screw-spindles to be operated by hand; but for the larger sizes some kind of motor must be provided, to furnish sufficient power for moving the heavy gates. Electric, hydraulic, and pneumatic power have all been used. In the case of hydraulic power, where the head employed is high enough, the pressure water may be taken directly from the penstock; but for lower heads a pressure-pump and weighted accumulator are required. Sometimes a by-pass is employed in connection with a gate-valve for the purpose of partially equalizing or neutralizing the pressure.

164. **Air-Valves.** At summits of a pipe-line and near stop-valves, air-valves should be placed for the purpose of permitting the escape of air in filling, the entrance of air on emptying, and occasion-

Fig. 88.

Fig. 89.

"Victor" High-Pressure Runners.
Fig. 88—For Heads of 100 to 2,000 feet; Fig. 89 –Runner with Scoop-Shaped Vanes.

ally the escape of air which may gradually accumulate at summits. They are usually designed to operate automatically.

165. **Blow-Off Valves.** These should be placed at all depressions of a pipe-line, for the purpose of cleaning out or of emptying sections of the line.

166. **Automatic Stop-Valves.** These should be placed at critical points of a line, so that, in case of accident to the pipe, the valves will gradually close, and thus prevent the loss of water and possible damage to property.

167. **Illustrations.** Some illustrations of turbine parts, accessories, and details will now be presented, with very brief explanation when necessary.

Fig. 88 shows a *Victor* high-pressure runner, intended for heads of from 100 to 2,000 feet. Fig. 89 shows another type of Victor run-

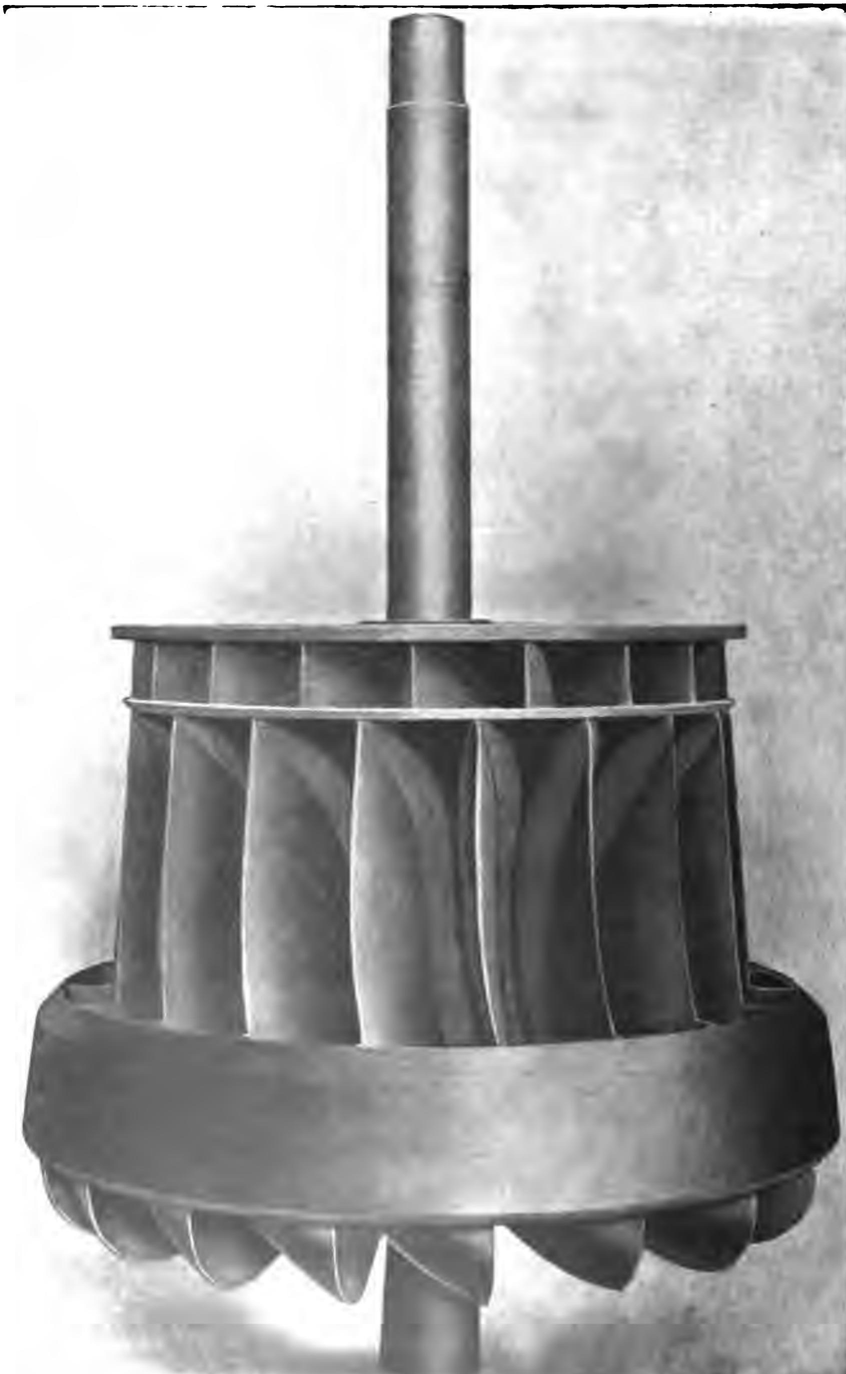


Fig. 90. Upright-Shaft "Samson" Runner.
Courtesy of James Leffel & Co., Springfield, Ohio.

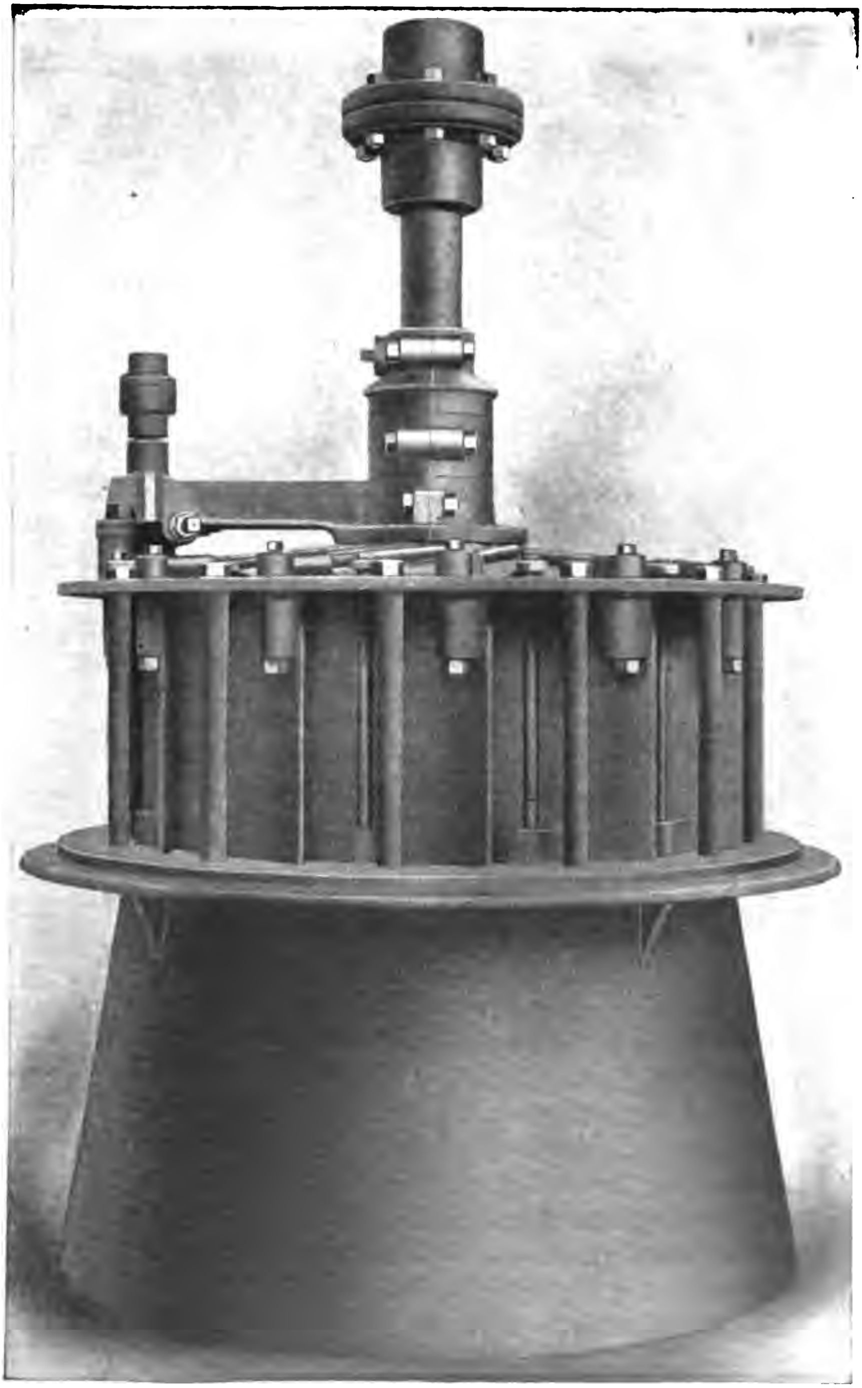


Fig. 91. Upright "Samson" Turbine Complete.
Courtesy of James Leffel & Co., Springfield, Ohio

ner, with characteristic scoop-shaped vanes. These wheels are cast in one piece, of cast iron or bronze. Fig. 90, the *Samson* runner, made by James Leffel & Company, Springfield, Ohio, is a *double* wheel, the upper portion of the wheel-passages being partitioned off by a diaphragm. Each wheel or set of buckets receives its separate quantity of water from one and the same set of guides, each portion of the water, however, acting on its own buckets. Fig. 91 shows the turbine complete, in its case. The small shaft to the left is for the purpose of operating the balanced wicket-gates. The horizontal projecting rim or flange serves the purpose of supporting the mechanism upon the floor of the turbine-chamber; the conical shell below the rim is a short, flaring draft-tube. Fig. 92 shows another type of Leffel turbine without and within a globe casing. The shaft and gearing for operating the regulating gates are clearly indicated. The penstock is supposed to be bolted to the flange of the casing. Fig. 93 shows a simple type of wheel with a quarter-turn draft-tube through which the water discharges, first horizontally, and then vertically downward. The shaft and gate-rods extend through stuffing-boxes in one side of the turbine-chamber (not shown); and the horizontal iron base-plate is frequently set directly upon its floor, usually with a draft-tube extending below tail-water. Fig. 94 represents two complete wheels on a horizontal shaft, one at each end of the cylindrical steel-plate case. Both wheels discharge horizontally toward each other, the water then passing downward through the central discharge pipe into a draft-tube. The large iron base-plate is set upon the floor of the turbine chamber, and the shaft may be extended in one or both directions, passing through stuffing-boxes in the sides of the turbine-chamber. The thin horizontal shaft on top of the cylinder has rigidly fastened to it two pinions, so that, by the turning of this shaft, the regulation of both gates is effected simultaneously through the action of the rods.

The types represented in Figs. 93 and 94 are supposed to be placed on the floor of a simple open or decked turbine-chamber, of wood or other simple construction. For high heads, particularly when economy in space is necessary, iron casings are employed. Fig. 95 shows the simplest style of horizontal-shaft wheel in an iron casing. The pulley shown is intended for rope transmission; but a flat pulley for belting may be substituted. The governing and controlling de-

RIVETING DECK BEAMS ON THE STEEL FLUME OF THE ONTARIO POWER COMPANY, NIAGARA FALLS, ONT.

Flume 18 feet in diameter, afterwards inclosed in a sheathing of concrete.

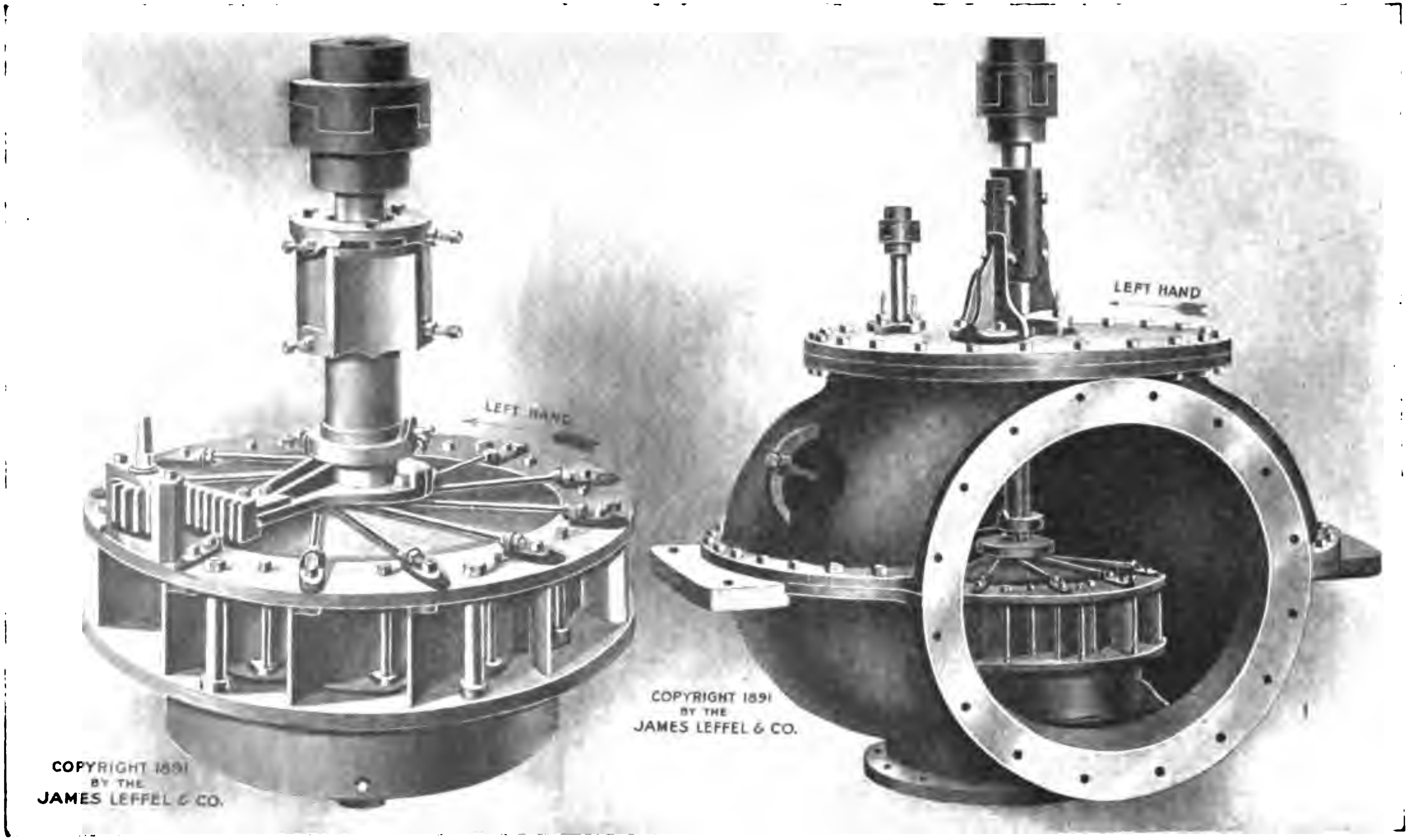


Fig. 92. Leffel Turbine without and within a Globe Casing.

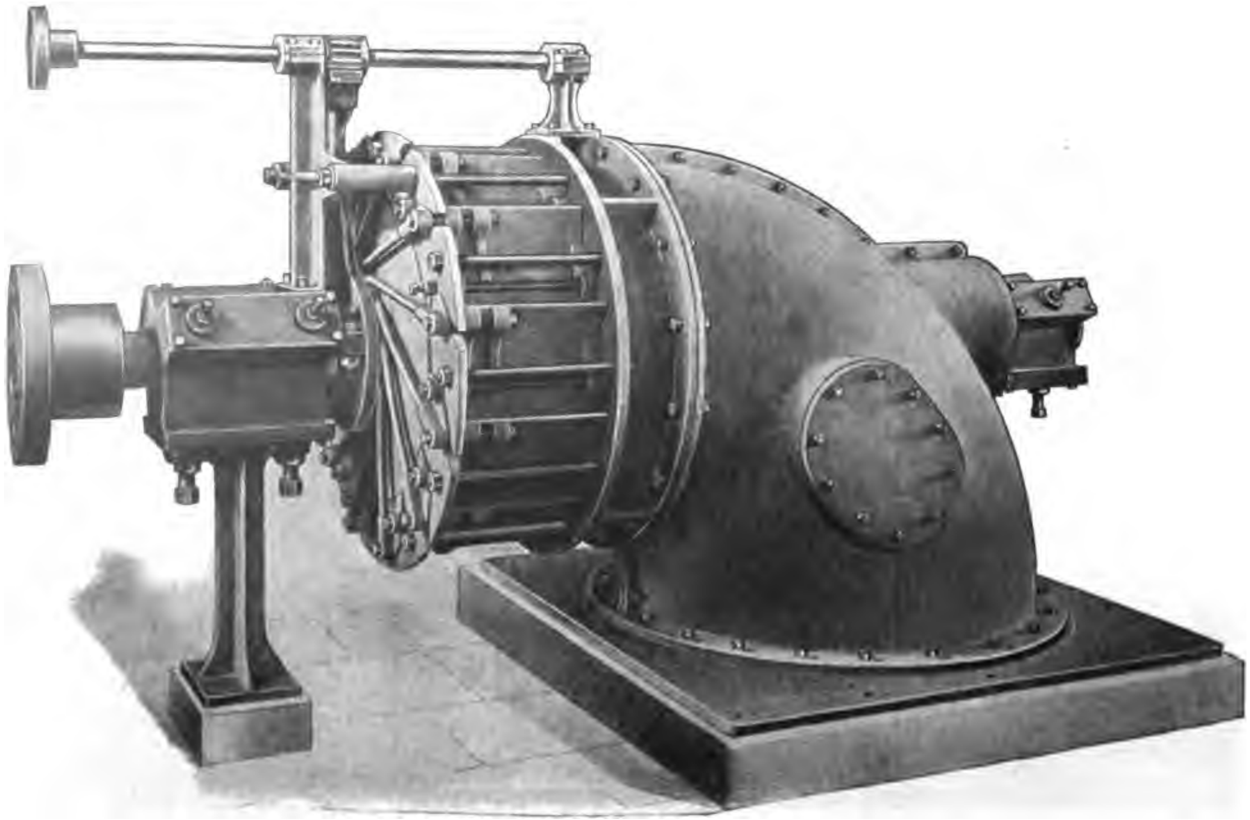


Fig. 93. Single-Discharge, Horizontal-Shaft "Samson" Turbine for Open Penstock, with Quarter-Turn Draft-Tube.
Courtesy of James Leffel & Co., Springfield, Ohio

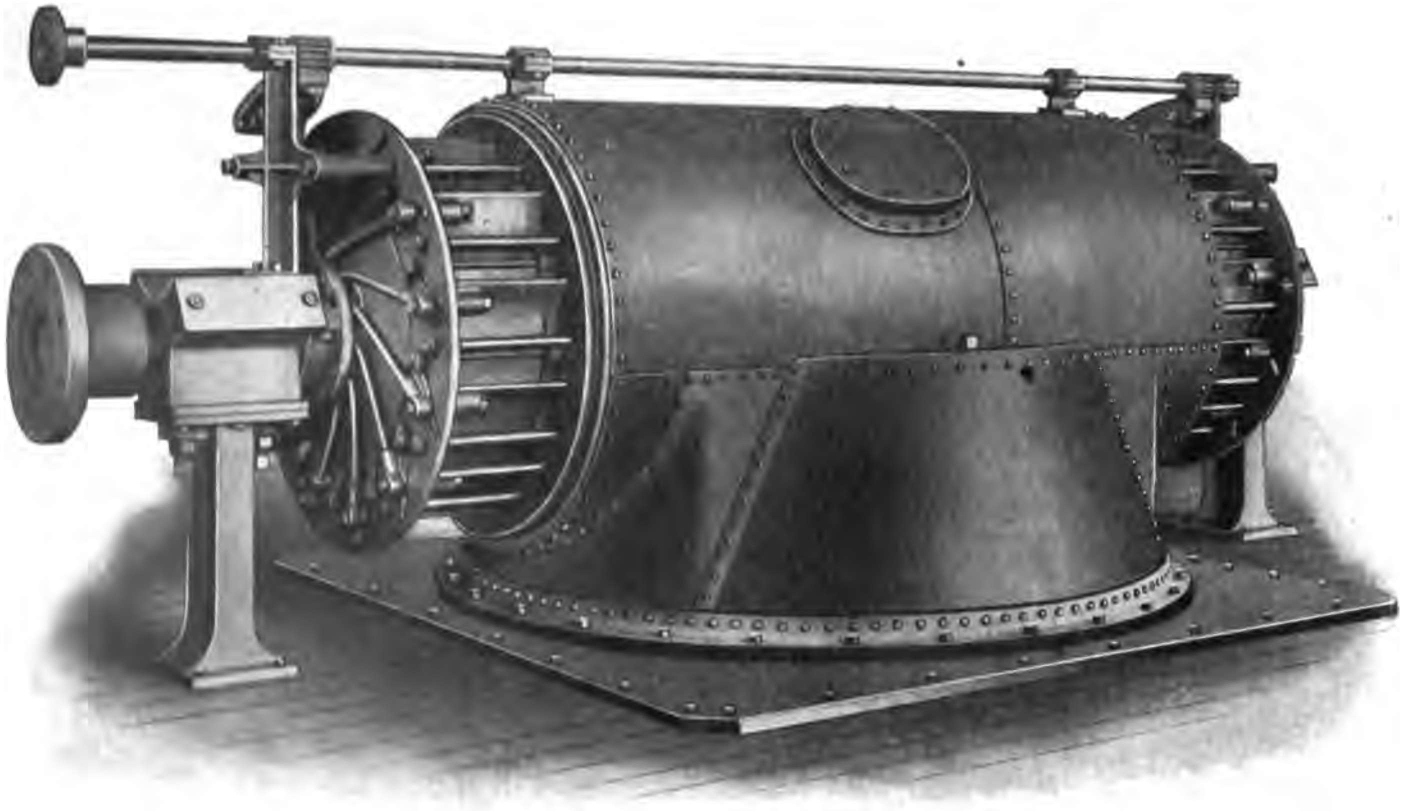


Fig. 94. Twin Center-Discharge, Horizontal-Shaft "Samsons" for Open Penstock.

Courtesy of James Leffel & Co., Springfield, Ohio.

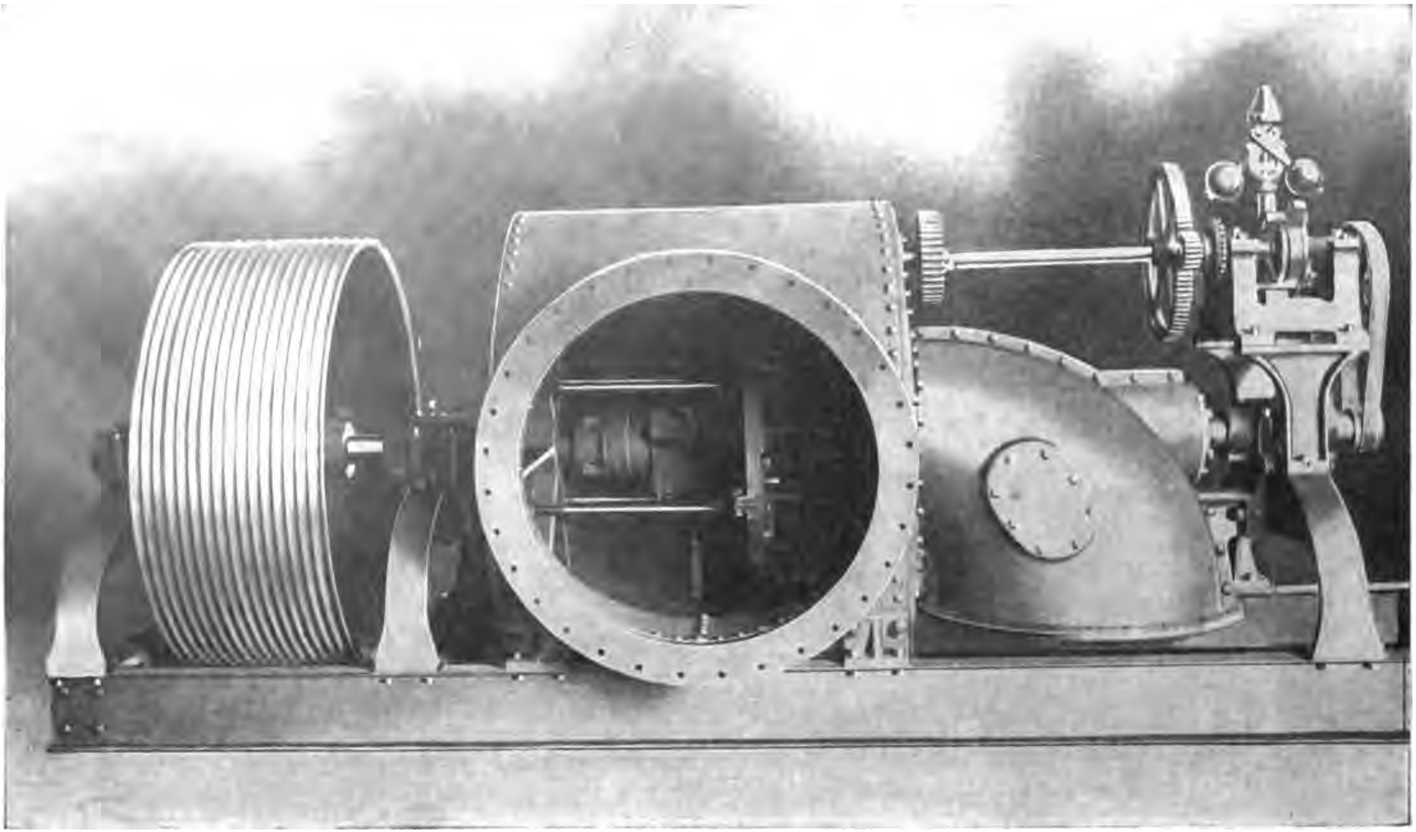


Fig. 95. Single-Discharge, Horizontal-Shaft "Samson" Turbine with Casing.
Courtesy of James Leffel & Co., Springfield, Ohio.

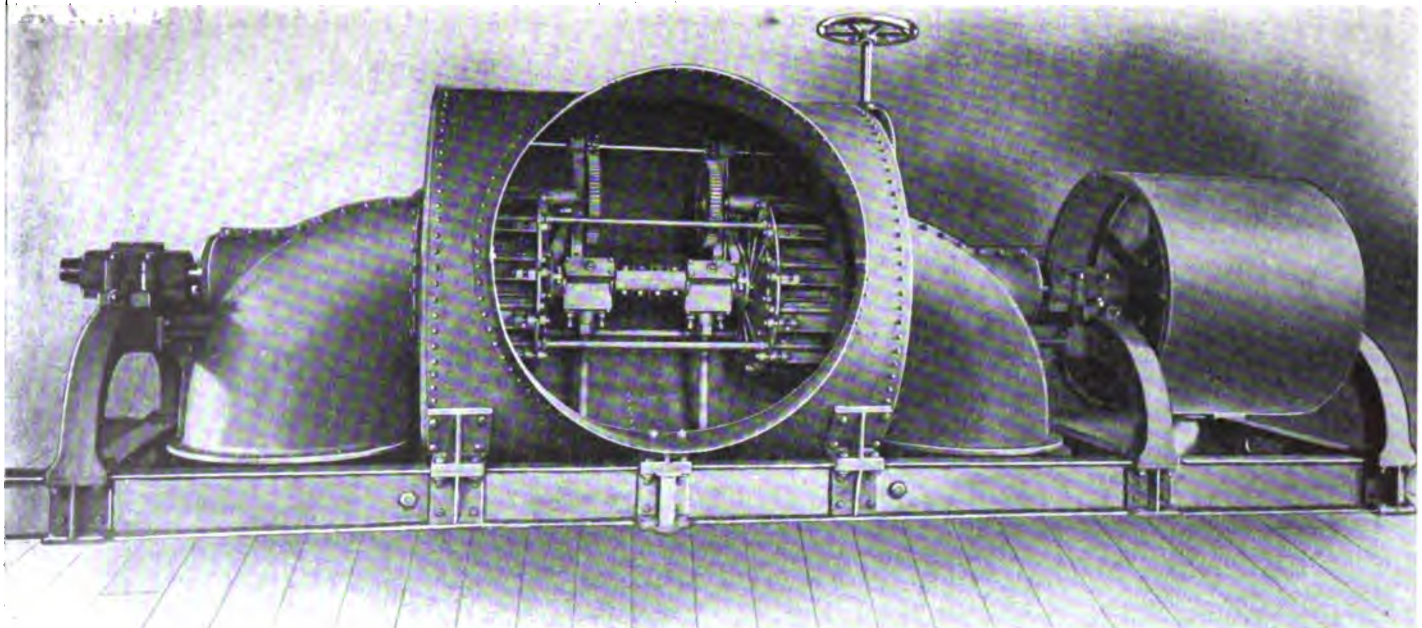


Fig. 96. Double-Discharge, Horizontal-Shaft "Samson" Turbine with Casing.
Courtesy of James Leffel & Co., Springfield, Ohio.

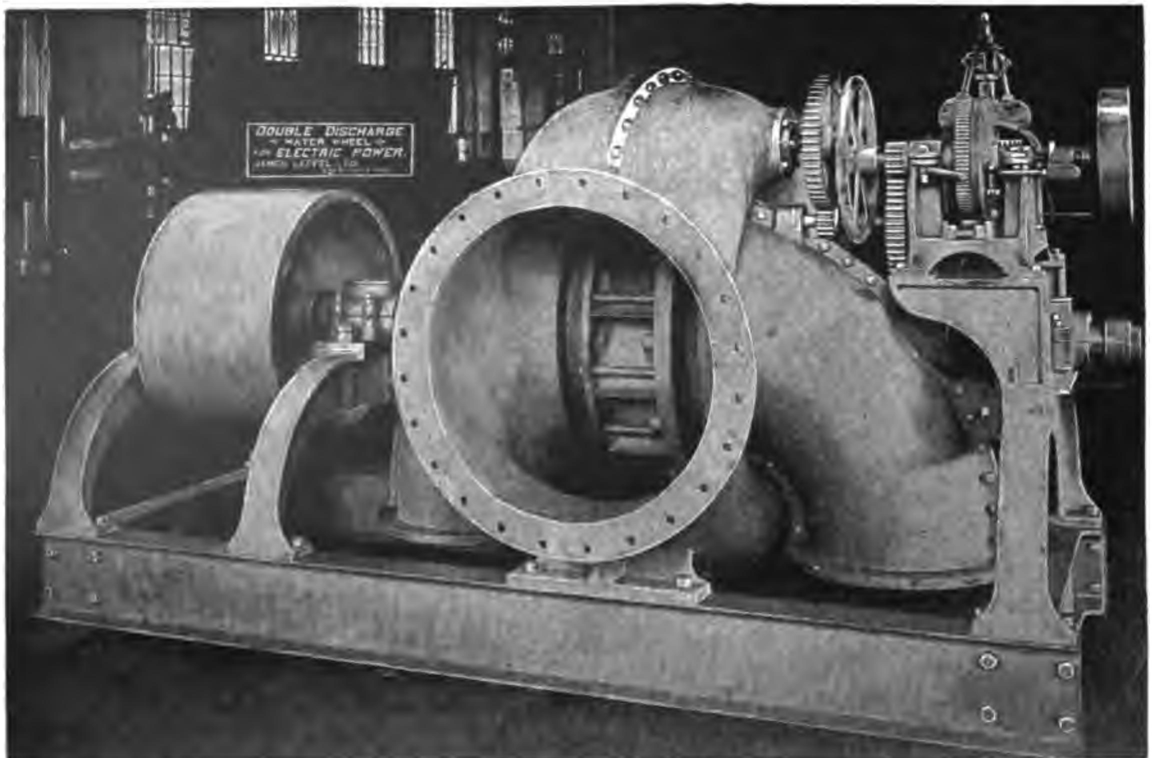


Fig. 97. Double-Discharge, Horizontal-Shaft Turbine with Globe Casing.
Courtesy of James Leffel & Co., Springfield, Ohio.

vices are seen on the right. Fig. 96 shows a casing containing two wheels on the same horizontal shaft, discharging in opposite directions through the curved elbows into the wheel-pit or tail-race.* The *Samson* turbines are of the horizontal-shaft type, usually having *one* runner built with two similar sets of buckets, taking the water equally divided from one set of guides, and discharging in opposite directions. Fig. 97 represents a "Niagara" type of turbine, the illustration showing it to be strongly built and very compact.

The Risdon-Alcott Turbine Company, of Mount Holly, New Jersey, manufactures a line of turbines of distinctive character. In Figs. 98, 99, and 100, are seen three types of these runners, showing very decidedly the peculiar spoon- or scoop-shaped vanes; they are cast in one piece, of cast iron or bronze. The top of the wheel is formed by a bell-shaped crown, which extends to the inner bottom of the buckets and forms their inner boundary. A band cast around the lower, outer portion of the buckets forms their outer boundary; the warped surfaces forming the vanes connect bell and band. In Fig. 100, the band is represented as transparent for the sake of clearness; and in Fig. 98, part of the band is supposed to have been removed for the same purpose. Three styles of gate are used with these wheels —the cylinder, the register, and the hinged. In Fig. 101 is seen an outside view of the wheel-case, etc.; *B-B* are the guide-vanes rigidly attached to the plate *R*; *C* is the vertical cylinder gate, to which are rigidly attached the horizontal projections *D-D*, fitting between the guide-vanes. These projections necessarily move with the cylinder in its vertical travel, and thus form in effect *movable roofs* for the guide-buckets. *S* is a short discharge tube, which may dip a few inches below tail-water, or to which may be attached a draft-tube; *V* is the turbine-shaft; and *W* is the shaft which operates the gate *C* through the intervention of the gearing, as shown; the supporting rim *R* rests on the floor of the turbine-chamber.

Figs. 102 and 103 show two turbines fitted with *outside* and *inside* register gates. The register gate consists essentially of a cylinder containing slots arranged parallel to the shaft; in one position of its circumferential motion, the passage of water is entirely cut off by reason of the slots coming opposite the guide-vanes, which are of consid-

* The types illustrated in Figs. 90—96 are manufactured by the James Leffel & Company, Springfield, O., who also manufacture the *Niagara* type of *Samson* turbines.

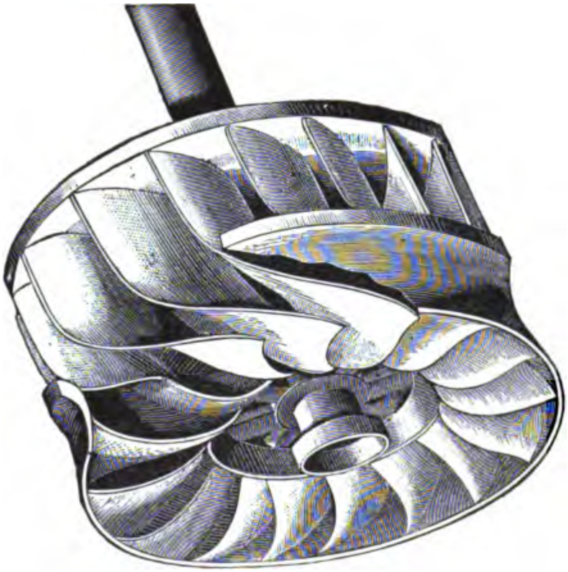


Fig. 98. Revolving Part of Risdon-Alcott Standard Turbine.

erable thickness; while a comparatively small movement from this position will leave the guide-passages fully open.

Fig. 104 shows two Risdon-Alcott turbines mounted on the same horizontal shaft, fitted with balanced *hinged* gates, discharging centrally into a common draft-tube. The various casings, etc., and different methods of mounting described in connection with the Leffel wheels, may also be found in the catalogues of the

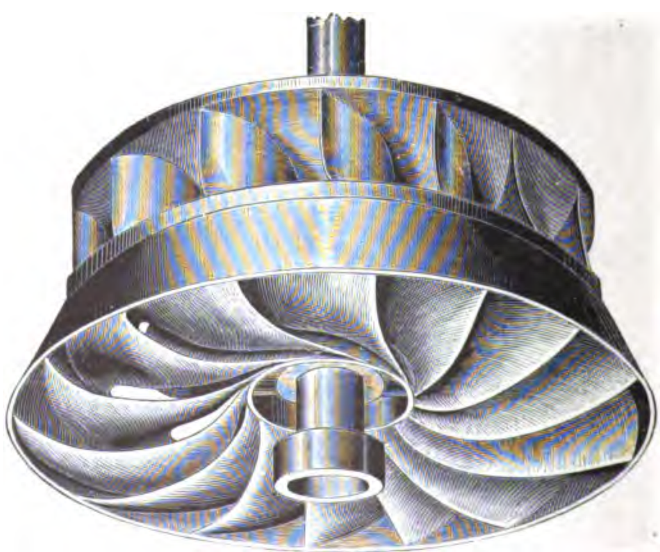


Fig. 99. Risdon-Alcott Double-Capacity Wheel.

Risdon-Alcott Turbine Company, and therefore need not be further illustrated.

Fig. 105 represents a vertical-shaft turbine with wicket-gate, built by the S. Morgan Smith Co., of York, Pa.; it is shown set in a

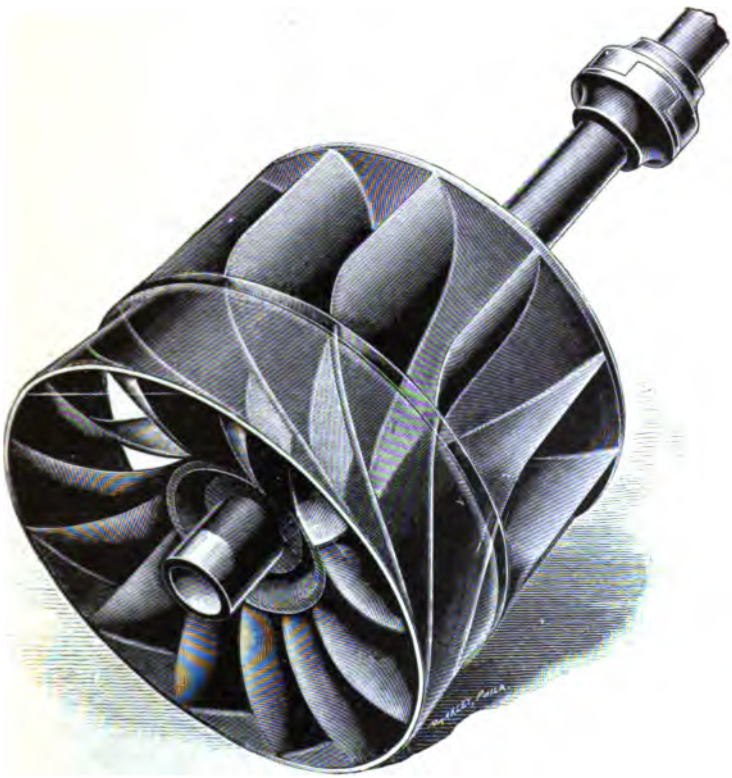


Fig. 100. Alcott Special Runner.
Risdon-Alcott Turbine Co.

steel turbine-chamber. Fig. 106 is an outside view of a turbine case, etc., from the same manufacturers, showing the cylinder regulating gate fully raised; the mechanism for operating the gate is clearly seen.

Figs. 107 to 110 show some of the usual methods of installing turbines.

Besides those already mentioned, there are several well-known American manufacturers of turbines and accessories—such as the Dayton Globe Iron Works Company, Dayton, Ohio; the Holyoke

Machine Company, and others. Figs. 111, 112, and 113 show some patterns of gate-valves; Fig. 114, a gate-valve with by-pass; and Fig. 115, a wicket-gate. Several other forms are in the market.

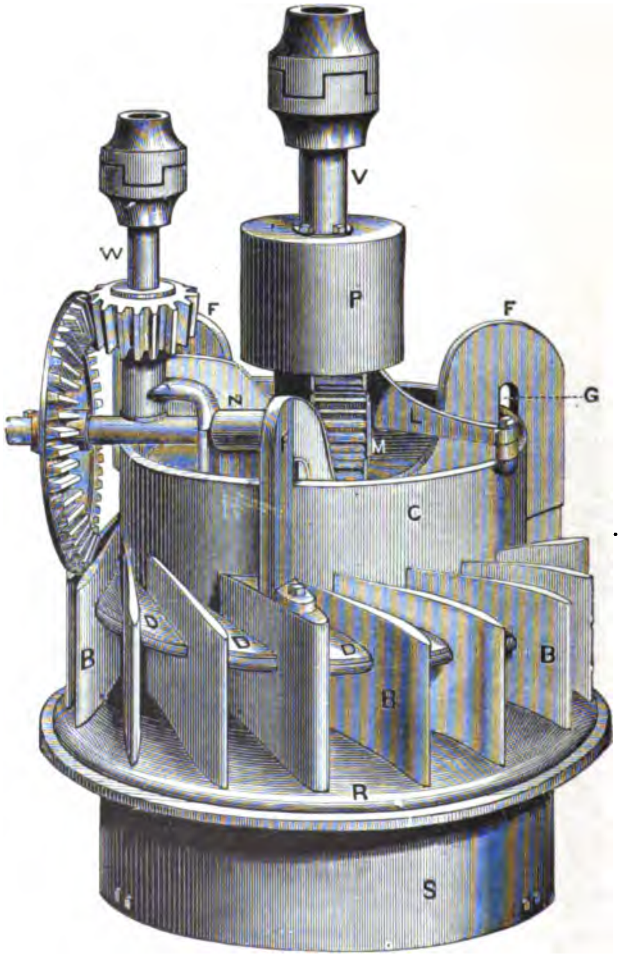


Fig. 101. Risdon Cylinder-Gate Turbine.
Risdon-Alcott Turbine Co.

Fig. 116 represents one form of safety relief-valve designed to operate at a pressure slightly greater than the normal; in the event of the water-flow being suddenly checked by the closing of the gate or by the operation of the governor, the excess pressure is relieved by the

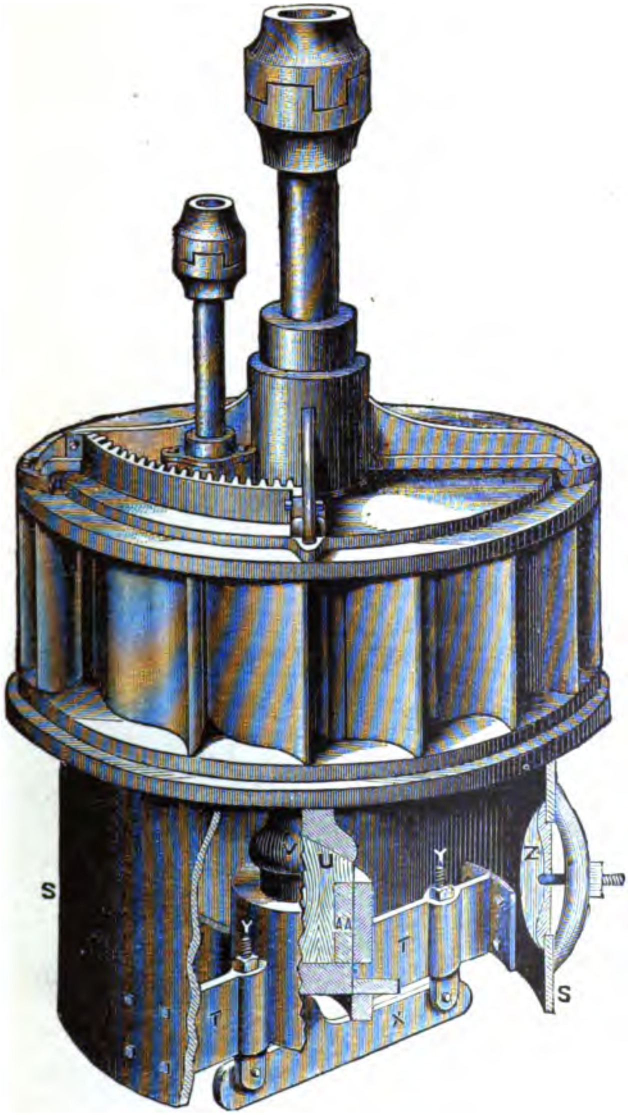


**Fig. 102. Risdon Register-Gate Turbine.
Outside Register Pattern.**
Risdon-Alcott Turbine Co.

momentary opening of the valves. They may be placed singly or in a battery.

Fig. 117 shows two types of safety air-valves designed to open

Fig. 103. Alcott High-Duty Turbine.
Inside Register Pattern.
Risdon-Alcott Turbine Co.

automatically in case the pipe-line should be emptied suddenly, thus permitting the air to rush in, and preventing possible collapse of the pipe due to the formation of a vacuum.

Fig. 118 illustrates a compensator consisting of a hydraulic plun-

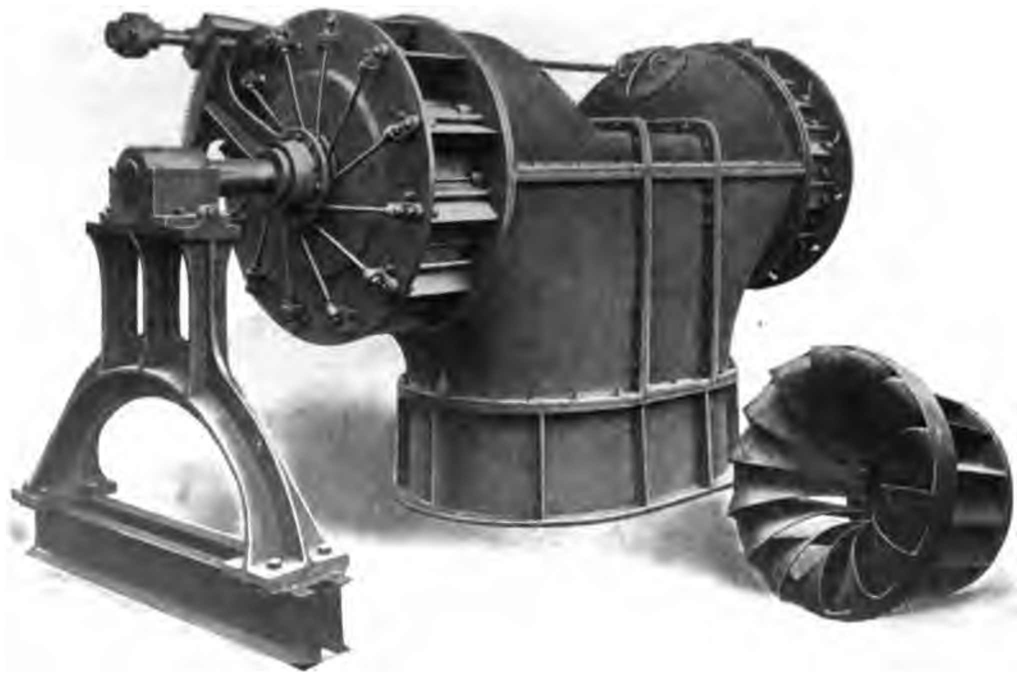


Fig. 104. Two Risdon-Alcott Turbines Mounted on Same Horizontal Shaft, Fitted with Balanced Hinged Gates, Discharging Centrally into Common Draft-Tube.

ger connected to the pipe-line, and balanced by suitable steel springs. The compensator is designed to take care of shocks in the pipe-line

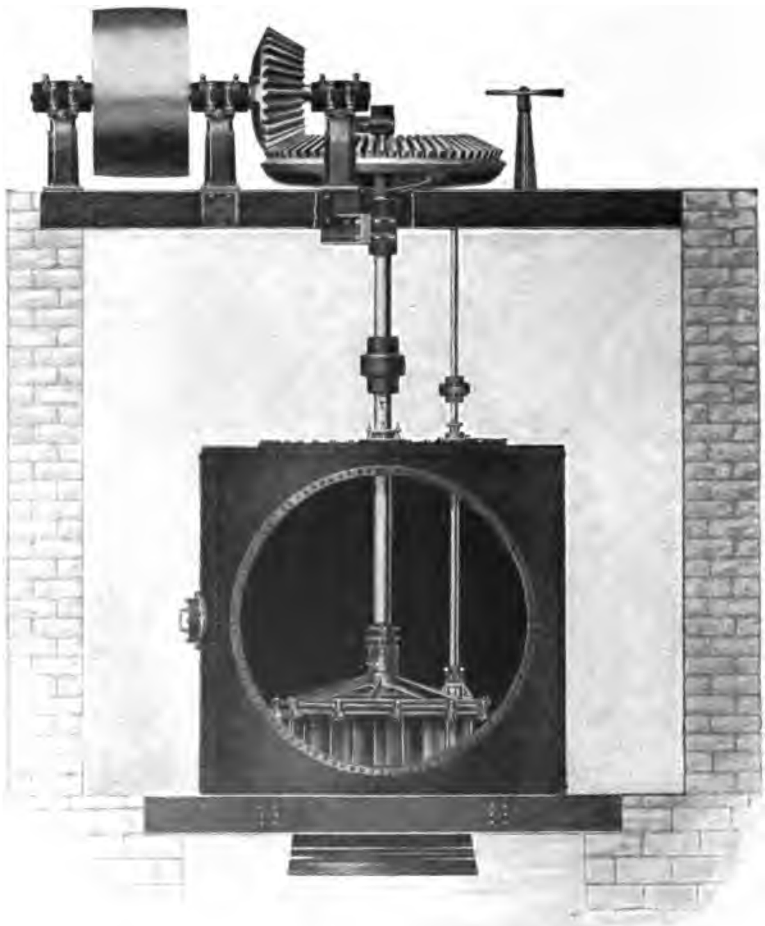


Fig. 105. Vertical-Shaft Turbine in a Steel Flume.
Courtesy of S. Morgan Smith Co., York, Pa.

resulting from checking the flow of water through sudden action of the governor or sudden closing of a gate-valve.

168. **Gauges.** In order to be able to obtain, without inconvenience, certain information relative to the operation of a water-power installation, gauges should be set up at various points of the works—

for example, a *pressure-gauge*, to indicate the pressure of the water near the entrance of the guide-buckets; a *vacuum-gauge*, to show the amount of draft or suction near the discharge openings of the runner-buckets; a gauge to show the pressure between the runner-disc and

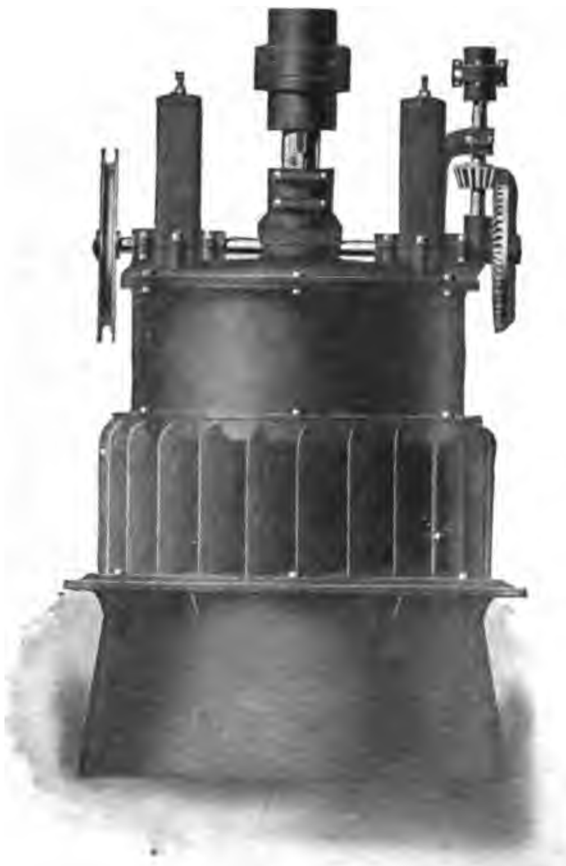


Fig. 106. Smith Turbine with Cylinder Regulating Gate Fully Raised.
Courtesy of S. Morgan Smith Co., York, Pa.

the head of the case or the dome; and, where a thrust-chamber or thrust-piston is employed, a gauge showing the pressure in the chamber, or behind the piston. A gauge should also be placed at the lower end of a long penstock, to indicate water-hammer, and the pressure fluctuations due to speed regulation of the turbine. The speed of the turbine at any moment should be indicated by a *tachometer* or gauge; and the gate-opening, or position of the regulating gates, should be shown by means of a dial and pointer.

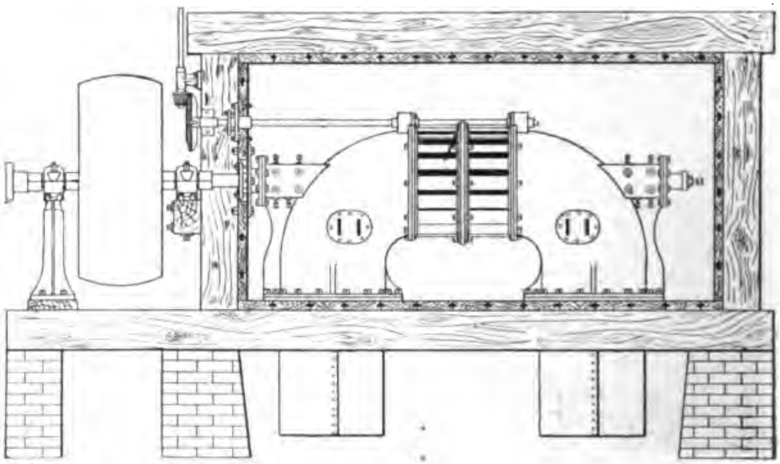


Fig. 107. Pair of Register-Gate Wheels in Wooden Penstock, Each Discharging through an Independent Elbow into its Own Draft-Tube.

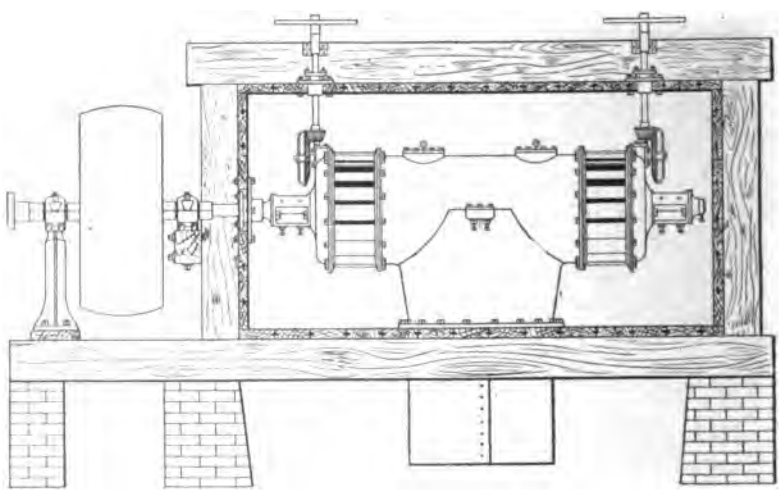


Fig. 108. Pair of Turbines in Wooden Penstock, Discharging through Single Draft-Tube.

169. **Transmission of Power from Turbine Shaft.** The simplest case is presented when the electric generators can be placed directly upon and revolve with the turbine shafts, as at the Niagara

BULKHEAD AND INLET PIPES FOR TURBINE PLANT

Conveying water to four pairs of 30-inch turbines at the power station of the Sacramento Electric Gas & Railway Company, Folsom, California. The turbines operate under a head of 55 feet, and are direct-connected to generators.

Courtesy of S. Morgan Smith Co., York, Pa.

Power Company's installations; in such cases the revolving armatures may be made heavy enough to act as fly-wheels. In other cases the power may be transmitted to other shafts by means of spur or bevel gearing, or by belt or rope transmission—with considerable loss of power. See Fig. 119.

170. **Connecting and Disconnecting Mechanism.** Mechanical devices for the purpose of throwing a wheel or shaft into or out of connection with the general system, involve problems which differ in

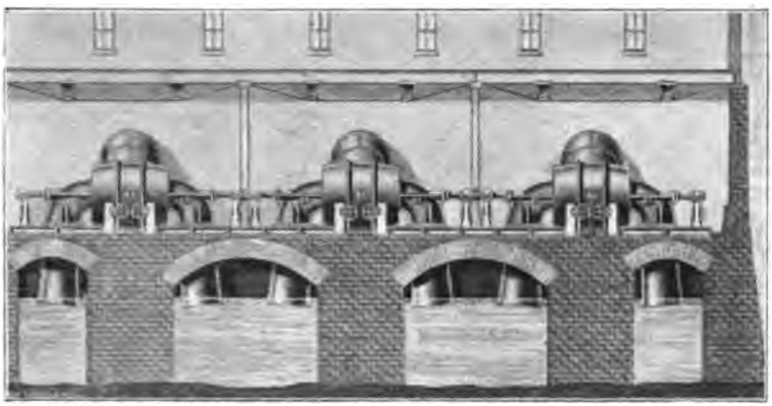


Fig. 109. A 2,000-Horse-Power Installation.
Comprises four 48-inch cylinder-gate wheels under 48 feet head, giving 1,600 horse-power; and two 36-inch cylinder-gate wheels under 27 feet head, giving 400 horse-power.

no essential particular from those met with in general mechanical engineering practice, and will therefore not be considered here.

GOVERNORS AND SPEED REGULATION

171. "Industrial operations require a uniform speed of shafting, although the quantity of work or the number of machines in operation may vary greatly from hour to hour, or from minute to minute. This condition necessitates an automatic device for controlling the admission of water to the wheels, diminishing the quantity when the velocity exceeds the normal, and *vice versâ*. The essential part of such a controlling and regulating device consists of an organ which moves in one direction and sets in motion the mechanism for partially closing the gate when the velocity exceeds the normal limit, and which moves in the opposite direction and sets in motion the mechanism for further opening the gate when the velocity falls below that limit."*

*Frizell, "Water-Power."

"The greatest difficulty encountered by the hydraulic-power engineer is the speed regulation of turbines under variable loads; and it has only been during the last few years that engineers have been

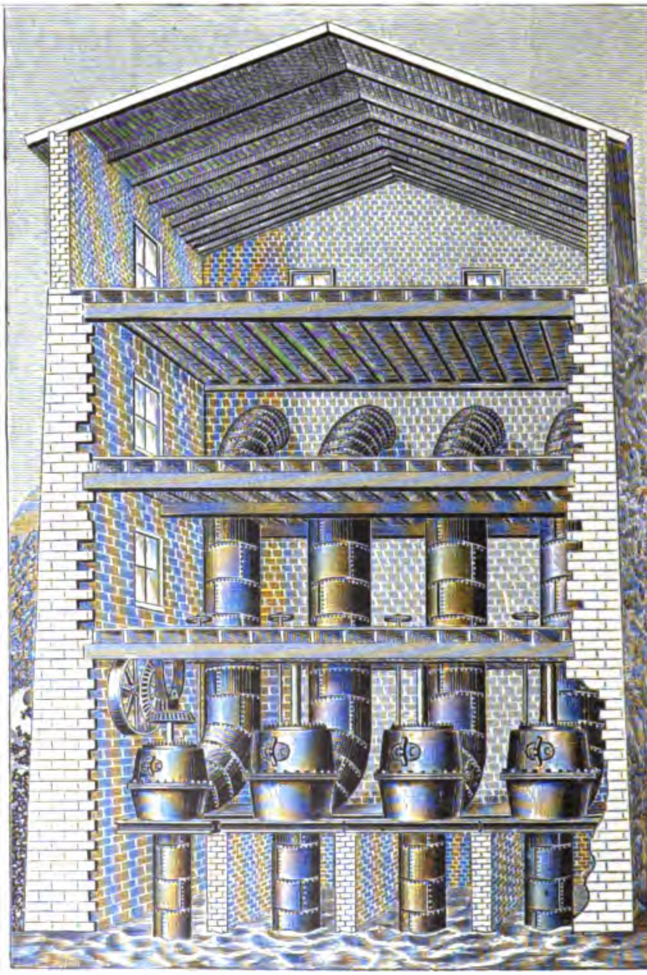


Fig. 110. An Installation of Vertical Turbines in Iron Cases with Draft-Tubes. Combined Power, 635 H. P.
Courtesy of Risdon-Alcott Turbine Co.

able to regulate the speed of turbines supplied by long penstocks as closely as is common in steam-engineering practice. The reason for this is that the weight of the moving column of steam will rarely exceed 100 lbs., while the velocity may be about 100 feet per second; on the other hand, the weight of the moving column of water may be millions of pounds, while the velocity is rarely over 10 feet per second. Thus the energy represented by the moving water-column may be hundreds or even thousands of times the energy represented by the moving steam-column.

"Every change of load or power developed requires a change in the engine cut-off or in the gate-opening of the turbine; and this, in turn, requires a change in the velocity of flow in the supply pipe or penstock, which means a change in the amount of energy represented by the moving column Moreover, steam is compressible and elastic; water is incompressible and inelastic; and if the load of the turbine, and thus the velocity required in the penstock, is suddenly decreased, the excess energy in the moving column must find some outlet, otherwise either the penstock or the turbine will be wrecked; while, if the load and the required velocity in the penstock are suddenly increased, the lack of energy must be supplied from some outside source.

Fig. 111. Straightway Valve of Single-Disc Type, with Rising Spindle, Intended for Pressure on One Side Only. Disc rises clear of opening, giving a perfectly free way for the water. Seats are of bronze, to be replaced when worn.

"Any decrease in the gate-opening, and consequent decrease in the velocity of the water in the penstock, will thus produce a temporary increase in the penstock pressure; and with the gates closing quickly, this increase in pressure may rise to the force and suddenness of a blow, usually called *water-hammer*. On the other hand, any increase in the gate-opening and in the velocity of the water in the penstock, will produce a temporary decrease in the penstock pressure. Such changes in gate-opening will frequently cause long-drawn pulsations in the penstock pressure, or surging of the water; and this action is often favored by badly arranged penstocks, relief-valves, standpipes, air-chambers, and connections, and aided by wave motion and eddies in the head-race near the penstock entrance.

"To obtain a good speed regulation, it is often necessary, espe-

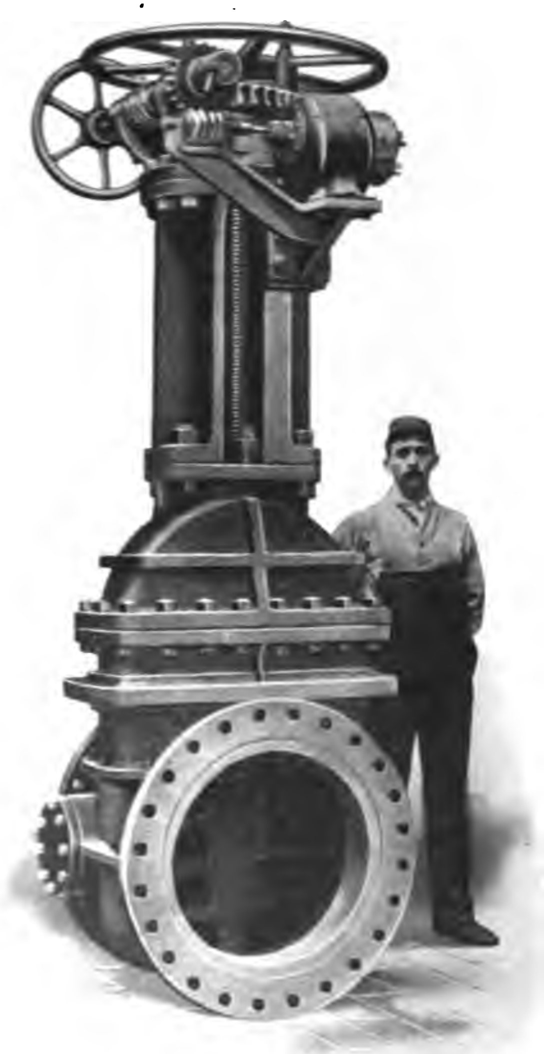


Fig. 112. A 24-Inch Gate-Valve of Single-Disc Type, with Outside Screw and Yoke and Rising Spindle. Arranged for operating by electric motor, and provided with roller bearings to take thrust from stem.
Pelton Water Wheel Co., San Francisco, Cal.

Fig. 113. Straightway Gate-Valve of Large Size, of Screw-Lift Pattern. Smaller sizes have a single instead of a double screw.
Dayton Globe Iron Works Co., Dayton, Ohio.

Fig. 114. Special High-Pressure Gate-Valve with By-Pass, and Roller Bearings on Stem.
Pelton Water Wheel Co., San Francisco, Cal.

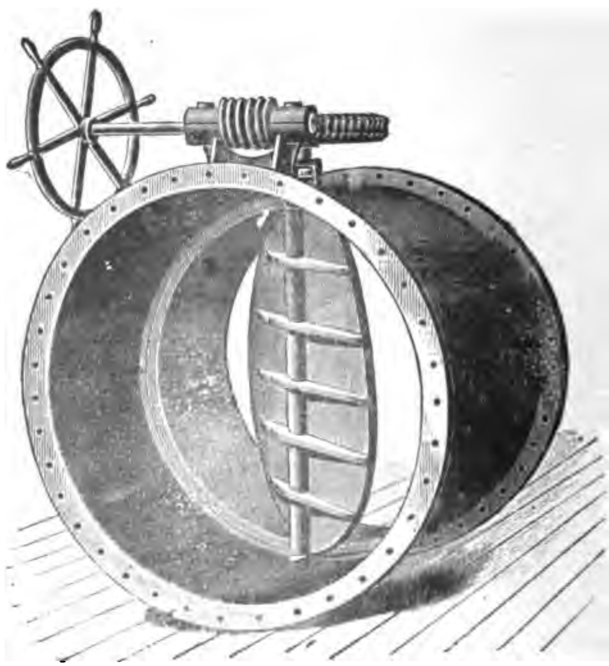


Fig. 115. Wicket-Gate.
Risdon-Alcott Turbine Co.

Fig. 116. Battery of Safety Relief-Valves.
Pelton Water Wheel Co.

Fig. 118. Spring-Balanced Compensator for Pipe-Lines.

Fig. 117. Safety Air-Valve for Pipe-Lines.

cially in connection with turbines supplied by long penstocks, to use some auxiliary device or devices for the escape and supply of energy, or for the escape at least, which may be briefly considered as follows:

172. "The *pressure-relief valve* serves for the escape of energy

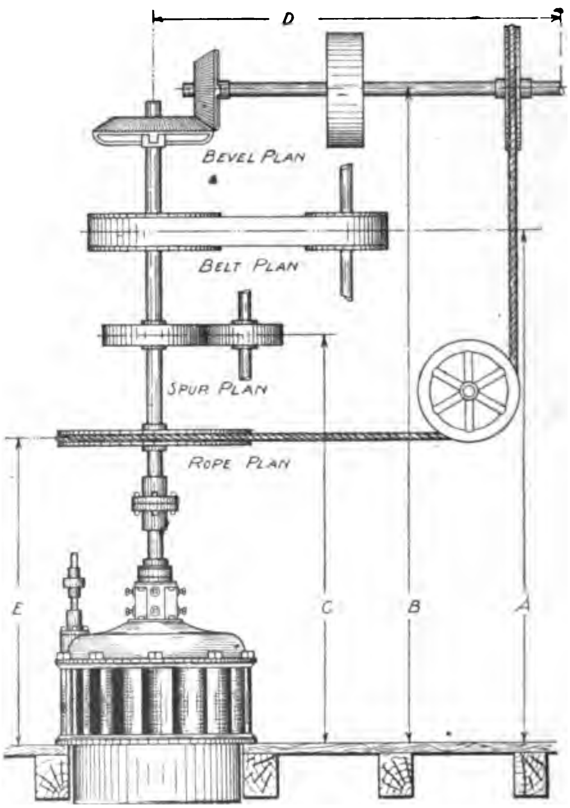


Fig. 119. Diagram Showing Various Methods Used for Transmitting Power.

when the pressure rises beyond a certain limit, in exactly the same manner as the safety-valve of a boiler. Relief-valves are well known; but nearly all of them are of poor design, being held closed by a single, short, helical spring of a few turns, and therefore can open only to a very small extent. A good spring relief-valve should have 3 to 6 helical springs, according to the size of the valve; and these springs should be long and have many turns, so as to permit the valve to open sufficiently, without requiring too great a rise in pressure in the penstock. Provision should be made for ascertaining at any time whether the valve is in working condition and set for the proper pressure.

173. "The *Lombard* pressure-relief valve, shown diagrammatically in Fig. 120, is a great improvement over the ordinary relief-valve. Its action is as follows: *A* is the end of a penstock, or a nozzle of a penstock, in which the pressure is to be relieved when a certain limit has been reached. The disc of the relief-valve *c* is held to its seat against the water-pressure in the penstock by the water-pressure behind the piston *e*, the pressure-water behind the piston being supplied from the penstock through pipe *f*. As the pressures per unit-area against the valve-disc and behind the piston are equal, the piston is made larger in diameter than the disc, so that the total pressure behind the piston will not only overcome the total pressure against the valve-disc, but also hold the latter firmly to its seat. The space behind the piston *e* is also connected through pipe *i* to the waste-valve *D*. This is a balanced valve, held closed by means of the spring *P*; while the water-pressure in the penstock, communicated through pipe *r*, and acting behind the piston *n*, tends to open the waste-valve. The force of the spring *P* can be regulated so that the water-pressure will overcome the force of the spring and open the valve at any desired pressure in the penstock.

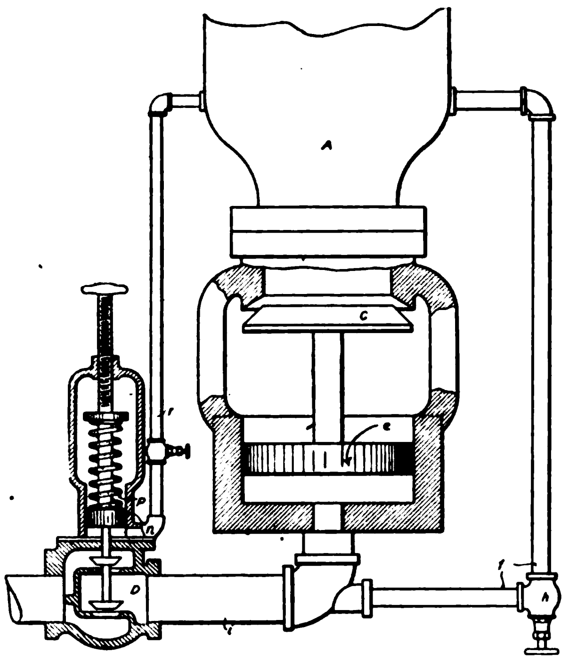


Fig. 120. Diagrammatic Representation of "Lombard" Pressure-Relief Valve.

"With the relief-valve closed and the water-pressure in the penstock rising above the normal to the pressure for which the spring *P* is set, the piston *n* will open the waste-valve, which will relieve the pressure behind the relief-valve piston *e*, and allow the pressure-water to escape. While water will begin to flow through pipe *f* as soon as the waste-valve is opened, yet the area of pipe *f* is so much smaller than the area of pipe *i* and the waste-valve opening, that the pressure behind the piston *e* will at once fall below the pressure which exists in the penstock; and therefore the pressure in the penstock forces the relief-valve disc *c* and piston *e* back, or, in other words, opens the relief-valve.

"The greater the rise in pressure is in the penstock, the greater will be the extent to which the waste-valve opens, and consequently the greater will be the reduction in pressure behind the piston *e*, and therefore the greater the extent to which the relief-valve opens.

"As soon as the pressure in the penstock has fallen to the pressure for which the spring *p* is set, the latter closes the waste-valve, the pipe *f* restores the full penstock pressure behind the piston *e*, and the latter closes the relief-valve. To prevent any surging in the penstock due to the closing of the relief-valve and the consequent retardation of the water, the relief-valve is made to close slowly, the rate of closing being adjustable by means of the valve *h*.

"Relief-valves must be prevented from freezing or from becoming incrusted with ice, as otherwise they may be rendered entirely useless.

174. "The *by-pass*, which may be employed where economy in water consumption is not demanded, consists of a valve or gate of sufficient area to pass the entire volume of water required by the turbine at full-gate opening, and moved in conjunction with the speed-regulating gates. With the turbine-gate fully open, the by-pass is closed; but when the regulating gates commence to close, the by-pass opens, and its passage area is increased in the same proportion as the gate-opening decreases, so that the combined area of the gate-opening and the by-pass is always sufficient to pass the entire volume of water required by the turbine at full-gate opening; thus the velocity of the water in the penstock and the amount of water discharged remain always the same, the discharge of the by-pass being run to waste. This arrangement not only permits the closest speed regulation with violently fluctuating loads, but also relieves the penstock from shock or water-hammer, and is therefore often used in connection

with impulse turbines working under high heads and supplied by very long penstocks.

"European engineers have abandoned the ordinary by-pass, on account of the great waste of water which its use implies, but frequently use the *temporary by-pass*, which is essentially the same device, except that the speed-regulating gates and the by-pass are connected in such a way that the temporary by-pass will open while the speed-regulating gate closes; but as soon as the closing movement of the regulating gate ceases, the by-pass at once starts automatically to close again slowly, being actuated by a spring, counterweight, or hydraulic pressure. The temporary by-pass does not open at all when the regulating gate closes very slowly. It will be seen that the temporary by-pass is similar in its effect to the relief-valve, except that the by-pass opens before a rise in the penstock pressure, due to the closing of the regulating-gates, takes place.

175. "The *standpipe* is frequently employed to aid the governor, and thus to improve the speed regulation of turbines; and is simply an open reservoir which, to a limited extent, will absorb or store energy, when the gate-opening is decreased in consequence of a reduction in the load of the turbine, and will supply energy when the gate-opening is increased in consequence of an increase in the load of the turbine. The standpipe is the best possible relief-valve, and should have its top edge a few feet above the high-water level in the head-race, and its diameter or capacity should be in accordance with the volume of water discharged by the turbine at full-gate opening, and with the length of the penstock.

"When the gate-opening of the turbine is suddenly reduced, the excess of water flows into the standpipe, causing the water therein to rise, and perhaps to escape over the top edge of it, until the water-column in the penstock has slowed down; while, when the gate-opening is suddenly enlarged, the additional water required is supplied from the standpipe, causing the water therein to fall until the speed of the water-column has increased to meet the demand. In connection with high heads, standpipes are rarely used, as they are not only very expensive but also less effective on account of the inertia of the water-column in the standpipe.

"Standpipes must be carefully protected from freezing, as otherwise they may be rendered entirely useless. A waste-pipe should be provided to carry off the water escaping over the top edge of the standpipe.

176. "*Air-chambers* are often used on penstocks; but while they may be useful to protect the penstock against the effects of water-hammer, they are of little or no value as an aid to the regulation of the turbines. To cushion the shocks in a penstock, an air-chamber should be of ample capacity; and as air is readily absorbed by the water, an air-pump should be provided to replace the air thus carried off. Gauge-glasses and try-cocks should be placed on each air-chamber, so that it may at once be seen whether it is effective.

177. "The blows struck by water in a penstock (or water-hammer), are first and most violently felt at the lower end of the penstock, and in the direction in which the water-column moves; and from there back up, so to say, with diminishing strength, toward the upper end of the penstock. Therefore all such safety devices as the relief-valve, by-pass, standpipe, and air-chamber should be at the extreme lower end of the penstock, and their discharge or connection should be in the direction in which the water-column moves. A standpipe should be connected with the penstock by a short, straight pipe of large diameter; and an air-chamber, by a short neck, also of large diameter.

178. "The *fly-wheel* is frequently employed in Europe, and to some extent in America, to aid the governor and thus to improve the speed regulation, especially in connection with turbines working under high heads; but a fly-wheel cannot, of course, protect a penstock against water-hammer. A turbine-runner has very little fly-wheel capacity; and the use of a fly-wheel will therefore eliminate the small variations in speed due to slight but sudden fluctuations in load, to water-hammer, to the surging of the water in the penstock and draft-tube, and to other causes, which momentary variations even the best governor cannot prevent. The amount of energy which a fly-wheel can absorb or give out is small, but it will at least retard the changes in speed of the turbine with changes of load. Where the turbines are used to drive dynamos, sufficient fly-wheel capacity may be given to the armature or revolving field to make a separate fly-wheel unnecessary. This plan was adopted by the Niagara Falls Power Company for its 5,000-horse-power alternating machines."*

179. **Mechanical and Hydraulic Governors.** The power to

* Thurso, "Modern Turbine Practice."

operate the turbine gates is sometimes furnished by the turbine itself; but on account of the size and weight of these gates in a large installation, considerable force must be exerted to operate them; and for this reason, they are not directly actuated by the sensitive centrifugal governor. An auxiliary *relay* motor, using either mechanical or hydraulic power, which is itself controlled by the centrifugal governor, is therefore interposed for the purpose of actuating the gates.

The *hydraulic relay* consists essentially of a piston and cylinder operated by pressure-water from the penstock, or oil from a pressure tank. The governor proper consists of the usual type of revolving centrifugal balls in gear with the turbine shaft, and so connected with the valves or devices of the motor or other mechanism which actuates the gates, that any change in the relative position of the balls brought about by a variation in the turbine speed, will bring into play the mechanism which operates the gates. (See also Article 83.)

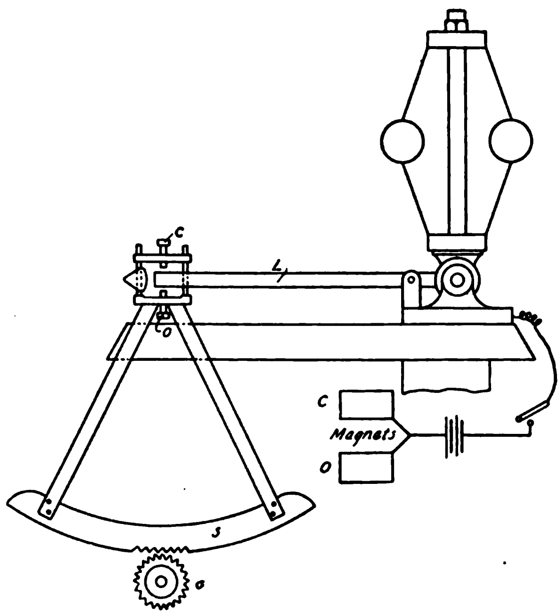


Fig. 121. Diagrammatic Representation of "Replogle" Centrifugal Governor and Return.

180. **Over-Governing or Racing.** When a change of speed due to a change in load takes place, the governor will set the regulating gate in motion; but, owing to the inertia of the water, a certain amount of time is required for the turbine to return to the proper speed; so that the gate would have traveled in the interval beyond the required position, and in a short time, under the influence of the governor, it would start moving in the opposite direction. To obviate this difficulty, the motion of the relay, and with it the movement of the regulating gate,

must be arrested before the latter has traveled too far, by a *return* device.

181. **Time of Closure.** Mechanical governors effect an entire

Fig. 122. Standard Type "Woodward" Governor, Suitable for General Mill Work.

closure of gate usually in 15 to 25 seconds, although some types have been constructed that require only about 3 seconds.

In the case of hydraulic governors, this time may be reduced to one second; such governors therefore afford very close speed regulation.

182. Fig. 121 represents diagrammatically the *Replogle Centrifugal Governor and Return,* a device well known in American practice. S is a toothed segment operated by the turbine-gate shaft G, which is also toothed for the purpose. L is a lever, tilted up or down by the governor-balls when a variation of speed occurs; this lever, in its motion,

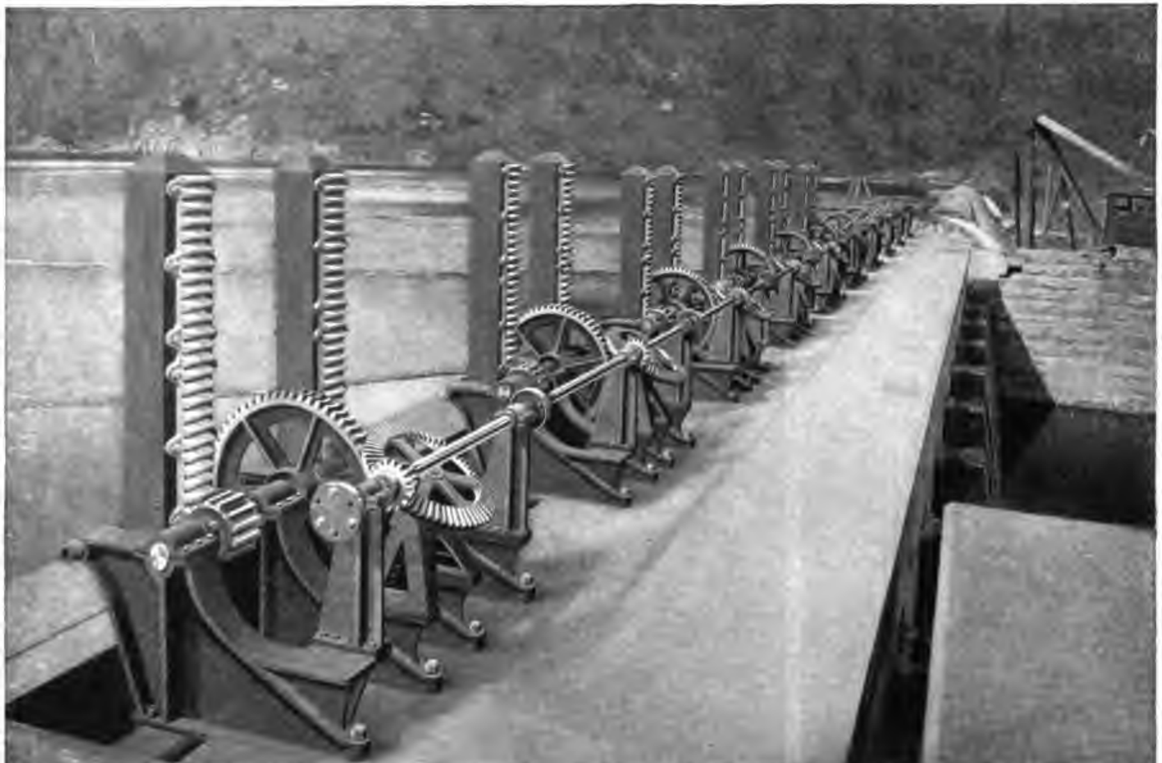
Fig. 123. Compensating Type "Woodward" Governor, Suitable for Electric and Heavy Mill Work.
Risdon-Alcott Turbine Co.

completes an electrical contact by touching points C or O. Such contact energizes one of two electromagnets in the circuit, by which means an auxiliary machine, or relay (not shown), is tripped, and begins to open or close the turbine-gate by turning G in the proper direction. The turning of G carries the rack S with it, which, by a suitable cam arrangement, breaks the contact between the lever and the point C

or *O*. This interruption of the electrical circuit cuts the auxiliary power out of action, thus stopping the motion of the gate before the increase or decrease of power due to the gate motion has given velocity to the turbine above or below the normal.

Later types of this governor have been still further refined.

Figs. 122 and 123 illustrate two types of *Woodward Governor.*



HEAD-GATE HOISTS OF THE HUDSON RIVER WATER POWER COMPANY AT SPIERS FALLS, NEW YORK

Ten large hoists operated by a single motor-driven shaft.

Courtesy of S. Morgan Smith Co., York, Pa.

PENSTOCKS LEADING WATER DOWN MOUNTAIN SIDE
Plant of the Puget Sound Power Company, Electron, W

WATER-POWER DEVELOPMENT

PART III

CONDUCTION OF WATER

183. **Head-Race and Tail-Race.** In general, comparatively narrow and deep races are preferable to wide and shallow ones, because of the smaller loss of head in the former case, particularly in localities where ice is likely to form in the winter months, which not only reduces the area of waterway, but offers considerable additional frictional resistance. In such localities it is advisable to protect that part of the tail-race which is under the power-house, together with the tail-race opening itself, against freezing and consequent accumulation of ice, by boarding up the upper part of the opening to within 1 or 2 feet of normal tail-race level, and attaching a floating strip of canvas or tarpaulin, or a hinged board, to the lower edge of this partition, the bottom of the board reaching to low tail-water level.

The velocity of the water in the races is usually between 2 and 3 feet per second, which is sufficiently low to allow the water to freeze over the surface, and thus prevent the formation and serious accumulation of anchor-ice and frazil.

The location and direction of the entrance to a head-race should be such as to prevent the carriage of sand in suspension into the head-race; and in order to keep out logs, ice, and floating debris, a heavy boom, or a crib with openings for the passage of water below the surface, should be placed across the entrance in such a direction that floating matter will have no tendency to lodge and accumulate against it, but will be deflected and carried downstream. Where possible, bends and curves should be avoided, because of the loss of head they occasion; when necessary, they should have long radii.

Where much sand is carried in suspension, the water may be allowed to flow through a *sand-settler*, which is sometimes merely a basin formed by an enlargement of the head-race, the increase in cross-section inducing a corresponding decrease in velocity, resulting in a deposition of the sand. Frequently grooves, or narrow upright boards, are placed in the bottom at right angles to the flow, to arrest sand rolling along the bottom. With open timber flumes, these sand-settlers may be merely large, shallow wooden boxes. When large volumes of water are handled, a ditch is often placed near the water-racks to catch the sand rolling along the bottom; and by means of a pipe, screened to keep out coarse material, and provided with a gate, the accumulated sand may from time to time be flushed out. With very large quantities of water carrying much sand, it will frequently be found more economical to allow the sand to pass through the turbines, and renew the worn guides and runners when necessary, than to attempt to free the water of its burden of sand.

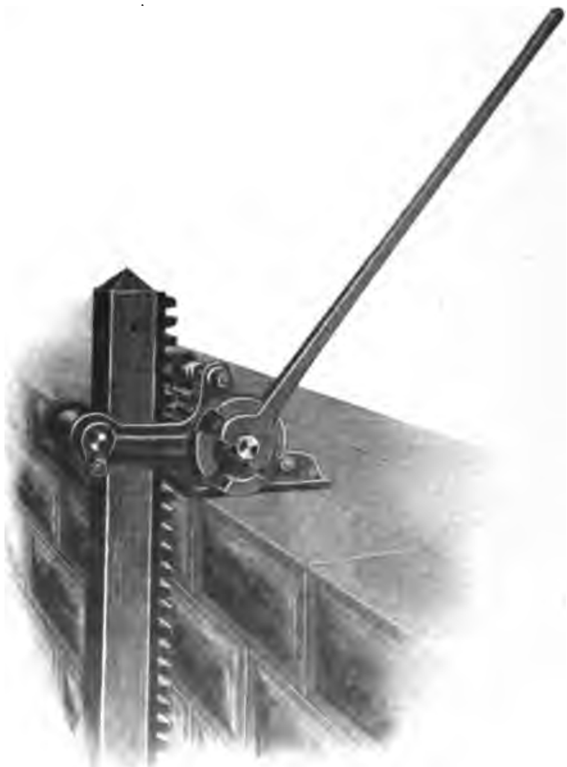


Fig. 124. Head-Gate Hoist.
Operation of opening and closing is performed by means of a lever inserted in a winch-wheel on the end of the pinion-shaft.
Dayton Globe Iron Works Co., Dayton, O.

Near the power-house end of the head-race, should be located a wasteway with sluice-gate, to discharge ice and other matter, and a boom to guide to this gate such floating matter as may pass down the head-race.

The tail-race should be designed to afford easy discharge of the water, so as to prevent the possibility of backing up around the turbine or draft-tube. For this reason the walls of the tail-race should be so shaped as to deflect the water with the greatest directness in the proper direction. When draft-tubes are employed, they should be carefully curved or inclined in the direction of flow; when several single

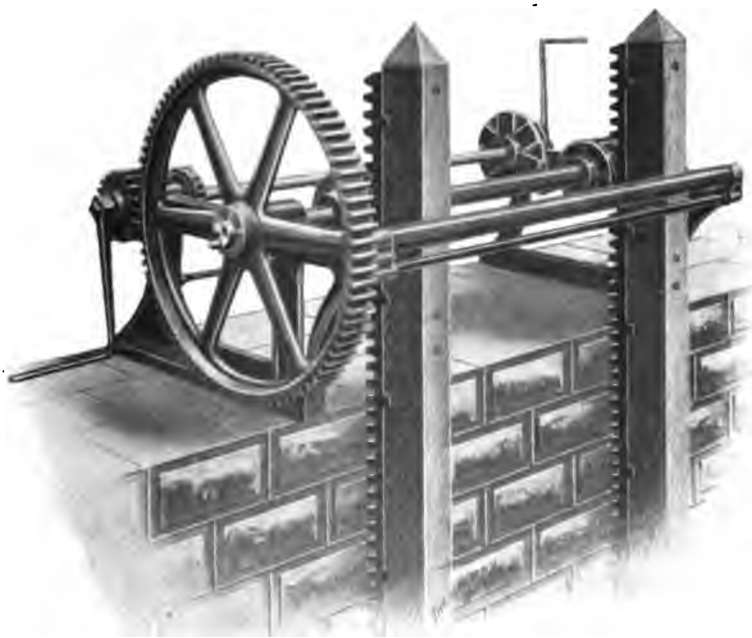


Fig. 125. Compound-Geared Head-Gate Hoist for Double-Stem Gate.
Shafts carrying spur gears and pinions are substantially mounted on cast-iron stands; wrought iron cranks are used for operating the hoist. Hoist of similar pattern may be used for a single-stem gate.
Dayton Globe Iron Works Co., Dayton, O.

draft-tubes discharge into the same tail-race, they should be located on one side. Double draft-tubes may be placed, one on each side; when so arranged, the obstruction to the flow of water will be much less than if they were placed in the center of the tail-race.

184. Water-Racks. These are screens through which the water is compelled to pass on its way to the turbines, so that all floating or suspended material larger than the clear opening between bars will be caught, and thus choking or damage to the wheels will be prevented.

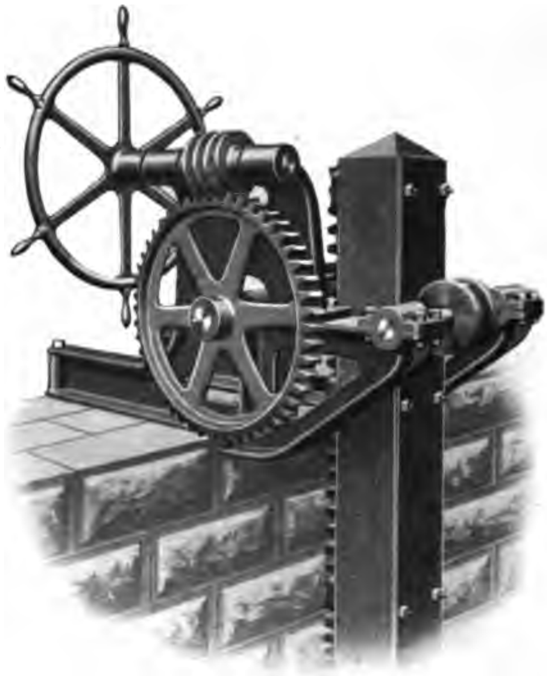


Fig. 126. Single-Stem Head-Gate Hoist.
With pilot wheel and worm-wheel operation.
Dayton Globe Iron Works Co., Dayton, O.

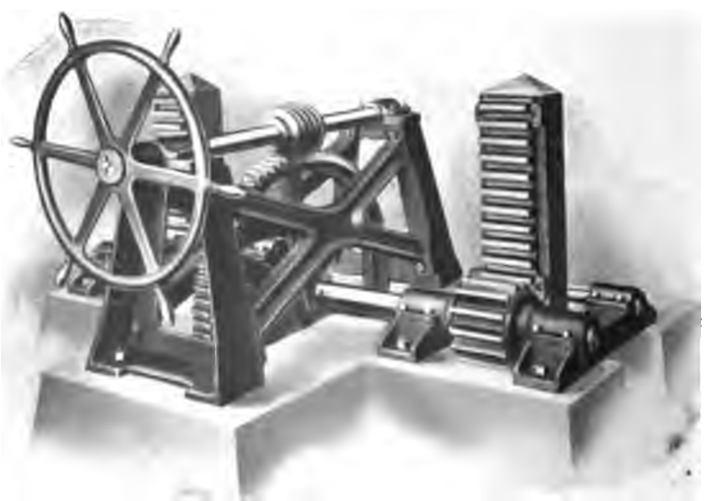


Fig. 127. Double-Stem Worm-Wheel Hoist.
Dayton Globe Iron Works Co., Dayton, O.

To insure this result, it is necessary that the clear opening between bars be somewhat less than the smallest dimension of the water passages in the guide- or runner-buckets; but this may not prevent temporary choking when the regulating gates are partially closed, a condition, however, quickly remedied by simply opening the gates. *Double racks* are sometimes employed, consisting of a coarse rack placed in front of the fine rack; this procedure is particularly advisable in the absence

Fig. 128. Head-Gate Hoist for Operating Gates under Pressure.
Upper end of gate stem is threaded to match a bronze nut attached to bevel gear shown in cut. Ball bearings are used to insure easy operation.
Dayton Globe Iron Works Co., Dayton, O.

of a protecting boom or crib. For the fine racks, the clear space is usually $\frac{3}{4}$ to $1\frac{1}{2}$ inches, the bars being of wrought iron or steel, $\frac{1}{4}$ to $\frac{3}{8}$ inch thick, by 3 to 4 inches wide. For coarse racks, the clear space is about 3 inches, the bars being of the same material, $\frac{1}{2}$ to 1 inch thick, by 4 to 5 inches wide.

In order that the water may find free passage, even though the rack be partially clogged, as well as because of the frictional resistance, the total clear area of rack should be considerably in excess of the total area of the penstock inlets; for this reason, and also for greater convenience in cleaning, the racks are usually placed in an inclined position. For greater ease in handling, repairing, etc., a rack built

Fig. 129. Filler Gate.
With frame for bolting to a timber head-gate. Allows passage of sufficient water to equalize pressure, so that head-gate need not be lifted under total pressure.
Dayton Globe Iron Works Co., Dayton, O.

Fig. 130. Head-Gate and Hoist.
Square, cast-iron gate.
Dayton Globe Iron Works Co., Dayton, O.

of removable sections 3 or 4 feet wide is frequently used, instead of a continuous rack.

185. **Head-Gates.** The purpose of the head-gate (see Figs. 124 to 136) is to control or shut off the water from a penstock, flume, open turbine-chamber, or forebay. It is usually a vertical gate sliding in guides, and, until quite recently, was constructed of wood

held together and braced by iron bolts and straps. Head-gates are now frequently built up of steel plate and structural steel—more particularly those of large size. Such gates are actuated by hand, by means of rack-and-pinion or screw-spindles, or by special devices operating under mechanical, electric, or hydraulic power. They are frequently counterweighted; and for large gates a *by-pass* or *balance-port* is usually employed, by means of which the water-pressure on the two faces may be balanced before moving the gate. In some cases, friction roller-bearings are employed to reduce the friction; and in others, by an ingenious contrivance, the gate is lifted from its seat in the preliminary action of opening. In order to reduce the wear, the gate is sometimes designed to slide on special guide-bearings, instead of on its seat, which it does not touch until reaching its position of complete closure.

Fig. 131. Head-Gate and Hoist.
Circular, cast-iron gate.
Dayton Globe Iron Works Co., Dayton, O.

A cylindrical gate, built up of plate and structural steel, and so pivoted that it is practically balanced, is shown in Fig. 133; it is also shown in position in the power house, Fig. 80.

Another form of head-gate which has not come into general use, though possessing many positive advantages, consists of a cast-iron cylinder, double-seated, the lower seat being formed by a ring attached to the floor of the head-race, and the upper seat by the edge or rim of a dome (forming the head), which is fastened by steel rods

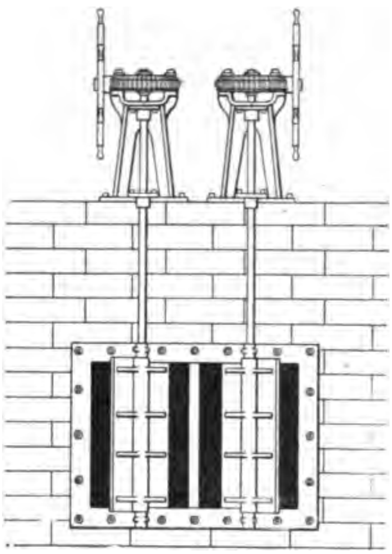


Fig. 132. Worm-Geared Swing Gate.

to the lower ring, as in Fig. 134. The cylinder itself is raised or lowered by means of a chain. In the center of the dome is a small filling-gate operated by a separate chain. It is thus practically balanced, and hence requires but little power in operation; moreover, the *lift* required to give a clear waterway equal in area to that of the circular opening which it controls, is but one-quarter of the diameter of such opening.

Where the necessary headroom is not available, *wicket-gates* turning on a vertical spindle may be used, as in Fig. 135. Such a gate requires less power to operate than a sliding gate, but is generally liable to greater leakage. When open, it presents its edge to the current, and so offers some, though no great obstruction to the current.

It is advisable to design gates intended to be used in cold climates, so that they may be entirely below the surface ice when closed, and entirely clear of it when open.

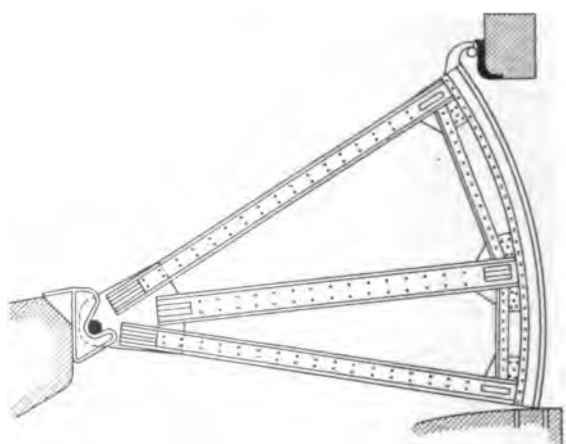


Fig. 133. Cylindrical Balanced Gate Built Up of Plate and Structural Steel.

186. Gate-Houses. In many important water-power projects, the flow into the canal is controlled by a series of gates, with their hoisting gear and appurtenances, all enclosed in a covered building.

187. The Penstock. This term is applied to the pipe which brings the water from the canal or other source of supply, to the turbine-chamber. When the source of supply is near the motor, it is a relatively unimportant detail of the system. On the other hand, it sometimes happens that this pipe is several miles long; in which case it assumes a position of primary importance; in fact it may become one of the controlling features or elements of the design.

Penstocks, or *feeder-pipes*, may be made of riveted wrought-iron or steel, of wooden staves, or of concrete-steel. They should always

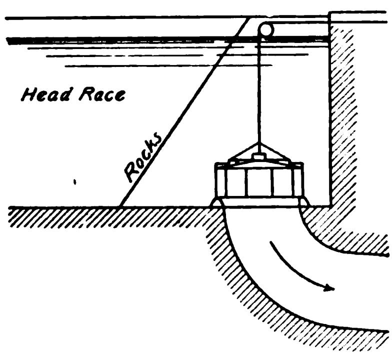


Fig. 134. Cast-Iron Cylindrical Double-Seated Head-Gate.

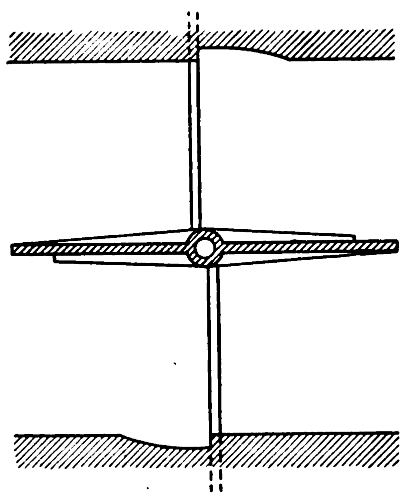


Fig. 135. Wicket-Gate Turning on Vertical Spindle.

be as short as possible, even when a shorter penstock involves a greater expenditure for excavation, etc. The shorter the penstock, the better it is for the speed regulation of the turbines, and the less steel-plate work has to be kept painted and repaired.

The following rules should be observed when determining the cross-sectional area of the conductors which convey the water to and from the turbines:

"The speed of the water should be gradually increased from the speed in the head-race, usually 2 or 3 feet per second, to the penstock speed, by means of a *cone* or *taper piece*, as in Fig. 136. Near the lower end of the penstock, the speed should again be gradually increased, so that the water will arrive at the guide-buckets with a speed equal to that with which it has to enter these guide-buckets. At the entrance of the draft-tube, or draft-tube elbow or tee, the water should have a speed equal to the absolute velocity with which it leaves the runner-buckets, and should then gradually decrease to a speed of about 2 or 3 feet at the lower end of the draft-tube. A speed of 2 or 3 feet is also usually chosen for the tail-race.

"In general it should be stated: Avoid changes of speed of the water where possible; but where such changes are necessary, make them gradually; also, avoid changes of direction of water; but where such changes are necessary, use curves of long radii.

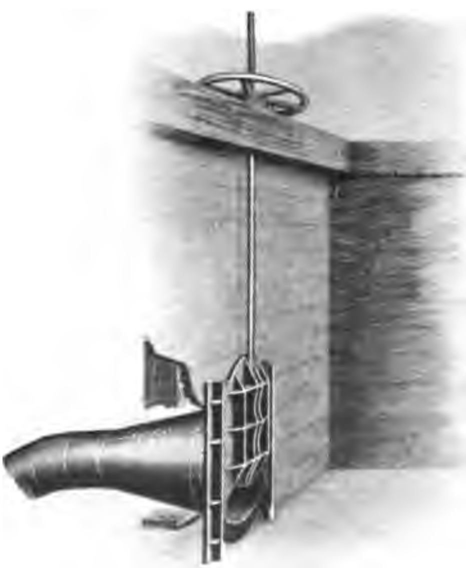


Fig. 136. Entrance Taper and Head-Gate in Flume. *Pelton Water Wheel Company, San Francisco, Cal.*

"The arrangement often employed, of having at the lower end of the penstock, and at right angles to the same, a drum or receiver of much larger diameter than the penstock itself, from which drum a number of turbines are supplied by branches set at right angles to the drum, must be condemned on account of the resulting abrupt changes in speed and direction of the water.

"All nozzles or branches of penstocks should be at an angle of not over 45 degrees to the penstock; or, in other words, the directions of flow of the water in the penstock and in the nozzle or branch should form an angle of not over 45 degrees with each other. Directly beyond each nozzle or branch, the diameter of the penstock should be reduced, to keep the speed of the water uniform.

"When determining the speed for the water in the penstock, all conditions should be carefully considered, and it should also be borne in mind that the friction loss in a penstock varies with the square of the speed.

"Conditions making a low speed advisable are: Low head, large diameter of penstock, great length of penstock, many bends in penstock, variable loads on the turbines, regulation of speed of turbines by changing the amount of water used.

"Conditions making a high speed permissible are: High heads, small diameter of penstock, short penstock, few or no bends in penstock, steady loads on the turbines, regulation of speed of turbines by by-pass.

"Many hydraulic engineers employ in all cases a penstock speed of 3 feet per second; but it is often of advantage greatly to exceed this velocity. From a great number of well-designed water-power plants constructed in America and Europe during recent years, the writer has deduced the following table of highest permissible speeds of water in penstocks of a length of 1,000 feet or less, with easy bends, and provided with proper arrangements for the protection of the penstocks against water-hammer:

PERMISSIBLE SPEEDS OF WATER IN PENSTOCKS

Diameter of Penstock (in feet)	4	5	6	7	8	9	10	11	12
Speed of Water (in feet per second)	12	11.5	11	10.5	10	9.5	9	8.5	8

"In penstocks of 1 or 2 feet diameter, speeds as high as 20 to 30 feet have been used. With very low heads, the penstock speed is often limited by the amount of head that it is permissible to lose in the penstock.

"The principal losses in the head of the water while entering the penstock and flowing through the penstock and draft-tube, are due to the following causes:

"(1) *Entrance Loss.* This loss may be kept low by having a large entrance connected to the penstock by an easy cone or taper piece. With the usual head-gate arrangement, such large entrance openings require very heavy and cumbersome gates for penstocks of large diameter; but there is no reason why this taper piece could not be partly or wholly in front of the gate and inside the head-race or forebay. The penstock entrance should always be as much below the surface of the water as circumstances will permit.

"(2) *Friction Loss.* This loss may be kept down by a low speed of water, and by smooth interior of the penstock and draft-tube.

"(3) *Loss Due to Changes in Direction of Flow.* This loss may be kept down by using as few and as easy bends as possible.

"(4) *Loss Caused by Changes in Speed of the Water.* This loss is due to the conversion of part of the energy in the water into another form, and may be kept low by having as few and as gradual changes as possible.

"(5) *Loss Due to the Speed of the Water While Leaving the Lower End of the Draft-Tube.* This loss is equal to the velocity-head, corresponding to the speed with which the water leaves the draft-tube, and may be kept down by making this speed low.

"Long penstocks, carrying water at high speed, should be provided with a safety-head, besides the usual devices for the protection of the penstock against water-hammer. For this purpose, a cast-iron or angle-bar flange is riveted to the lower end of the penstock, to which flange the head closing the lower end is bolted. The flange-bolts should have a factor of safety of not more than about half the factor employed for the rest of the penstock. Between flange and head a packing of dry white pine should be used, which, when water is admitted to the penstock, swells and makes a tight joint. Where the end cannot be used for this purpose, large nozzles may be riveted to the penstock, located as nearly as possible in the line of the water-hammer, and closed by heads secured as already described. The end of the penstock, or the nozzles, should be so situated that, should the heads blow out, no damage will be done by the jet of water issuing from the opening. This arrangement will not only save the penstock and turbines from being wrecked in case of severe water-hammer, but also the power-house from being demolished by the water set free.

"Ample air-inlets should be provided at the upper end of the penstock, as otherwise—should the safety-head by some chance give way, or the turbine-gates or turbine stop-valves, if such are employed, be opened while the head-gate is closed, but the penstock full of water—the penstock might collapse by the vacuum created in its interior. Care must be taken to prevent the water in the vents or air-inlets from freezing, as this would render them useless.

"A penstock which is carried for a considerable distance at about the same elevation as that of its inlet, and with so little slope as to be nearly horizontal, and then descends to the power-house on a steep grade, is liable to collapse when the turbine-gates are opened quickly, as the water in the inclined part has the tendency to increase its speed more quickly than the water in the horizontal part, and may thus break away from the latter and cause a vacuum in the penstock. An air-inlet valve will prevent this, but it is better to have a small compensating or equalizing reservoir (or stand-pipe) at the junction of the horizontal and inclined part of the penstock. Such a reservoir may be built of steel plate, concrete, or masonry, and will not only prevent the collapse of the penstock from the cause above named, but will also greatly improve the regulation of the turbines, and decrease the water-hammer in the penstock, acting, in fact, in the same manner as a standpipe.

"*Expansion-joints* in steel penstocks are not as important as is often asserted, since most penstocks contain bends which permit of a limited movement large enough to compensate for expansion and contraction; but in a straight steel penstock rigidly held at each end, the strains due to changes in temperature are very heavy, and in such case expansion-joints must be provided.

"The lower end of a penstock should be held very securely in all cases, to prevent forces due to temperature changes and other causes from throwing the turbines out of alignment, cracking the power-house walls, etc.

"Steel-plate penstocks are usually made in small and large courses, and lap-riveted. Butt-strap joints, with a single butt-strap on the outside, offer less frictional resistance to the flow of the water, but are more expensive. A manhole should be provided at the upper end of a penstock, and at the lower end also, if required. When repainting the inside of a penstock, or repairing the same, the water leaking through the head-gate should be prevented from running down the penstock; and for this purpose a small outlet-nozzle, about 6 inches in diameter, and closed by a blank flange, is provided at the lower side of the upper end of the penstock; and by building a small dam of clay in the penstock, just beyond this nozzle, the leakage is prevented from flowing down the penstock. All openings in penstocks for large nozzles, branches, manholes, etc., should be reinforced by steel-plate rings riveted around the openings, to make up for the material cut away by the opening.

"Penstocks should be calked both inside and outside, and the plates thoroughly cleaned by scrapers and wire brushes before painting.

"Masonry piers are often damaged by the expansion and contraction of the penstock they support, and the paint is rubbed off the penstock where it rests on the piers. Such unprotected places are hidden from view by the masonry, and are apt to corrode very quickly, as water is always retained between the surfaces of contact of the masonry and the penstock. It is therefore preferable to use steel piers on concrete or masonry bases. Such steel piers are cheaper than concrete or masonry piers; they leave every part of the penstock accessible for painting and repairs, and are free to swing on their bases, like inverted pendulums, to accommodate themselves to any movements of the penstock caused by changes in temperature. The uprights or posts of these piers are provided with bolt-holes, to fasten to them the studs for a housing over and around the penstock when desired.

"Except where the distance between the penstock and the ground varies considerably, the steel piers are all made the same; and the variations in the height of the penstock above the rock or solid ground are made up in the height of the concrete or masonry bases. The uprights of the steel piers are anchored to the bases, or, if the latter are of small height, through the bases to the rock below.

"A penstock running down a steep mountain-side must be prevented from sliding down the slope. Where concrete or masonry piers are employed, it is often sufficient to rivet short pieces of heavy angle-bars to the penstock, and to have these bars bear against the up-hill side of the piers; but with steel piers the penstock must be anchored to the rock or to special anchor-piers. It is well to have, in any case, a specially heavy concrete or masonry pier at the lower end of the penstock, to prevent the latter from throwing the turbines out of alignment.

"In a climate like that of the northern part of the United States and of Canada, penstocks must be covered or boxed in, to protect them from the extreme cold; otherwise ice will form on their inner surfaces.

"During midsummer, the heat of the sun's rays acting on an

Making Brush Mats for Dam.

View Showing Brush Mat as Used in Constructing Dam.

CONTROLLING FLOOD WATERS OF THE COLORADO RIVER

empty penstock will often injure the paint, cause it to blister off, and perhaps overstrain the penstock itself; and a covering will therefore prove an advantage both in cold and in hot weather. Even in a well-protected penstock, ice will be formed in severe weather, when the water in it is allowed to remain stationary for more than a few hours at a time.

"Where the ground under a penstock consists of earth, it is pref-

Fig. 137. Redwood Stavepipe under 160 Feet Head. Santa Ana Canal, California.

erable to bury the penstock below the frost-line, like the water mains in a city street.

"A buried penstock is free from the bending strains occasioned in a penstock carried on piers by reason of the length between the piers being unsupported; but a buried penstock of large diameter will require stiffening angles to be riveted to the upper half of its circumference, to prevent it from collapsing by the weight of the earth above it.

"The cost of burying a penstock will be about the same as when masonry piers are used.

"Under the penstock, in the center of the ditch, should be a drainage-ditch about one foot square in cross-section, and filled with pebbles or broken stone, as used for concrete-making. The penstock should rest on short wooden blocks; and the main ditch should be left open during the first year, or for one winter season at least; after which the penstock is carefully inspected, recalked where leaky, and repainted inside and outside; and then the earth is packed under and around the penstock, and the ditch filled in, the wooden blocks being removed as the work proceeds.

"**Wooden Penstocks.** Wooden penstocks or *stavepipes* deserve a wider application than they have so far found in the Eastern States. Wooden penstocks are cheaper and will last longer than steel penstocks, need less protection against extremes in temperature, and require no painting. Their interior surfaces are smoother than those of steel penstocks, and therefore offer less resistance to the flow of the water. Such pipes have been built up to 9 feet diameter.

"Wooden penstocks are made of staves from 2 to 4 inches thick and from 6 to 8 inches wide, planed to the proper shape, and held together by round or oval iron or steel rods, connected by hooplocks of various designs. The staves must be thick enough, or the hoops spaced closely enough, to prevent the staves from bulging out between the hoops. Thick staves are usually provided on one edge with a bead of from $\frac{1}{16}$ to $\frac{1}{8}$ inch in height, by $\frac{1}{2}$ to $\frac{3}{4}$ inch in width, located next to the inner side of the stave, as with such a bead it will require less strain in the hoops to make the penstock water-tight.

"The joints at the ends of the staves are usually made by steel tongues driven into kerfs. These joints must be well broken.

"Curves in wooden penstocks require a long radius, and therefore their horizontal and vertical alignment must be located on the ground, like a railroad line. The minimum radius, in feet, that can be used in a wooden penstock is about $R = 12.5 \times D_p \times t_s$, in which D_p is the inside diameter of the penstock in feet, and t_s the thickness of the staves in inches. Where a smaller radius is required, a section of steel penstock has to be inserted in the wooden one for the purpose.

"The wood employed should be clear and sound, and free from pitch, so that the staves will become saturated by the water. The wood used for such stavepipes is, in the order of its value for the purpose: California redwood, Douglas spruce (also called Douglas fir), spruce, white pine, southern pine, and cypress.

"The staves of a wooden penstock that is not left empty long enough to allow the wood to dry, will last much longer than the hoops; and the hoops may be renewed, when destroyed or weakened by rust, by placing new ones between the old hoops, if the soundness of the

Fig. 138. Redwood Stavepipe (52-inch) Crossing Warmsprings Canyon, near Redlands, California.
Courtesy of U. S. Geological Survey.

Fig. 139. Side View of Sterling Flume in Provo Canyon, Utah.
Courtesy of U. S. Geological Survey.

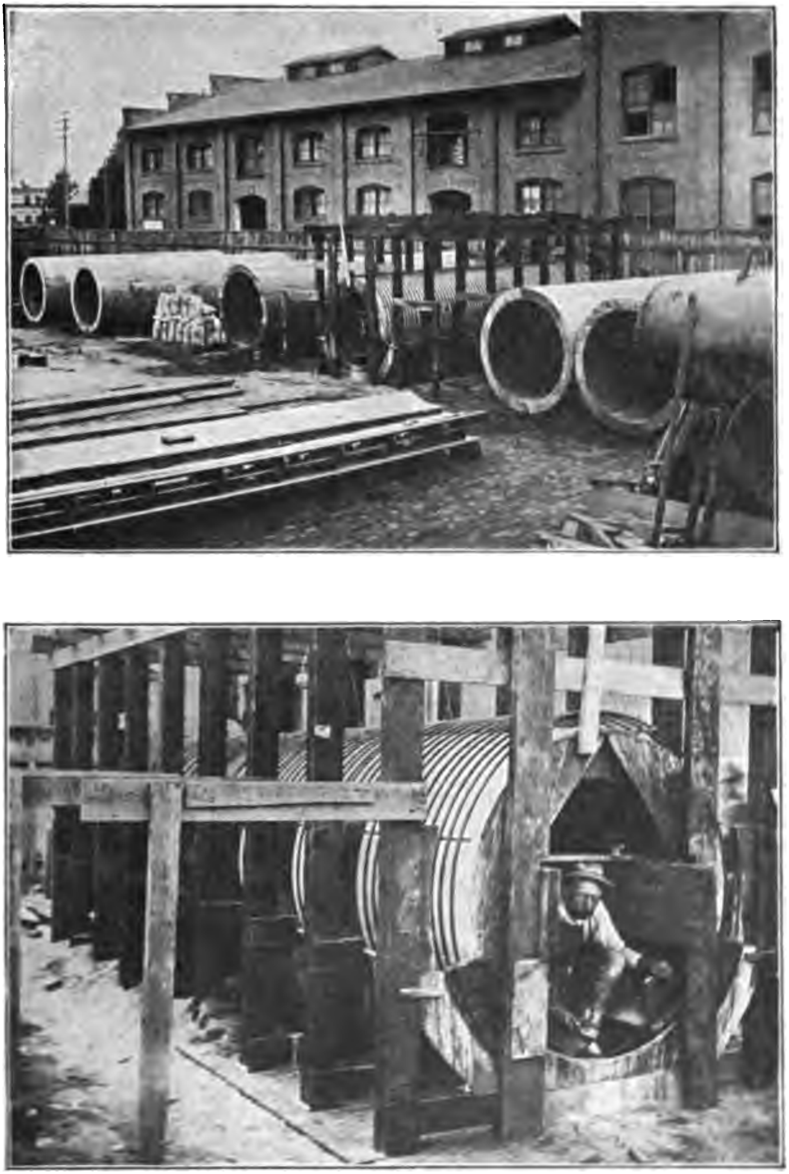


Fig. 140. Reinforced-Concrete Pipes as Made and Tested by U. S. Geological Survey.

staves will warrant it. A stop-valve should be used at the lower end of a wooden penstock, with no head-gate at the upper end, to insure the penstock being always full of water, which may be shut out by the use of stop-logs in case of necessity. A wooden penstock buried in the ground may be left empty for some time, without danger of the staves drying out.

"In heads of 200 feet and more in height, wooden penstocks are not economical, as the hoops require as much metal as the plates for a steel penstock.

"Penstocks constructed of concrete and steel also deserve a wide application, and should outlast both the steel and wooden penstocks, as the steel rods are protected by the concrete.

"Instead of welding together the ends of the embedded hoops, these ends may be run past each other for a distance of from 30 to 40 times the diameter of the hoop-rod; or an inch or so of each end of the hoop-rod may be bent back flat on itself, and the ends run past each other for a distance of from 20 to 30 times the diameter of the hoop-rod. For small concrete penstocks, steel wire wound spirally can be used to form the hoops.

"For heads of 200 feet and more in height, penstocks built of concrete and steel are not economical, as the hoops require as much metal as the plates for a steel penstock.

"Standpipes may be built either of steel plate or of concrete and steel. An excellent arrangement is to have a concrete base straddling the penstock, and the stand-pipe placed on top of this base, like a steel chimney or stack."*

Figs. 137, 138, and 139 are illustrations of wood stavepipes.

Figs. 140, 141, and 142 illustrate steel concrete pipes as made and tested by the United States Geological Survey.

Riveted-steel pipes are shown in several of the illustrations of power plants appearing in subsequent articles.

188. **Canals; Flumes.** The older method of applying the power of water to industrial purposes consisted in conducting the water by means of *canals* or *flumes* to the various establishments in which the power was to be utilized. This method required special physical conditions of a very favorable nature, in order that great expense in construction might be avoided. The later method is to convert the

*Thurso, "Modern Turbine Practice."

A. **Pipe under Test.**

B. **Method of Raising End of Pipe.**

Fig. 141. Reinforced-Concrete Pipes as Made and Tested by U. S. Geological Survey.

energy of the water into mechanical power, and the latter into a form suitable for transmission, in a single power-house, from which central station the power may be conveyed long distances, and applied to various machines located to the best advantage without reference to the waterfall itself.

Shafting, wire rope, compressed air, and water under pressure, have all been more or less utilized for the transmission of power;

Fig. 142. Construction of Pipe by Movable Form.
Courtesy of U. S. Geological Survey.

but, since the great development of electricity in comparatively recent times, this agent is most generally employed for that purpose.

It does not usually happen that a fall available for power production (either in natural fall, or one created by damming a stream) occurs in a single vertical plunge; usually there are several rapids of greater or less extent between the dam or actual waterfall and the proposed site of the power-house, necessitating the construction of a canal or flume of greater or less extent, if the entire fall is to be utilized. Even with a waterfall in which the total available head occurs in a single drop, a canal or flume is generally employed to convey the water to the several wheels or groups of wheels.

In a canal system, whether to be used for power purposes,

Fig. 143. Concrete-Lined Section of Truckee Canal, Nevada. Sixth section, looking upstream.

Fig. 145. Concrete-Lined Section of Truckee Canal, Nevada. Lining in rock cut, seventh section, looking downstream.

Fig. 144. Concrete-Lined Section of Truckee Canal, Nevada. Ninth section, rounding Wadsworth Point at mouth of canyon.

Fig. 146. Concrete-Lined Section of Truckee Canal, Nevada. Junction of earth and concrete-lined section, showing warped surface in latter.

Fig. 147. Santa Ana Canal, California.
Gravel concrete lining.

Fig. 148. Riverside Canal, California.
Sand deposit left on bottom after a year's service.
Courtesy of "Engineering News."

water supply, or irrigation, there are many important features requiring very special attention, some of which have already been referred to—such as the *headworks*, with the corresponding *regulator-* or *head-gates* for controlling the supply of water into the canal head; the *diversion dam* or *weir*, of greatly varied construction, to raise the level of the water adjacent to the headworks and thus induce a proper flow into the canal; *escape-heads* (*wasteways*) and their gates, to empty

Fig. 149. Bear Valley Canal. Redlands. California. Boulder-lined and plastered.
Courtesy of "Engineering News."

the canal quickly in case of accident or danger, or to dispose of surplus water in times of flood or excessive rains; *sand-gates*, for scouring out deposits of sand or silt; *vertical falls* and *inclined chutes*, to compensate for excessive grade; and other features controlled by local conditions. Such features are treated in detail in connection with the subject of Irrigation.

The phenomena of erosion of bed and banks, of sedimentation of suspended matter, of capacity and velocity of flow with various cross-sections, and influence of kind of lining, together with their inter-

Fowler Switch Canal, California, Showing Effect of High Velocities.

Bear River Canal, Utah, Looking North.

Fig. 150. Views of Western Irrigation and Power Canals.

Courtesy of U. S. Geological Survey.

Fig. 151. Tunnel of Bear River Canal, Utah.
Courtesy of U. S. Geological Survey.

Fig. 152. Santa Ana Canal (Cement-Lined), California; Capacity, 240 Second-Feet.
Courtesy of U. S. Geological Survey.

Fig. 153. End View of Sterling Flume in Provo Canyon, Utah.
Courtesy of U. S. Geological Survey.

Fig. 154. Flume of Kern Valley Power Development Works. California.
Courtesy of U. S. Geological Survey.

Fig. 155. Flume at Sanger. California.
Courtesy of U. S. Geological Survey.

IMPROVEMENT OF THE MISSISSIPPI RIVER, MEMPHIS REACH

Old-style revetment of upper bank, completed except ballasting.

Fig. 156. Flume across Pecos River.

Courtesy of U. S. Geological Survey.

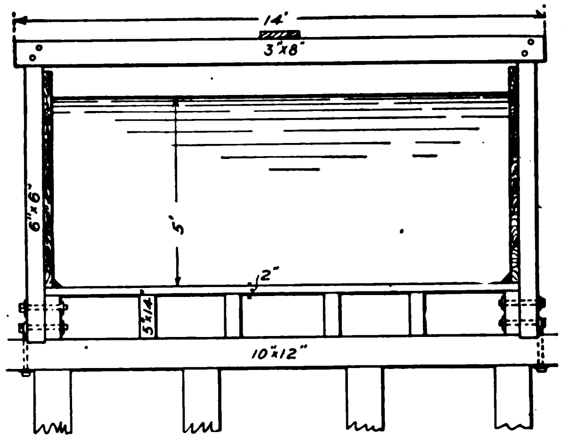


Fig. 157. Cross-Section of Flume Resting on Piles.

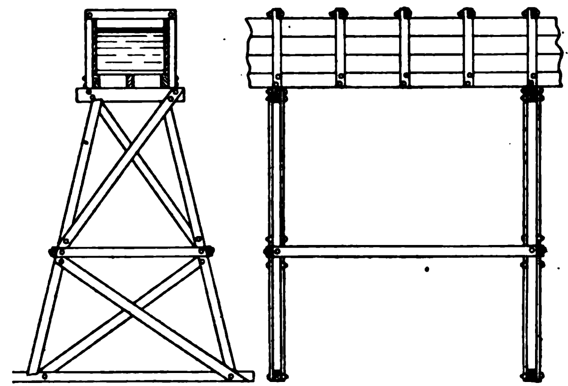


Fig. 158. End and Side Elevations of Flume on Trestle.

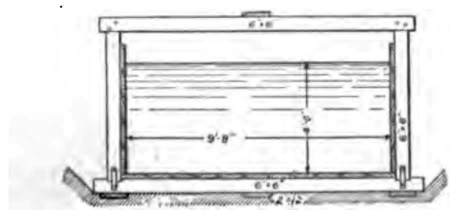


Fig. 159. Flume on Steep Hillside.

Fig. 160. Wasteway No. 1, Truckee Canal, Nevada. Showing mechanism for operating Taintor gates.

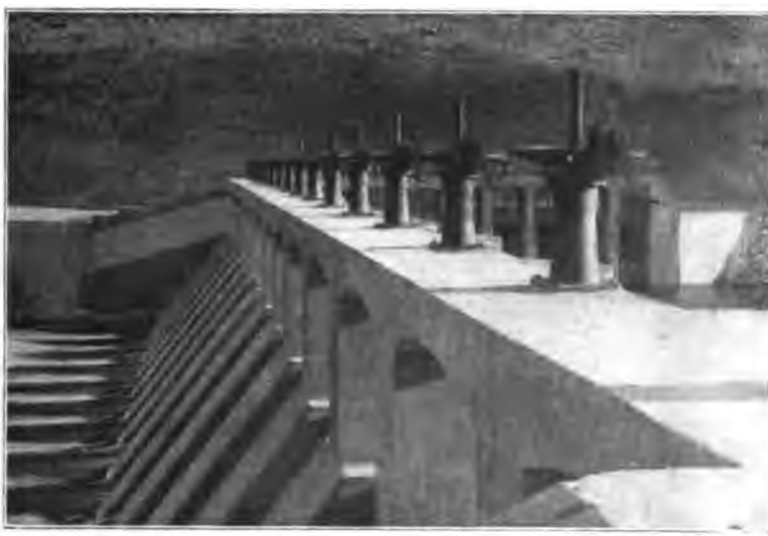


Fig. 161. Diversion Dam and Gates, Heading of Main Truckee Canal, Truckee-Carson Project, Nevada. View looking south along the dam, showing front face of dam; also the gate-operating mechanism.

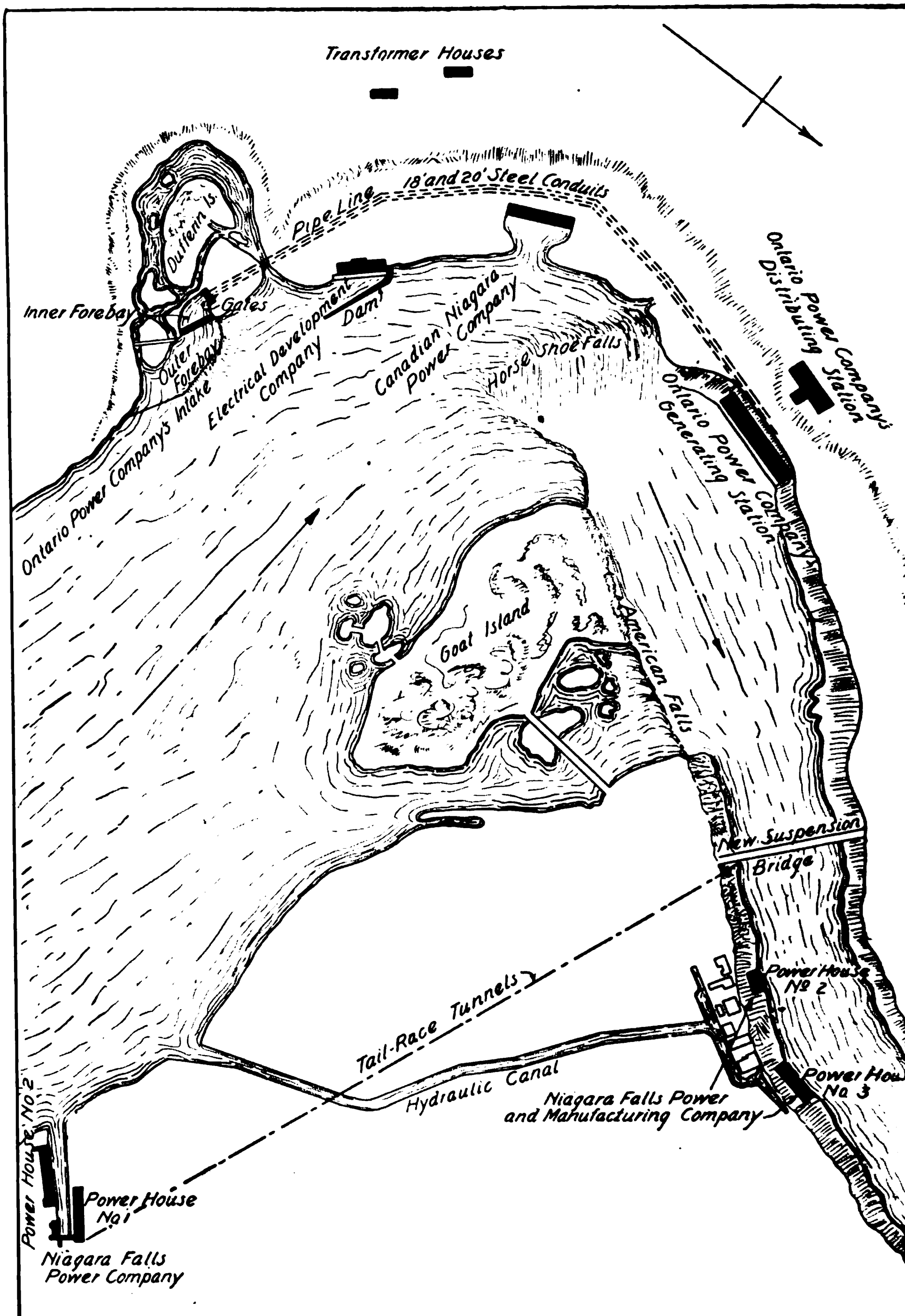


Fig. 162. Map of Niagara Falls and Vicinity, Showing Location of Power-Development Enterprises

relations and mutual dependence, are discussed in connection with Hydraulics and Water Supply, and will not here be considered further than to present some typical illustrations (see Figs. 133 to 147).

HYDRAULIC POWER INSTALLATIONS

189. **Niagara Falls.*** Along the boundary between Canada and the United States, there exists a chain of great lakes having a surface area of some 90,000 square miles, which receive the drainage from a catchment area of about 240,000 square miles. Between Lakes Superior and Huron there is a drop in elevation of the water surface of about 18 feet at the Sault Ste. Marie; between Lakes Erie and Ontario, which are connected by the Niagara River, there is a total drop in elevation of 326 feet. The distance between these two lakes is about 30 miles, but almost the entire fall occurs in the last 15 miles. There is a fall of about 56 feet in the rapids above the Niagara Falls; about 160 feet at the Falls; and about 110 feet below the Falls. The entire drainage of the upper lakes flows from Lake Erie through the Niagara River into Lake Ontario, and thence, by the St. Lawrence River, into the Atlantic Ocean.

These lakes form great natural storage reservoirs, so that the volume of flow and the levels in the Niagara River are remarkably uniform. In extreme cases the river level above the Falls varies $3\frac{1}{2}$ feet, the variation being chiefly due to wind holding back the outflow from the lakes. Below the Falls, the river level varies at most 15 feet, due chiefly to ice-blocks formed in the lower river.

The minimum flow of the Niagara River, as given by the government engineers, is 178,000 cubic feet per second; the mean flow is 250,000 cubic feet per second. The minimum flow, with the total fall of 326 feet, represents about 6,600,000 gross horse-power; with a fall of 216 feet (*i.e.*, from the upper river to the foot of the Falls), this would be about 4,360,000 gross horse-power. With the mean flow of 250,000 cubic feet per second, the corresponding figures would be about 9,290,000 and 6,156,000.

The Falls comprise the Horseshoe Fall, about one-third of a mile wide, on the Canadian side; and the American Fall, about 600 feet wide, on the opposite side, the two being separated by Goat Island.

*Proceedings of the Institute of Mechanical Engineers, Feb., 1906. Also *Engineering Record*, Jan., 1900; Nov., 1901; Nov. and Dec., 1903; Feb. and Oct., 1904; Apr., 1905.

Below the Falls, the river flows through a gorge or ravine 600 to 1,200 feet wide, and 200 to 300 feet deep, eroded by the action of the river itself. Fig. 162 represents the conditions and locations of the various installations.

The importance of the Falls as a source of energy was recognized from an early period. The first important effort to obtain power was made in 1853, when construction on the so-called "Hydraulic Canal," 36 feet in width, 8 feet in depth, and 4,400 feet in length, was begun from a point above the upper cataracts to a basin at the top of the bluff, located on the side below the Falls. This canal was completed in 1861. On the bluff were constructed mills, having turbines supplied with water from the basin and discharging it through short tunnels on the face of the bluff. In these cases, only part of the available fall was utilized, water being plentiful and the cost of excavating pits for the turbines considerable. In 1885, about 10,000 horse-power was utilized in this way, or the whole available supply of the hydraulic canal as then constructed.

190. **Niagara Falls Power & Manufacturing Company.** In 1877 the Hydraulic Canal and all its appurtenances were purchased by the present owners. In 1892 the company commenced an enlargement of its canal, and it has made notable improvements from time to time. The plan adopted at that time was to widen the original channel to 70 feet, and to make the new part 14 feet deep, thus providing an additional capacity of about 3,000 cubic feet of water per second, giving a surplus power, after supplying the old leases, of about 40,000 horse-power. In 1895–96, a second power-house was erected for the purpose of supplying power to customers. For this new plant a branch canal was taken to a forebay, 30 feet wide and 22 feet deep, near the edge of the bank. From this forebay, penstock pipes of flange steel, 8 feet in diameter, conduct the water down over the high bank a vertical distance of 210 feet, to the site of the power-house on the sloping bank at the edge of the water in the lower river.

The first portion of the power-house, 60 by 100 feet, was completed in 1896. Because of the fluctuations of the water in the lower river, it was necessary to place the floor of the station on which the generators stand, about 20 feet above the ordinary water level. As it was desired to couple the generators directly to the ends of the water-wheel shafts, it was necessary to place the water-wheels also at this elevation, and to employ draft-tubes, in order to obtain the full head available. It was also required that the wheels should run at a given speed suited to the speed desired for the generators. To fulfil all these conditions, turbine wheels mounted on horizontal axes were adopted. The specifications for these wheels required that each should furnish 1,900 horse-power, measured on the shaft of the wheel, when run at a speed of 300 revolutions per minute. The head under which the wheels work is generally 210 feet; but they were required to have sufficient capacity to deliver 1,900 effective horse-power under a head of 205 feet; and all parts were to have sufficient strength to withstand the pressure due to a head of 220 feet without undue strain. They were required to show a percentage of useful effect of not less than 78 per cent, at any point between full and three-quarters discharge, under any head from 205 to 225 feet and running at a constant speed of 300 revolutions per minute; and not less than 60 per cent under the same conditions, from three-quarters to one-half discharge.

The apparatus for regulating the speed of the wheels consists of a hydraulic piston which applies its force in either direction to a rack connected with a pinion in the gate-rigging of the turbine. The force which operates the hydraulic piston is air, compressed under about fifteen atmospheres. This compressed air is contained in a cylinder, and the pressure is maintained by a pump which constitutes a part of the machine. The machine is provided with a high-speed ball governor actuating a balanced piston-valve. The governor has an "anti-racing" appliance by which the governing machine is checked before it has carried the gate too far in either direction.

The second section of this power station was completed in 1900, making the present size 120 by 100 feet. This portion of the station contains five turbines, each having a capacity of 2,500 horse-power (Fig. 163). They are fed by a new 11-foot penstock consisting of a vertical portion about 200 feet high, with an arm on each end, that at the top having a length of about 68 feet, and that at the bottom about 115 feet. The penstock is built up in sections of 5 feet, the sections lapping inside and outside alternately. The thickness of the plates varies from 5-16 inch at the top, to $1\frac{1}{8}$ inches at the bottom. Most of the sections are made up of the two plates. The thinner plates have lap joints with two rows of rivets; while the thicker plates have butt joints and double cover splice-plates with three rows of rivets on each side.

A vertical recess about 15 feet square and 50 feet high was cut out of the solid rock at the base of the cliff, and in it was set the bottom

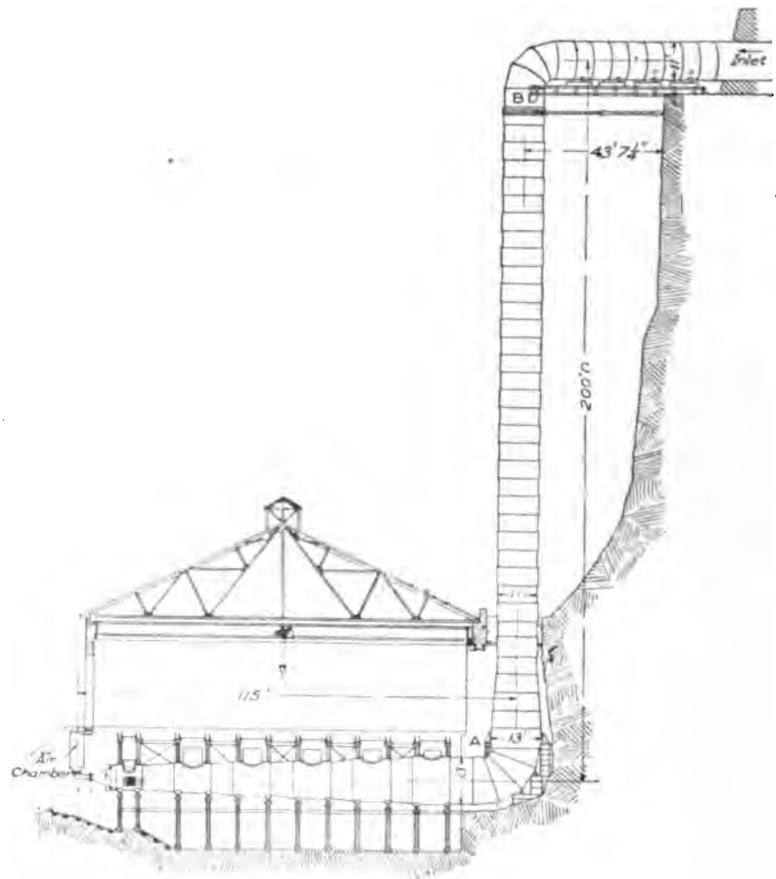


Fig. 163. General Dimensions of Penstock of the Niagara Falls Hydraulic Power & Manufacturing Company.

of the vertical portion of the penstock. At the lower elbow, the penstock increases to 13 feet in diameter, and then decreases, as it passes the turbines, to a diameter of 7 feet at the outer end. The upper end has a bell-shaped inlet, 22 feet wide, built into the masonry of the forebay at an oblique angle. Heavy cast-iron brackets are riveted to the top of the vertical portion of the penstock on each side, and support one end of a pair of plate-girders 30 inches deep and 49 feet long, which have 8-inch transverse I-beams across their top flanges to support the horizontal portion. About 6 feet below the bottom of the plate-girders, the penstock is encircled by a pair of bent 10-inch

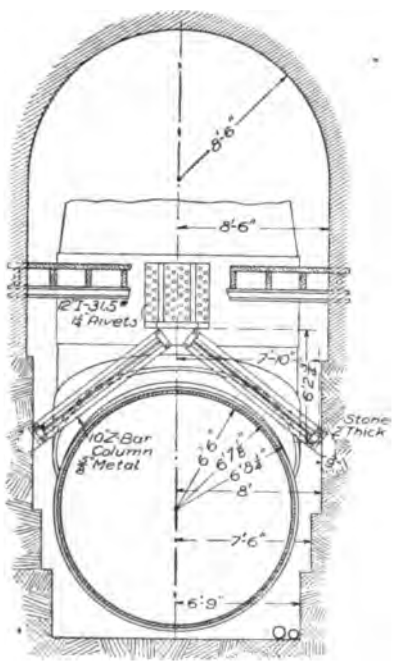


Fig. 164. Penstock Support, Plant of Niagara Falls Hydraulic Power & Manufacturing Company.

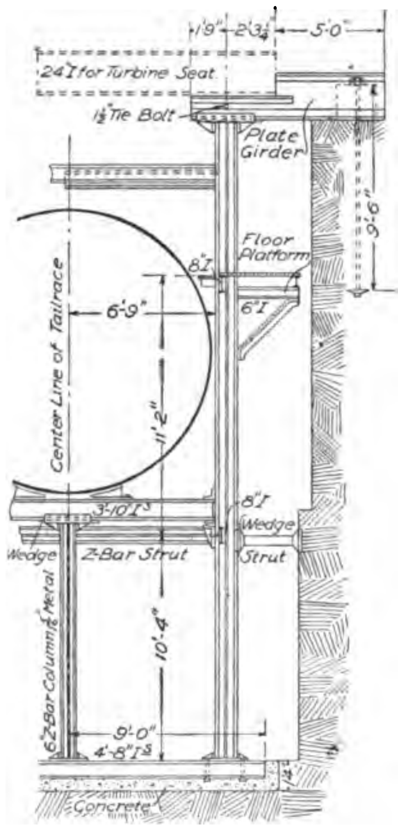


Fig. 165. Iron Work in Tail-Race, Plant of Niagara Falls Hydraulic Power & Manufacturing Company.

I-beams with horizontal webs, between which plates are riveted to afford a pin connection for two sets of eye-bar anchors which guy the penstock horizontally to 2-inch eye-bolts drilled and cemented into the side of the cliff. At the lower elbow, the horizontal end of the pipe is seated on a bed of cut-stone masonry, and is bedded in cement mortar. The convex side rests on flanged cast-iron angle-blocks riveted to the pipe and seated on stone piers. Large right-angled brackets of cast iron, reinforced by heavy angles, are riveted around the pipe just above the elbow; and the one on the convex side rests on solid masonry, while the one on the concave side is supported by two inclined braces whose lower ends rest on castings let into the vertical rock wall.

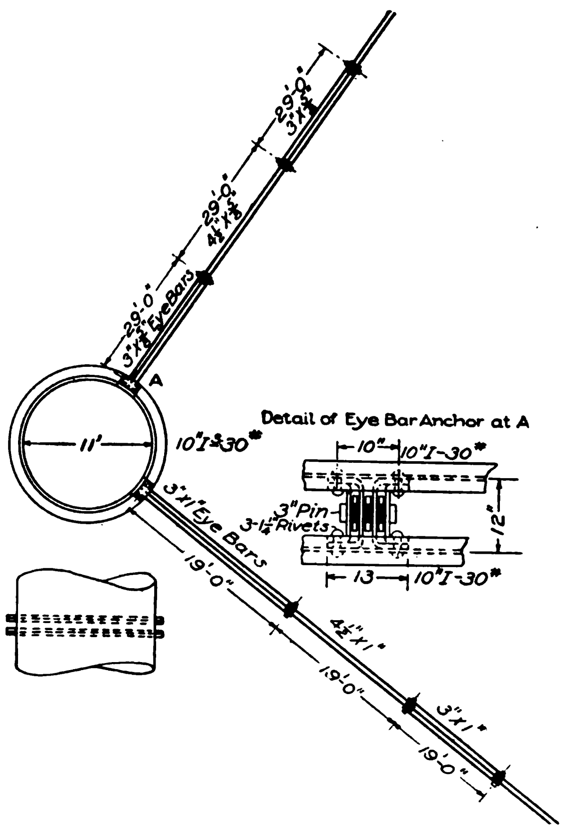


Fig. 166. Anchor at Top of Penstock, Plant of Niagara Falls Hydraulic Power & Manufacturing Company.

The tail-race is a rock cut 17 to 21 feet wide and 19 feet deep below the center line of the lower arm of the penstock, which is supported on steel columns. The lower horizontal part of the penstock is made in sections which have alternate parallel and tapered sides. The former are uniformly about 10 feet long, and are fitted on top with vertical branches 5 feet in diameter. These branches are flanged to the inside of the penstock and riveted to it and to outside reinforcement collars, which are 8 inches wide and 2½ to 3¾ inches thick. A cross-section near the end of the penstock shows the knee-brace supports of the beam platform, the suspension rods by which the penstock is tied to the beams each side of the vertical branch, the anchorages of the double draft-tubes to the masonry, and the transverse and longitudinal adjustable rods with which they are tied together just above high-water level. See Figs. 164, 165, 166, and 167.

The valve-gates are set horizontally, with their axes inclined 45 degrees to that of the penstock, and are operated by pneumatic pressure. The interposition of the tapered sections between the turbine connections reduces the diameter of the penstock in proportion to the diminished flow of water required as the successive turbines are passed, and brings it down to 7 feet at the last turbine, which is supplied through two side tubes. At this point the penstock is guyed laterally by four 5 by 1-inch horizontal eye-bars on each side, which are anchored to steel bars drilled and cemented into the solid rock. The penstock terminates here with a conical section 7 feet long, which tapers to 18 inches, and connects by a cast-iron elbow with a vertical air chamber about 11 feet in extreme length, and 4 feet in internal diameter. It has an air-valve on top, and gauge-glasses on the side. The tapering portion is fitted with thirty spring relief-valves, set in three groups, having 6-inch openings and set to open at a pressure of 100 pounds per square inch (Fig. 168).

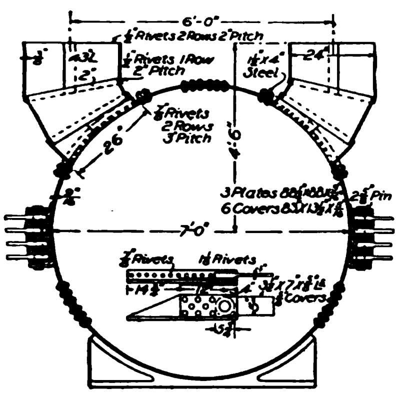


Fig. 167. Anchor Connections, End of Penstock, Plant of Niagara Falls Hydraulic Power & Manufacturing Company.

Fig. 168. Relief-Valves on End of Penstock.

The turbine wheels are made of bronze, and are located in the draft-tube casing, one on each side of the casing proper; each pair weighs about 5,000 pounds. From the sides of the turbines, the discharge pipes

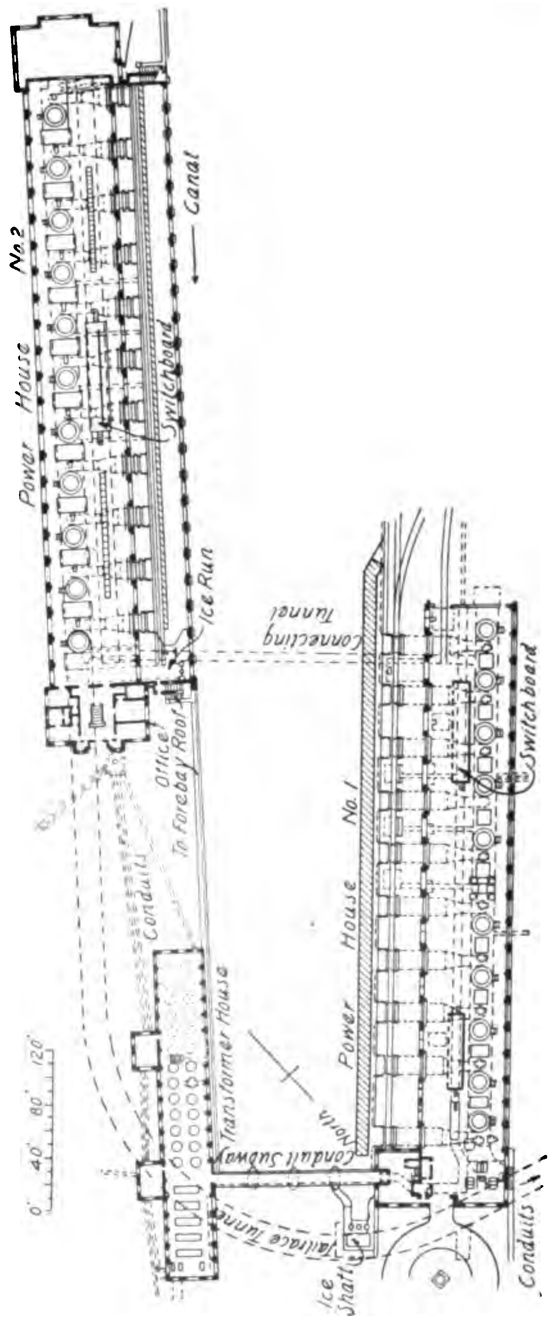


Fig. 169. Plan of Power-Houses, Canal, and Upper End of Tail-Race Tunnel of the Niagara Falls Power Company.

project laterally and then downward to connect with the draft-tubes, which are 22 feet 8 inches long. A third power-house capable of developing 100,000 horse-power, also situated at the foot of the bluff, completes the installation of the Niagara Falls Power & Manufacturing Company.

191. **Niagara Falls Power Company.** This company began work in 1890, and first delivered power in 1895. The installation comprises two power-houses. Power-House No. 1 has ten 5,000 horse-power units, each unit consisting of twin outward-flow reaction turbines with vertical shafts, on the

upper ends of which are placed the electric generators. Power-House No. 2 is in its main features similar to Power-House No. 1, but it contains eleven 5,500 horse-power simple inward-flow turbines with draft-tubes.

The engineering history of the power company began in 1889, when the Cataract Construction Company proposed to divert water from the upper river into an open canal at a point somewhat over a mile above the Falls, to deliver the water to wheels in a pit at the side of the canal, and to conduct the water from the wheels to the river below the Falls through a tunnel 7,000 feet long, driven through the rock at a distance of nearly 200 feet beneath the city of Niagara Falls.

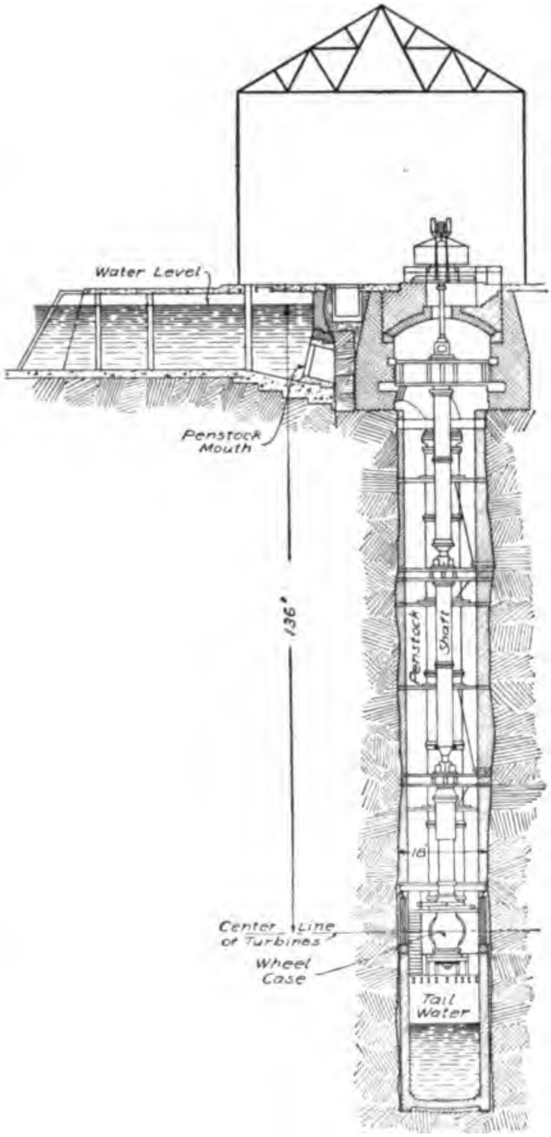


Fig. 170. Transverse Section of Wheel-Pit, Power-House No. 1, Niagara Falls Power Company.

192. *General Considerations.* The canal is located on the New York shore about 1¼ miles above the crest of the American Fall, at a point where the mean water elevation is 561.2 feet. The discharge

portal of the tunnel is about 1,100 feet below the American Fall, at the mean elevation of 343.4 feet. This makes a gross head of 217.8 feet. The canal, which was excavated to give an average depth of 12 feet of water, and projected 600 feet beyond the original shore line into the river with embankments formed from the excavated material, has now a length of about 1,500 feet, and a width of 180 feet at the mouth, and 100 feet at the inner end. The wheel-pits—in which are located the penstocks conducting the water to the turbines, and the shafting connecting generators with turbines, and over which are built the power-houses—are located on opposite sides of the canal, as shown in the accompanying plan (Fig. 169). The pit for Power-House No. 1 is 178 feet deep, 18 feet wide and 425 feet long, and houses 10 units of 5,000 horse-power (Figs. 170 and 171); the pit for Power-House No. 2 (Fig. 172) is 178.5 feet deep, 20 feet wide, and 468 feet long, and accommodates 11 units of 5,500 horse-power. In both cases, about 27.2 cubic feet of wheel-pit excavation were made per horse-power developed.

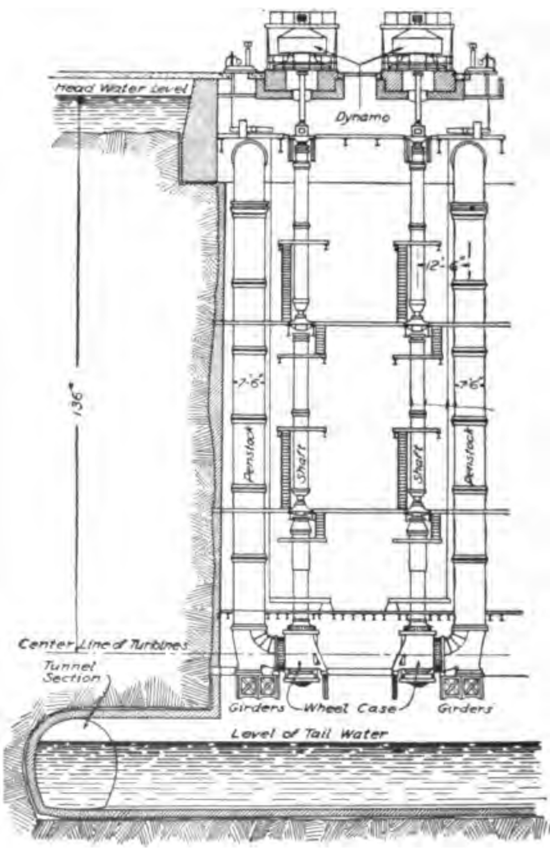


Fig. 171. Longitudinal Section through Portion of Wheel-Pit of Power-House No. 1, Niagara Falls Power Company.

Each of the ten turbines of the first plant passes an average amount of 430 cubic feet of water per second; and each of the second

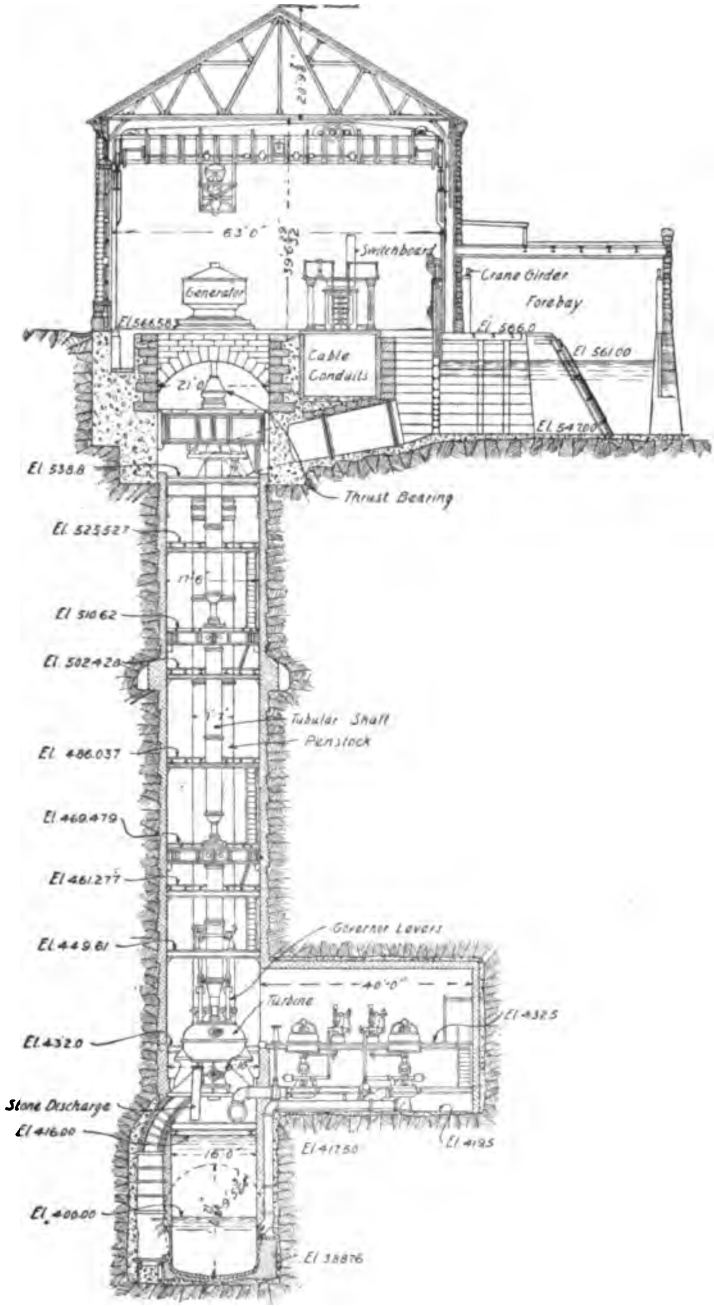


Fig. 172. Cross-Section of Power-House No. 2 and Wheel-Pit, Niagara Falls Power Company.

Fig. 178. Interior of Forebay of Power-House No. 2, Niagara Falls Power Company, Niagara Falls, N. Y. Showing racks protecting entrance to turbine chambers.

IMPROVEMENT OF THE MISSISSIPPI RIVER

Lock construction at Moline, Ill. View of wing dam under construction, looking east from cofferdam.

plant, 445 cubic feet. Each unit being supplied from a 7½-foot penstock, the velocity in the penstock is 9.75 feet in the first case, and 10 feet in the second. With 10 wheels in operation in each plant at one and the same time, the total quantity of water to be taken through them is over 10,000 cubic feet per second, equivalent to a velocity in the canal at its entrance of 4 feet per second. The tail-race tunnel, through which all the water must pass, has a maximum height of 21 feet, and a width of 18 feet 10 inches, and a net cross-sectional area of 335 square feet. The mean velocity through it with the given quantity of water is, therefore, about 29 feet per second. The tunnel being about 7,000 feet long, a considerable portion of the gross head must necessarily be employed in effecting the removal of the water at the required rate; it was accordingly given a slope which averages 6 feet in 1,000, placing the floor of the tunnel at the wheel-pit at the elevation of 387 feet, or 43.6 feet higher than the river level at the portal. A further reduction due to the location of the turbines in the first plant about 37 feet above the wheel-pit bottom (no draft-tubes seeming applicable at the time), together with a minor loss in the flow through the canal and intake racks, reduced the gross head to 136 feet. In the second plant, the effective head based on the use of draft-tubes was taken at 145 feet.

193. *The Canal and Forebays.* The canal was excavated to allow a depth of 12 feet at low water, as already stated. The walls of the canal are of solid masonry 17 feet high, 8 feet thick at the base and 3 feet at the top, laid in an ordinary Portland cement mortar composed of 1 part cement and 2 parts sand. Besides the two power-houses which have to be supplied with water, the canal is also tapped for a supply to a separate wheel-pit owned by the International Paper Company, whose property adjoins that of the Niagara Falls Power Company. Hydraulic power is sold in this case, as, at the inception of the work, electric power could not be furnished in time; and the power company provides for the disposal of the water through a 7-foot tunnel running from the wheel-pit of the International Paper Company to the main tail-race tunnel.

The points of intake to the power-houses, or outflow from the canal, are distributed in two groups to avoid local high currents as much as possible, the two power-houses being, partly for this reason, located diagonally across the canal. There is a separate opening for each penstock in each of these power-houses, but the intakes are materially different. Racks of the usual flat-iron bar construction guard the entrances to the penstocks; but in the later installation they are enclosed by a portion of the power-house, in a covered forebay (Fig. 173). While the extra cost of a covered forebay is considerable, its provision was deemed advisable as a means of fighting ice. Under ordinary circumstances, there is a floating timber boom extending across the mouth of the canal; but cakes of ice, especially with a large flow of water, find their way under the boom and into the canal. During the present year the boom has been located a considerable distance in front of the canal, to divert ice more readily toward the Falls.

Fig. 174. Arched Entrances to Inner Forebay of Power-House No. 2, Niagara Falls Power Company. Entrances in lower wall, submerged when in operation.

Much trouble used to result from allowing floating ice to enter the canal, and a large corps of men was constantly required in cold weather, several shifts per day, cleaning the racks at Power-House No. 1. These racks consisted of three parts—a bottom section; a top section, dropping from the top of the canal wall to a level about one foot below water; and a middle, removable section entirely submerged, which could be temporarily hauled out of position for purposes of cleaning, etc.

Though a secondary boom was provided, extending into the canal a few feet in front of the rack, and across the entire series of openings, cakes of ice would succeed in getting under this boom, and would frequently also be drawn under the top section of the racks, owing to the suction of the inflowing water. As the wheels of the first power-house were not designed to allow for an accumulation of ice, as will be explained, it can readily be seen that the passage of much ice into any penstock was likely to be a serious affair.

It was these general considerations that affected the design of the wheels of the second power-house, and the arrangement of the intake at that building. Here the secondary boom is replaced by the outside wall of the covered forebay, the water being admitted into the forebay through arched openings (Fig. 174), the crowns of the arches being about 4 feet below normal water level, this distance being assumed great enough to prevent blocks of ice diving through the arches.

In the design of the new power-house, moreover, an ice-run has been provided; and blocks of ice collecting in the canal are directed toward the run, and drawn by the current of water in the ice-run down a shaft and into the tail-race tunnel. In a similar way, there is now an ice-run for Power-House No. 1, secured by utilizing a shaft that had been sunk in connection with the driving of the extension of the tunnel to the second power-house. It will thus be seen that much has been done to overcome the ice troubles, and that in this connection the tail-race tunnel has to take care of the additional amount of water, if it is necessary to prevent any great lodgment of ice in the intake canal.

194. *General Comparison of the Two Plants.* A comparison of the design of the two stations can be made from the accompanying cross-section sketches. The main difference is the covered forebay in the case of the new plant, as already stated. At the normal level of the headwater, the width of the power-house is the same in each case, about 110 feet; but in the older plant, the outlying position of the racks makes the total width at the foot of the racks somewhat greater. Otherwise the general features of the two plants are the same; the width of the two buildings is in each case about 70 feet, their height to the ridge of the roof, 60 feet; the distance from floor to under side of the roof trusses, 40 feet; the width of the finished wheel-pit, 17 feet 6 inches; and the general arrangement of the main machinery the same. Each turbine in both plants is supplied from a 90-inch penstock, and drives a long vertical shaft, which is a tubular affair direct-connected to the electric generator shaft in the power-house.

The notable difference in the hydraulic-machinery exists in the turbines, which are of the inward-flow type in the new plant, and of the outward-flow type in the old; in the thrust bearings, which are of a disc type in the new, as against the usual collar type in the old; and in the exciter equipment, which comprises a group of dynamos direct-connected to turbines in an underground chamber in the new plant, as compared with direct-connected units having the exciters in the power-house on the top of long shafts, and the exciter turbines in the bottom of the wheel-pit, in the old plant. While the governors are different in the two plants, the scheme of levers, rods, and counter-weights of the governing system is essentially the same.

195. *Intake of Power-House No. 2.* The arches through which the water enters the forebay of the new plant from the canal are sprung from 5 by 6-foot piers, spaced 20 feet apart, and leaving openings 14 feet wide and 10 feet to the crown of the arch. This gives an area of flow of about 125 square feet; and as there are two arched openings per unit, the inlet area is 5.7 times that of the penstock, not allowing for the water taken from the forebay for the exciter turbines and other purposes. Immediately inside the arches are grooves on each side for stop-logs for shutting out water from the forebay at any time. A short distance beyond these, the water passes through the racks, which are of the usual construction, inclined 30 degrees with the vertical. The rack bars are of mild steel, 3 by $\frac{3}{8}$-inch in size, separated by pieces of $\frac{3}{4}$-inch gaspipe into spaces 1 13–16 inches wide. They are built in three sections as regards height—the upper, with rounded top, being about 8 feet long and extending 3 feet below normal water level; the middle, 10 feet long; and the bottom 2 feet 10 inches long. The center sections are removable, sliding between I-beams and channels; and for handling them, there is a 5-ton Niles electric crane traveling the entire length of the forebay. The rack structure is supported on 15-inch I-beams on 8-foot centers, and the bars are divided into two groups between them. The total wetted area of openings between the rack bars is substantially equal to that of the aggregate area of the arched openings. Between the racks and the head-gates, which are in the power-house proper, are two sets of grooves in the buttresses against which the racks are supported. These allow for closing the inlet to the corresponding penstock with stop planks, in the event of a need for repairs to the gate.

196. *Head-Gates.* The gates in the new power-house, like those in Power-House No. 1, may be operated both by hand and by electric motor. They are lifted by screws; but the new gates are provided with wicket-gates for passing the water into the penstock and relieving pressure before opening, and are not provided with the roller bearings used instead in the old plant.

Ice-Runs. The ice-run from the canal at Power-House No. 2 is also provided with gates, so that no water may be wasted in this way during summer, as it is undesirable to have more water than necessary discharged into the tail-race tunnel. The ice-run has direct connection with the canal, and also with the covered forebay, the latter inlet to the ice-run being provided to take care of any ice which may succeed in entering the forebay. These gates are also designed to be lifted either by motor or by hand. They are of the lifting-screw type, and the motor drives a horizontal shaft which carries at each end a pair of bevel gears. The horizontal gear in each case revolves on ball bearings, and the screw is lifted and lowered through it.

197. *Penstocks and Wheel-Pits.* The intake proper to each penstock, that guarded by the head-gate, is 14 feet wide, and under normal conditions carries 14 feet of water. The mouthpiece to the penstock starts a few feet back of the gate, with a flaring elliptical entrance, and, pitching 2 inches in a foot, joins the 90-inch circular penstock 14 feet beyond. The elliptical entrance is 12½ feet wide and 8½ feet high. The inclined portion of the penstock, and the mouthpiece, are bedded in concrete, and are arched over by brick-work which supports the substructure of the power-house proper. The junction between the walls of the chamber behind the gate and the penstock mouth is made with cement mortar, and the joint between the inside masonry wall of the chamber and the brick wall of the cable conduit behind is waterproofed with asphalt.

From the mouthpiece to the turbine, the penstock comprises a short length of straight riveted-steel pipe; a riveted-steel elbow at the top of the wheel-pit; six vertical sections of straight pipe, each 15 feet 5½ inches long and 90 inches in inside diameter; and a cast-iron elbow at the bottom connecting into the turbine. The upper elbow is provided with cast-iron brackets, and the portion of the penstock above it is supported from it by the brackets, which bear on steel beams bridging the wheel-pit. The rest of the penstock is carried by the cast-iron elbow at the bottom. There is a stuffing-box of cast iron in the pipe immediately below the upper elbow, to allow for expansion; it consists of two rings of cast iron bolted to the abutting ends of the penstock, one sliding on the other, as shown in an accompanying detail (Fig. 175). The stuffing-box was required to stand a hydraulic test of 25 pounds. An idea of the size of the bottom supporting elbow of the penstock can be obtained from the fact that the metal is 2¼ inches thick, and it is reinforced by circumferential ribs 4 by 4 inches in size.

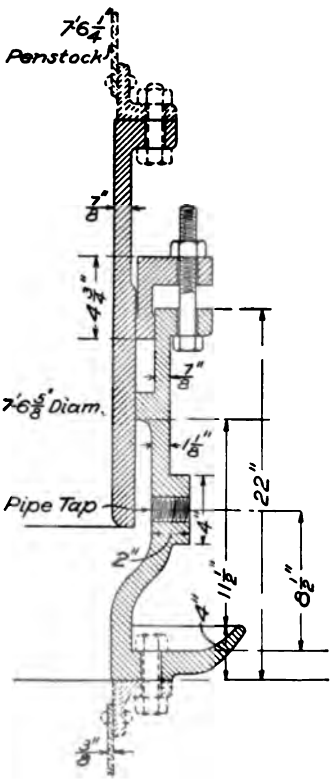


Fig. 175. Penstock Expansion Joint, Niagara Falls Power Company.

The steel of the penstock is ⅜ inch thick in the upper part, and ½ inch in the lower; and each section has two 18-inch manholes. There is a series of floors or decks and intermediate platforms in the wheel-pit, as indicated in the drawings, and the penstock and shaft are thus accessible throughout their height. The pit is lined with brick throughout, and cast-iron brackets are laid in the walls to support the various beams. Drainage pipes emptying into the tail-race were provided behind the walls, laid in broken stone, wherever ground-water was likely to collect.

198. *The Tail-Race Tunnel.* The tail-race tunnel, from a point near the wheel-pit of Power-House No. 1 to the portal at the lower river, was built in a straight line for construction and hydraulic reasons. Surveys for this purpose, which were started in the latter part of March, 1890, marked the beginning of actual work on the Niagara Falls Power Company's plant. The work was executed from the discharge portal, and from two shafts which were sunk at points 2,600 and 5,200 feet from the portal.

The tunnel is lined throughout with brickwork, partly on account of the poor character of the rock, and also because of the decreased friction to the flow of water thereby secured. The invert was laid last, and has a face of vitrified paving brick. The sides and arched top are usually four rings of brick 16 inches in thickness, but are sometimes six and eight rings thick; and the space behind was filled with rubble masonry. The bricks were laid in a mortar of 1 part Portland cement to 3 of sand, except where the flow of water is very great, in which case the proportions are 1:2, and in some cases 1:1.

One of the most interesting points in connection with the tunnel is the provision of an ogee discharge. It provides for lowering the grade of the invert about 11 feet below the average low water of the river, to allow one-half or so of the flow from the tunnel to discharge below the surface. The ogee surface starts at a point 95 feet from the portal, and the 10½-foot drop takes place in this distance. This portion of the tunnel, to the elevation of the spring line, is lined with steel boiler-plate riveted to steel ribs 3 to 4 feet in depth, which are bedded in Portland cement concrete. For the last 25 feet of the tunnel, granite masonry is used instead of the brickwork, and the arch and the face of the portal are also of granite masonry, carried to 38 feet below the water surface to a ledge of white sandstone.

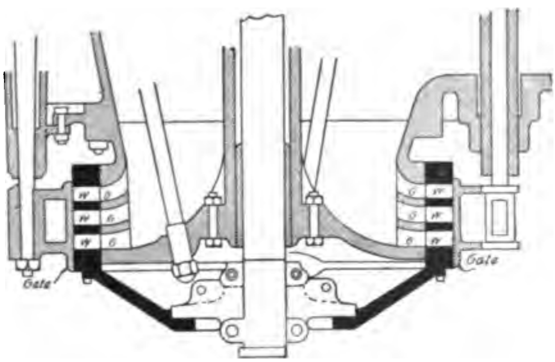


Fig. 176. Lower Part of 5,000-Horse-Power Turbine, Power-House No. 1, Niagara Falls Power Company.

199. *The Turbines.* The distinct difference in the turbines of the two plants is that in the new plant the turbines are of the single, inward-flow or *Francis* type, while those of the old plant are of the twin outward-flow or *Fourneyron* type, each wheel divided into three parts by horizontal partitions. In the former, as already stated, the design has been made to utilize draft-tubes, while in the older machines the water was discharged freely into the air directly above the

tail-race. The new wheels were designed by Messrs. Escher, Wyss & Company, to give 5,500 horse-power each, under a head of 145 feet at a speed of 250 revolutions, this capacity of the turbine to provide for overload in a 5,000-horse-power generator. The wheels of the

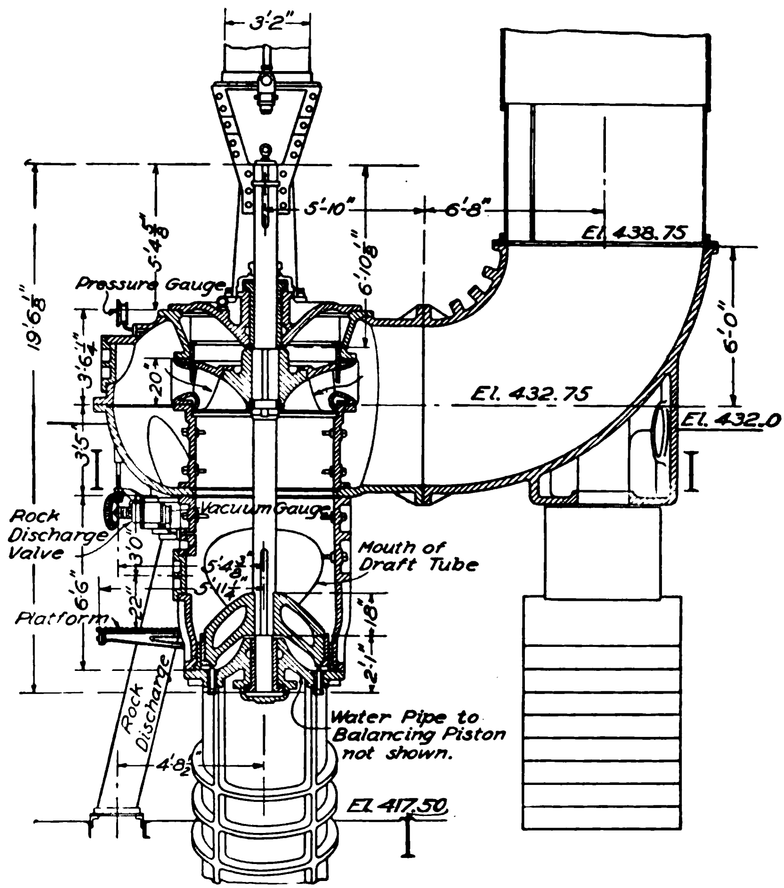


Fig. 177. Detail of 5,500-H.P. Turbine in Power-House No. 2, Niagara Falls Power Company.

first power-house were designed by Messrs. Faesch & Piccard (now Piccard & Pictet), of Geneva, Switzerland; and, under a head of 136 feet (the distance from the surface of the headwater to the center between the upper and lower wheels of the turbine), give 5,000 horse-power, corresponding to a discharge of 430 cubic feet per second at an efficiency of 75.5 per cent.

In both cases there is a balancing disc or piston which may take up practically the entire weight of the revolving parts; and there is a thrust bearing near the top by which the revolving parts are hung, this bearing taking the difference between the total weight of the rotating parts and the thrust on the balancing disc. In the first plant the balancing piston is above the upper of the twin turbines, and is in communication with the turbine chamber; but in the second plant a special piston below each of the single turbines is provided, and there is a separate supply of pressure water for this purpose. In the old plant a type of step or collar thrust bearing is used, but this has required careful watching whenever the upward water-pressure thrust is reduced. The bearings for the new plant are of the disc type. The chief source of trouble with the step bearings in the old plant arose from a diminution of the upward thrust on the shaft, due to a throttling of the supply of water in the penstock by the collection of ice in it. It was this source of trouble that led to a design of balancing in the new plant from a separate, positive supply of water. In the new wheels there are 25 blades in the fixed or guide wheel, and 21 blades in the running wheel, which is of manganese bronze cast in a single piece and weighing nearly 4,000 pounds. It has a diameter of 5¼ feet; but the total diameter of the turbine, which should include the casing around the guide-wheel, is 12 feet. The guide-wheel of the Faesch & Piccard turbine has 36 buckets, and the turbine wheel 32, of bronze cast solid with the rim. The outside diameter of the wheel of this turbine is about 7 feet.

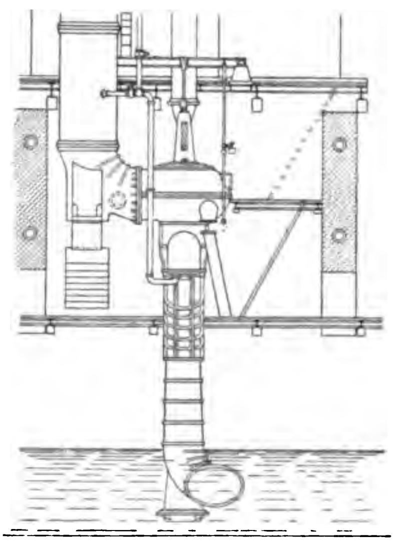


Fig. 178. Arrangement of Turbines and Draft-Tubes, Power-House No. 2, Niagara Falls Power Company.

200. The accompanying drawings (Figs. 176 to 182) will serve to show some of the details of the design of the wheels in both power-houses. In the first three units installed in Power-House No. 1,

the cast-iron elbow at the foot of the penstock was carried on built-up girders, these in turn resting on brickwork walls 2 to 2½ feet thick; but the vibration to which they were subject was found so great that the succeeding units in Power-House No. 1, and those in Power-House No. 2, all have the cast-iron penstock elbows provided with feet cast solid with the elbows, and seated on brackets embedded in the walls of the wheel-pit. This plan of bridging the tail-race is necessary in order that the bottom of the wheel-pit shall be as little obstructed as possible; and on this account, also, the draft-tube from each turbine in Power-House No. 2 bifurcates, the two branches passing down in recesses in the walls, discharging into the tail-race at the opposite sides. The pair of draft-tubes for each turbine starts from a Y-piece; and they are of heavy, ribbed cast-iron anchored to the rock, and serve as supports for the turbine casing and such an amount of the dead weight of the water as is not carried by the elbow at the bottom of the penstock.

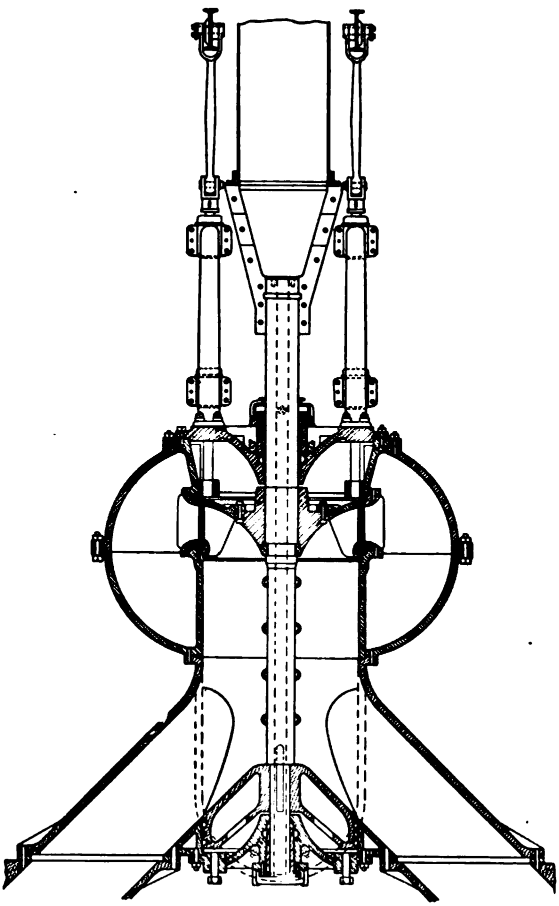


Fig. 179. One of the 5,500-Horse-Power Turbines, with Balancing Wheel. Showing Double Draft-Tubes, Power-House No. 2, Niagara Falls Power Company.

Another feature of the new turbines is the enlarged casing around the guide-blades, which allows for collecting stones that may be carried down the penstock, and from which they may be discharged into the tail-race as desired, through a pipe dropping from the bottom. In Power-House No. 1, it has sometimes been necessary to shut down the wheel to remove ice collecting in the turbine and in the lower part of the penstock.

The manipulation of turbine gates is similar in the two plants, in that the governors are connected to the gates by a series of levers and suspender rods extending from the governors in the power-house to the gates at the turbines.

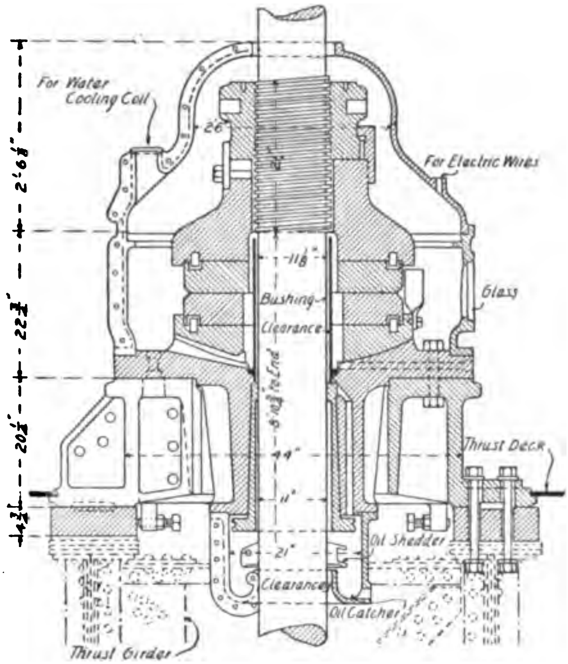


Fig. 180. Thrust or Hanging Bearing.

The turbine shaft in the new plant, like that in the old, is built up of sections of pipe 3 feet 2 inches in diameter, with walls $\frac{3}{8}$ inch thick, connected by solid pieces of shaft 11 inches in diameter at the points where bearings are employed. Besides the two bearings at the turbine, and those at the generator at the top of the shaft, there are three guide-bearings, and the thrust or hanging bearing (Fig. 180). The sections of tubular shaft are fixed to the solid pieces by hollow steel cones bolted around the 11-inch shaft, and to a ring of angle iron riveted to the tubular shaft. The guide-bearings are located roughly 40 feet apart, and are 20 inches long. They are lubricated with oil under slight pressure; and a feature of their design is the use of oil shedders and oil catchers, to prevent oil from dripping downward along the shaft.

201. *Thrust Bearings.* The choice of a thrust bearing has been the subject of a considerable amount of experimental work largely carried on at the power-house. The plans of Messrs. Escher, Wyss & Company called for a disc type of bearing, in which the shaft was hung by a revolving disc carried on a stationary disc. Three different types were constructed and placed on the shafts of the first three units of Power-House No. 2. The discs were all made of close-grained charcoal iron, of 25,000 pounds tensile strength, and with the mating faces scraped to a bearing. Their differences arise from the method of lubrication. For wheel No. 11, the first in the new plant, forced lubrication was adopted; for No. 12, a combination of forced and self-lubrication; and for No. 13, self-lubrication alone.

In the combination bearing, arrangements are made to utilize two different pressures, the low-pressure oil to come from a general lubricating system, and the high-pressure from an individual pump on the thrust deck, of 25 gallons capacity per minute at 400 pounds pressure. The oil is introduced into the stationary disc at two diametrically opposite points, and the oil forced between the surfaces. Each disc has two circumferential grooves, one near the outer, and the other near the inner edge. Connecting these in the stationary disc, are grooves $\frac{3}{8}$ inch deep, and branching grooves $\frac{5}{16}$ inch deep. The connecting grooves of the rotating disc are bent backward as regards the direction of rotation, and are $\frac{1}{2}$ inch deep near the inner ring (Fig. 181).

The thrust discs of the bearings in all cases are enclosed in a casing which is provided with two sight-holes in diametrically opposite positions, the sight-holes being fitted with $\frac{1}{4}$-inch plate glass to allow for observing both the condition of the bearing and the temperature of the oil by means of a thermometer hanging in the oil, illumination being secured by suspending an incandescent lamp within the casing. The stationary disc is fastened by steel dowel-pins to a third disc which has a spherical seat scraped to fit a support which in turn is bolted to the thrust girder. This spherical disc makes it possible to take up slight deviations from the vertical, where a more rigid construction with the thrust disc bearing might cause trouble. The radius of the spherical disc and ball seat is 3 feet 4 inches. The bottom of this is grooved at six points to allow the oil to pass from the outer or discharge chamber to the space where it can reach the bear-

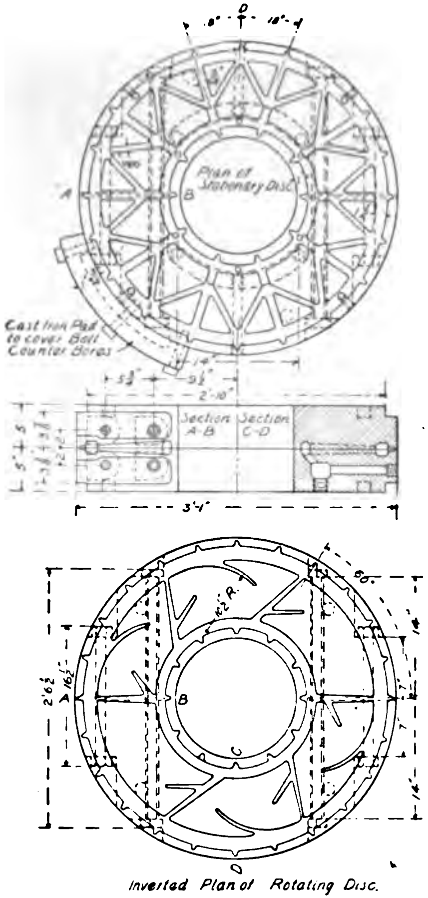


Fig. 181. Oil-Grooves in Thrust-Bearing Discs.

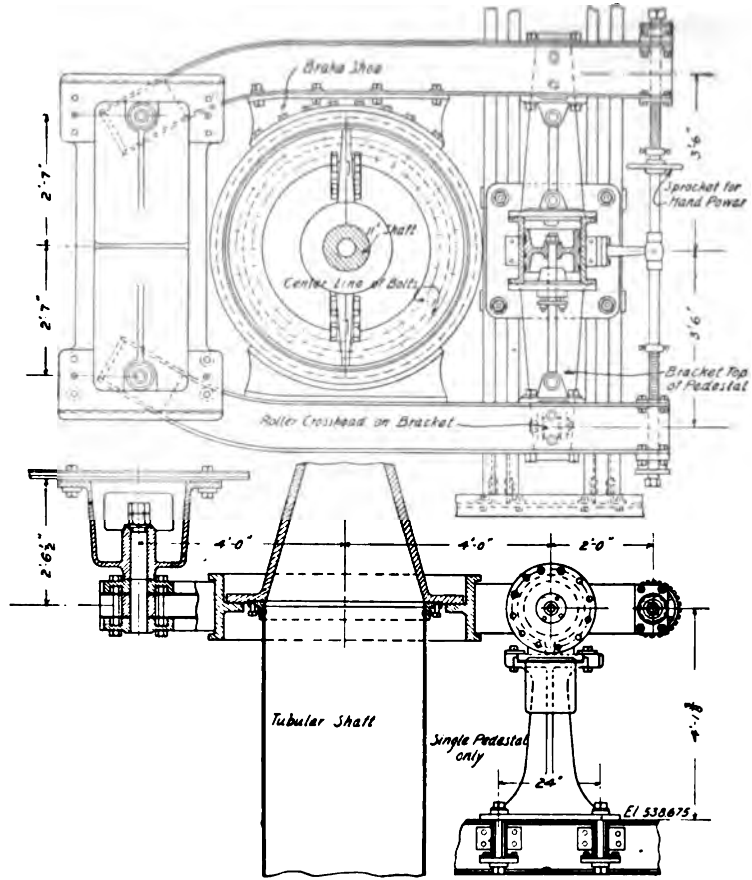


Fig. 182. Brake on Main Shaft under Thrust-Bearing Girders.

ing surfaces at the inner ring. As a result of the trials, all units were equipped with the combination bearing.

Brake. The brake on the shafts of the new units is different from that in use in Power-House No. 1. It is hung from the thrust-bearing girder, and clasps a flanged collar, or brake-wheel, bolted to the end of the adjacent section of the tubular shaft. It is operated either by compressed air at 100 pounds pressure, or by hand. Its office, in addition to providing a means of stopping the machines quickly in case of accident, is to bring the rotating members to a stop on an ordinary shut-down, in a reasonable length of time. For example, without applying the brake, the machines would tend to rotate for 30 to 45 minutes after the closing of the gates to the turbine; while, with the brake, full stop is obtained in less than a minute.

202. The brake consists of two levers on opposite sides of the brake-wheel, and a mechanism for drawing the levers toward each other to bring two oppositely located brake-shoes to a bearing on the brake wheel. Each lever has the fulcrum at one end, the point of power application at the other, and the shoe at the mid-point. The shoes are lined with maple, and together have a bearing surface of 180 degrees of the circumference of the wheel, or 7.2 square feet. The air-piston has a diameter of 13 inches and a total stroke of 6 inches, and the force exerted to draw the levers together is multiplied by two at the shoes. The brake-wheel is 5 feet in diameter, and 13 inches wide. The compressed air is controlled from the generator floor, as is also the hand operation of the brake. For the latter a sprocket-and-chain transmission turns a screw-shaft to bring the levers together. Coil springs on the shaft keep the brake-shoes normally free from the brake-wheel. See Fig. 182.

203. In these power-houses, as already explained, the weight of each turbine wheel and shaft is about 35 tons, and that of the revolving-field ring of the dynamo about 35 tons—altogether about 70 tons, which could not be carried on any pivot or collar-bearing at the speed of the turbines. In Power-House No. 1, water-pressure acts on the balancing piston, or cover of the upper turbine, and is relieved from acting on the lower turbine, giving an upward force of 65 to 70 tons to balance the weight wholly or partially. The excess pressure is taken by a collar-bearing. In Power-House No. 2, another arrangement was necessary. A special piston is here provided,

4 feet 6 inches in diameter, the water-pressure on which, due to the head, partially balances the weight. The excess or unbalanced load in this case is taken by the oil bearing shown, oil being forced between the bearing surfaces.

In Power-House No. 1, the speed regulation is accomplished by means of a sensitive governor acting on a ratchet wheel connected

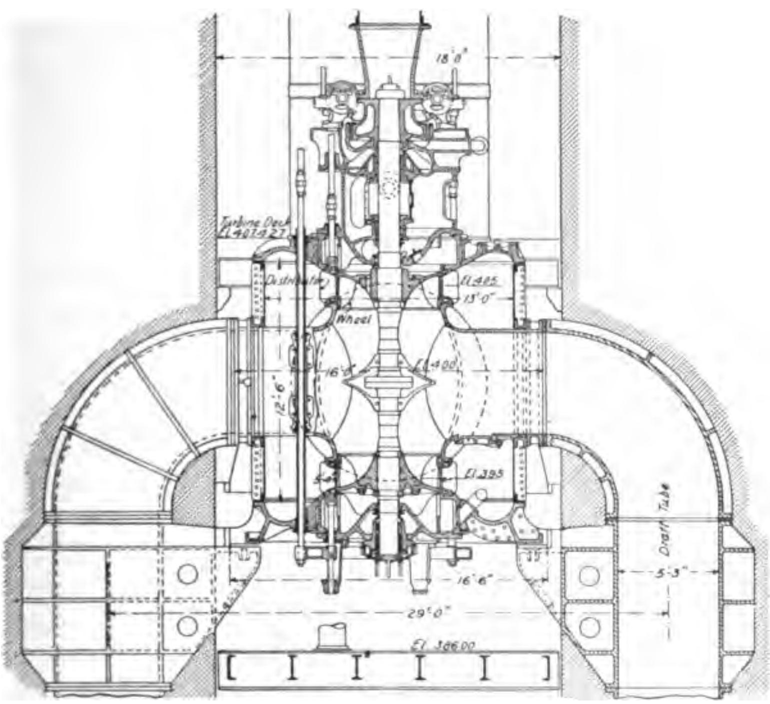


Fig. 183. Detail of 10,000-H.-P. Turbine, Canadian Niagara Power Company.

with the sluices. In Power-House No. 2, the relay is of the hydraulic type. In this case the sensitive governor in action opens a valve, thus actuating a ram driven by oil from an oil reservoir, at a pressure of 1,200 lbs. per square inch.

204. **Canadian Power Company.** The installation of this company will be worked in combination with that of the Niagara Falls Power Company, so that, when necessary, it can assist the works on the other side of the river. The works were not started on the Canadian side until it was thought that all the problems involved had

been satisfactorily solved by experience on the American side. When complete, the power-house will have eleven units of 10,250 horse-power each (Fig. 183), on 133 feet effective fall. The adoption of these large units has resulted in a material reduction in the size of the wheel-pit, canal, and power-house, for a given power development, as compared with the American plant.

In 1906, five turbines of the double inward-flow type with draft-tubes, had been installed. The tail-race tunnel is 2,200 feet long, 21 feet high by 19 feet wide, and the water in it will flow with a velocity of 27 feet per second. The wheel-pit is 570 feet long, 165 feet deep, and 18 feet wide. The weight of turbine-wheel, shaft, and field-ring is 120 tons, and this weight is carried by a balancing piston, on which water acts at the pressure due to the fall.

205. **Electrical Development Company of Ontario.** The success of the Niagara Falls Power Company has stimulated other enterprises of similar magnitude. The Electrical Development Company of Ontario obtained rights, and are erecting an installation of 125,000 horse-power on the Canadian side (see general plan, Fig. 162). In order to construct a masonry intake dam in the river, a cribwork cofferdam 600 feet long was built in some of the worst of the upper rapids, laying bare eleven acres of the river-bed. This temporary dam was at the worst part, in water 24 feet deep, and flowing at probably 30 miles an hour. Its construction was an engineering feat of the greatest boldness. Within it, is now (1908) being erected a concrete gathering dam with granite coping, to direct water into the intake, while floating ice will pass over the dam and back into the river. The water, before entering the power-house, must pass through submerged arches and screens. The tail-race tunnel, 26 feet high, 23½ feet wide, and 1,900 feet long, passes right under the upper rapids and discharges underneath the Horseshoe Fall. A drift-way to the mouth of the tunnel was first driven, and then the tunnel excavated back from the mouth. The excavated material was thrown down from the mouth of the tunnel into the lower river, where it has disappeared. The wheel-pit is 416 feet long, 27 feet wide, and 150 feet deep. It is to receive eleven double turbines of 12,500 horse-power each. A feature of this installation is the two-branch tail-race tunnel, one branch on each side of the wheel-pit, five turbines discharging into the one, and six into the other.

EAST BREAKWATER, HARBOR OF CLEVELAND, OHIO

206. Ontario Power Company. The plans of this company differ essentially from the others, and are no doubt partly conditioned by the fact that the ground nearer the Falls was already occupied. The intention is to develop 200,000 horse-power. The intake—at the top of the rapids on the Canadian side, near the Dufferin Islands—is specially designed with reference to ice difficulties. The openings in the intake dam have a curtain dipping 9 feet into the water, below which the flow to the turbines takes place, the floating ice being carried past. A second curtain on the same principle is constructed between the forebay and inner basin, and the ice in the outer basin is carried forward over the lower part of the outer dam. The ice in winter is a serious difficulty at Niagara. Cake ice floats down from the upper lakes, and "mush" ice is formed in the turbulent rapids, primarily by the freezing of spray and foam. For ice in this latter form, there are screen frames.

From the intake, three great steel conduits, 18 feet and 20 feet in diameter, convey the water round the other power-houses to the top of the bluff below the Falls. These conduits, of which one is already (1908) constructed, are of ½-inch steel plates, stiffened with bulb irons, and encased in concrete. The velocity in the conduits will be 15 feet per second. There is a spillway at the end, formed by a weir, to prevent water-hammer in the pipes. The flow over the weir passes down through a helical culvert or spillway in the rock to the lower river. From the conduits the water will be taken down to the turbines through twenty-two steel pipes, 9 feet in diameter, passing down the face of the bluff.

The power-house is on a platform at the foot of the bluff, and just above the level of the lower river. The turbines are to be inward-flow, twin turbines, each of 12,000 horse-power, under 175 feet head. The axis of the turbines is horizontal, and the shaft is 24 inches in diameter. The turbine wheels are 78 inches in diameter, and have movable guide-blades, which are more efficient than the cylindrical sluices used in the other power-houses, though of course economy of water is not of great importance at Niagara.

207. Snoqualmie Falls Power Company.* Within the past few years a number of plants have been established on the Pacific slope, to utilize natural water powers for generating electricity to be trans-

* *Engineering News*, December, 1900.

mitted to distant points, and there used for lighting and power purposes. Among the most interesting and important of these plants is that at the Snoqualmie Falls, in Washington. For this plant, no long flume or pipe-line is required to develop the necessary head of water, as the Snoqualmie River has at the Falls a vertical drop of 270 feet, giving an available energy of 30,000 to 100,000 horse-power. In this respect the plant resembles those at Niagara Falls. In the placing of the electric machinery, however, there is an essential difference; for, while the Niagara Falls plants have this placed in a building above ground, the Snoqualmie Falls plant has the water-wheels and electrical machinery all installed together in a large underground chamber whose floor is directly above the tail-race tunnel, which extends to the river below the Falls. The force of the water is used to drive impulse wheels on horizontal shafts, instead of turbines on vertical shafts, as at Niagara Falls. Another notable feature of the plant is the use of aluminum wire for the long-distance transmission lines. The entire plant represents an investment of about $1,000,000.

208. The great fall of the Snoqualmie River is about 34.5 miles northeast from Tacoma (in a straight line), the same distance southeast from Everett, and 25 miles west from Seattle, being situated in the foothills of the Cascade Range. The river proper commences about three miles above the fall, at the junction of three forks which flow westward down the slopes of the range. Below the fall the river runs almost due north, and finally flows into Puget Sound near the city of Everett. The flow of the river is about 1,000 cubic feet per second at its lowest stage, increasing to over 10,000 cubic feet per second at its flood periods. The river does not freeze during the winter, and there is neither floating ice nor anchor ice to be dealt with.

An investigation showed that by the construction of dams or dikes, some of the large lakes on the watershed could be utilized as impounding reservoirs, so as to ensure a uniform flow sufficient to develop nearly 100,000 horse-power throughout the year, should a demand for so much power eventually be found. It was also determined that by the erection of a 50-foot dam above the headworks, a reservoir could be formed, having an area of 15 square miles and an average depth of 25 feet. This would almost double the power, should the growth of the industries served make this desirable in the future.

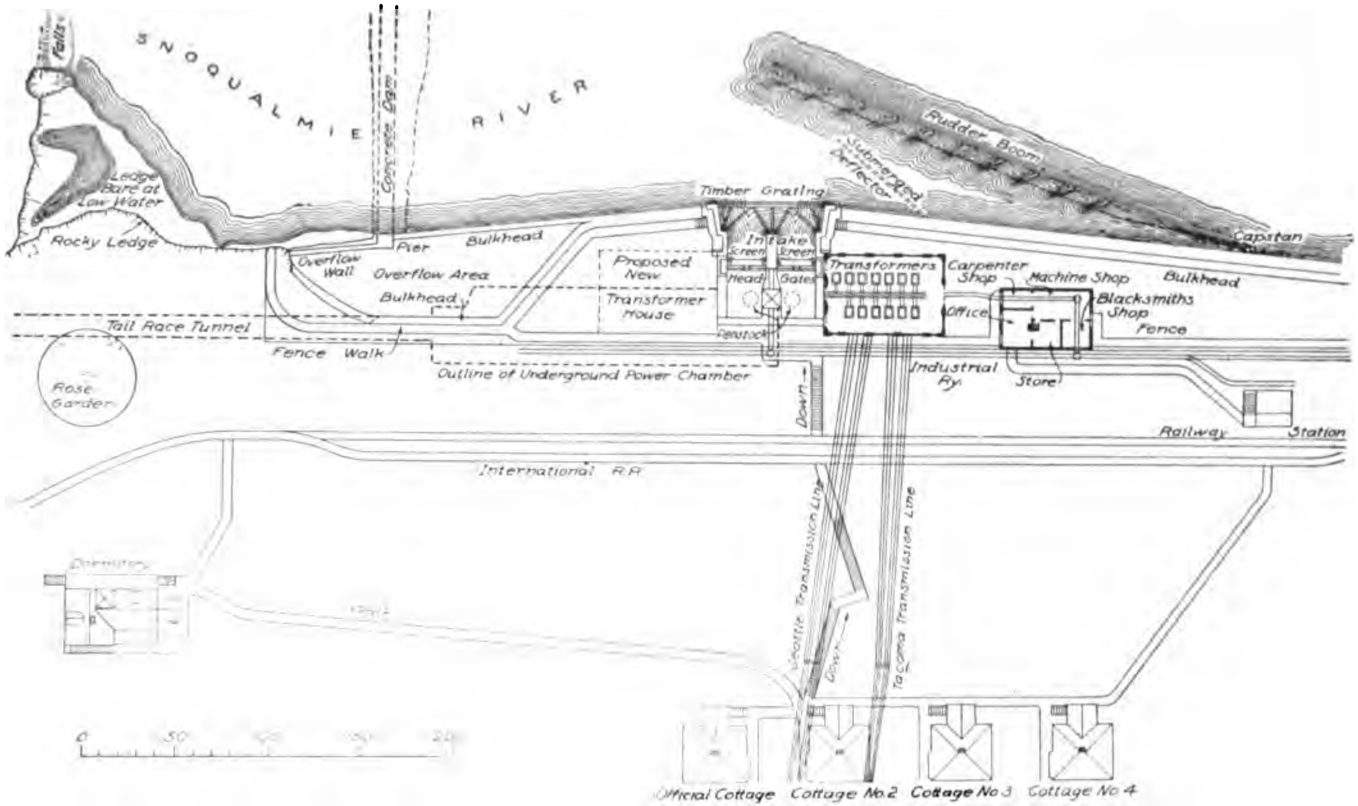


Fig. 184. Plan of Headworks of the Snoqualmie Falls Power Company, Washington.

The rock at the Falls is basaltic, with no regular cleavage. It is hard and non-absorbent, and is apparently divided by seams into great ledges. These conditions led to the adoption of the plan of placing the machinery in an underground chamber, as already noted. It was at one time proposed to build a power-house near the base of the fall; but this would have been at a disadvantage, on account of the clouds of spray, which keep everything damp, and which coat all of the surroundings with ice in cold weather.

209. A plan of the headworks above the Falls is shown in Fig. 184, which also shows the position of the underground power chamber and the tail-race tunnel. The intake bay is a rectangular chamber, about 60 feet long (parallel with the river), and 20 feet wide. It has walls, and a center pier of concrete masonry 6 feet thick and 25 feet high, built upon solid rock formation, its floor being on a submerged reef about 5 feet above the river-bed. This bay is protected from the river by a timber grating across the opening, supported by a steel girder construction bearing against the walls and pier. The timbers are 12 by 12-inch, laid horizontally with 12-inch spaces between them, through which the water flows into the intake. This grating protects the works from floating trees and logs; while just inside the intake are inclined steel screens made of flat bars on edge, which serve to exclude the smaller debris. The intake thus has two head-bays, separated by the pier.

210. A rudder boom 300 feet long is moored above the intake, and extends beyond it. By turning the capstan at the head of the boom, the rudders are thrown out, and cause the boom to swing out into mid-stream, so that it serves as a fender to deflect floating logs, etc., from the intake. The river is 150 feet wide from the head-bay to the opposite shore, and about 15 feet deep at ordinary stages. The face of the intake was continued 400 feet upstream and 200 feet downstream, in the shape of heavy retaining walls built of sawed cedar timber, tarred. The space behind them is filled with excavated rock, and has a top dressing of soil for a lawn and shrubbery.

At the end of the lower bulkhead, a submerged concrete dam is built across the river, resting on the rock bottom, and this raises the low-water elevation of the river 6 feet at the intake. This dam, whose location is shown in the plan, Fig. 184, is of the form shown by the elevation and section, Fig. 185. It was first framed of heavy timbers sheeted over with 6-inch planking, and then filled in solid with concrete. It was built about the time of low-water flow, portions of the river-bed being laid bare by cofferdams. Preparatory to the construction of the dam, the river-bed was thoroughly cleaned of loose rock, and was roughened by occasional blasts so as to afford a good footing. In addition to this, pieces of steel rail were driven in holes drilled 2 feet deep, the rails extending up into the concrete body of the dam. Old railway cables were also embedded in the concrete to perfect the bond. The dam has a batter of 2 on 1 upstream, and ½ on 1 downstream, with a level crest 8 feet wide. At each end of the dam is an abutment pier of concrete 8 feet square, these being 210 feet apart. The dam was built on natural rock ledge, about 3 feet above the river bottom. It is always submerged from 2 to 10 feet, according to the stage of the river, and varies in height from 3 to 10 feet, and in width on bed rock from 16 to 35 feet, according to the conformation of the river bottom. The lower bulkhead is only 5 feet higher than the dam, so that flood waters have a very considerable increased sectional area of discharge, as shown by the plan. The capacity of this spillway is such as to insure the complete discharge of an extreme flood without the river backing up to an unusual elevation. The top of the upper bulkhead is above flood level.

211. The general arrangement of the plant, with its underground power chamber, is shown in Fig. 186. About 300 feet above the Falls, a shaft 10 by 27 feet was sunk in the bed of the river on the south side descend-

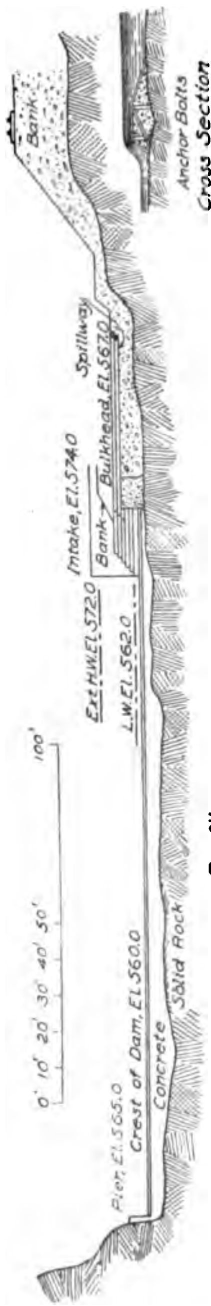


Fig. 185. Section Showing Submerged Dam of the Water-Power Plant of the Snoqualmie Falls Power Company, Washington.

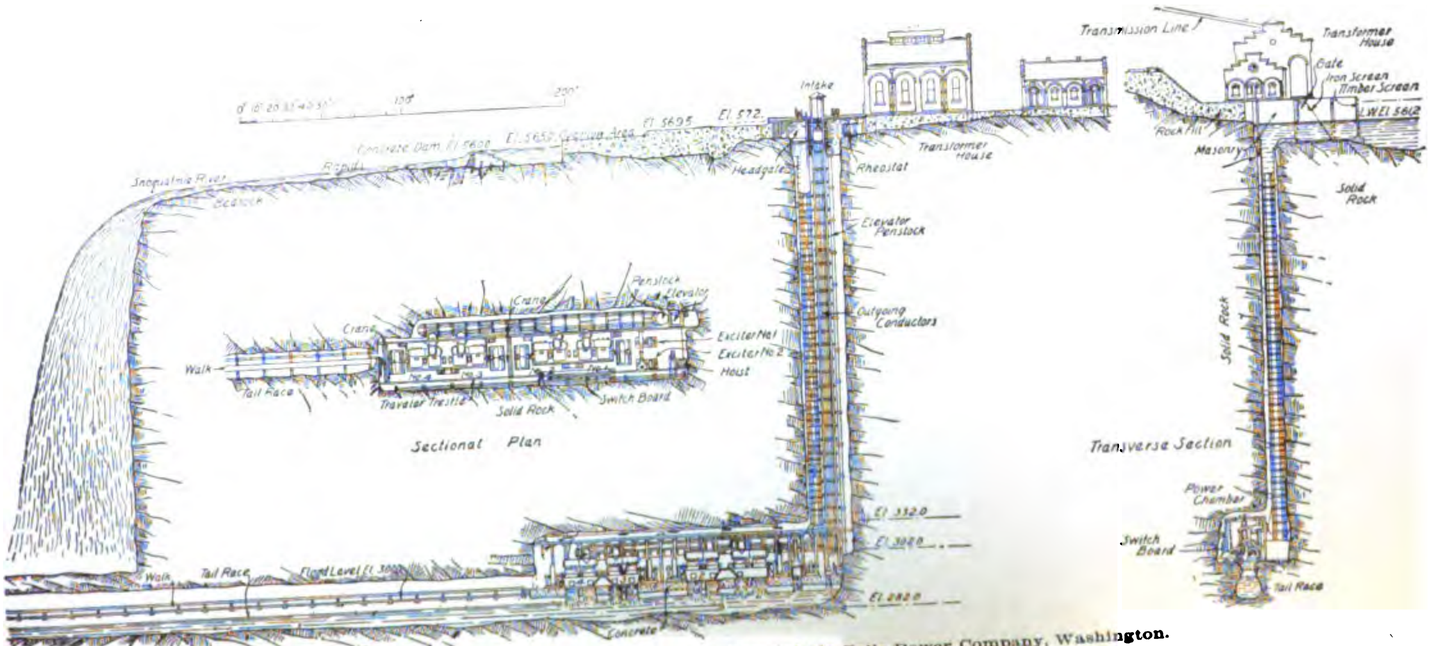


Fig. 186. Sectional Elevations of the Plant of the Snoqualmie Falls Power Company, Washington.

ing 270 feet to the level of the river below the Falls. While this shaft was being excavated, a tunnel 12 feet wide and 24 feet high, with a fall of 2 feet in its entire length, was drifted in from the face of the ledge below the Falls, to an intersection with the bottom of the shaft, a distance of 650 feet. Beginning at the foot of the shaft, and extending over and along the tunnel, a chamber 200 feet in length, 40 feet wide, and 30 feet high, with the floor at the elevation of high water below the Falls, was excavated out of the solid rock (Fig. 187).

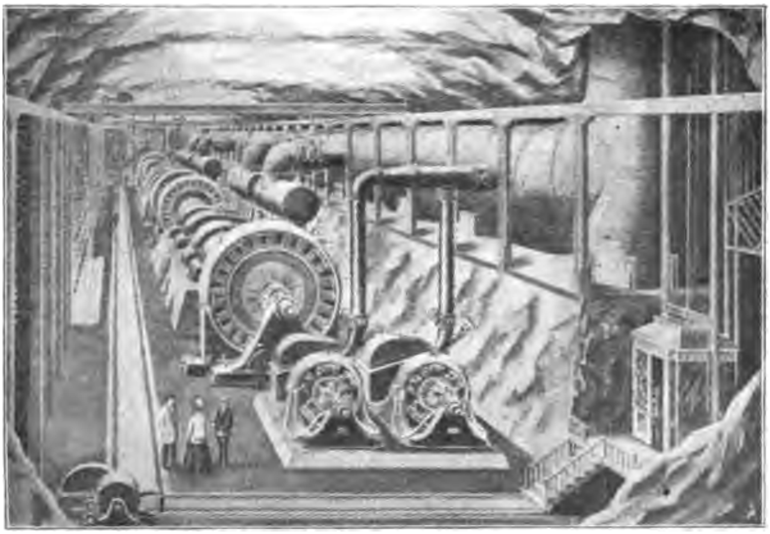


Fig. 187. Interior of Power Chamber at Snoqualmie Falls.

This chamber forms the power-house, or machinery room, in which the water-wheels and electric generators have been installed. At average stages of the river, the water is about 12 feet deep in the tunnel; while during flood seasons the tunnel is nearly filled. The tunnel extends under the floor of the chamber, forming a tail-race with concrete roof 5 feet thick, The walls of the chamber have been left rough and whitewashed, while the floor is covered with concrete.

The chamber is ventilated by natural draft through the tail-race and up the shaft, the draft being so strong that it was necessary to curb it. The chamber is said to be cool and perfectly dry, the temperature remaining the same (about 55°F.) throughout the year.

This low and uniform temperature contributes to the high efficiency of the generators.

Each intake bay contains a massive head-gate, moving vertically, which controls the flow of water through an opening 8 by 12 feet through the shore wall into the penstock. The gate is raised and lowered by mechanism connected with the piston-rod of a hydraulic cylinder. The shaft is 10 by 27 feet, and at the top has three compartments; the two end compartments are for the penstocks; while the center one, enclosed at the top by a steel bulkhead, forms a shaft 8 by 10 feet for the hydraulic elevator and the main cables forming the outgoing conductors, and also for raising and lowering machinery, etc. The steel bulkhead which encloses this center shaft extends from the bottom of the intake bay to the surface of the ground. It is built up of steel plates, and is stiffened by horizontal frames of I-beams on the outside, riveted to the plates and to each other at the ends. Below it, the elevator shaft is timbered and sheathed with plank. At the surface, this shaft is surmounted by a small building. The penstock already built is steel pipe $7\frac{1}{2}$ feet in diameter, passing through a concrete roof which keeps the shaft water-tight. The plates are in 8-foot courses, and are 1 inch thick for the lower half of the pipes; in the upper half, the thickness decreases from $\frac{7}{8}$ to $\frac{1}{2}$ inch at the top. The joints are heavily riveted, and calked water-tight. At a depth of 250 feet, the penstock reaches the chamber, and connects with a horizontal cylindrical receiver which rests on a rock bench in the north side of the chamber, 12 feet above the floor. This receiver extends almost the full length of the chamber. Its diameter is 10 feet for half its length, and then reduces to 8 feet. It is built up of 1-inch plates 8 feet wide. The penstock and receiver weigh 225 tons, and the weight of the water column in the penstock is 340 tons. A small independent penstock supplies water to the elevator machinery, as shown in Fig. 186.

At four points in the length of the receiver are 4-foot branches extending from the side, each branch being fitted with a gate-valve. These valves weigh 23,000 lbs. each, and are said to be the largest valves in the world operated under such high pressure. Each branch has a cast-iron elbow turning downward and opening into the horizontal cylindrical receiver of a water motor. These elbows have an inside diameter of 4 feet, and the metal is 2 inches thick, each casting weighing 8,000 lbs. Owing to the size and form, special care had to be taken to avoid internal strains in the castings which might lead to rupture, especially in view of the high pressure which they have to withstand.

212. The main generating plant consists of four electric generators, each driven by a 45-inch Doble water-motor of 2,500 horsepower, coupled directly to it. Each motor consists of a shaft carrying six tangential-jet wheels, with two nozzles to each wheel. Each of the four elbows above referred to is bolted to a flanged ring on a horizontal cylindrical receiver, 48 inches in diameter and 20 feet 8 inches long. This is made of two $\frac{1}{2}$-inch steel plates 10 feet wide, and of sufficient length to make the shell with only one longitudinal seam, which is double-riveted. The heads are of dished steel plates. The receiver is supported by six pipes, each of which carries two nozzles delivering jets at right angles to each other, the nozzles entering the side and bottom of the wheel-casing. The use of the receiver effects an even distribution of the flow from the elbow to the several nozzle-pipes, and also a steady and uniform rate of flow. From the buckets of the motors, the water falls through draft openings in the floor directly into the tail-race channel.

To handle the volume of water necessary to develop the power in each unit, requires 12 jets $3\frac{7}{8}$ inches in diameter, discharging against six wheels. For convenience of bearing and shaft design, these wheels are divided into two groups of three wheels each, each group being in a separate housing, with a bearing between. This arrangement makes two groups of three vertical nozzle-pieces each.

213. One of the special features of this plant is the needle-regulating tips used on the nozzles. They not only throw a perfect and unbroken stream, but give absolute control over the quantity of water applied to the wheels, and therefore over the power output of the unit. As these tips are controlled by the governor, the arrangement gives an excellent degree of speed regulation with variable load, at high efficiency. The full size of the jet is 3 inches; it is a solid, smooth stream, delivered with a head of 253 feet, entirely free from swirling or other disturbance. This form of nozzle maintains the same condition from full-jet size to 1-10 of the jet area. The regulating nozzles are operated from two long rocker-shafts, one controlling the upper, and the other controlling the lower nozzle-tips. Both

rocker-shafts are operated by a Lombard governor, which is connected to the rocker-shaft by cranks and connections so arranged with clutches that either or both rocker-shafts can be disconnected from the governor and operated or regulated by the hand-wheel on the pedestal stand. By this governor arrangement, with the regulating nozzle-tips, the wheels use water in proportion to the power developed, so that they operate with high efficiency at part as well as at full load.

The wheels are encased in sheet steel housings with cast-iron fronts, three wheels in each, so that there are two housings to each unit. The housings are made with the upper half removable, to provide access to the wheels when desired. The cast-iron front of each housing is made of such form as to provide a deflector guard, which takes care of the water thrown from the wheels by the centrifugal action, and directs this water into the tail-race, and thus prevents its being driven around the housing by the air currents created by the rapidly revolving wheels. In the top housing is a guarded opening to permit the indraft of air to replace that driven out of the housing and down the tail-race by the rush of the water and the action of the wheels as centrifugal blowers. To prevent water from splashing out where the shaft passes through the side of the housing, the opening is protected with patented centrifugal discs and guard-frames. Although this arrangement prevents the outflow of water, it permits a large and free indraft of air at this point also to replace that driven out by the action of the wheels and the water.

Each wheel unit weighs about 100,000 lbs., in addition to the weight of water in the distributing receiver and the nozzles; and in view of the high speed of the parts, and the power developed, careful design and construction were required for the foundations, which are of concrete, built solidly into the floor and one side wall of the chamber; and the lower part of the steel wheel-housing is firmly built into the concrete walls. The waste water drops from the wheels directly into the tail-race.

A tunnel is provided under the governor platform for the lower rocker-shaft and connections that operate the adjustable tips of the lower nozzles, and thus makes this operating gear accessible. The foundation for each unit is divided into two compartments corresponding to the two wheel housings, and a 2 by 3-feet doorway is formed in the front wall of each. Four steel rails are built into the concrete across the opening into the tail-race, to support a temporary floor, when it is desired to enter the foundation for the purpose of inspecting the wheels or nozzle-tips without removing the top wheel-housing.

214. **Hydraulic Plant at Vauvry, Switzerland.*** This installation is situated on the left bank of the river Rhone, a short distance above the point where that stream empties into Lake Leman. The water is taken from Lake Tanay, at an elevation of 4,644.5 feet, and is delivered to the wheels in the power-plant at an elevation of 1,528.8 feet, which represents a gross head of 3,115.7 feet—probably greater than that of any other hydraulic plant in the world. The installation is intended to supply electricity for lighting and power purposes to a large number of Swiss towns and villages in the valley of the Rhone.

Lake Tanay, the source of the water supply, has an area of about 112 acres, and receives the drainage of an area of about 1,875 acres, which, it was estimated, would yield a steady stream of about 12.2 cubic feet per second throughout the year. The main structural features of this plant comprise the supply pipe-line, and the power-house, with its machinery. The head of the pipe-line is at an elevation of 4,559 feet, which is 65.6 feet below the normal level of the lake, and 84.2 feet below its maximum level; and it terminates in a vertical shaft about 241.7 feet deep. From the shaft a short gallery or tunnel 984 feet long, with a sectional area of 10.75 square feet, is built on an almost level grade. This tunnel is provided with bulkheads, pipes, valves, and other apparatus for regulating and controlling the supply of water taken from the lake; and at its end the pressure pipe-line begins. For 328 feet, this line is a steel pipe 2.62 feet in diameter; then, for 984 feet, it consists of a masonry tunnel; and finally, for 3,936 feet, it is again a steel pipe, 2.62 feet in diameter. At the end of the last section, the pipe branches into three pipes, each of 1.64 feet diameter. One of these pipes extends to the power-plant; but the others are plugged, and will be built to the power-house only when the demand for power necessitates their construction. At the point of junction of the single and triple pipes, the

* *Engineering News*, November, 1902.

head of water is only 689 feet; but from this point on, the descent is very abrupt. At this point, also, there is a retaining valve, and a vertical standpipe (regulator) 1.31 feet in diameter, and 82 feet high, which relieves the water-hammer in the pipe-line above.

215. The steep-grade pipe-line, from the junction point just mentioned to the power-house, is 6,363 feet long, and has a fall of 2,952 feet. As stated, the pipe is 1.64 feet in diameter at the junction point, and it continues with this diameter for 2,682.8 feet, varying in thickness of shell from about .275 to .45 inch, finally terminating in a Y, each branch of which is provided with a valve. From the branch of the Y, two 1.12-foot pipes extend for a distance of 4,264 feet, with shells varying in thickness from about .3 to .7 inch. The transverse joints are made as shown at *a* in Fig. 188; and in order that the joints may be tight, a suitable gasket is inserted before tightening up the bolts. As the pipe-line lies in a trench following the surface grades, it has many bends; and to provide for these, the wedge-shaped pieces shown at *b* in Fig. 188 are inserted at the joints. The sections of pipe used varied in length from 16.4 feet to 32.8 feet, and weighed from about 1,760 pounds to about 2,500 pounds. Each branch of the pipe-line has at its lower end a slide-valve provided with a by-pass, to permit it to be operated by hand.

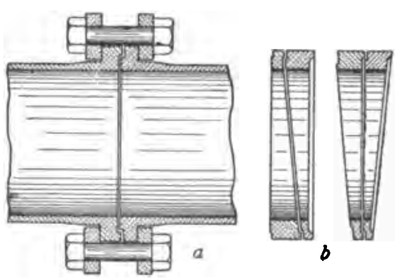


Fig. 188. Detail of Joint Connections for Pipe-Line of Water-Power Plant at Vauvry, Switzerland.

216. The power-house is a steel building 45.9 by 216.5 feet in plan. The two pipe-lines described terminate underneath its main floor; and each supplies water to two wheels, all of which are of the impulse type. Two makes of wheels were installed, as shown in Figs. 189 and 190. Each wheel is supplied with water through two nozzles, one above the other, the upper one of which is provided with a device actuated by a governor, for controlling the supply of water. A vertical diaphragm divides the end of each nozzle into two openings, as shown. The pipe from which the nozzles are supplied rises vertically from the main, and is opened and closed by a hand-valve, above which it divides into two branches, one leading to each nozzle,

and each opened and closed by a hydraulic valve controlled from the operator's platform on the floor above.

Each wheel is mounted on the shaft of an alternator, and operates at 1,000 revolutions per minute. The dynamos are each of 500 horse-power.

217. The *Mill Creek No.* 3 *Power-Plant of the Edison Electric Company, Los Angeles, Cal.,* which went into service in March, 1903,

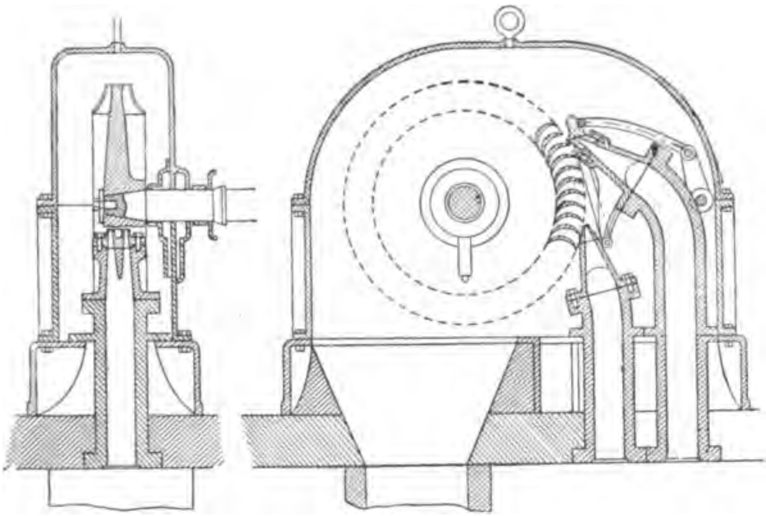


Fig. 189. Detail of Duvillard Wheel for Vauvry Water-Power Plant, Switzerland.

is remarkable for the high head used. All the water usually flowing in Mill Creek at Akers Narrows, is diverted by a masonry dam, and conducted through 5 miles of pipe to a reservoir 1,960 feet above the power-house in Mill Creek Canyon. The conducting pipe slopes 0.2 foot per 100 feet, and is designed to carry 20 cubic feet of water per second. It contains 5 inverted siphons of steel pipe, aggregating 3,585 feet in length, and 25,190 feet of concrete pipe 3 inches thick and 32 inches in inside diameter, and passes through 10 tunnels having an aggregate length of 7,500 feet.

From the reservoir the water descends through a steel pressure-pipe, varying in diameter from 26 to 24 inches, and in thickness from No. 14 B. W. G. to $\frac{7}{8}$ inch. The pipe is protected from rust by a heavy coat of asphaltum, applied by dipping. At the lower end it branches, leading the water through 18-inch and 14-inch lap-welded pipe to the four generating units, which are housed in a concrete building with steel roof-trusses and galvanized-iron roof. Of these generating units, three were made by the Abner Doble Company.

218. Each Doble unit consists of a 1,300-horse-power Doble tangential water-wheel and a 750-kilowatt generator, mounted on a single shaft. This shaft has a speed of 430 revolutions per minute, and is mounted in three bearings which rest on a single cast-iron base-frame set in concrete. Each wheel is provided with a Doble needle-regulating and deflecting nozzle, with hand-operated balanced needle. With this apparatus the station attendant can set the needle by hand every half-hour at the most economical point in order to carry the load which from experience he is led to expect during the next half-hour. The governor takes care of all sudden fluctuations of load by deflecting the nozzle momentarily, so that all or part of the water issuing passes under the water-wheel, and wastes its energy against the "Vortex" baffle-plate installed in the tail-race.

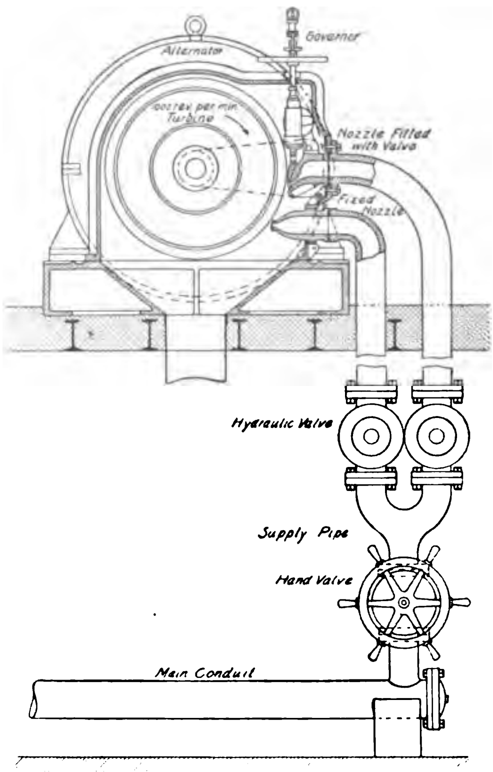


Fig. 190. Detail of Wheels Supplied by the *Société de Constructions Mécaniques de Vevey*, for Vauvry Water-Power Plant, Switzerland.

219. The static pressure due to the head of 1,960 feet is over 850 pounds per square inch, and the spouting velocity of the jet is about 4 miles per minute. The generating units deliver current at 750 volts to the switchboard, whence it passes through transformers, and out over the 33,000-volt 86-mile transmission line to Los Angeles.

220. The *De Sabla Power Plant*, in Butte County, California, was erected by the Valley Counties Power Company in 1903, and is now an important source of supply for the California Gas & Electric Corporation's extensive transmission system.

Water is taken from Butte Creek, through a 12-mile ditch, and also from a branch of the Feather River, both conduits discharging into a regulating reservoir at the head of the pressure line. Besides taking up variations in load, this reservoir, containing over a day's supply, will permit repairs to be made on the ditch without shutting down the wheels. From this reservoir, two 30-inch steel pressure-pipes over 6,000 feet in length conduct the water down to the power-house, the total effective head being 1,528 feet. One pressure line supplies two 2,000-kilowatt hydro-electric units, and the second line supplies a 5,000-kilowatt unit. Hydraulically operated piston gate-valves of a special design are installed in the branch pipes leading to the units.

221. Each of the 2,000-kilowatt units consists of an alternator directly driven by a 3,700-horse-power Doble tangential water-wheel, the speed being 240 revolutions per minute. The 5,000-kilowatt alternator is direct-connected to an 8,000-horse-power Doble tangential water-wheel, the speed being 400 revolutions per minute. Regulation of this plant is secured by hydraulic governors, which deflect or raise the nozzles as the load varies.

All three units are of the two-bearing type, the water-wheel being mounted on the extended end of the generator shaft, and overhanging one bearing. Each water-wheel is provided with a Doble needle-regulating and deflecting nozzle.

222. The large wheel was the most powerful single water-wheel constructed at the time it was placed in operation, in September, 1904. It delivers 8,000 horse-power from a single jet of water, the jet having a spouting velocity of approximately 20,000 feet per minute. The general design of this unit is shown in Fig. 191. The shaft is 20 inches in diameter in the middle portion and 16 inches in the bearings, the latter being 60 inches long and of ring-oiling and water-cooled construction.

223. The transmission voltage is 55,000 volts; and current has been delivered from this plant, over the lines of the California Gas & Electric Corporation, a distance of 378 miles from the power-house—the present record (1908) for long-distance transmission.

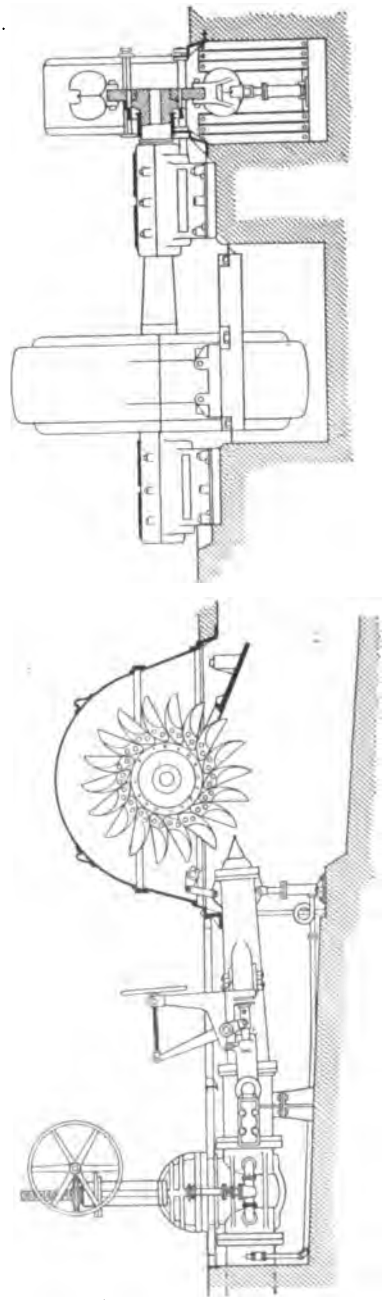


Fig. 191. Type of Large Hydro-Electric Unit in the De Sabla Power Plant of the Valley Counties Power Company, Butte County, California.

224. Considerable interest now centers in a new hydro-electric unit about to be installed in the De Sabla Plant, the hydraulic end of which will consist of a 9,000-horse-power Doble tangential water-wheel driven by a single jet of water, and will embody the same general features of design as the 8,000-horse-power De Sabla wheel.

225. **Centerville Power Plant.*** The California Gas & Electric Corporation has recently placed in service in the hydro-electric generating station at Centerville, Cal., a 9,700-horse-power hydraulic reaction turbine, designed to operate under an effective head of 550 feet (Fig. 192). The station in which this unit is installed is a part of the extensive generating system of that corporation, from which power is distributed to practically all of the cities within a radius of upwards of 100 miles from San Francisco. The development is about 200 miles northeast of that city, on Butte Creek. The 14,000-kilowatt De Sabla hydro-electric station of the system, of which the one at Centerville is also a part, is approximately 8 miles

* *Engineering Record*, March and April, 1908; *Engineering News*, March, 1908.

IMPROVING THE HARBOR OF CLEVELAND, OHIO

Top of west breakwater after the destructive storm of November 10-11, 1898.

upstream from the latter, and has been in service for several years.

226. The flow of Butte Creek at the De Sabla station, although comparatively large, has been increased considerably by the construction, on the watershed above that point, of several reservoirs for impounding a part of the flood waters. The water discharged from this station is diverted into an open canal about 8 miles long, having

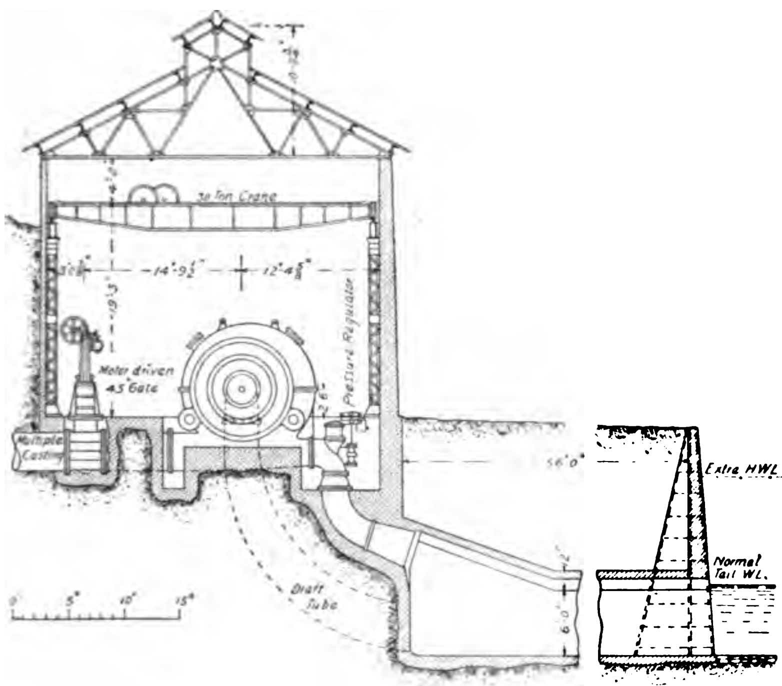


Fig. 192. Section through Centerville Station of the California Gas & Electric Corporation.

a capacity of 175 cubic feet per second, which leads to the Centerville station. It is so located that a difference in elevation of 591 feet is obtained between the end of the canal and the tail-race at the Centerville station.

227. An overflow concrete dam was erected across the creek at the upper end of the canal, to divert water into an approach to the latter. The flow into this approach is controlled by two large sluice-gates; it is 150 feet long, and is built with converging sides, its width being somewhat greater than that of the canal at the upper end. The

bottom of the approach is also built on a sufficient grade to bring it to a level 4 feet below the bottom of the canal at the end of the approach, thus forming a settling basin in which the coarser materials carried by the stream during certain flood conditions are deposited. These materials are readily sluiced out of this pocket, through a gate in one side of the lower end of the approach. In order to facilitate this sluicing, the opposite side of the approach is built on a curve which throws the force of the stream through the gate when the latter is open.

Some of the clay and fine sand are not intercepted in this basin, and a certain amount of sand may be blown, or will fall, into the canal. Provision is made to remove most of these materials from the water by means of sand-boxes built in the bottoms of the flumes which carry the canal over the water-courses that are crossed. These boxes are merely depressions in the bottom of the flume, and are placed at intervals of approximately three-quarters of a mile. Each box is arranged with a sluice-gate in order that the sand and debris that collect in it may be removed easily.

A settling basin, 18 by 50 feet in plan, and 9 feet deep, is placed about 300 feet upstream from the lower end of the canal, to remove finally all the grit that is carried in the water, which enters this basin through a screen of iron bars placed across the canal at an angle of 30 degrees. These bars are spaced 2 inches apart, so that they intercept practically all floating matter. The lower end of the bars is 1 foot above the bottom of the canal, thus permitting sand and gravel to flow along into the basin, which is built by offsetting one side of the canal, and is lined with rubble masonry laid in cement mortar. The bottom drops away from the canal bottom gradually to a depth of 9 feet, the downstream end of the basin being vertical up to the level of the canal. The materials deposited in the basin are flushed out through a sluice-gate placed in the downstream end of its outer side; and the opposite side of the basin is curved toward the gate to induce a current toward the latter, which delivers into a natural water-course.

228. Three steel-pipe pressure lines lead from the end of the canal, down to the power station. The upper ends of these lines open in a reinforced-concrete forebay, 18 by 25 feet in plan; and the connections are made trumpet-shaped, to accelerate the water gradually in order to reduce the entrance losses.

The Centerville station was originally installed to supply power to mines in the vicinity, and at first contained a 400-kilowatt generator, direct-connected to a tangential water-wheel. A 900-kilowatt generator, direct-connected to a 1,500-horse-power Doble tangential water-wheel was afterward added to the station. The 400-kilowatt unit was removed at the time the large turbine was recently installed. As a result of the increase in capacity at different times, three pressure lines have been laid to convey the water from the end of the canal to the generating station, all three of which lines were built separately to meet the increases in the capacity of the station. These pipes, one of which is 42 and 36 inches, and the other two each 24 inches in diameter, are 2,565 feet long, and are laid down the side of the mountain, which rises quite abruptly above the station building. They are all of riveted steel, the thickness of the plates varying with the difference in head. The upper end of each of the three pipes is controlled separately by means of a gate-valve at the forebay. The lower ends of the pipes are joined at the rear of the station building by a three-way connection. A gate-valve is placed in each pipe just back of this junction, so that any pipe may be cut out of service without interfering with the other two.

229. *Turbine.* The turbine is of the radial inward-flow, single axial-discharge, Francis type, with a horizontal shaft direct-connected to a 5,500-kilowatt alternating-current generator revolving at a speed of 400 revolutions per minute. It is joined by means of a 45-inch gate and taper-piece to the Y-casting which is the confluence of the three pipe-lines previously mentioned. The rated capacity is 9,700 horse-power; but the wheel has never exceeded 8,200 horse-power, owing to the limited capacity of the generator. It has a cast-steel spiral casing, made in two parts, and provided with a quarter-turn discharge to the draft-tube, and with a pressure-regulator. The runner is of cast steel, with 20 vanes; 24 pivoted guide-vanes are provided, and are connected to a shifting ring, which is in turn connected to the governor rocker-shaft by lever arms. A thrust bearing takes up the end thrust which arises with sudden changes of load, and is supplied with oil under pressure. A ring-oiling bearing of the ordinary pedestal type on the discharge side of the wheel, supports the other end of the shaft. The pressure-regulator or relief-valve is governor-operated, and is designed to relieve the pipe-line and wheel-casing of excessive

pressure and water-hammer when the vanes are closed. If the guide-vanes are suddenly closed, the pressure-regulator discharge is opened, but has the general tendency to close, and does so gradually through the agency of a relay valve and dashpot arrangement. If the vanes are closed slowly, the regulator does not operate. The dashpot can be cut out, and the pressure-regulator will then act as a by-pass, being closed when the vanes are open, and *vice versâ*. Tests were made of the relief-valve at the time the turbine was first put in operation. The machine was running at full load, discharging about 155 second-feet; the vanes were suddenly closed, the relief-valve opened at the same time, and closed after a period of 30 seconds. The total rise in pressure was 15 pounds above the static, or 28 pounds above the working pressure. With conditions at 1,000-kilowatt load, the time of closing was 5 seconds, and the rise in pressure above static was 41 pounds, or 42 pounds above the working pressure. The guide-vanes and relief-valve are operated by a "Type N" Lombard governor, which is connected to the rocker-shaft by suitable pinions and segment, the relief-valve being lever-connected to the bell-crank of the shifting-ring lever.

230. From the hydraulic standpoint the striking feature of the Centerville plant is the use of a regular reaction type of turbine for the 5,500-kilowatt unit at an effective head as great as 550 feet. The ordinary practice under such conditions has been to use impulse wheels of one form or another, on account partly of the speed conditions, and partly because of the difficulty of getting a turbine to stand up under the high velocity of the water, often carrying some sand in suspension. For the development of moderate power at any convenient speed, the impulse wheel is extremely well suited; but as the output rises, there comes a necessity for more and bigger nozzles, met commonly by the use of double or triple nozzles, and often by the addition of another wheel on the same shaft. By the time this is done, it is fairly evident that further increase could well be made by making the entire periphery of the wheel active, and this leads naturally to regular turbine construction, as it has in this case. Given the requirement of large output, the turbine meets the needs of the case admirably. The next question is the endurance of the turbine at such heads. The older turbines of cast iron proved generally inadequate. The Centerville turbine, in all essential

parts, is of steel, the runner being a solid steel casting on a forged steel shaft. Governing is accomplished by shifting the guide-blades, which are forged solid with their pivots and held between steel rings. Especial care has been taken in the hydraulic works, to free the water from grit; and there seems to be no reason why the wheel should not stand up in service quite as well as an ordinary impulse wheel.

231. **Plant at Electron.** The Puget Sound Power Company's hydro-electric plant (Fig. 193) is located on the Puyallup River, 32 miles from Tacoma and 48 miles from Seattle, Washington. This river has its origin in the glaciers and snow peaks of Mt. Rainier, the highest mountain in the United States; consequently an unfailing source of water is assured from the melting snow and ice.

The water-power scheme consists of the diversion of the Puyallup River, and the conduction of its flow by means of a flume 10 miles long to a reservoir located on a high plateau, and thence by steel pipes to the Pelton wheels, which operate under a head of 865 feet. The flume and reservoir are constructed with a view to the ultimate development of 60,000 horse-power; and the present equipment (1908) consists of four direct-connected Pelton wheels, each driving a 3,500-kilowatt General Electric generator at 225 revolutions per minute; and two Pelton wheels, each direct-connected to a 150-kilowatt exciter. Each wheel unit has an overload capacity of 7,500 horse-power, making the present output of the station 30,000 horse-power. This power is transmitted to Tacoma and to Seattle, being used for the various industrial enterprises in that section, and particularly for operating the extensive system of suburban electric roads in the vicinity of Seattle.

232. The ultimate installation is to consist of eight units, and the entire equipment is so arranged as to provide for complete pilot control of both water-wheel and electrical apparatus, which is accomplished from the switchboard at one end of the building. The complete equipment of eight units will require a building considerably over 200 feet in length. It was found necessary to reduce the length of each unit to the minimum, and for this and other reasons the Pelton "double-overhung" construction was adopted. This construction consists of one water-wheel overhanging each end of the shaft beyond the bearings, of which there are two for each unit,

Fig. 193. Power Station of the Puget Sound Power Company at Electron, Washington. Located 48 miles from Seattle, 32 miles from Tacoma, on the Puyallup River, which originates in the glaciers of Mount Rainier (Tacoma). From the reservoir up the mountain side, fed by a flume 10 miles long, water under a head of 865 feet is delivered to the Pelton water-wheels connected to the generators, through riveted-steel penstocks (one of which is seen at the right). Power is transmitted to Tacoma, Seattle, and other points, for street-car, lighting, and other purposes.

placed one on each side of the engine-type generator, which is in the center of the bed-plate; it enables the bearings to be more equally loaded than if the entire output were obtained from one wheel. Each of the two wheels on each unit is required to develop 3,750 horse-power capacity when operating under an effective head of 865 feet, at 225 revolutions per minute. Each wheel is provided with a 24-inch single-disc, bronze-mounted gate-valve, with 5-inch by-pass arranged for operating normally by electric motors from the switchboard, and also provided with gear and worm-wheel for quick and slow motion by hand. There is one combination needle and deflecting nozzle, with ball-and-socket joint, for each wheel, the weight of the swinging portion being suitably counterbalanced, and the position of the needle controlled by a hand-wheel. The power developed on each wheel is controlled in two different ways—*first*, by the deflecting portion of the nozzle, which is actuated by an automatic governor, thus limiting the quantity of water impinging on the wheel; and *second*, by varying the flow of water through the nozzle by means of the needle device above mentioned. Sudden changes of load are taken care of by means of the governor and deflecting nozzle; and consequently there is no variation in velocity of water in the main pipes; hence, no danger of water-ram. The adjustment of the needle to vary the flow through the nozzle is a comparatively slow operation and therefore cannot injure the pipe-line.

233. The *Pike's Peak Hydro-Electric Company*, of Colorado Springs, Colorado, has the distinction of operating a water-wheel plant under the highest head available in the United States. In fact, there is but one installation in the world utilizing a higher head, and that for only a small amount of power—namely, the Vauvry plant in Switzerland, already described.

The plant in question is located on the outskirts of the town of Manitou, Colorado, and consists of three Pelton units, each direct-connected to a 750-kilowatt electric generator running at 450 revolutions per minute. The net head on the Pelton wheels is 2,150 feet, equivalent to the enormous pressure of 935 pounds per square inch.

234. The wheels are mounted in the pulley compartment of the generator, and are provided with combination needle and deflecting nozzles operated by hydraulic governors. The gates, nozzles,

and other pressure parts are of cast steel, designed with a large safety factor, and were subjected to a cold-water test of 2,000 pounds per square inch before installing. The wheels proper consist of cast-steel discs with gun-metal buckets, fine-ground and machined inside. Each wheel has an overload capacity of 1,500 horse-power.

Current is transmitted to Colorado Springs for power and lighting purposes, and is also largely consumed by the many mines and mills in that vicinity.

235. The *Rio das Lazes** hydro-electric station of the Rio de Janeiro Tramway Light & Power Company, is to have an initial installation of 54,000 horse-power, consisting of six 9,000 horse-power units.

A concrete dam, 115 feet high and 92 feet thick at the base, was built at a series of falls on the Rio das Lazes, 56 miles from Rio de Janeiro, Brazil, to develop a storage reservoir for the station. Two 8-foot riveted-steel supply lines about 6,000 feet long, lead from this dam to a cross-receiver near the power-house; one 12-inch and six 36-inch pipes extend from this receiver to the station, at which the total head varies from 950 to 1,000 feet. These pipes are all of welded steel plates; the thickness of the plates for the 36-inch pipe varies from 0.4 to 0.7 inch, and that of the 12-inch pipe, from 0.25 to 0.3 inch, depending on the head. The receiver is connected with a small service reservoir at the same elevation as the main storage reservoir, this small reservoir acting as an equalizer to maintain and regulate the flow in the supply lines.

At the power-house, each 36-inch pipe-line connects with the penstock of a 9,000-horse-power vertical-shaft impulse wheel. The main wheels are each direct-connected to a 6,000-horse-power Westinghouse generator, the units being designed to operate at 300 revolutions per minute. The 500-horse-power wheels are each direct-connected to direct-current generators, which supply excitation current to the main units, and operate at 500 revolutions per minute.

236. Each main wheel has four variable-orifice needle-nozzles, through which water is supplied to the runner of the wheel. The operation of these nozzles is controlled by a special type of oil-pres-

* *Engineering Record*, October, 1907.

sure governor of the fly-ball type. This governor is geared to the shaft of the unit, and actuates a valve in a pipe supplying oil to a cylinder that directly controls the position of the needle-nozzles. Each wheel is also provided with a relief-valve, which is connected to the governor in such manner that it is opened only when all or a relatively large percentage of the load on the generator suddenly drops off. Dependence for speed and pressure regulation is placed entirely on the governor and the relief-valve, respectively, as the nozzles are not deflected. The whole arrangement is therefore water-saving, since the relief-valve is wasting water for a few seconds only when the load is decreasing rapidly.

237. The wheels of the main units have a guaranteed efficiency of 82 per cent at full load. The governor is guaranteed to prevent a speed variation of more than 10 per cent when the full load is suddenly thrown off, the difference in speed between full load and no load being only 2 per cent. The relief-valves will prevent an increase of more than 3 per cent in the water-pressure in the penstocks.

The runners of the impulse wheels of the exciter units are mounted on an overhang of the shaft of the direct-current generators. Each of these wheels has a single needle-nozzle, controlled by an oil-pressure governor similar to those for the main units.

COST OF WATER-POWER*

238. Water-power is generally, though not necessarily, the cheapest form of power development. The water itself costs nothing, but it cannot be used as nature provides it. The cost of preparing it for proper use in wheels, and of providing for its control, will involve considerable expense, as a rule; and the interest on this expense will correspondingly be the largest charge against power cost, and will, so far as its importance is concerned, be commensurate with the fuel charges in gas, steam, or oil engines. The cost of developing water-power depends almost entirely on the local situation, and cannot be reduced to any formula or rule. In the early days of water-power development—say over 1,000 horse-power—only low heads were utilized, and the wheels were located at the side of the dam, making the total development involve little more than a timber

*Articles 238 to 240 are abstracted from a paper by Prof. Charles E. Lucke, of Columbia University, New York, on "Power Costs."

and stone dam, with a house at one end. As the demand grew for more power from the same stream, it was necessary to construct a canal for the purpose of bringing the water around the original power-house to some other house. Thus the expense began to increase materially, from the difficulty of bringing the greatest quantity of water to the wheels under the largest heads by long pipe-lines, or combinations of pipe-lines, canals, flumes, tunnels, and vertical shafts. The simpler development complete cost about $40.00 per horse-power, without electrical equipment. This has now increased so that a minimum of $75.00 per kilowatt is considered a very good proposition; while a maximum of $200.00 per kilowatt is not by any means prohibitive, including electrical equipment. The mean is somewhere near $100 for very large installations favorably placed, like those at Niagara Falls. Taking these two limits, and a gross interest charge of 5 per cent, with an average depreciation of 4 per cent, and with insurance and taxes at 1 per cent, there is a total charge of 10 per cent on the first cost, or $7.50 per kilowatt-year, as a low limit, and $20 per kilowatt-year as a high limit. The operating expenses, labor, oil, waste, repairs, etc., may be expected to cost from $1 to $5 per kilowatt-year, which places the cost of electric power at the bus-bars between the rare low limit of $8.50 per kilowatt-year, and the high limit of $25 per kilowatt-year for full load, 24-hour power, for these rates of charging expenses.

Besides the increase in development cost, there is another item that sometimes enters; and that is an increase in land expense or land damage, which may be large in settled communities. The increase in cost is not all due to increase in cost of machinery; this has probably decreased, not only from better methods of manufacture and more competition, but also from the use of higher heads and better wheels and electric generators.

239. Just how much can be paid profitably for the development of water-power, either now or in the future, will be measured solely by the power cost of that one of the competing systems—steam, gas or oil—most available in the same locality, or, if not in the same locality, at some point within the limits of electrical transmission. To the cost of generating water-power, must be added transmission cost, involving fixed charges on lines, transformers, switchboard, and other equipment, together with their maintenance and operating

charges. All of these together may add to first cost $30 per kilowatt, and increase the power cost for a 150-mile transmission $5 per kilowatt-year.

After analyzing the details of power cost for oil, gas, and steam development, Professor Lucke continues:

240. These costs may now be summarized for comparison, as follows:

COMPARISON OF POWER COSTS

Conditions Assumed: Stations consisting of six units, two in reserve, and four working on 24 hours rated load, with the exception of the water-power. First cost and fixed charges are based on the capacity of 150 per cent of the output.

	Water Power	Oil Engines	Gas Engine and Producer	Steam Plant
First cost per k.w.-rating....	$ 75.00–$200.00	160-k.w. units $217.00	600-k.w. units $270.00	5,000-k.w. units $110.00–$150.00
Fixed charges, rate per cent ..	10 per cent	10 per cent	10 per cent	10 per cent
Fixed charges, per k.w.-year..	$7.50–$20.00	$21.70	$27.00	$16.50–$22.50
Operating and mfg. costs per k.w.-year	$1.00–$5.00	$56.94	$38.54	$52.50
Total power costs per k.w.-year	$8.50–$25.00	$78.64	$65.54	$69.00–$75.00

From these figures it appears that we have not yet reached the limit of cost of development of water-powers which may be advisable. It apparently would pay to spend even more money than $200, the present maximum per kilowatt for water-power development, if there were no other considerations entering. Among the chief considerations of this kind, may be set down that of transportation of products from the works, and of raw material to the works; but this must be considered against the question of transmission of current from the waterfall to a convenient point of transportation.

241. "Whenever the development of a water-power for the purpose of selling water or mechanical or electrical energy is under consideration, the most important question to be decided is: What is the limit of cost per horse-power that may be expended for a development and still leave the plant a financial success; or what is a reasonable price to be charged per horse-power per year?"

"A great amount of data has been published in regard to the cost of hydraulic power and power-plants; but, as water-powers present an infinite variety of conditions, such prices of other plants should be used only with the greatest precaution. A few general figures, intended to apply to conditions at present prevailing in the northern part of the United States and in Canada, may be given here.

242. "A water-power electric plant, including transmission line and substation, where such are required, but without the local distribution, should not cost more than $100 per electrical horse-power, if situated in a remote location or in a farming district; but $150 to $200 may be expended per electrical horse-power for power-plant, transmission, and substation, if the power can be sold in a large city or industrial district.

"The price charged for power water per gross horse-power per year, delivered at or near the customer's turbines, may be taken at from $5 to $15, the lower figure being for remote locations, low heads, and large powers, and the higher figure for the reverse conditions. The price of $15 to $25 per mechanical horse-power per year at the power-house, or of $25 to $50 per electrical horse-power per year delivered to the customer, may be taken as the limits paid at present. Here, again, the lower rate is for remote locations and large powers, and the higher for the reverse conditions.

"It is also safe to state that in a climate such as that of the northern part of the United States, and in Canada with its long and severe winters, it does not pay to develop a water-power if the power produced will cost more than 75 per cent of the amount for which steam-power could be produced in the same locality.

"In Canada, with the great number of water-powers yet undeveloped or only partly utilized, it must be regarded as poor policy to install a larger plant than can be run at all stages of the water, or to have a great proportion of power dependent upon storage lakes during the months of low water.

"A water-power requiring an auxiliary steam plant during the low-water season, can only pay if either the cost of the development is exceptionally low, or the locality very favorable for the sale of power, or both."*

* Thurso, "Modern Turbine Practice."

IMPROVEMENT OF THE MISSISSIPPI RIVER, PLUM POINT REACH
Weaving a connecting mat to prevent erosion.

RIVER AND HARBOR IMPROVEMENT

RIVER IMPROVEMENT

To quote a well-known author on Civil Engineering:

"There is no subject falling within the province of the engineer's art, that presents greater difficulties and more uncertain issues than the improvement of rivers. Ever subject to important changes in their regimen (*i. e.*, the relations which are found to exist between the cross-section, longitudinal slope, nature of the bed, and volume of water), as the regions by which they are fed are cleared of their forests and brought under cultivation, one century sees them deep, flowing with an equable current, and liable only to a gradual increase in volume during the seasons of freshets, while the next finds their beds a prey to sudden and great freshets, which leave them, after their violent passage, obstructed by ever-shifting bars and elbows."

This being so, whatever the nature of the engineering project being undertaken for the improvement of a river, it will have in the main as its object the removal of these obstructions so as to increase either the depth or width, or both, of the channel, to aid navigation and thus permit the use of vessels of heavier draught; or the protection of the banks to prevent the destructive effect of the freshets causing these obstructions.

The River Survey. Such an undertaking requires a careful and minute study of the characteristics of the river itself; and the more complete this is, the more satisfactorily may the improvements be planned. The collection of this information would usually be made by means of a survey, where no records exist, to determine the many and varying factors which may control or help to affect the nature of the improvement. This is usually a difficult proposition, and may require both a nautical and an engineering survey. In the former, it is necessary to determine the outline of the channel, and its depth, the nature of the banks, currents, tides, winds, etc.; in the latter, the volume of water discharged at varying stages, the nature of the bed, the geology of the watershed, topography, etc. Currents are of greater importance to the engineer than to the navigator; and their determination, the most important and difficult part of an engineering survey. The survey need not necessarily be made with the level, transit, or other engineering instruments; but if not so made, the knowledge must be supplied from maps, charts, statistics, and records that have been compiled from data so collected.

Specifically such a survey would require an investigation of the main stream from its mouth to the source, together with the determination of the flow of the streams entering it, and should include a complete analysis of the area of the watershed, both as to topography, hydrography, rainfall, run-off, and stream-flow. While some of these investigations may more properly come within the province of the water-supply engineer, they nevertheless form the basis for the work that the engineer engaged in River Improvement must undertake.

The *watershed* of a river is that area, tributary to it, from which the rain falling upon it will be directed to the stream. The amount of water flowing in the stream will therefore depend upon several things—*first*, and most important, the amount of rainfall precipitated on the watershed or basin; and *second*, the portion of this which reaches the stream. The latter is controlled or affected by many factors, the more important of which are the evaporation that takes place, the nature and amount of the vegetation covering the watershed, and its geology.

The *topography* of a country or district refers to the relation that exists between hills and valleys, streams, roads, railroads, property lines, fence lines, and such other data; and which, when located on a map, will indicate to the observer the changes in elevation of the ground, and the positions of the natural and artificial features of it.

Hydrography concerns itself particularly with the position of streams, their means of supply, volume, currents, slopes, cross-section, dry-season and freshet flow, and other similar characteristics.

The rainfall is pre-eminently the most important of all in its effect upon stream-flow. Where this is unknown in a district tributary to a river under investigation, it may be necessary to make an extended series of observations to determine its amount. Usually, however, this will be found to have already been done, since in all

IMPROVING THE HARBOR OF CLEVELAND, OHIO

Laying concrete blocks in repairing west breakwater. This work necessitates the employment of divers.

civilized countries statistics are now kept, and have been for a long period of years, of the rate of rainfall in different localities.

Rainfall follows no set law; and where a river passes through extensive territory, the rate is generally as variable as the climatic conditions of the country; and hence, with no data at hand, it would require a determination in each separate district. Such records as these have no value, however, unless kept over a long period of years, as the amount of precipitation in any one year is no indicator of what it may be in any other.

Run-off is a term which refers to that portion of the rainfall which actually finds its way to the streams. No definite means for determining this has been established, as so many variable factors enter into the problem; but the United States Government has made elaborate investigations to formulate, if possible, a law for some of the less variable ones. It is quite evident, however, that it will depend largely upon the nature of the surface of the watershed, the slope, presence or absence of vegetation, etc.

That the geology will affect the stream-flow is clear. For example, if rain falls upon an impervious strata of rock, practically all of it will be diverted to the water courses; while, on the other hand, if it falls upon a porous formation, such as limestone or sandstone, only a small proportion may be delivered to the stream.

Vegetation is known beyond question to have a marked effect upon the regularity of the flow of rivers. It is observed that as a country becomes more thickly populated, the land more thoroughly cleared of timber and underbrush, the more speedily does the rainfall reach the river, and the less regular is the flow of the stream. In consequence of this, rivers passing through such districts approach the conditions of a torrent at some periods, and are almost dry at others. In Switzerland, such a deep appreciation is had of the effect on stream-flow of growing timber situated on the watershed, that laws have been passed by the Government for the promotion and encouragement of its growth on the rugged mountain sides. In our own country, timber preserves have been established, not only for the lumber they contain, but also for the beneficial effect they will have in regulating the stream-flow. The particular feature of timber land in relation to the regulation of the flow of rivers, is that the leaves and branches overhead produce shade which prevents speedy evaporation or the melting of the snow, while the roots and tendrils hold the water and give it off only gradually.

It is not necessary here to discuss in great detail these or other features of the question relating to stream-flow; but it is necessary to point out that such factors have their effect upon the rapidity and amount that a river discharges.

Besides the necessity of determining such factors as have been mentioned above, it is important to know the nature and amount of the traffic which a river carries, and its relation to the improvements proposed.

Flow of Rivers. The velocity of a river and the amount of water it discharges may be estimated approximately by a study such as that indicated in the preceding pages; but there are more usual and satisfactory methods, and they will here receive some consideration.

It is customary in the preliminary survey, if the river is small, to select some convenient stretch with a fairly straight course, uniform channel, and regular flow, where a gauging station may be located and observations conducted for the purpose of learning the nature of the profile, the velocity of flow, quantity of discharge, and other like data.

Here a wire or rope will be stretched across from bank to bank, at right angles to the axis of the stream, so that soundings with a rod may be taken at intervals along it, and the cross-section or profile thus plotted. These intervals will depend upon the regularity of the bed, the more regular it is in outline, the fewer soundings being found necessary. A determination of the velocity of the current will also be made at this point, either by means of a weir, by observations on floats, or by the use of the current meter (see Fig. 1).

While a scheme of this kind may be perfectly satisfactory for streams of small dimensions, and is, in fact, the method adopted by the United States Geological Survey in its gauging of streams of this character, for a large river it may be absolutely insufficient.

Were the river a mile wide and one hundred feet deep, it is quite evident that to stretch a rope across would be impossible, as would also the measurement of the depth by a rod. Here, too, floats prove unreliable, though they are frequently used, because of contrary currents; and the measurement of the fall is hard to obtain, because of surface undulations or local elevations arising from the curvature of the bed or the effect of the wind.

Other methods must therefore be employed to determine the profile, while the gauging of the velocity should be undertaken by means of the current meter.

To arrive at an accurate determination of the cross-section of a large river it is usual to establish transit stations on the shores in

Fig. 1. Stream Measurement.
Courtesy of U. S. Geological Survey.

known positions; and from these to take observations on a boat which contains the sounding party, by reading the horizontal angle with reference to some base line. The sounding party consists of several members—one to row, one to heave the lead and read the depth on the chain, and a third to keep notes and to signal to the transitmen so that they may sight on his flag and thus locate the spot where the sounding is being taken. Various modifications of this general method occur, but the principles are the same.

Under some circumstances it may be deemed advisable to dispense with the above methods for the determination of velocity; and resort is then had to the use of formulæ. In this respect, that best known is the one deduced by Chezy in 1775, after a thorough analysis of all available data. The formula,

$$v = c\sqrt{rs}$$

is one for the mean velocity, and the factors appearing in it have the following significance:

c = An experimental coefficient depending upon the nature of the river bed;

r = Hydraulic radius.

s = Slope $= \dfrac{h}{l}$ = Height divided by Length.

It is quite evident, then, that the important quantity in the determination of the mean velocity by this formula, is the factor c, since r and s are easily measured; and hence the accuracy of c will determine the reliability of the formula in expressing the approximate mean velocity.

The factor c is generally derived from what is known as Kutter's formula, in which,

$$c = \frac{\dfrac{1.811}{n} + 41.65 + \dfrac{0.00281}{s}}{1 + \sqrt{\dfrac{n}{r}}\left(41.65 + \dfrac{0.00281}{s}\right)},$$

and in which it is seen that c is made to depend upon r and s, and also upon a quantity n, which is called the *friction factor*, and which expresses numerically the condition of the roughness of the channel.

In most cases, however, it is more expedient and much more satisfactory, particularly in the case of large streams, to gauge the flow by means of floats of various types, or by the use of a current meter. Floats were extensively employed on the Mississippi River in some of the earlier surveys, where elaborate pains were taken to secure accurate values.

Where either floats or the meter are used, the vertical cross-section of the river at right angles to its axis is divided into vertical strips of such width that the velocity in each may be in approximate conformity with the general conditions of flow of the stream. The various velocities in each one of these strips is measured, or at least such velocities are measured that a mean velocity may be deduced; and the sum of these mean velocities for all the strips, divided by the number of sections, will give the mean velocity for the entire cross-section. See Fig. 2.

It should be stated here that usually, in the measurement of streams, it is the mean velocity that is desired, or the velocity which would result if the discharge were to be divided by the area of the cross-section. The mean velocity is, however, a difficult quantity to determine; for not only do the velocities vary on the surface from

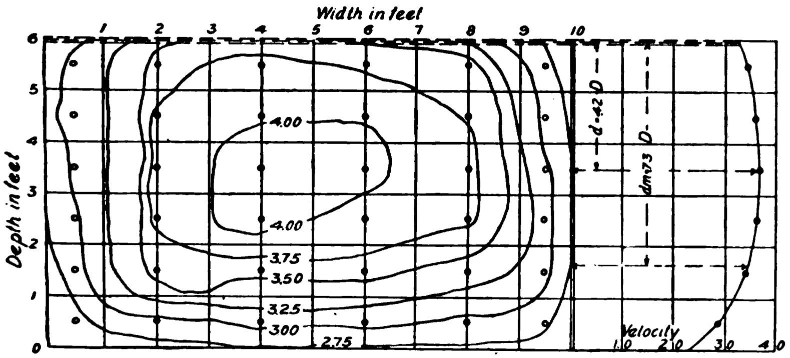


Fig. 2. Velocity Curves in Cross-Section.

shore to shore, but they also vary between the surface and the bottom. The greatest surface velocity will usually be found over that part of the channel where it is deepest; while the greatest velocity in any vertical between the surface and the bottom, will be found at a point about four-tenths below the surface. The variation in the surface velocity is due to the greater friction that the water encounters near the banks than in mid-stream; while the variation in velocity in a vertical, is due to the greater friction that the lower layers of water meet with in flowing over the bed, than those layers above, which flow merely over other layers of water.

For this reason, and because the surface floats feel the effects of winds, currents, eddies, etc., they will give only approximate values, even if the observations be taken in various sections of the river.

Mid-depth floats are very much better, since they are unaffected by the wind, and record a more normal velocity. Such a float consists of two parts—that which is submerged and is carried by the current; and that attached to the former by a thin cord or wire, and floating on the surface, thus serving as an indicator to locate the one submerged. The wire or cord should be as thin as possible, to avoid any loss of velocity due to friction; and the surface attachment should be as small as may be permissible under the circumstances (see Fig. 3).

Neither the surface nor the mid-depth float gives the mean velocity of flow; but formula may be applied which will reduce these observations to the quantity desired, though the result will be but an approximate value.

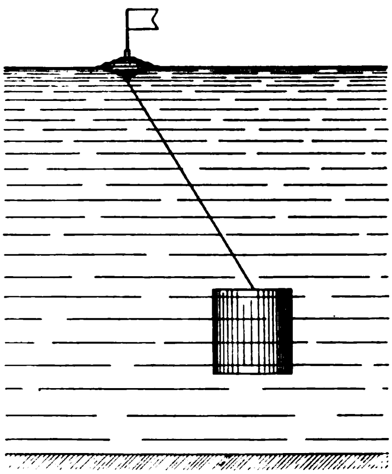


Fig. 3. Mid-Depth Float.

Another form of float is that known as the *rod float*. It consists of a hollow rod or cylinder of tin, and is weighted with stone or shot, so as to stand vertical in the water and remain immersed to a depth almost equal to that of the stream. In view of the fact that this float almost reaches the bottom, its speed will represent the mean velocity of all those taken in a vertical plane. An observation on any of these floats is made simply by noting the time required for it to pass a given length of the stream.

Current meters are better adapted to the conditions found in large rivers, as they give more accurate results than floats. The meter is constructed in various forms, but generally consists of a set of revolving cups attached to a vertical rod (Figs. 4 and 5). These are made to point upstream by a vane or tail which trails behind the cups, so that as the water strikes them they will be made to revolve. This revolution is recorded by an automatic electrical instrument, having a dial with pointers, which indicates the speed. Readings are taken systematically throughout the entire cross-section; and from this record a mean velocity is calculated. This question of velocity is of prime importance in the discussion of river improvements, since upon it depend the transporting and eroding powers of the river.

Fig. 4. Price Electric Current Meters.
Courtesy of U. S. Geological Survey.

FLOODS

The foregoing remarks on the flow of rivers refer primarily to [illegible] should be remembered that any improvement [illegible] and is desired to remain as a permanent

Fig. [illegible]. [illegible] Position.

piece of work, must be designed to withstand the extreme conditions of flood, as well as the less severe one of normal flow.

Floods depend primarily upon the amount, suddenness, and duration of the rainfall, but the nature of the watershed also has a bearing on the subject, so that all those factors which have been referred to as affecting the flow of rivers (*i. e.*, geology, vegetation, slope, etc.) may be applied with equal force in the discussion of the subject of floods.

When an improvement of a river is contemplated, it is necessary to know at what seasons of the year the floods are apt to take place, the amount of water they carry, and any other characteristics, so that precautionary methods may be adopted in the design to insure it withstanding the destructive effects of the water. At the same time, similar data should be collected with regard to the dry-weather or

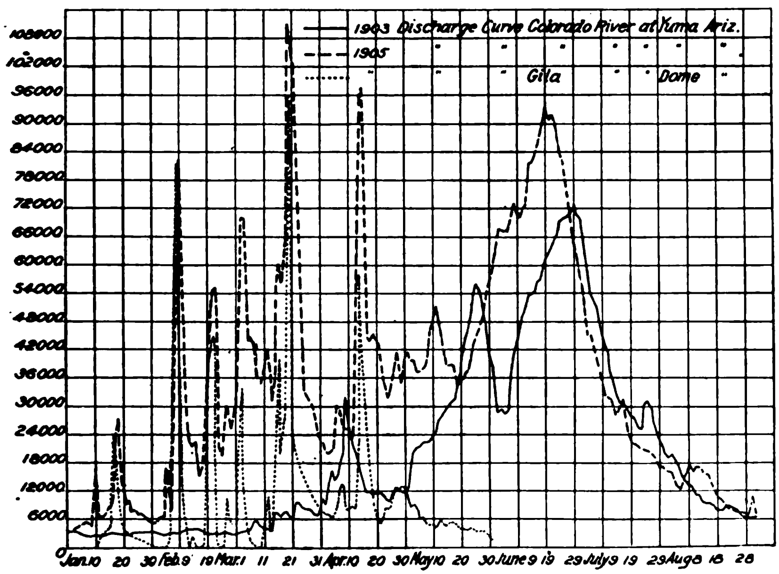


Fig. 6. Diagram Showing Flood of Colorado and Gila Rivers.

minimum flow and the frequency with which such conditions occur, to study its possible effect upon the protective works.

In tropical climates, where dry and wet seasons alternate, these variations in the flow of a river are not only much more common but also much more pronounced than in a cold or temperate region. In the former case, the rivers are very likely to vary in their flow from a small stream to a raging torrent—a notable example of which in our own country is the Colorado River, one of its chief characteristics being the extreme variability of its flow (see Fig. 6).

The most serious floods are those of the spring season, and are generally produced by a heavy rain falling on melting snow. A similar condition may result when an excessive rain occurs during a very short period of time. Winter floods are less destructive in agricultural districts than those of summer, since during the former season the ground is frozen hard and contains no crops to be damaged.

Under normal conditions, the channel of a river is eminently fitted to care for the amount of water which it is required to discharge; but at flood periods, unless the banks are high or well protected, the river is likely to overflow and to cause destruction of life and property in the vicinity. A large part of the work of river improvement concerns itself, therefore, with the study of means to protect the banks so that this loss may be avoided.

In many cases, flood conditions will have but little effect upon navigation—rather aiding it, in fact, than hindering it, by furnishing a greater depth and width of channel. On the other hand, however, currents are sometimes produced which make navigation dangerous; besides which, the elevation of the water may be so great that the passage of boats beneath bridges may be made impossible.

Floods, unless confined to the river's channel, may cause great damage; but where so confined and controlled, they may serve to remove from the bed of the stream the sediment which has been deposited there by the slower-moving normal current. These destructive floods frequently occur along the Ohio and Mississippi rivers. Thus, the Ohio has experienced several floods of 50 feet in height; and in 1884, the river at Cincinnati rose to an elevation of over 70 feet.

Rivers fed from lakes seem to be less likely to experience floods than those fed from springs, streams, and the melting snows of mountains. On the other hand, it has been observed that the flood period of the Mississippi does not occur when the flow from the mountains is large, but rather when it is small.

FLOOD CONTROL

To prevent havoc from floods, provision must be made to care for the surplus water in some satisfactory manner. On the lower Mississippi where these destructive risings of the river have done damage probably in excess of what has occurred in the case of any other river in the world, and where, in consequence, the subject has received considerable scientific attention, the three following methods, which are indicative of good engineering practice, have been suggested as a means of protecting the bottom lands and thus avoiding the losses consequent upon overflow:

(1) To modify the actual relations existing between the accelerating and retarding forces in the channel in such a manner as to enable the former to carry off the surplus water without so great a rise in the surface. To this class of protective works belong *cut-offs.*

(2) To reduce the maximum discharge by diverting the excess of water to *tributaries, artificial reservoirs,* and *artificial outlets.*

(3) To confine the water to the channel and allow it to regulate its own discharge. Such a procedure as this requires *embankments* or *levees.*

The more important methods of those enumerated are levees and artificial reservoirs, so that the others will receive but brief mention.

Cut-Offs. A *cut-off* or *straight cut* is either a natural or artificially constructed by-pass, which diverts the water in the main channel from the upper to the lower side of a bend. It is easy to see that such an artifice reduces the length of channel; lowers the surface of the water below the point where it first enters the cut-off; and thus increasing the fall, helps to remove the sediment deposited on the bend. Under some circumstances, cut-offs have been in operation on the Mississippi and its tributaries, either by the action of the river itself or by design; but no particular advantages may be said to have resulted.

With this means of control, it will be necessary to make the cut-offs continuous from the mouth of the river to that point where it is proposed to give relief; for, where the water from the cut-off re-enters the main channel, an increased flow is again occasioned; and farther downstream it will be as necessary to introduce cuts as it was above, unless the river itself is capable of handling this maximum discharge. Such a system, too, may be all right for small streams where the high stage lasts for only a few hours; but where this period extends over several weeks, as it does on the Mississippi, it is not so effective.

It is assumed, in this system, that the greatest velocity of the water in the part where the slope has been increased by a cut-off, will bring a larger volume in floods to the portion below the cut, where the slope has not been increased, and where, therefore, the water will rise higher than before. It is hence necessary that a second cut be introduced below the first; and so on to the mouth.

Diversion to Tributaries. Diverting the flow of floods to tributaries has often been suggested as a means of getting rid of the surplus water; but it has never been applied on a large scale. There are several locations along the Mississippi where the application is quite feasible; but other means have been employed in view of the great expense attached to the undertaking. The plan would generally require the construction of a long diverting canal or channel; and this, owing to the fact that different drainage areas are generally at some distance apart or else separated by intervening hills, would ordinarily increase the cost of construction to a prohibitive amount.

It has been proposed, however, with regard to the upper Mississippi, that such a channel be cut through the prairie between the "Great Bend" and the Mouse River, which drains into another system, that of the Red River of the North. This would direct the flood flow of the Missouri to the northward rather than allow it to enter the Mississippi.

There are several factors to be taken into consideration before such a scheme as this can be pronounced a success:

First—The item of expense that would attach to the construction of such a canal, which in the above-mentioned case would be 40 miles long;

Second—The difference in elevation between the two streams, which of necessity would have to be sufficient to guarantee a flow in the desired direction;

Third—The periods at which maximum flood occurs in each river, to see that such times do not coincide, giving the greatest flow in each at the same time.

Storage Reservoirs. These furnish a legitimate means of providing for the surplus water of the flood period, as by them it may be impounded when least needed, and gradually fed to the watercourses when their discharge becomes diminished.

Preferably, such reservoirs should be located in the naturally formed pockets of the hills, along the upper levels of the river, and near the source; for here nature makes the construction of a dam to hold back the flood flow a comparatively easy matter. On the other hand, also, near the mouth, the land adjoining a river is generally flat, so that fewer opportunities for a reservoir site present themselves, and usually these can be constructed only at much greater expense.

With such reservoirs, the water on the lower part of the basin has an opportunity to discharge into the main stream without the augmentation of the excess from above; and hence the river is better able to handle it.

The following abstract, because of the completeness of the treatment, is given in some detail. It is taken from the U. S. Government report on the subject of Storage Reservoirs, made in 1898 by Captain H. S. Chittenden:

Natural Reservoirs. Nature presents abundant examples of the effective control of stream-flow through the agency of reservoirs. There are, indeed, comparatively few streams whose flow is wholly uninfluenced by such action. The most perfect example in the world, both as to the magnitude of the stream and the completeness of control, is the St. Lawrence River, embracing the great chain of North American lakes. Considering only that portion of the system which lies above the Falls of Niagara, let the flow at the outlet be compared with that of other streams of similar magnitude. For this purpose, take the Niagara River at Buffalo; the Ohio at Paducah, Ky.; the Missouri at its mouth; and the Mississippi just above the mouth of the Missouri. The following table gives the area of watershed in square miles, and the mean annual discharge in cubic feet per second, of each:

	Niagara	Ohio	Missouri	Mississippi
Watershed	265,095	205,750	530,810	171,570
Discharge	232,800	307,000	100,000	130,000

The maximum and minimum discharges, except for the Niagara, show a much greater divergence, the ratios of maximum discharge to minimum discharge for 1883 being as follows: Niagara, 1.19; Ohio, 28.22; Missouri, 29; and upper Mississippi, 10.29.

This striking dissimilarity in the regimen of streams of similar magnitude, and, with one exception, of similar climatic conditions, is entirely due to the reservoir action of the Great Lakes.

The mean annual fluctuation of Lake Superior, Lake Michigan, Lake Huron, and Lake Erie, represents an annual storage of 2,419,000,000,000 cubic feet of water, equivalent to above 153,000 cubic feet per second for a period of six months.

In addition to the annual fluctuation, there is constantly going on a periodic change which often requires several years to complete the cycle. As an illustration of this characteristic, take the period of eight years from 1872 to 1879, during which the mean annual level of the four upper lakes rose for a period of four years and fell during the following three years. The total storage represented by this rise of mean level was 4,000,000,000,000 cubic feet. The fall in mean level following the rise was equivalent to 3,627,000,000,000 cubic feet. After this fall, the mean level began to rise again.

The foregoing figures convey some faint idea of the magnitude of the storage of the Great Lakes, and of the way in which it operates to preserve a balance, not only between the wet and dry seasons of each year, but between those cycles of wet and dry years which are continually recurring. These reservoirs absorb the flood waters of spring, and pay them out in the following dry season, thus preventing floods on the one hand and low water on the other. And while these seasonal changes are going on, the lakes respond to the varying conditions of longer periods, levying upon years of more than average precipitation in order to maintain a flow in the outlets during the years of deficiency which are certain to follow.

The result of this storage action of the Great Lakes is to produce a river system radically different in its general characteristics from nearly all other streams. Such conditions as high and low water, as elsewhere understood, are here entirely unknown. Commerce pursues its way through these lakes and rivers without serious hindrance except when ice closes the way; and the river and harbor engineer has little to do with low-water problems or protection against floods, but rather with the deepening of harbors and connecting channels for an ever-increasing size of vessels and volume of commerce.

While it is impracticable to imitate nature on the scale of her own work in the construction of reservoirs, her example has nevertheless been followed very extensively on a smaller scale. In fact, works of this character have been built for a variety of purposes since the remotest antiquity. Many reservoirs have had as a prominent reason for their construction the prevention of floods in the valleys below them, although this has seldom if ever been an exclusive reason. In all examples of reservoir construction, however, the purpose has been to correct the inequalities of nature—to prevent the rapid and destructive flow of rivers at seasons when not needed, and to augment and reinforce that flow when the need does exist.

The largest artificial-reservoir system ever yet constructed is that at the headwaters of the Mississippi River. It is dotted with an immense number of lakes, the total number having been estimated as high as a thousand. Some of the larger of these afford exceptionally favorable opportunities for the inexpensive storage of water. The dams required are low structures; but the area over which the water is raised by them is so extensive that the cost per unit of volume stored is probably the smallest ever yet realized.

These remarkably favorable natural conditions for the storage of water have long attracted public attention; and in 1881 actual construction was begun.

The average annual storage of the reservoirs constructed before 1898 was estimated at about 40,000,000,000 cubic feet, equivalent to about 5,200 cubic feet per second for a period of ninety days. This supply is estimated to increase the gauge height at low water at St. Paul, 357 miles below, from 1 to 2 feet.

The effect upon the navigable stage of the river would, of course, vary with the locality considered, and would diminish rapidly with the distance downstream. But considering that such an improvement is of the most permanent character, depending upon the maintenance of the dams for its perpetuity, the above cost cannot be considered excessive when compared with the vast outlay for the more temporary improvement of these rivers by present methods. A permanent increment of from 10,000 to 20,000 cubic feet per second to the low-water stage of even so large a stream as the Mississippi River, is not to be passed over as a matter of small importance.

Every reservoir built along the course of a stream is, to some degree, a protection against floods in the valley below. The extent of this protection depends, of course, almost entirely on the ratio of its capacity to the flood discharge. A reservoir that can store the entire flow of a stream is an absolute protection against floods for a considerable distance below. It is difficult to propose any general rule for the extent of this control; but, assuming a general similarity of watershed, it would seem not unreasonable to say that it ought to be decisive to at least such a distance below as will give an additional watershed to a stream equal to twice that above the reservoir. This is simply saying that, in the general case, the reduction of a flood wave by one-third of its volume will rob it of its destructive character.

But in a great many cases this control extends very much farther. For example, in the case of a flood caused by the rapid melting of snows in the mountains, reservoirs situated below, capable of impounding this flood, will protect the entire valley as far as its destructive influence would otherwise have reached. When it is remembered that the volume of a destructive flood is only a part—probably always less than half of the total flow of a year—it will be admitted that a storage capacity equal to one-fourth of the run-off, well distributed throughout a watershed, will practically eliminate the evil effects of floods on its streams, and supply a percentage sufficient for the purposes of irrigation.

It is not necessary, though important, that a reservoir should be empty when a flood comes. Even if full, it still moderates the flow of the stream below, the effect varying directly with the superficial area of the reservoir when full, and inversely with the capacity of the spillway. In this respect it acts precisely as does a natural lake. For example, if the spillway of a reservoir or the outlet of a natural lake be of such an increase of discharge, every increment of this depth of outlet means also an increment of the same depth over the entire reservoir. A flood passing such a reservoir will be reduced by the storage resulting from this increment; and before it can produce a full discharge, it must fill the reservoir to the necessary height above the bottom of the spillway. A large reservoir is, therefore, even when full, always a perfect protection against sudden floods. In the case of long-continued floods, it greatly retards the arrival of maximum effect, and gives ample notice of its approach.

In fact, this is a very important feature of reservoir action, even where the capacity of the reservoir is not sufficient entirely to prevent the flood. It does prevent freshets—that is, sudden floods—and in smaller streams it is often the suddenness quite as much as the magnitude of floods that causes damage and loss of life.

A reservoir ceases to be any protection if a flood continues long enough to fill it to such a height that the discharge at the outlet is equal to the entire inflow.

In the case of floods, which are the results of combinations of discharges from the various tributaries, reservoirs may actually operate to increase the combination. Take, for example, the natural reservoirs at the sources of the Mississippi. While they restrain the flood excess in that stream, they keep up a heavy flow for some time after the flood has passed. If this larger flow happens to come in with a flood crest at the junction of some tributary below, it will actually increase the combination over what would have been the case without the reservoirs. In the French investigations, the dams proposed for restraining floods were to have open sluiceways without means of closing them. In the ordinary flow of the stream all the water could pass through; but they were to be so proportioned that when the flow should pass a certain point, the surplus would be retained in the reservoir, the outflow being always limited by the capacity of the open sluices. The arrangement was, therefore, precisely like that of a natural lake without a dam across the outlet. The outflow

IMPROVING THE HARBOR OF CLEVELAND, OHIO

West breakwater, at angle between shore and lake arms, under construction. Concreting between blocks. Decking over filling stone.

could never be entirely restrained, and it would increase in proportion to the height of water in the reservoir.

Now, in the case of a large stream, where flood combination is the really dangerous thing, it was found that these reservoirs, had they actually been constructed, would have increased certain floods. They would have maintained a heavy retard flow on some tributaries, which in their natural condition would have entirely run out before the arrival of floods from other tributaries. As it happened, this retard flow in the one case would have come upon a flood crest in the other, and would actually have increased the natural combination. This, of course, could not be true of reservoirs with closed sluices, unless, as above stated, the reservoirs were entirely filled with the flood passing over them.

It is therefore clear that the efficiency of reservoirs in moderating great floods would have to be a matter of judicious management in controlling combinations, quite as much as of actual capacity.

For weighty reasons, very few, if any, reservoirs have been built for the exclusive purpose of protection against floods in the valleys below them; but there are numerous examples where this has been an important consideration in their construction.

Various works have been constructed in Europe; but all have other motives, in addition to that of flood protection, to justify their construction.

The systematic creation of a comprehensive system of reservoirs on any river for the sole purpose of mitigating the severity of floods, has never been undertaken. The subject has, however, received exhaustive study; and examples of such studies are to be found in France.

Particular emphasis should be placed upon these studies, because they disclose the true obstacle to the use of reservoirs for the sole purpose of flood prevention. It is the cost, not the physical difficulties, which stands in the way. It may be stated that, as a general rule, a sufficient amount of storage can be artificially created in the valley of any stream to rob its floods of their destructive character; but it is equally true that the benefits to be gained will not ordinarily justify the cost.

The reason for this is plain. Floods are only occasional calamities, at worst. Probably, on the majority of streams, destructive floods do not occur, on the average, oftener than once in five years. Every reservoir built for the purpose of flood protection alone would mean the dedication of so much land to a condition of permanent overflow, in order that three or four times as much might be redeemed from occasional overflow. One acre permanently inundated to rescue three or four acres from inundation of a few weeks once in three or four years, and this at a great cost, could not be considered a wise proceeding, no matter how practicable it might be from engineering considerations alone. The cost, coupled with the loss of so much land to industrial uses, would be far greater than that of levees or other methods of flood protection.

In fact, examples of natural reservoirs, while they show conclusively the vast beneficial influences of large reservoirs upon the flow of streams, also disclose the fatal obstacle to their successful imitation by man. In only very few places has nature prepared sites where man can erect works which will create large bodies of water; and even if she had done so, the gain from utilizing them would not equal the loss. The reservoir system of the Great Lakes involves the perpetual withdrawal from agricultural and industrial uses of an area nearly twice the size of the State of New York. It will be found in general, that the surface of the earth where reservoirs could be built on an extensive scale is liable to be of more value in its present condition than it ever could be if covered with water.

The construction of reservoirs for flood protection is not, therefore, to be expected, except where the reservoir is to serve some other purpose as well; and inasmuch as such purposes are not ordinarily extensive enough to develop systems of reservoirs, upon which, rather than upon isolated works, the control of great floods depends, this large control is hardly one of the possibilities of the future. The only probable exception is that of a reservoir system on the watershed of the Missouri River.

For flood protection in isolated cases, however, on a relatively small scale, reservoirs will undoubtedly continue to be built, particularly when they serve other purposes as well. From this point of view, they will always be projects of public importance.

Artificial Outlets. This method consists in reducing the flow of water in the main stream by conveying it to the mouth through other channels. It is claimed by some engineers, that such a reduction of flow in the main stream would increase the amount of sediment deposited in the channel, and thus, instead of depressing it, only tend to raise it and the surface level of the river. On the other hand, investigations made on the Mississippi River Survey tend to contradict this, and show that no such deposition of material occurs.

On that river, such outlets have been practically established by the water breaking through the banks or levees (which breaks are called *crevasses*) and being allowed to drain off into the adjoining, low-lying, swamp land. Here it is not so much the question of how to get rid of the water from the river, but what to do with it when it has gotten into the neighboring bayous and swamps, for the natural drainage proves insufficient, and the back-water rises until plantations are threatened. It therefore requires a special channel to tide-water; and this usually means a prohibition, due to excessive cost. In this particular instance, therefore, it was deemed very much better to protect adjoining lands from floods by an extensive system of levees; and to this end such a system has been and is being constructed.

Levees. Levees are banks of earth built on the shore line (or near it) of rivers or harbors, to prevent the inroad of the water. This method of protection was used by the Romans, Egyptians, and other nations of antiquity; while to-day, wherever rivers are found, the same principles are applied to keep the waters from overflowing the banks or shore. Those countries most actively engaged in this sort of engineering work are the United States, England, France, Spain, Italy, Germany, and Holland. In the latter country, these protective works are called *dikes*, but the significance of this term must not be confused with the later application of the word.

In the United States, the most important application of this principle of engineering is found in connection with the Mississippi River. Until 1884, this work was carried on as required, by the different States bordering the shores of the river; but owing to the unsatisfactory and unscientific results, it was taken out of the hands of the States by the Federal Government. It has now assumed enormous proportions, and annually large sums of money are expended; in fact, up to 1908, $15,000,000 had been spent on improvements of this nature.

The principal factors to be considered in the construction of levees are: Economy of design and maintenance; the permanence of the structure; and its suitability to form part of a system when enlargement is considered necessary.

This last consideration is of particular importance, since the increased elevation of the river bed resulting from the deposition of sediment, must be met by an increase in the height of the levees; and where this modification may take place without radical change in the original plan, great saving will result.

The works should therefore follow some definite plan, and, above all, conform to the principles underlying the flow of rivers. A complete study of the river's currents is hence necessary, as well as a study of the places where erosion occurs, how it is occasioned, the nature of the banks, their height, and the material composing them.

Angles should be avoided as much as possible; and where curves are encountered, they should be made as easy as is consistent with good practice. Generally speaking, the line of a levee follows the river banks; but in the location, regard should also be had for a high and firm foundation, and the line should be selected with that continually in view.

The size and section of a levee will vary with the conditions to be met in that particular locality—that is, specifically, the sort of foundation, the nature of the material of which the levee is to be constructed, and the exposure of the position to tides, winds, and waves.

In America the more usual dimensions are 8 to 10 feet on the top, with a slope of 1 to 3 for ordinary earth, diminishing to proportions of 1 to 5 for sand, with a top width of 15 feet.

If it is necessary to make the levee more than usually high, what is known as a *banquette* is constructed on the rear face, to reinforce the bank. This is really nothing more nor less than a bench or terrace to increase the width of the lower portion and the stability of the whole. They are usually about 20 feet wide, and have a rear slope something less than that used on the river face.

In foreign countries, the cross-section and mode of construction vary considerably from those in the United States and from one another.

It is stated that for a cross-section of equal strength in all its parts, side slopes of one to one are all that are required; but in practice this must be modified according to the angle of repose of the

Recreation Pier on Delaware River, at Chestnut Street, Philadelphia, Pa.

Recreation Pier on Delaware River, at Race Street, Philadelphia, Pa.

VIEWS OF WATER-FRONT OF THE CITY OF PHILADELPHIA, PA.

Courtesy of Geo. S. Webster, Chief Engineer, Bureau of Surveys, Dept. of Public Works.

material. Steeper slopes than 1 to 3 cause the material on the landward side to slip, while on the river face the material is too easily washed away by the current or the waves.

Where the exposure to winds is very great, the front slope is often made as flat as 5 to 1, the back slope being then reduced to 2 or 2.5 to 1. It is found that a flat slope is a great protection against the wash of waves, and that a well-sodded *buckshot* levee, with a slope of 5 to 1, will stand a pretty stiff wind. If the sod be once cut through, however, and a hole made in the clay, the latter is liable to be undermined, and the superincumbent masses of earth fall in huge blocks.

The standard section adopted by the Government for all usual conditions, according to Coppee's work, is as follows: Crown, 8 feet; front of river slope, 3 to 1; back slope, 3 to 1. Where the levee is over 11 feet in height, a banquette, at an elevation of 8 feet below the top of the main levee, is added. The slope of the crown of the banquette is 10 to 1; width of crown, 20 feet; and back slope, 4 to 1. Where the foundation is bad or the material weak, the banquette section, and perhaps the front slope of the main levee, are increased.

The specifications require the levees to be constructed in 2-foot layers, with scrapers, on a well-grubbed and thoroughly plowed foundation containing a small exploration muck-ditch filled back with strong material, the best to be found in the vicinity, and sodded at 2-foot intervals with Bermuda grass.

On that portion of the Mississippi known as the "Fourth District," the dimensions of the standard adopted vary with the height, and are intended to conform more nearly to the supposed theoretically perfect section. These variations may be further modified, as in the other districts, when required by abnormal condition of foundation, material of construction, wave wash, etc.

For levees from 5 to 10 feet in height, the crown is 8 feet; the river slope is 3 to 1; and the land slope is 4 to 1 to within 5 feet of the crown; thence to the crown it is 2½ to 1.

For levees from 15 to 20 feet in height, the crown is 8 feet; the river slope is 3 to 1; the first 8 feet of the land slope from the ground is 6 to 1; the next 6 feet, 4 to 1; and thence to the crown, 2½ to 1.

In the upper districts 10 per cent of the height, both in wheelbarrow and team work, is required for shrinkage.

These standard sections are expected to withstand the water to within 3 feet of the crown of the levee, without excessive saturation or change of form, and to give unqualified protection under all normal conditions of foundation and materials of construction.

When subjected to water above the 3-foot line, though they are intended to remain intact, they cannot be considered, either theoretically or practically, standards of excellence.

Without taking into account the effect of waves on exposed levees, which necessitates recourse to special slopes and methods of protection, planking, revetments, etc., the whole question of standard section depends on the permeability of the embankment and foundation, that is, the extent of seepage or percolation, and the best form and method for overcoming it in different materials.

In buckshot or clay, which is practically impermeable, the section might be given a strictly theoretical form, dependent alone on the height of the water and the weight of the buckshot, allowing some crown merely for increasing the height in time of excessive flood, the slopes being plane surfaces with an inclination sufficient to insure the required weight to counteract the hydrostatic pressure and the angle of repose of the material.

In cases of permeable materials, light clays, sand, and loam, the levee becomes partly saturated when subjected to high water, the line of demarcation between saturated and dry soil descending in a hydraulic gradient varying in inclination with the soil of which the levee is composed, and being probably very irregular in trace because of the lack of homogeneity of the material in the body of the levee.

In surface soils, subject to direct rainfall or to percolation from adjacent watered areas, the ground-water stands at a level dependent on the composition of the soil, both physical and chemical, the natural and artificial voids, and the hydrostatic pressure. In nearly all soils remote from intersecting fissures, wells, or streams, the line or plane of saturation is parallel with the surface of the ground, following the inclination of hill and valley. Where wells, fissures, or river beds occur in the surface soil, the line of moist material, or plane of upper surface of saturation, is inclined towards the fissure, well, or river, the degree of inclination depending on the consistency of the soil.

The power of soils to resist the pressure of water is due to their specific gravity, fineness of comminution, cohesiveness, and the irregularity of individual particles. Coarse, sharp sand has greater resisting power than that composed of fine, smooth, rounded particles.

The author estimates the strength of materials, as found in this levee district, to resist deformation due to seepage, or their value for levee purposes, in about the following order:

1. Buckshot and gravel tamped in shallow layers.
2. Buckshot artificially mixed with sharp sand in shallow layers.
3. Buckshot or clay.
4. Heavy, strong soil.
5. Coarse, sharp sand.
6. Light soils.
7. Fine sand, rounded particles.

One of the most important factors to determine with regard to a levee is the *height*. This needs special consideration, since failure is a foregone conclusion if water is allowed to overtop the crest. It is therefore necessary that all levees be built to a height above that of the highest flood level; and, depending upon circumstances, this will vary from two to four feet above that elevation.

The height is dependent, in a measure, upon the position of the levee, and the direction of the current. For example, an embankment so placed that it lies at an angle to the direction of the current will need to be much higher than one parallel to it, for with the former the water tends to bank up against the levee, while with the latter such is not the case.

The material of which a levee is constructed is usually that nearest at hand, but some materials are better adapted to such use than others. Clay is the most satisfactory, as it is not only impervious to water, but also resists wave action well. Sand is not so good, as it is more easily washed away, is less impervious, and requires a greater amount of material, since the slope at which it will stand in a bank is less. Frequently gravel is used as a facing to prevent the washing away of material.

The following, taken from an article by Starling on "Holland Dikes," shows the practice in that country, where the engineers are probably the greatest authorities in the world on the subject:

"The earth of which the embankment is to be composed must be such as to cohere readily with itself and with the soil beneath it. The more cohesion the soil has, the more it is to be preferred, and the more will its different parts unite and form a compact mass which can oppose resistance to the water, and thus furnish a tighter dike. Clay is thus the most suitable. . . . Sand has very little coherency, and does not afford a water-tight and strong bank. Peat and swamp soil have too little specific gravity, often less than water itself, and should therefore, as well as sand, be rejected. Mould or arable land, though far inferior to clay, is still much better than peat or sand, as it packs closely by reason of the smallness of its particles, and is especially suitable for dressing slopes that are to be sodded, as grass grows very well upon it. Clay cannot always be had in a pure state or in sufficient quantities, so that inferior earths must sometimes be mixed with it. But if the precaution be taken to work the best and purest clay on and near the outside, and the inferior sorts in the body of the levee, such sorts may be used without great danger. There are examples that consist of very sandy soil with a covering of only about 3 feet of clay on the outer slope, yet they furnish very satisfactory embankments. It is easy to be seen, however, that such a dressing of clay must be treated with the utmost care, and the slightest injury to the outer slope must be immediately repaired, for if enough of the clay be removed to permit the water to come in contact with the peat, very little confidence can be placed in the construction.

"Great importance must be attached to the care with which embankments for water are built. If made simply, as those for railroads are constructed, they will be more or less permeable; and when the water comes against them, settlement and deformation will result. It is necessary, then, that precautions be taken in cleaning up the foundation and in rolling or tamping the material in place so as to make it compact and close-grained.

"Before beginning the construction of a levee, it is important that all vegetable matter, trees, etc., be removed from the site. This should include the roots, as well as the trunks and branches. After a thorough cleaning, the ground should be plowed or spaded deeply, in order to secure a more perfect bond with the new structure; and if the top soil is unsuitable, it should be removed before bringing on new earth. Generally the specifications provide for cutting a muck-ditch near the center line of the levee, at the discretion of the engineer.

This ditch and all excavations made in removing stumps, trees, etc., are filled up with approved material, well tamped."

The following statement from the United States Government specifications shows the method adopted in building levees:

"The embankment will be started full out to the side stakes, and be carried regularly up to gross fill, in layers not exceeding 2 feet in thickness when built by scrapers, and 6 inches when built by wheelbarrows. In wheelbarrow work, the earth will be carefully tamped, either by wheeling over the embankment or by employing one rammer to two wheelbarrows. When the embankment has been brought up to the proper height, it shall be dressed, and planted with living tufts of Bermuda grass, 4 inches square, and not more than 2 feet apart, well pressed into the earth and lightly covered with soil, to the satisfaction of the engineer in charge or of his designated agent. The contractor will cut down all trees, both great and small, to a distance of 100 feet from the base of the levee on both sides.

"Only clean, unfrozen earth, free from all foreign matter, shall be used in constructing the embankment. It will be procured on the river side generally. In no case must it be obtained within 40 feet of the base of the levee on the river side, or within 100 feet on the land side, and the side slope of the pit next to the embankment is not to be steeper than 1 on 2. At intervals, traverses must be left across the borrow-pits, to prevent the flow of a current along the levee. The distances between the traverses will not be more than 500 feet. They shall be at least 10 feet wide on top, with slopes of 1 on 2. Borrow-pits must not exceed 3 feet in depth on the side next to the levee, but they may gradually deepen with a slope of 1 on 50 when on the river side, and 1 on 100 when on the land side of the levee. All existing levees, or parts of old levees, must be left, unless written permission of the engineer in charge is given for their removal.

"Generally, if it be possible, the levee is built a year or so earlier than it will probably be needed, in order to give it time to settle thoroughly and to be completely covered with sod. With certain kinds of soil, there is an objection to scraper-built levees—namely, that they are very liable to be cut and washed into gullies by rain before the sod has had time to grow. These soils are loam and mixed sand. When put up with wheelbarrows, banks of such material are at first comparatively loose and porous, and absorb water like a sponge. By the time they have settled fully, the sod has grown. When the earth, however, has been put up with scrapers, it is very hard, and sheds water like the roof of a house. The material being light and friable, however, the rain soon cuts channels which it uses regularly, and the gullies which are the result of this action sometimes cut almost through the crown of the levee. A slope thus eroded has to be re-dressed before it is sodded, and the new dressing is liable to be washed away also. In spite of this objection, scraper-built levees are generally preferred; and some engineers place such restrictions on wheelbarrow work as almost to prohibit it. The shrinkage exacted is generally one-fifth for wheelbarrow work untamped, or one-tenth if it be tamped; and one-tenth for scraper work.

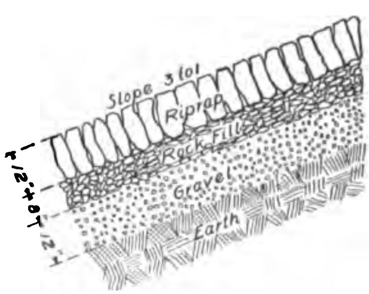


Fig. 7. Simple Form of Bank Protection

"Heretofore the construction of levees has been carried out almost entirely by hand methods, owing probably to the remoteness of the sites and the difficulties attendant on the use of machinery in such work. Recently, however, steam dredges have been employed with success sufficient to demonstrate that by the use of suitable machines satisfactory construction can be obtained, and at a great reduction of cost. To obtain economical results, the dredges would be used as much as possible while the river was at bank stage, but would also be provided with pumps by which the borrow-pits in which they were digging could be flooded so as to keep them afloat."

River Bank Protection. While the erection of embankments on the shores of a river serves the purpose of confining the water to the channel itself or to a given cross-section, in no sense do they fulfil the function of preventing the currents from eroding the banks, carrying off the material, and undermining them until at last failure finally occurs.

With this particular object in view, therefore, special works of a protective nature must be undertaken. These will tend to reduce the amount of silt carried by the river and consequently the amount deposited; to preserve a permanent river, bed and channel; to protect property, levees, wharves, landings, etc.; and to prevent cut-offs.

Where protection can be given so as to prevent erosion, the flood current will help to remove any deposit that may have collected in the channel, and thus prove to be a helpful rather than a destructive agent.

In its simplest form this improvement is secured by placing loose stone along the submerged face of the bank, and by sodding that portion of it which lies above the normal water level, as a precautionary measure during flood stages.

The more complex structures are known as *revetments*, and are divided into two general classes. In the first, the bank for its entire length is covered by a protective apron of material; while in the second, only portions of it receive this treatment.

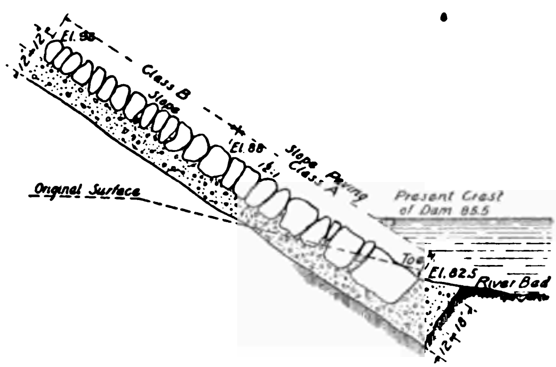


Fig. 8. Bank Protection by Means of Stone and Gravel, where Bottom is Soft.

All such works consist of two main parts — that below the water level, which performs the function of a foundation; and that above, which rests upon the former and is continually exposed during the dry season. Whatever its nature, the foundation requires care in its construction, since, being hidden from view, it is hard to locate spots where deterioration has commenced. Upon its stability also depends that of the superstructure.

Frequently the method employed will consist simply in depositing stones with more or less care against the bank in question, and continuing this process until from the bed of the river to a considerable elevation above flood level, it has been completely covered. In case the bottom is soft and the stones settle, it is the practice to dig a trench along the foot of the bank so that the rock may be deposited in this, and serve as a foundation or *toe* for that above. Such a procedure, if properly executed, guarantees stability, and uses less material than in the preceding case (see Figs. 7 to 10).

Round or sheet piling is also frequently employed. It has the advantage of using even less material and being more stable. It is subject to decay as all wood is, if alternately wet and dry; for this reason, therefore, all timber should be carefully inspected during the dry season.

A revetment of dry masonry is sometimes used, and less frequently the stone is laid with mortar; but the advantages gained by its in-

Fig. 9. Bank of Mississippi River Covered with Rubble to Protect it against Erosion.

creased stability are offset by the excessive expense, except, perhaps, where the bank lies within the limits of a city or town.

Dry masonry consisting of either field stone or roughly cut flat stone placed by hand, is frequently laid on a foundation of gravel or sand, the sand or gravel adapting itself to the irregularities of the bottom of each stone and thus securing an even upper surface. This method is often employed on canals where the banks of the prism need some sort of protection, and where erosion would otherwise occur as a result of the wash of small steam craft or the rubbing of tow ropes.

Stone, however, is not the only means of protecting the banks of a river, as both in this country and abroad extensive works of defense have been constructed of brush, saplings, poles, etc. This method has the particular advantage of being cheap, and of using material

CUT AT BAS OBISPO, PANAMA
On line of the Isthmian Canal.

usually close at hand; besides which, it gives very satisfactory results. The names applied to the material so employed are *fascines* and *mattresses;* the former consist of round bundles of long willow or other branches, tied together; while the latter consist of branches or poles woven together in horizontal layers.

In Holland, the fascines used vary in length from 8 to 13 feet, and from 1⅓ to 1⅔ feet in diameter; while in Germany they are 13 to

Fig. 10. Bank Protection on the Mississippi River.
Placing the broken stone in position. Partially sunken mattress shown in foreground.

16 feet in length, and from 1 foot 2 inches to 1 foot 10 inches in diameter.

Mattresses are formed into various shapes; but they are usually horizontal layers of brush, woven together by poles, wire, timber, etc. Iron wire has been found to rust readily, so that galvanized wire or wood has been substituted, with very satisfactory results.

The use of mattresses has been developed to a very great extent along the Mississippi, where cottonwood and willow are abundant and stone usually conveniently located. In this region, it is the method pre-eminently in favor for protecting the banks from scour; and the practice is here followed and developed to a greater extent than anywhere else in the world. See Figs. 11 to 16.

A mattress 300 feet wide by 120 feet long represents a superficial area of about 8 acres; and when one realizes that this vast willow carpet, over a foot thick, is placed on the bottom of the river in depths of from 40 to 100 feet, and against currents of from 5 to 8 feet per second, the difficulty of the enterprise will be appreciated. Though much of the revetment from Cairo to New Orleans has needed repairs from year to year, and in some reaches has required renewal as a whole, it may be said to have been eminently successful in the protection of

Fig. 11. Mooring a Mattress to the Bank of the Mississippi River.
The mattress is subsequently sunk by loading stones upon it, and makes an effective foundation for jetty construction in the improvement and control of the river.

cut-offs and outlets, and fairly so in the control of bank-caving and the resulting change in position and flow of the river.

At some points where the material of the bank was friable and the currents very strong, the earlier forms of revetments proved too light and were entirely swept away, the shore line continuing to move back. Also, considerable reaches of protection work needing repairs and reinforcement at the ends, have been destroyed because of the lack of funds. But in the later work the results have been beneficial and satisfactory, and the loss but slight.

The method of constructing these revetments as practiced on the Mississippi River is as follows: The bank is cleared 50 feet back from the top, and graded on a slope of 1 to 4 from the low-water line. This slope is then dressed by filling the holes and removing tree stumps, etc. At the upper end of the section to be protected, a cluster of piles is driven, called an *abutment;* and below them, along the shore, single piles, spaced 100 feet apart, are driven to hold the mattress in position for the full length of bank to be protected. The mattresses are built on barges, floated to position, and then sunk by placing rock on top.

The following description, based on the Government report, illustrates the method of constructing the mattresses themselves:

Fig. 12. One of the Huge Mattresses Used in Jetty Construction on the Mississippi River.
Mattress 100 ft. wide, in position in shallow water, covered with the broken stone used to sink it. The cross-sections are produced by the scantlings used to tie the mattress together.

Hardwood poles, as large as can be conveniently handled by a gang of men, and reasonably straight, are laid in two lines on ways over and parallel to the inner gunwale. These poles lap each other 10 to 15 feet, the two lines breaking joints. Where they lap, they are spiked together, and they are also tied together by No. 12 galvanized wire at intervals of 10 feet. Two ties are made at the laps. This line of poles is as long as the mattress is wide. About 7 feet 6 inches apart on these poles, and at right angles to them, the butt ends of weaving-poles made of live willow or cottonwood brush from 4 to 6 inches in diameter and 25 to 30 feet long are fastened with spikes and wire. Another set of poles similar and parallel to the first are placed on these, and securely spiked and wired. To facilitate weaving, the top and the bottom of the weaving-poles are shaved and the knots trimmed.

A cable made of eight strands of No. 12 wire is fastened around the head of the mat at every third weaving-pole, and run up alongside of it, the end being fastened thereto by two staples.

These cables are 24 feet long, with an eye in one end, to which, after each shift of the mat, a new length is looped in weaving. The continuous cables are thus formed in the mat, greatly strengthening

Fig. 13. Sinking a Mattress by Throwing Stones upon it from a Scow.
The mattress is to serve as the foundation of a jetty for river-bank protection on the Mississippi.

it longitudinally. When this head is finished, lines are connected to it from the shore, passing under the mooring barges.

The brush used for weaving is live, straight willow of any length over 25 feet, and from 2 to 4 inches thick at the butt.

The butts are placed over one weaving-pole, and project two feet beyond, being woven at the other end over the next pole, under the third, over the fourth, and so on, the light ends being always left on top. A strip 5 feet wide is thus woven. In the next strip, the butts are reversed, the butts changing direction every 5 feet. When the mattress is woven within 2 feet of the end of the poles, giving about 22 feet length of mattress, it is swung into position with the accompanying barges. The head-lines on the barges and mattress are slackened until the barges are nearly normal to the shore, with their inside edge resting against the pile abutment. The slack in the moor-

IMPROVING THE HARBOR OF CLEVELAND, OHIO

Outer or lake face of shore arm of west breakwater. The projecting iron rods were used to hold the forms for the concrete, and are to be cut off.

ing cables is now taken in from the bank, and the strain equalized. They are then fastened permanently with clamps, as are also the mattress head-lines. An entire shift, 22 feet long, is then launched, a new set of weaving-poles being spliced to the projecting ends of the first set. This is continued as described, to within 2 feet of the top of the second set of poles, when another launch is made; and so on until the full length of the mattress is obtained.

When three shifts have been launched, the construction of a top grillage or framework is begun. This consists of a line of poles laid over and parallel with the weaving-poles, lapping each other, butts to tops, from 6 to 8 feet, and wired to the weaving-poles every 4 feet

Fig. 14. Building a Mattress from a Trestle where Water was Too Shallow to Allow of Bringing in a Mattress which had been Built on the Regular Mattress Building Ways.

by lashings 2 feet long, made of two strands of No. 10 wire; transverse poles 8 feet apart for the first 100 feet, and thereafter 16 feet apart, are placed in similar manner, and fastened to the longitudinal ones at the intersections by two-foot lashings made of four strands of No. 12 wire.

The purpose of the grillage is to make pens in which the stone is retained, so as to strengthen the construction.

The first set of transverse poles along the inner edge are hardwood, set 8 feet apart throughout the length of the mat, and are used to connect the shore mat, which is also being built, to the river mat.

This shore mat is constructed of hardwood poles of the size of the weaving-poles, lashed and spiked to the river mattress, with willow or cottonwood poles spliced to these until they reach up the slope about 40 feet. Alongside, and fastened to each of the hardwood poles, is a [illegible] wire, one end of which is fastened to [illegible] the other to the willow poles extended [illegible] are laid longitudinally willow

[illegible] Mississippi River [illegible] mattresses.

[illegible] Mississippi River [illegible] several [illegible]

[illegible] apart, beginning with the first set about 4 [illegible] The [illegible] poles are wired to the [illegible] The longitudinal poles are carried on lines 8 feet apart, up to the top of the slope; and on their lower side, 8 feet apart, are driven stakes 2 feet 6 inches above ground, to the tops of which is loosely fastened a lashing wire whose bight has been passed under the pole. These stakes are used down to the pole nearest the water edge. Upon this framework is laid willow brush diagonally, with the butts toward the top of the slope, and breaking joints throughout. A second layer of brush is put on in the opposite direction, the two thus being at right angles to each other. On top of these layers a second pole framework, fastened similarly to the first, is placed and fastened down firmly by the lashings mentioned above as being tied to the stakes. As fast as the river work and shore work are finished, transverse cables are run across the entire width of the mat at 16-foot intervals, carried to top of bank, hauled taut, and fastened to trees, stumps, or "deadmen" placed for the purpose. These are fastened to the mat every 16 feet, with lashings.

When 400 or 500 feet of river mattress have been completed, longitudinal cables are run out from the mooring barges and securely attached to the mat at 16-foot intervals. One of these is placed close to the edge of the mat, the others at 30, 37, 38, and 42 feet respectively. The mats are ballasted and sunk by placing stones on them; and, when sufficiently submerged, the scows are floated over them, and the material dumped from them.

Owing to the undermining of the outer mattresses, these are generally more firmly built, and consist of bundles of fascines bound together.

Another method of protection similar to the above is by means of a spur revetment, where the mattresses are built out into the stream at right angles to the shore. These are placed at intervals along the bank that is threatened, and seem satisfactorily to check the destructive effects of the current.

Dikes. Dikes are a form of structure "for guiding a stream along a caving bank and protecting the same from undermining, for confining and directing the water at bars and shoals, and for closing secondary arms of a river." See Fig. 17.

There are three general forms—*spur-dikes*, *longitudinal dikes*, and *submerged spurs*—the two former being frequently used in combination.

Spur-dikes are used a great deal in Germany, though only to a

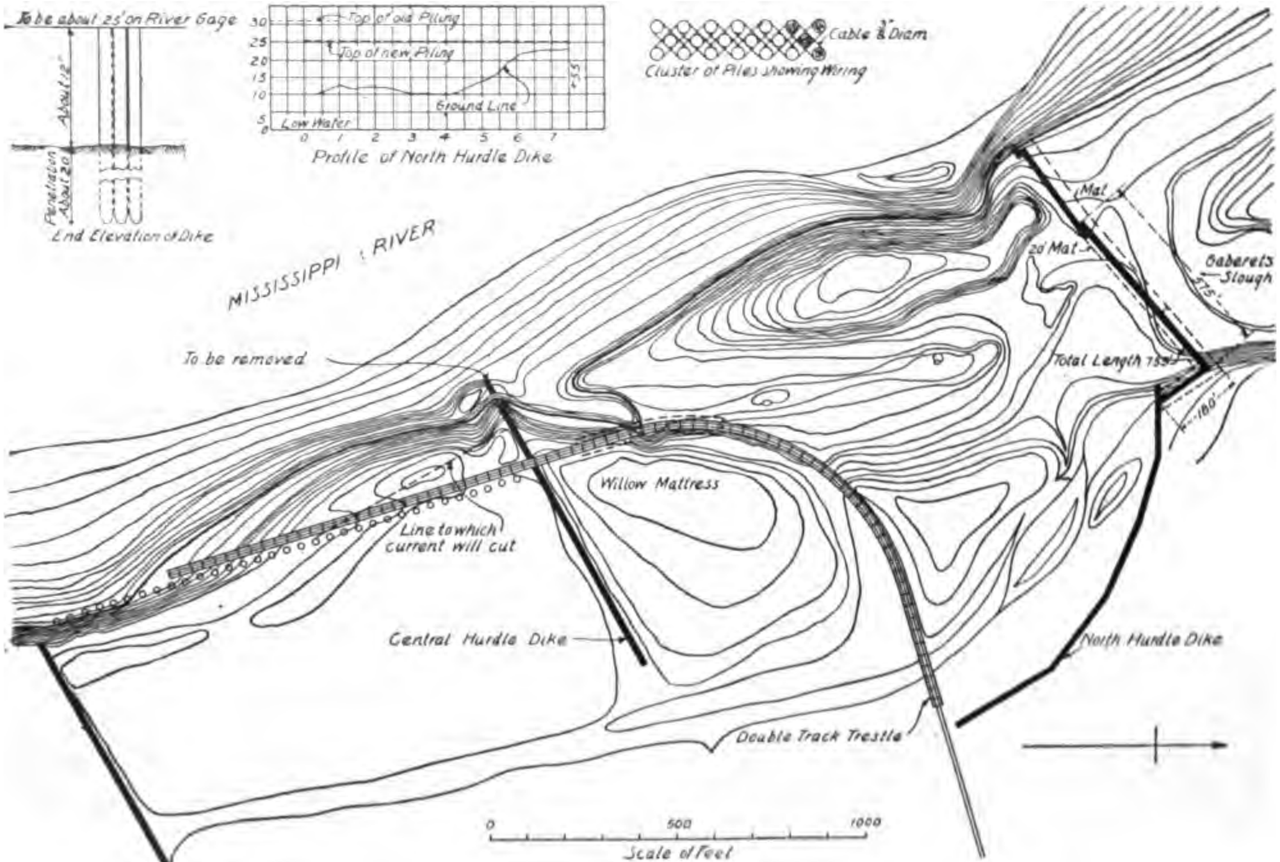


Fig. 17. Protection of a Trestle by Means of Dikes.

small extent in the United States. Their object is to improve navigation, and this is accomplished by placing them along the shore at intervals. They run out into the stream at a small angle with the shore, and thus tend to confine the current to a smaller cross-section. Their form of construction causes the deposit of silt between the spurs, which material tends to consolidate and help protect the dikes themselves from floods and ice. They thus form a new bank which is continuous between the heads of the dikes. This system generally requires more than one to produce any increase in the depth of the channel, as the water is confined only at the outer end, or "head," of the dike.

Longitudinal dikes are made parallel to the shore or the current, their object being the same as in the case of spur-dikes. They are used on the Ohio to some extent, where they are built of wooden cribs filled with stone. Generally they begin at the bank and extend downstream, with a gentle curve away from the shore, when they run parallel to it for some distance. Behind them, as with spur-dikes, the space fills with silt and mud, and thus tends to form a new bank. Such construction is highly advantageous where it is desired to scour out bars, and thus secure greater depth in the channel at low water. During normal flow, the tops are submerged. Occasionally they become undermined by the swift current that they produce; besides which they are expensive to maintain, and need constant attention.

Submerged spurs are built in the bottom of a stream or river, to correct the influence of scour produced by one or the other of the two forms of dike already described. Their object is to produce a silting-up of the bottom, or to prevent the bottom from being scoured out below a certain depth, and thus endangering the bank or other protective works

EROSION AND TRANSPORTATION

Where the flow of water in a river channel is uniform, and the channel comparatively straight and regular, there is little tendency for the stream to erode the banks; but where bends and curves occur, this tendency does exist, and sometimes to a marked degree. Under such circumstances, the greatest depth is near the outer portion of the bend, while the shallow portion is opposite. This is due to the centrifugal force of the current, which resists the change in direction and scours away the outer bank.

This erosion, or wearing down, of the river banks is continually going on. The greater the velocities, the greater the transporting and erosive power of the stream. As these higher velocities exist in the mountainous districts, where the slopes are greatest, here the larger material will be eroded or moved; and where the velocities are reduced in the flatter portions of the stream, the larger portions are deposited, until finally, where the current is very slow, only the finest silt is carried.

The following table shows the velocities of water necessary to the transportation of various materials:

TRANSPORTING POWER OF WATER AT VARIOUS VELOCITIES

Potter's clay	0.26 foot per second
Sand deposited by clay	0.54 " " "
Large, angular sand	0.71 " " "
Gravel, size of peas	0.53 " " "
Gravel, size of beans	1.07 feet " "
Round pebbles, size of thumb	2.13 " " "
Angular flint stone, size of hen's egg	3.20 " " "

The diameter of a body that may be moved by a current varies as the square of the velocity; and the weight of such bodies, as the sixth power of the velocity.

The force exerted in moving sand or pebbles by water is proportional to the square of the velocity and to the area exposed. Thus the force required to move a body of diameter d will be:

$$F = cd^2v^2$$

in which c = some constant. If motion just occurs, then F is proportional to the weight of the body also, because the frictional resistance of one body upon another varies as the normal pressure or weight. The weight of a sphere varies as the cube of the diameter, so that:

$$d^3 = cd^2v^2,$$

or,

$$d = cv^2.$$

Since d varies as v^2, the weight, which is proportional to the cube of the diameter, must vary as v^6; and consequently an increase in velocity causes greater increase in transporting capacity.

As the weight of such material in water is about one-half its weight in air, the frictional resistances to motion are slight; and therefore the pebbles, etc., are easily moved by moderate velocities.

It may be said, therefore, that ordinarily small material such as loose earth will be carried or rolled along the bottom by velocities of 2 feet per second or less.

The formation of bends and bars in a river may be indicated by quoting the illustration given by the Russian engineer Janicki.

Let us examine how the bends and the bars are formed. To better understand the details of the process, let us suppose a plain, absolutely regular, and not horizontal, but slightly inclined in a certain direction. Let us suppose, further, that in the direction of this inclination a canal is dug having a certain cross-section, a regular bottom, a slope parallel to that of the ground, and regularly constructed side-slopes. Into this canal we shall admit a river at its maximum discharge. This river has a certain velocity of current which undermines the banks and cuts out the bottom of the canal, and we shall accordingly witness the following phenomena:

The water begins by detaching a few small particles from the bed and cutting away the two banks equally; but, as in nature no ground is absolutely homogeneous and of the same tenacity throughout, it finally happens that at some point one bank yields sooner than the other. This first undermining gives rise to a slip or mound which destroys the symmetry of the original profile, and deflects the current toward the opposite shore. Soon this second shore, in its turn, crumbles away just where the deflected current strikes it most powerfully. This new mass of fallen earth cannot remain at the foot of the bank whence it came. The current here being already increased, it is carried a little lower downstream, and there forms a bar, which in turn directs the current toward the bank whence the trouble first started. If we add to this that the current, once turned aside from the original straight line, wanders farther and farther away by the very force of inertia, and that this deviation continues until, from the very fact of its digression, the slope and velocity of the current are reduced and come into equilibrium with the resistance of the banks, we shall see how bends are slowly and gradually formed, making detours both to right and left of the line of maximum slope. The elongation of bends ceases only when the slope is so diminished from the increased length given to the course through which the water must flow that there is no further tendency to produce scour of bottom or banks. Theory and experience teach us also that velocity of current is determined, not only by the inclination of water-surface, but also by the form of the bed—that is, by the form of its cross-section. For any given soil and any given inclination, there is but a single form of flowing cross-section which will give the maximum velocity of current with the least resistance. In the above assumed case of an artificial river whose curves are freely developed, the form of cross-section will undoubtedly vary according as we consider the bed at the head of the bend, at its apex, or at the point of passing from one bend to another; these variations, however, will be constant at similar points, for the slope, by reason of the increased length, has become almost uniform; and the other two factors—velocity of current and nature of the bottom—being likewise uniform, the depths of the sections and their widths will have a constant maximum limit.

But what will happen if one of the banks is higher or more solid than the other, and if, from this or any other special condition of the surface of the adjacent bottom land, the river cannot sufficiently increase the length of one or more of its bends? It cannot maintain a very steep slope; the nature of the soil forbids that. In such a case it is evident that the stream will more and more scour out its bed, and deposit the debris in those places favored by the topographical features of the valley—that is, where it is possible to build up the bottom. In other words, the result will naturally be an elevation of the bed of the stream; and this elevation will act as a cross-dike, damming the river like a weir; it will withstand and considerably diminish the force of the current; the water will flow over it in a thinner sheet, and, following the exterior slope of the bar, will fall into the lower pool by a route much shorter than through a bend, and without a tendency to cut away its bed; for we know that, with a given slope, the velocity of bottom flow diminishes with the depth of the sheet of water.

If the discharge of our artificial river always remained the same, it would finally come into equilibrium with the resistance of the soil throughout its whole length. The bends and the bottom, when they had once assumed their proper form, would retain it, whatever might be the consistency of the soil. But the discharge of rivers often varies through wide limits. Into our artificial canal, we let loose a river at flood-height. If the discharge should gradually diminish, the water-level would fall. At deep places the section would always be sufficient for the passage of the water in spite of any diminution of the slope caused by lowering the level, and the regimen of those parts would not vary; but wherever the bottom had previously been raised, these elevations would act more and more as if they were dams that closed the whole width of the river. The water would flow over them, would attack them, and dig out a new low-water channel which would have a width and a depth adapted, according to hydraulic principles, to the nature of the obstructions, to the fall (from the upper level to the lower), and to the low-water discharge.

When the water again rose, the swell, starting at points above, with a current increased with the slope, would bring down new material and fill up the low-water channel already formed, and thus reconstruct the bar to its former height. This work of lowering and reconstructing bars is repeated at each freshet. The longitudinal profile of a river taken during low water shows that it is composed of a succession of pools where the fall is generally less than the mean fall, and also that the pools are separated from each other by bars where the fall is greater than the mean fall of the river.

We have taken a purely theoretical river for an example; and we have supposed that in the beginning it had a regular bed situated in a plain where the soil was homogeneous. If we now take into account the diverse topographical features, and the geological complexity which ordinary river valleys present, we shall have that variety of cross-sections, of surface slopes, and of more or less pronounced bends which is found in free rivers.

Our attention is thus called to the intimate relationship that exists between all those phenomena which at first view appear so entirely distinct from one another. No one can anywhere interfere with the curvature of a river, its slope, or the depth of its cross-section, without immediately causing, either above or below the point, some change in the pre-existing conditions of its equilibrium. This equilibrium, we must not forget, is not a static but a dynamic equilibrium; and, therefore, in the present condition of the science, it is very difficult to determine its exact conditions in advance.

We have shown that the want of solidity in the soil is the natural regulator of the rapidity of the current. This lack of solidity, consequently, leads to the formation of bends and bars, which re-establish the equilibrium between bed resistance and velocity of current.

The status, the general character of a river, therefore, depends on the united action of these three factors—the *discharge*, which is variable periodically with the changes of the seasons, and which also undergoes permanent changes as the result of deforestation of the watershed and other conditions produced by the work of man; the *magnitude of the slope*, which is likewise variable; and the *nature of the soil*, which is variable in different localities.

For a river to be navigable, it is necessary that it should have a sufficiently deep channel throughout its entire length. A river may have much water; but if the fall is considerable and the soil unstable, it cannot have a deep channel. On the other hand, there are rivers with a relatively small discharge and great fall, which are yet quite suitable for navigation, owing to their hard bottom, which effectively resists erosion and transportation of its material, and thus preserves a relatively permanent river bed.

Shoals are portions of the river bed which resist erosion and therefore produce shallow spots.

Bars, on the other hand, are formed by the deposition of transported material, either from the natural conditions of the current or from the artificial ones produced by the construction of piers or breakwaters. The effects of such constructions vary, of course, with the nature of the material of which the river banks and bed are composed, but more particularly with the angle at which the constructions are laid relatively to the axis of the stream.

When river banks are protected, the silt carried down during the flood is finally deposited, on the lowering of the stream, in the bed of the channel, rather than over the adjoining lands as would be the case were it unconfined. In consequence of this, and by successive depositions during long periods of time, the bed of the stream becomes elevated above its former level, so that the banks must be raised at the same time to keep pace with the silting-up of the stream.

In Japan there are cases where this process has been in progress for so many years that the river now flows at such a considerable elevation above that of the surrounding country that railway lines are preferably carried beneath the stream by tunnels, rather than over them by bridges. This is the extreme condition, of course; but observations show that the same operation is going on along the course of the lower Mississippi River.

HARBOR IMPROVEMENT

Harbors. A *harbor* may be said to be "an indentation or inlet on the shore of a sea or lake, so protected from winds and waves, whether by natural conformation of the land or by artificial works, as to form a secure roadstead for ships." In its general sense, a harbor will include all those works which are required or used to facilitate the repairs of ships, loading or unloading. Thus it will be seen that the subject of "Harbors" embraces the questions of anchorage, piers, docks, buoys, lighthouses, etc.; while "Harbor Improvement" will concern itself with the betterment of the same.

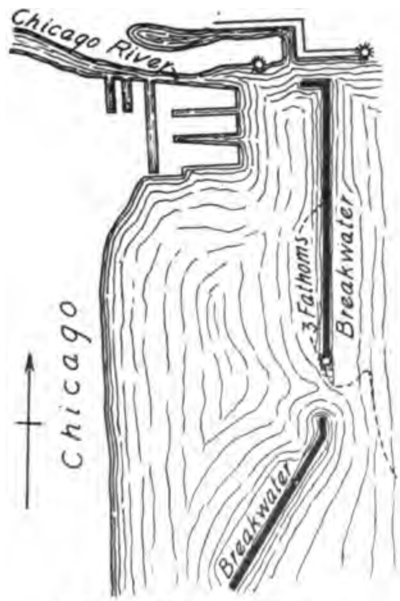


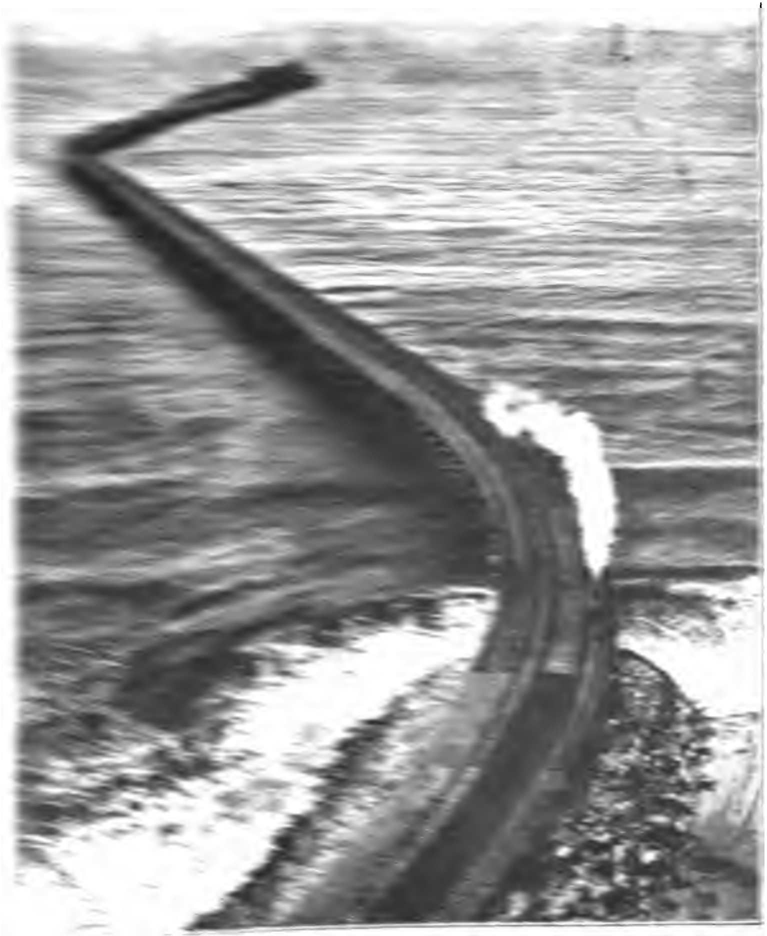
Fig. 18. Harbor of Chicago, Ill.

All harbors may be divided into two general classes—(1) those designed to serve *commerce*; and (2) those constructed as places of *refuge*. These may each be again subdivided into *natural* and *artificial* harbors.

Commercial Harbors. These are chiefly for the purpose of loading and unloading the freight carried by ships. They may consist of almost any arrangement of piers and breakwaters to enclose and tranquilize the water, together with the quays, wharves, docks, etc., which are maintained in conjunction with them. Frequently on exposed coasts, what is known as a *compound* or *double* harbor will be constructed, and this consists of outer protective works, with an inner basin for the mooring of ships. See Fig. 18.

Harbors of Refuge. The term *Harbor of Refuge* may be applied in both a generic and a specific sense. Generally speaking, it refers to a port in which craft may obtain shelter, no matter what the conditions of the weather, and which is located on a portion of the coast much frequented, but where no such protection has been supplied by nature. See Fig. 19.

meet the



wide at base. ... dressed ... 300 acres with

... various berths for all sorts

(3) It should possess facilities for furnishing supplies and making small repairs;

(4) It should have natural capabilities for the construction of works of defense at a small cost.

On the other hand, in a *special* sense, a harbor of refuge is one which is located primarily with a view to its strategical possibilities, and rather as a harbor of refuge for battleships, both from the enemy and from the weather as well. Such a harbor would naturally be a national undertaking, and should possess the following characteristics:

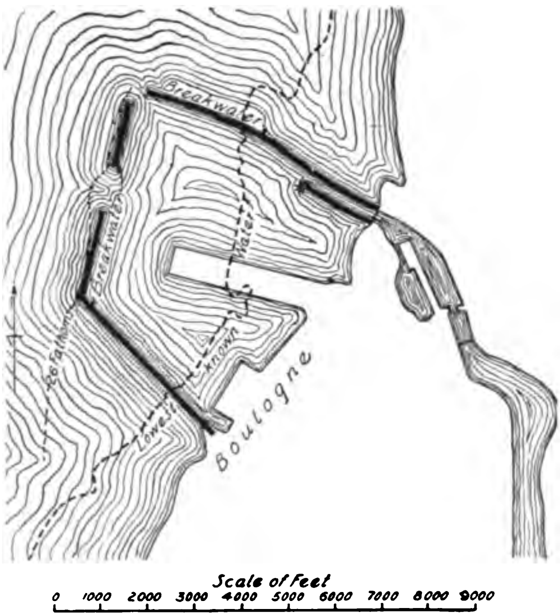


Fig. 20. Harbor of Boulogne, France.

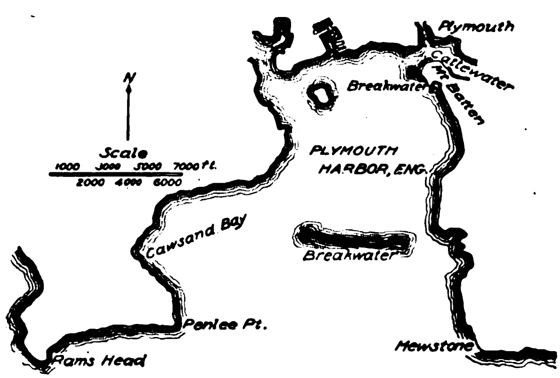


Fig. 21. Harbor of Plymouth, England.

(1) Its position should be centrally located with regard to the defensive works and the shipping of the coast at that particular point; and it should be situated where the enemy can be conveniently observed.

(2) The anchorage area for heavy-draught warships should be much larger than in the case of the general harbor of refuge.

(3) Such a harbor should be connected with the great railway systems of the country, so that troops, coal, supplies, repairs, etc., can be readily transported there.

[illegible] will depend upon the local needs and requirements; but under any circumstances, they, as in the case of the others, should be easy of approach and afford satisfactory anchorage. See Figs. [illegible]

[illegible] Harbors. It seldom happens that a harbor will be found so satisfactory that no improvements whatsoever will need to be made. But while this, in some cases, will be but a simple matter, in others it may mean practically the entire reconstruction of the port. Thus, for example, the Harbor of New York, one of the finest in the world, needs constant attention in the matter of dredging to prevent the channel from becoming filled up and thus precluding the entrance

Fig. [illegible] Harbor of Cleveland, Ohio.

of deep-draught vessels to the Upper Bay and the docks of the city itself.

The improvement of the mouth of a river for harbor purposes is one of the most important—and one of the most difficult—undertakings in harbor engineering. Generally speaking, the mouth of a river [illegible] because of the bars or shoals which form there. These are caused by the sudden deposition of the material carried by the river in suspension; for, when the current strikes the larger body of water, its velocity is at once checked, and the silt sinks to the bottom.

Waves also tend to increase the amount of deposition, not only because they obstruct the river's current, but because they also carry the material back to the beach-line itself.

Cases occur where prevailing winds blowing toward shore, in combination with long periods of drought, will cause the waves to deposit the silt to such an extent as completely to block the channel of a river. Where the river's mouth is wide, it usually happens that more than one bar — sometimes many bars — with corresponding channels will be formed.

It should therefore be the purpose of the harbor engineer to use the forces of nature as much as possible in the design of such works, whether these be protective in character or not. This will consist:

(1) In directing the currents so that they will tend to scour out and deepen the channel rather than silt it up: in this connection, the transporting power of the ebb-tide should be taken into consideration, so that the material

Fig. 23. Harbor of Milwaukee, Wisconsin.

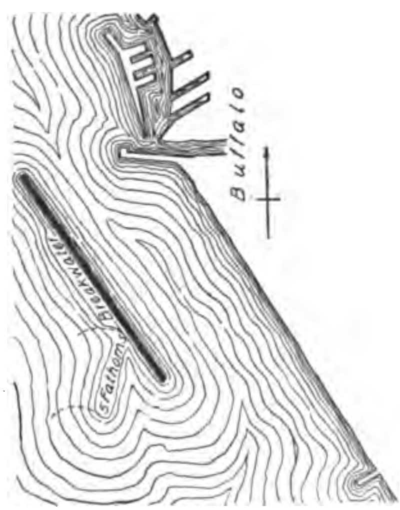


Fig. 24. Harbor of Buffalo, N. Y.

in suspension may be carried out to sea rather than back into the harbor at flood-tide.

(2) In increasing the amount of water flowing over the bar at ebb-tide, by enlarging the harbor through dredging, and preventing any obstruction to the flow through the agency of curves: it has been proved to demonstration over and over again that reclamation or other works reducing the capacity of the tidal compartment, and consequently the quantity of tidal water passing through a harbor-entrance, injuriously affect the depth of water at the entrance in proportion to the amount of water excluded, the converse being equally true.

(3) In protection of the outgoing current from being deflected or retarded by prevailing seas: this is frequently done by means of breakwaters.

Such harbors as indicated will almost always require periodic dredging, and generally the presence of breakwaters or training banks. See Figs. 22, 23, and 24.

It has been observed that when fresh water meets salt, as in the case of a river flowing into the ocean, the former tends to flow over the latter. It is evident, therefore, that, while the tide is at flood, the river will flow on top of the ocean water and thus perform no service in the function of scouring. In consequence, the ebb-tide alone must be depended upon to bear away the material deposited.

WINDS, WAVES, AND TIDES

Winds. Winds cause the waves; and the latter, the destruction of protective works. There are periods of the year when gales are more frequent than at others; and during these, more than ordinary care should be observed and provision made to guard against destruction. The time varies in different localities.

The strength, duration, frequency, and sequence of winds are also matters to be determined and studied, if protective works are to be properly constructed.

Apparently no exact relation exists between the velocity of the wind and the pressure it exerts, as the latter seems to vary with the size and form of structure against which it acts; but the following table shows the pressures that are most frequently used for varying velocities of wind, in the design of structures that must withstand such action:

WIND VELOCITIES AND PRESSURES

Velocity (Miles per hour)	Pressure (Lbs. per square foot)
10	1
20	2.5
30	5.0
40	8.0
50	13.0
60	18.0
70	25.0
80	32.5
90	41.0
100	50.0

CONSTRUCTION OF CONCRETE SUPERSTRUCTURE TO BREAKWATER, MARQUETTE, MICH.

Waves. While the theory of wind-waves is too abstruse to be discussed fully in this treatise, it is a fact that a knowledge of wave intensity is an absolute necessity in the proper design and construction of sea-walls, piers, breakwaters, etc.

It is a mistaken notion that in a wave the water itself travels; for, as may be readily demonstrated, the movement of a chip on the water is absolutely uninfluenced by the wave action, and the object remains practically stationary, while it is the crest of the wave that moves on.

When waves pass into shallow water, the friction along the bottom so retards the lower particles that the crest overturns and *breaks.* Thus the length of waves diminishes as they approach shore (*i. e.*, the distance from crest to crest diminishes); and their velocities also in like manner. As the pressure exerted on any structure by a wave depends upon its velocity, the longer the waves the greater the damage that may be created.

Waves generally break when they enter water which has a depth little if any in excess of their height (*i. e.*, trough to crest).

Those waves that are small in length as compared with the depth of water in which they act, may be termed *oscillatory.*

The highest waves are never greater than 60 feet; and under most circumstances and in most places they are considerably less than this. Such waves usually occur in mid-ocean or away from harbors, so that they need not be considered in their destructive effects on harbor works.

The intensity of the stroke of waves is due in a great measure to the non-elasticity of the water.

Tides. The variation of the tides, as is quite evident, affects the height to which protective works in harbors must be raised, so that their study becomes a very important matter.

Tides are due to the influence of gravitation which both the sun and moon exert upon the earth. The latter is the more powerful factor in this regard. The word *tide* is used to denote the action of the water of the ocean in rising and falling, which occurs generally twice a day.

The time at which a tide occurs, and its height, will be found to vary from day to day. The variations are known as *spring tides* when the water is highest, and *neap tides* when lowest. As high water of spring tides rises higher than that of neaps, so the low water of springs generally falls lower.

The *range* of a tide is the vertical rise between low and high water.

Half-tide is the mean distance between high and low water of springs and neaps.

The difference of rise between spring and neap tides varies in proportion to the range of the tides. As an average, it may be taken that the rise of neap tides above low water of spring tides is 75 per cent of that of a spring tide.

New-moon tides are about one-seventieth of the total rise higher than full-moon tides.

When the sun is vertically over the equator, and its path coincides with it, the tide-producing effect is at a maximum. At these periods, which occur in March and September, the new moon and full moon have also most influence. On the other hand, in June, when the sun is farther from the equator, the rise is a minimum.

BREAKWATERS

Breakwaters are generally divided into two main classes—the *vertical* type and the *mound* type. The former (see Figs. 25 to 28)

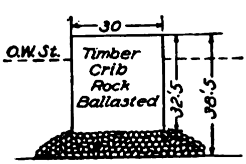


Fig. 25. Section of Breakwater, Harbor of Chicago, Ill.

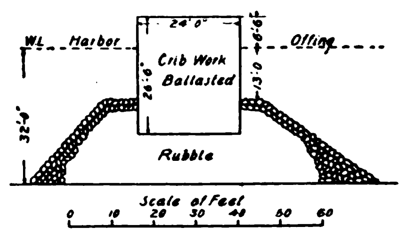


Fig. 26. Section of Breakwater, Harbor of Milwaukee, Wisconsin.

may include those where the face is slightly inclined, and may be constructed of:

(1) Timber cribwork filled with stones.

(2) Outer walls, of masonry or concrete, dry or in cement, with the interior composed of dry rubble.

(3) Concrete blocks, laid in horizontal courses or in connection with an interior and base of cyclopean character.

(4) Mass concrete.

(5) The same as No. 4, but with a base formed of concrete in bags.

(6) Concrete blocks laid in sloping courses.

Fig. 27. Improvement of Harbor of Buffalo, New York. Breakwater under construction, showing timber cribwork extension at south end. The substructure cribs loaded with stone, are sunk in place, and the erection of the superstructure is shown under way.

... exposed face to break ...

... they are formed of:

... deposited loosely and allowed ...

... Construction of breakwater, ... concrete superstructure.

... deposited ... and brought to high-

BREAKWATERS OF VERTICAL TYPE

... type of breakwater is that ... material than any other form per unit of

Fig. ... Section of Breakwater, Harbor of Buffalo, N. Y.

... are usually selected where material is ... may control in the design, such as ... foundation; and these must of course ... considered.

BUILDING THE PANAMA CANAL

Drill men at work in cut at Haut Obispo.

Fig. 31. Improvement of Harbor of Buffalo, New York. Showing method of constructing breakwater.

Fig. 32. Breakwater under Construction, Harbor of Buffalo, N. Y. Showing size of and method of placing facing stone.

Against a breakwater of the vertical type, waves tend to rise higher than where the other form is employed, so that it may be stated as a general principle that they are elevated to double the height of their crests above normal water surface. At the same time that this occurs, a wave is sent downward along the wall, which strikes the bottom and scours out the material of the foundation. The consequence of this action is that vertical-faced breakwaters are subjected to tremendous forces during the heaviest storms, from which damage frequently results, which the mound type of breakwater does not meet with.

Fig. 33. Breakwater, Harbor of Buffalo, N. Y., with Facing and Top Stones Complete.

Timber Cribs Filled with Stone. These are cheaper than breakwaters of stone, and are more rapidly constructed. Such are the breakwaters at Boulogne (Fig. 34) and Calais, France, and Dover, England. Frequently these are constructed as temporary affairs until more permanent ones may be built of stone, and are usually quite satisfactory. It is imperative that the timber work receive careful attention, for that which is above low water is subjected to decay, while that below may be subjected to the attack of wood-boring animals (notably the *teredo navalis*), as is well known to be the case on the Pacific Coast.

This type of breakwater is best suited to shallow water, or where the wave action is less intense than in exposed positions. They are frequently formed by piles being driven into the bed of the harbor, the stone being filled in among them. The purpose of the timber is always to hold the stone in place, while the stone itself supports and strengthens the crib.

Breakwaters with Side Walls of Masonry and Interior of Rubble. As in the previous class, this is best suited to harbors where the exposure is slight. The method of construction is to build the exterior walls first, and fill in between them with loose stone which lends bulk and stability to the structure; but, as this leaves the outside shell unsupported for a considerable length during construction, it permits of the possibility of the breakwater being destroyed before completion, should a heavy storm arise. Furthermore, masses of water falling from above on the unprotected rubble interior, fill it with water and exert an excessive outward lateral pressure which may cause failure.

Breakwaters of Concrete Blocks. This method possesses several advantages over the use of loose stone or block stone put in place, because each block is better able to withstand the wave action than an equal amount of small material.

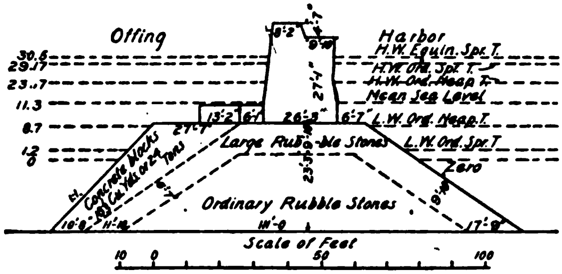


Fig. 34. Section of Breakwater, Harbor of Boulogne, France.

In this type it is very necessary that the foundation be leveled off and properly prepared, as the first course of concrete blocks must be laid horizontally, and have close joints. The blocks are placed in position by the aid of divers, those blocks below the water being laid dry. Owing to this fact, the blocks are less likely to bear against each other uniformly, and hence they tend to break. For the same reason (*i. e.*, lack of cement in the joints), no tendency toward settlement should be permitted, as this will permit the blocks to slide or move under the force of the waves.

Cement or mortar is generally used between the joints when the masonry has risen above the low-water mark; and the topmost course, or capping, is usually formed of mass concrete deposited in place.

Sometimes, in this type, the foundation is formed of mass concrete which is brought up to the level of low water; and then, on top of this, are placed the concrete blocks laid in mortar.

Breakwaters of Mass Concrete. This system consists in depositing wet concrete in temporary timber frames below or above water level, and is employed on both large and small work alike. The expense and time required to level off the foundation in the previous system, are obviated in this.

It has the disadvantage, however, of being essentially a fine-weather method of construction, as both the frames and the wet (or "green") concrete will not withstand storms.

Usually the framework is made of wood or iron verticals resting in iron shoes, if the bed be soft, or in holes if of rock, to which horizontals are attached. Planks are then fastened to these, to form the sides of the frame; and the concrete is deposited therein. To prevent the bulging of the walls of the frame during the deposition of the liquid concrete, they are secured together by iron tie-rods. The frame should be as water-tight as possible, so as to prevent the cement from being carried off by currents while being put in place; and to secure this end, the bottom and sides are sometimes lined with canvas.

On a level foundation, the same frames may be repeatedly used; but where it is irregular, it will be necessary to construct new ones for each section, thereby increasing the expense.

To prevent the loss of cement during deposition, the concrete is frequently put in place by loading it into skips with a flap bottom, lowering this under water to within a few feet of the bottom, and then opening the skip so that the concrete is discharged. As little disturbance as possible of the concrete and of the water in which it is to rest, is the most desirable condition.

This system is generally preferred, though sometimes the concrete is deposited by using a chute. To keep the water out, the chute should be maintained full of concrete all the time.

For purposes of economy, large pieces of stone may be incorporated with the concrete; but the amount should never exceed about one-sixth of the entire mass, and the stones should always be separated from each other by at least 9 inches or a foot of concrete. The concrete should have the proportions of about 1 part of cement to 3 or 4 parts of stone.

Breakwaters Composed of Concrete Bags. This system is sometimes employed for under-water work, and consists simply in placing the concrete in canvas bags, so as to keep it and the cement in place. By this means an even foundation may readily be secured, even where the bottom is somewhat irregular.

Such construction is frequently employed as an apron to prevent the scour of foundations, the bags being deposited by means of skips with bottoms opening outward and of a capacity of from 5 to 20 tons. In the construction of the Aberdeen breakwaters, Scotland, bags of concrete have been used weighing as much as 100 tons.

The material should be deposited in place as soon as possible after mixing, so as to prevent it from acquiring a set, and, when so deposited, it should be lowered as close as possible to the place where it is to rest, before being released. Dropping the bags through considerable heights is to be avoided as much as possible.

The richness of the material used is generally about 1 part cement to 4 or 5 parts sand and broken stone.

This system is best adapted to localities but little exposed, or where the foundations are deep, as the bag work is not well suited to resist the wave action. It may be so employed, however, and even brought up to an elevation above low water, if the exposed face is protected by a deposit of loose blocks of stone.

Breakwaters of Concrete Blocks Laid on Inclined Courses. This system consists in laying blocks of concrete with the joints at an angle to the vertical, and may be used either where the breakwater is to have vertical faces or where there is to be a loose stone base. The foundation must first be leveled off in this case, which is accomplished by means of concrete in mass, in bags; or by the use of broken stone.

Blocks weighing from 20 to 120 tons each have been employed in this system. The amount of inclination of the blocks will vary; but a slope of 10 to 30 degrees from the vertical usually proves satisfactory. Frequently the blocks are arranged with a sort of tongue and groove which fit into each other to secure greater stability.

The object to be attained in this method is the closeness of joints and the prevention of the slope from flattening.

It compares as a system very favorably with others of this class, as there is little risk during construction, little injury from settlement, and it may be rapidly constructed.

BREAKWATERS OF MOUND TYPE

Loose Stone. Such a type requires an excess of material per unit of length, so that it is absolutely necessary that the materials of construction be abundant.

The effect of the tides on such structures is very considerable, so that, as the range increases, it becomes more and more necessary to increase the cross-sectional vertical area of the breakwater. The waves sweeping over such structures tend to change the slope of the rubble stone more and more, until the material has at last, perhaps after years, come to rest in accordance with the forces of nature. This slope will depend not only upon the height of the waves, the depth of water, etc., but upon the material itself and the height to which it is carried above low water.

It is found that the waves on the harbor side affect the slope of this material to a depth of 10 or 15 feet below low tide, and the tendency is to reduce it almost to the horizontal or to as small a slope as 1 in 72. At Plymouth, England, the breakwater on its outer face has the following slopes: Below low tide, it has a slope which varies from 1 in 3½ to 1 in 5 to a depth of about 10 feet; and from there to the bottom, where the wave action is ineffective, it assumes that of the angle of repose of the material, which is about 1 to 1½. Above low tide, it is almost horizontal, which extends right up to the high-water mark; on the harbor side, the slope is approximately that of the angle of repose.

Where a vertical wall is placed on top of such a breakwater—for example, for the purpose of forming a quay—the action of the water is only increased in its tendency to disturb the material in the mound, which tendency does not exist when the wall is absent. This is caused by the downward action of the waves and the force of recoil.

The best practice would indicate that a rubble mound surmounted by a superstructure should either be carried well above high water, and be maintained at that level, so as to make the waves break and expend themselves on it; or it should be kept at such a depth below low water as not seriously to affect the character of the waves, or to be exposed to their disturbing action.

In rubble-mound breakwaters, it is probably better to mix the material as to size than to have it uniform, as in the latter case the interstices left are too large and permit of wave action therein. The largest stone should, however, be placed on the outer slope.

Mound Breakwaters of Concrete Blocks Thrown Together Irregularly. Not many of these have been built, as the expense is too great; but they are often found in combination with rubble, where they form the upper part of the breakwater or the outer face. They are found to some extent in the harbors bordering the Mediterranean Sea, particularly in Italy.

Where the blocks are large, as in this type, they take a steeper slope than rubble; and, having more voids, they resist and check the wave action better.

BUOYS, LIGHTS, AND BEACONS

The function of providing lights and buoys for a harbor generally devolves upon the local authorities, though in some cases it may resolve itself into a national undertaking.

The necessity for such lights and buoys to mark a channel or indicate the shoals on a coast, is readily appreciated when it is realized that vessels now proceed into port under full steam whether it be day or night, and consequently a channel must be made as easy to navigate in the dark as in daylight.

Unfortunately no general set of regulations govern the lighting and buoying of harbors, so that a uniform system does not exist throughout the world, nor even in the same country. While this is of little consequence where a vessel is being guided by a local pilot, it is sometimes a hardship, particularly to smaller-sized vessels visiting the port. For the purpose of establishing uniformity in this matter, however, the following regulations are in force wherever the British Admiralty, Board of Trade, and Lighthouse Board has jurisdiction; and the code has been recommended to all local harbor boards also for adoption.

The method of lighting and buoying estuaries and tidal channels is not subject to any uniform regulation; consequently a considerable diversity of practice prevails amongst local authorities. To vessels in charge of local pilots, this is not of consequence; but it is obvious that if one universal system were in use, it would be of great advantage to ships failing to obtain pilots, and to smaller craft navigating without their aid.

To obtain uniformity as far as possible, a conference was held by representatives of the Board of Trade, the Trinity House, and the Admiralty; and the following code of regulations was agreed to for adoption in all cases where the buoys come under their direct management and control, and was recommended for adoption by all local authorities. In framing these regulations, the shape of the buoys was more relied on than the color, as distinguishing the different parts of the channel. It is unfortunate, as likely to lead to confusion, that in determining the right and left-hand side of the channel, the well-known rule that the right bank of the river is that to the right hand when going *down* the stream from the source to the mouth should have been departed from, and the right-hand side settled as that on the right hand when going *up* the channel.

REGULATIONS FOR BUOYING CHANNELS

The term *starboard hand* shall denote the side which would be on the right hand of the mariner either going with the main stream of flood or entering a harbor, river, or estuary from seaward; the term *port hand* shall denote the left hand of the mariner under the same circumstances.

Buoys showing the pointed top of a cone above water shall be called *conical*, and shall always be starboard-hand buoys, as above defined.

Buoys showing a flat top above water shall be called *can*, and shall always be port-hand buoys, as above defined.

Buoys showing a domed top above water shall be called *spherical*, and shall mark the ends of middle grounds.

Buoys having a tall, central structure on a broad base shall be called *pillar-buoys*, and, like other special buoys, such as bell-buoys, gas-buoys, automatic sounding buoys, etc., shall be placed to mark special positions either on the coast or in the approaches to harbors, etc.

Buoys showing only a mast above water shall be called *spar buoys*.

Starboard-hand buoys shall always be painted in one color only.

Port-hand buoys shall always be painted of another characteristic color, either single or parti-color.

Spherical buoys at the ends of middle grounds shall always be distinguished by horizontal stripes of white color.

Surmounting beacons, such as staff and globe, etc., shall always be painted of one dark color.

Staff and globe shall only be used on starboard-hand buoys; staff and cage on port-hand; diamonds at the outer ends of middle grounds, and triangles at the inner ends.

Buoys on the same side of a channel, estuary, or tideway may be dis-

YACHT PASSING THROUGH CORINTH CANAL, GREECE

This sea-level canal, opened August 6, 1893, connects the Gulf of Lepanto with the Ægina. Length, about 4 miles; width, 78 ft.; depth, 26 ft. diminishing near Corinth end to 18 ft.; cost, $2,500,000. Cuts 185 miles off the route from Venice to Constantinople, and 100 miles from Mediterranean ports; and is a great help to commerce between the Adriatic and Black Seas.

tinguished from each other by names, numbers, or letters, and, where necessary, by a staff surmounted with the appropriate beacon.

Buoys intended for moorings, etc., may be of shape or color according to the discretion of the authority within whose jurisdiction they are laid; but for marking submarine telegraph cables, the color shall be green, with the word *telegraph* painted thereon in white letters.

Wreck buoys in the open sea, or in the approaches to a harbor or estuary, shall be colored green, with the word *wreck* painted in white letters on them.

When possible, the buoys shall be laid near to the side of the wreck, next to mid-channel.

When a wreck-marking vessel is used, she shall, if possible, have her top sides colored green, with the word *wreck* in white letters thereon, and shall exhibit:

By day: Three balls on a yard 20 feet above the sea, two placed vertically at one end and one at the other, the single ball being on one side nearest to the wreck.

By night: Three white fixed lights similarly arranged, but not riding light.

In narrow waters, or in rivers, harbors, etc., under the jurisdiction of local authorities, the same rules may be adopted, or, at discretion, varied as follows:

When a wreck-marking vessel is used, she shall carry a cross-yard on a mast, with two balls by day placed horizontally not less than 6 feet nor more than 12 feet apart, and two lights by night similarly placed. When a barge or open boat only is used, a flag or ball may be shown in the daytime.

Fig. 35. Gas-Lighted Bell Buoy.

The position in which the marking vessel is placed with reference to the wreck, shall be at the discretion of the local authority having jurisdiction.

Buoys. There are various kinds of buoys, the following being the better-known—*can, cone, nun, spherical,* and *pillar;* and they are so designated, depending upon their shape.

The *cone buoy* is practically a metal cone, which floats with the base in the water, and the cone projecting above.

Fig. 36. Cunard Line Steamship *Mauretania.* Built in 1907 at the yards of Swan, Hunter & Wigham Richardson, Wallsend-on-Tyne, England. Largest ship afloat. Length 790 ft. (5 ft. greater than that of her sister ship the *Lusitania*); width, 88 ft.; depth 60 ft. 6 in.; engine horse-power, 70,000; displacement, 46,640 tons. The vessel is here shown making her way to sea at night from New York Harbor along new Ambrose Channel, guided by light-buoys.

The *can buoy*, on the other hand, has the cone immersed; and the base with a flat top is the part that is seen above the water.

The *nun buoy* is pointed at both ends, with the largest diameter at water level, and because of its lightness and simplicity, is particularly useful in shoal water.

The *spherical buoy* has a sphere showing above water.

The *pillar buoy* is practically the same as a can buoy with a pillar resting on top of it.

These buoys are now universally made of steel.

Fig. 37. "Subaqueous" System of Harbor Lighting. Invention of Mr. Leon Dion, of Wilkesbarre, Pa. Incandescent lamps, with reflectors designed to concentrate the light into parallel beams, are connected at intervals to a cable conductor fed from a generator on shore.

The purpose of these various-shaped buoys is to expose as much of the body of the buoy as possible, so that it will be easily distinguished; and it is a question which shape accomplishes this best. Long, narrow buoys have a tendency to lie over in the flowing tide, while flat-bottomed ones seem less inclined to do so.

Most buoys exposed to heavy seas are divided into two compartments so that in case of injury to one, the other will maintain it in its position.

Bell-buoys (Fig. 35) are used to mark dangerous parts in the channel, or shoals, and have the advantage over the others, of being as useful in foggy as in clear weather.

Fig. 38. Cape Hatteras Light Station, North Carolina.

All buoys should be cleaned and painted at least once a year, and, where conditions require it, oftener. Tar withstands the salt water better than paint, and a coat of neat cement is still better, as it absolutely protects the metal of the buoy.

To maintain buoys in position, they are moored by chains held

PLACING CONCRETE BLOCKS IN CONSTRUCTION OF BREAKWATER AT MARQUETTE, MICH.

by anchors or sinkers. The chain is usually from two to three times the depth at high water, the reason for this being that it is then less subject to sudden jerks during bad weather. The sinkers used are generally of cast iron, and weigh from 1,400 to 1,600 pounds, while the anchors are of the "mushroom" type.

The Trinity House (English Lighthouse Board) buoys are held in position by cast-iron sinkers. These weigh from 14 to 16 hundredweight for the 8-foot and 10-foot buoys. The mushroom anchor, commonly used, is a very effective mooring, and rarely drags. The mushroom part is of cast iron, and the shank of wrought iron. The large size 2 feet in diameter, with 2¾-inch shank 3 feet 6 inches long, and 3½-inch eye-bolt, metal in thickest part 1¾ inches, weighs 1¾ hundredweight; the second size, 1 foot 6 inches in diameter, with 2-inch shank, weighs 1¼ hundredweight; and the third size, 14 inches, with 1¾-inch shank, weighs ¾ hundredweight. A buoy, when properly moored, seldom breaks adrift. The greatest danger arises from ice, when large drifts are sent out of the rivers at the breaking-up of a long and hard frost.

Lights. For lighting the channel leading to New York Harbor between Sandy Hook and Coney Island, a system of electric buoys has been adopted. The buoys, made of wood, 50 feet long and 15 inches in diameter (to be replaced by buoys of riveted steel), are shackled to a cast-iron sinker weighing 2 tons. A cable from the shore is led up to an electric light on top of each of the buoys, the power being supplied from one station at Sandy Hook.

From some 6,000 observations made by the English Lighthouse Board to determine the relative efficiency of the various illuminants—oil, gas, and electricity—the following conclusions have been drawn:

(1) That the electric light was the most powerful under all conditions.

(2) That the quadriform gas apparatus and triform oil apparatus were of about the same power when seen through revolving lenses, the gas being a little better than the oil.

(3) That through fixed lenses, the superiority of the gas lights was unquestionable, the large size of their flame and their nearness together giving the beam a more compact appearance.

(4) That the Douglas gas-burner was more efficient than the Wigham burner.

(5) That for the ordinary necessities of lighthouse illumination, mineral oil was the most suitable and economical illuminant.

Fig. 39. Diamond Shoal Lightship.
Moored in about 180 feet of water, about 14¼ miles off Cape Hatteras Light. It has a 12-inch steam chime whistle, which blows a 5-second blast, with silent intervals of 55 seconds.

Fig. 40. Braddock Point (N. Y.) Light Station, on Lake Ontario.

(6) That for salient headlands and places where a very powerful light was required, electricity offered the greatest advantages.

Where harbors are approached by a channel, it is usually customary to mark such by buoys, and by lights at night. These are separated at a distance apart of from 0.1 of a mile to 1½ miles, depending upon the circumstances, and are placed so as to indicate the channel, shoals, bends, obstructions, etc.

Lights are divided into classes depending upon the height and diameter of the light. The first three classes are for the sea or coast, while the fourth class is the largest used for a harbor.

To avoid mistakes, a red tinge may be given the light by using a colored chimney or a sheet of colored glass; but it has been proven that a white light can be seen much farther than one of red, and a red light farther than one of green.

The height at which a light should be placed above sea-level to be seen a given number of miles away, is determined by multiplying the square of the distance in miles by $\frac{4}{7}$. Thus, if it is desired that a light be seen 15 miles away, it must be placed at a height equal to $15^2 \times \frac{4}{7} = 225 \times \frac{4}{7} = 128.6$ feet.

Originally, lights used as signals were made by burning wood on the exposed or dangerous headlands; but to-day, either oil, gas, or electricity is used in some sort of lantern which increases the power of the ray enormously. Thus, the burner invented by Sir James Douglas is able to increase the intensity ten times.

Various kinds of oil have been used in such lamps; but mineral oil seems best to satisfy the needs. It gives a more brilliant light than vegetable oil, does not congeal at low temperature, and requires less trimming of the wicks.

In harbors the lights used are: *Beacon lights*, to mark prominent headlands; *floating lights*, to mark the channel; and *leading lights*, to mark the entrance to a river, or the course in a channel.

Floating lights (that is, *lightships*), in English waters, are always painted, and have a ball at the masthead. Such vessels are usually of small tonnage, and are anchored by means of a mushroom anchor. During fog, horns, bells, or gongs are sounded to indicate the position. An American lightship is shown in Fig. 39.

Gas buoys (Figs. 35 and 45) are frequently used by employing a compressed oil gas which may be stored in a reservoir in the buoy,

[illegible]

[illegible]

Fig. 3. [illegible]

[illegible]

Fig. 5. Spar-Shaped Gas Buoy.

Fig. 46. Cape Henry Lighthouse, Virginia.

Fig. 47. Grassy Island Light Station, Detroit River.
South Channel Range. Window beneath balcony faces south along range line.

Fig. 48. Rebecca Shoal Lighthouse, Florida.

Fig. 49. Sand Key Light Station, Florida.

and which, burning day and night, may last for a period of weeks without replenishing.

LIGHTHOUSES

Lighthouses are usually towers built of iron or masonry, well above the height to which waves may reach, and in such a manner that their force may not destroy the structure. The waves that

Fig. 50. Hog Island Shoal Lighthouse, Rhode Island.

Fig. 51. Lighthouse for Promontory of Lao T'ieh Shan, China.

assail them are always of less intensity than those found in the open sea, as the location of a light is always placed on some rock-bound coast, or shoal, near shore. It is necessary in their construction, however, to know the height of the waves in relation to their fetch, and also to determine their effect. The rate of increase in the height of waves has been found to be proportional to the square root of the distance from the windward shore; and the greatest force ever said to be exerted on exposed rocks is estimated to be 3½ tons per foot. The force of winter gales is about three times that of summer.

One of the best-known lighthouses in the world is the Eddystone Light, which lies on the Eddystone Rocks about 14 miles from Plymouth, England. It was one of the first lighthouses of the building of which we have record, and has been reconstructed many times. In 1698 it had a height of 80 feet; but owing to damage, it had, in the following year, an extra shell of 4 feet built around it. In 1700 the height was increased to 120 feet, but during the year 1703 the tower was destroyed. In 1706 it was reconstructed, then having the form of a frustum of a cone, and being 92 feet high. It was principally of timber construction, the lower part, especially, being of oak. Above this a stone structure rose. It rested upon a stepped level base, was circular in plan, depended for fixedness upon its weight, and had a uniform exterior.

In 1756 Smeaton rebuilt the light, which consisted entirely of masonry and which was completed in 1795. It was the first masonry structure of this nature, with dovetailed joints, each stone weighing about one ton. In 1882 a new tower was erected.

It has been found that the level of the plane of danger due to the waves depends upon the relation between the height of waves and the nature of the rocks against which they strike both above and below water. Consequently the design of any lighthouse tower will not serve as a model for any other, except where the conditions are identical. The following may be said to be the general rules that are observed in construction:

(1) The tower should have a low center of gravity, and sufficient mass to prevent its being upset by the waves.

(2) It should be throughout circular in the horizontal plane, and either straight or continuously curved in the vertical plane, so as to present no abrupt change of outline which would check the free ascent of the rising waves, or the free descent of the falling waves, or the free vent of those passing round the tower. All external scarcements in the vertical plane, or polygonal outline in the horizontal plane, are therefore objectionable.

(3) Its height should be determined by the distance at which the light requires to be seen by the sailor. The rule for determining this has already been given.

(4) Where the rock is soft, or consists of ledges which are easily torn up, the tower should spring from the foundation-course, at a low angle with

Fig. 52. Bar Point Light Station, Detroit River.

Fig. 53. Buffalo (N. Y.) Breakwater, North End Light.

the surface of the rock, so as to prevent its being broken up by the reaction of the waves from the building; or, in other words, the tower must have a curved profile. But special care should be taken to sink the foundation courses below the surface of the rock, as the superincumbent weight decreases with the sine of the angle of inclination of the wall. If the rock overhangs, owing to the wearing action of the waves, the tower should, if possible, be built at a distance from the place where this dangerous action is in progress.

(5) Where the rock is hard and of ample area, the tower may be of such a curved form as will best suit the economic arrangement of the materials, so as to avoid an unnecessary thickness of the upper walls.

(6) When the rock is hard, but of small dimensions, the diameter above the base should not be suddenly reduced; and if the rock is hard, but of yet smaller dimensions, a cylindrical form of greater height should be adopted, so as to thicken the walls and to increase the weight and therefore increase the friction, which is directly proportional to the weight of the blocks of masonry. In all cases where the rock is small, the thickness should be decreased by steps or scarcements, *internally* but never *externally*.

(7) The level of the top of the solid part of the tower, and the thickness of the walls above it, should, in different towers having the same exposure, be determined in each case by the level and configuration of the rock and of the bottom of the sea.

(8) The best position of the tower is not necessarily the highest part of the rock. It should, in each case, be selected so as to secure the greatest protection in the direction of the maximum fetch and deepest water near the reef.

(9) The tower, if possible, should not be erected across any chasm which divides the rock, nor in the direction of any gully which projects into it, especially if it be of converging form, which would concentrate the wave action.

(10) No permanent fixture of the tower to the rock is required for increasing the stability of the structure. The foundation course (unless where a curved profile is adopted) becomes, indeed, in the end the most stable of all, because it has the greatest weight above it to keep it in its place.

(11) The stones should, however, be sufficiently connected together and fixed to the rock, in order to prevent their being washed away during the construction of the work, when they have no superincumbent weight to keep them in their beds.

(12) The tower should rest on a truly level base, or on level steps cut into the rock.

(13) The pressure of all the materials within the tower should act vertically, so as not to produce a resolved force acting laterally as an outward thrust.

(14) The tower should be of such height and diameter, with walls of such thickness, as to prevent the masonry being disturbed by the impact of the waves.

(15) The entrance door should be placed on that side of the tower where the length of the fetch is shortest, or where, from the configuration of the reef and the depth of water, the force of the waves is least.

(16) The materials should be of the highest specific gravity that can

Fig. 54. Twin-Screw Lighthouse Tender "Zizania." Showing Method of Placing and Removing Buoys.

Fig. 55. Cape Charles Light Station, Virginia.

be readily obtained; and, in some special cases, lead, or dovetailed blocks of cast iron set in cement, might perhaps be employed.

Illumination. The purpose is to distribute the rays equally, either constantly or periodically, over the entire horizon or else over certain angles of the horizon.

Fig. 56. Light for Lighthouse on Cape Villano, Spain. Showing apparatus giving double flash every 15 seconds.

Fig. 57. Third-Order Lightning Light, Toledo (Ohio) Harbor. Fresnel lenticular apparatus, showing clockwork for revolving.

A *fixed* light is one which is always of the same intensity and constantly in view about the whole horizon. The apparatus for such a light should therefore bend the rays in a vertical plane, but should not interfere in the horizontal one, if the rays of light are to be sent in the direction desired—*i.e.* downward and outward over the water.

A *revolving* light, on the other hand, illuminates each point of

Fig. 58. Prince of Wales Dry Dock, Southampton, England. One of the six large docks owned by the London & Southwestern Railway Company. At the time of its construction, the Prince of Wales Dock (opened about 1890 by His Majesty, King Edward VII, then Prince of Wales) was the largest graving dock in the world. At the same port, a larger dock (begun in 1900) has since been constructed 800 ft. long (clear inside gates), 125 ft. wide at cope level, tapering to 90 ft. wide at bottom, 33 ft. deep, giving a flotation depth over keel blocks of from 29 ft. 6 in., neap tides, to 33 ft., spring tides, at high water.

the horizon successively. A revolving light, though supplied by a flame of the same intensity as a fixed light, will therefore be far stronger than the latter, as it does not lose power by being diffused, but gathers the rays into beams of greater intensity. Metal or glass agents are usually employed in all lighthouses; but by the use of glass alone, it has been found that fully 25 per cent of light may be saved.

All apparatus may be classified under the *catoptric system*, where metallic reflection is used; the *dioptric*, where the material employed is wholly glass, producing refraction and total reflection; or the *catadioptric*, which is a combination of the two.

Beacons. These are constructed generally of iron columns set into rock, or of stone or rubble. The first light of this nature was constructed in 1851, on a sunken reef in Stornoway Bay, Hebrides Islands, Scotland.

These beacon lights are frequently kept burning for weeks at a time, either by oil or gas, without being trimmed. Pintsch gas is often employed with them, a buoy or beacon being charged with gas sufficient to burn night and day for three or four months. The pressure is kept constant by an automatic regulator, and the beacons have done eminent service.

DOCKS

Docks may be classified as either *dry docks* or *wet docks*. A *dry dock* (see Figs. 58 to 62) is a water-tight chamber, supplied with gates which, when the vessel has assumed its position in the dock, are closed tightly and the water pumped out, thus leaving the hull of the ship exposed so that repairs may be made. With this type, the ship is floated into position; while with some other forms (Figs. 63 and 64) the dock itself is submerged, as a sort of caisson, and when in a proper location beneath the ship to be docked, the water is gradually exhausted, air admitted, and the boat raised.

Dry docks—or *graving docks*, as they are frequently called—vary in length and width, depending upon the size of vessel they are supposed to accommodate. One of them (Fig. 58) has been built as long as 860 feet, and 90 feet in bottom width. The sides of the dock consist of a series of masonry steps called *altars*, against which timbers are placed to support the vessel in position while the keel rests on blocks.

Such docks are usually constructed of masonry, though they are

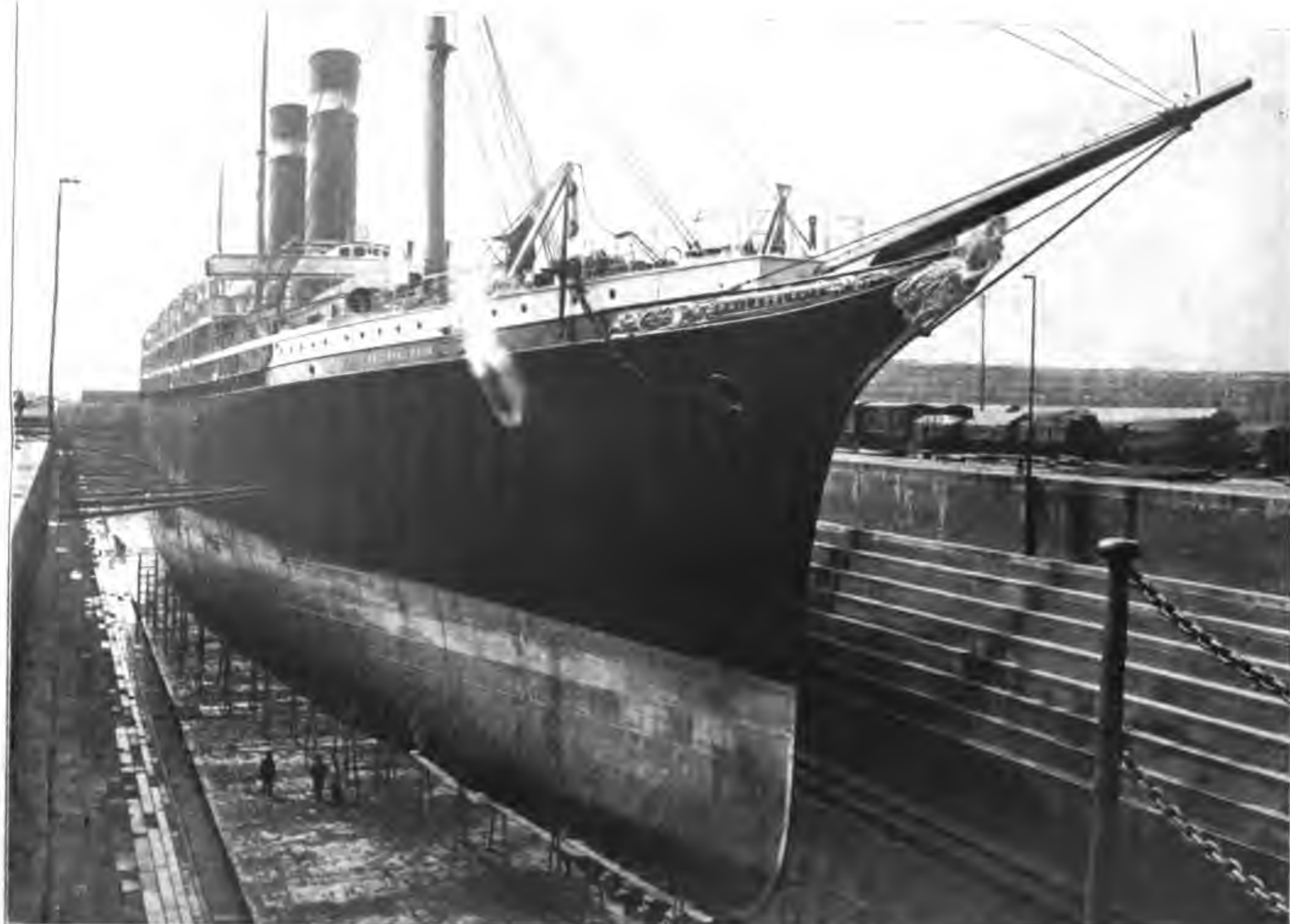


Fig. 59. SS. "Philadelphia" in Prince of Wales Dry Dock, Southampton, England.

sometimes built of wood. It is particularly necessary, whatever the construction, that care be exercised to prevent leakage.

Dry docks have been constructed of steel, and are then usually in the form of caissons which may be floated beneath the vessel, the water exhausted, and the ship raised. The particular advantage belonging to this latter form is that it is independent of the tide.

The general advantages connected with graving docks as compared with slips are:

(1) Although costly, they are, if well built, more durable than other methods for docking vessels.

(2) The dry docks require little supervision or management, as they are easily controlled.

(3) They are easily worked whether there be large or small vessels using them.

(4) They afford easier means of filling a vessel with water than any other form of dock, which is sometimes a necessary undertaking to detect leaks.

(5) Where double gates are provided, the water in the dock tends to scour the forebay or entrance.

(6) It is much easier to dock a vessel in a dry dock where the current is rapid or strong.

(7) Graving docks need not interfere with the set of the currents.

(8) The strains on the ship are less.

Slips consist of a cradle or carriage which is worked on an inclined railway, at various inclinations, and which extends some distance above high water. The cradle is let down into the water beneath the ship, which is put in place, and the cradle is then drawn up till it catches the ship, when both are hauled out of the water.

Wet docks are great enclosures or basins furnished with gates which during high tide are kept open, and closed during the ebb, so that the water may be maintained at a constant depth in the basin. That they may be serviceable at all periods of the tide, locks are often supplied at the entrance so that at low water the boats may be lifted up into the dock.

Wet docks are best located and more necessary:

(1) Where the rise of the tide is considerable.

(2) Where the nature of the trade requires ships of great length, as they are very liable to injury from taking the ground, and becoming strained.

(3) Where the bottom is hard and uneven.

(4) Where the harbor is open to the entrance of surface waves of considerable height, or of a ground swell, because of the lack of protection.

(5) Where there is ample supply of fresh water to fill the basin or dock.

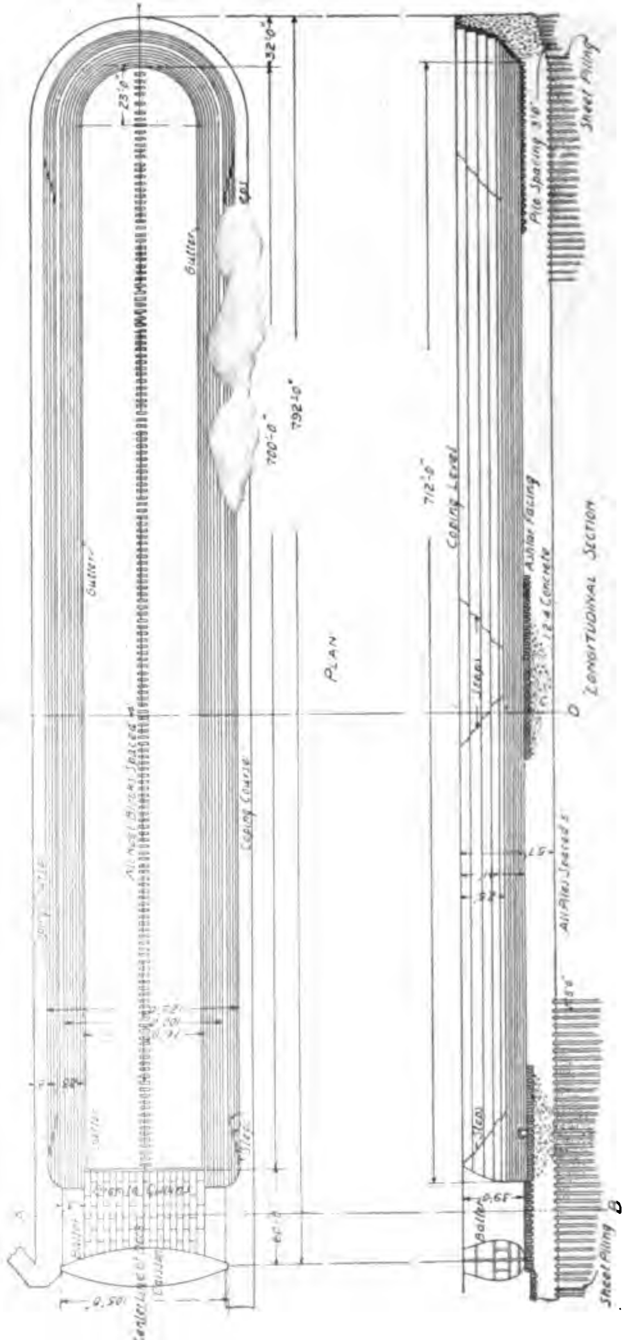


Fig. 60. Typical Plan and Sectional Elevation of a Dry Dock.

WHARVES AT ISMAILIA ON THE SUEZ CANAL, EGYPT

In the design of a wet dock, particular attention should be given to the exposure to the sea or river, so that the gates which control the entrance may be sufficiently strong to resist the action.

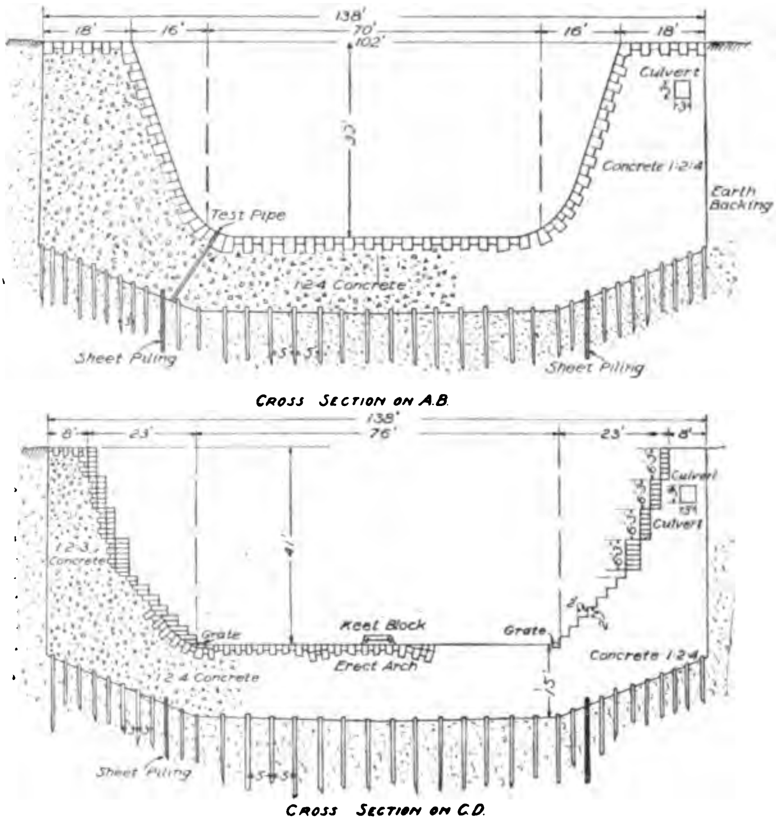


Fig. 61. Transverse Sections of Dry Dock.
See plan and elevation on opposite page.

The following formula is sometimes used to determine the width of the entrance:

$$b = \frac{60}{h},$$

in which b = Width of entrance, and h = Height of the waves; or again,

$$b = \frac{60}{1.5\,D \times (2.5 - D)}$$

may be used when h has not been observed, and where D is the length of the fetch in miles.

The capacity is an important factor, and may be said to be the

Fig. 62. SS. "Oceanic" in Liverpool's Largest Dry Dock.

number of vessels that may be contained within each acre. This varies with the size of craft and with the exposure, depth, etc.

The form or plan of a wet dock will depend upon the nature of the ground and other local conditions; but in general it should be largest at the entrance, so as to permit of easy egress and ingress. The proportion of water area to length of quays will depend upon the shape of the dock.

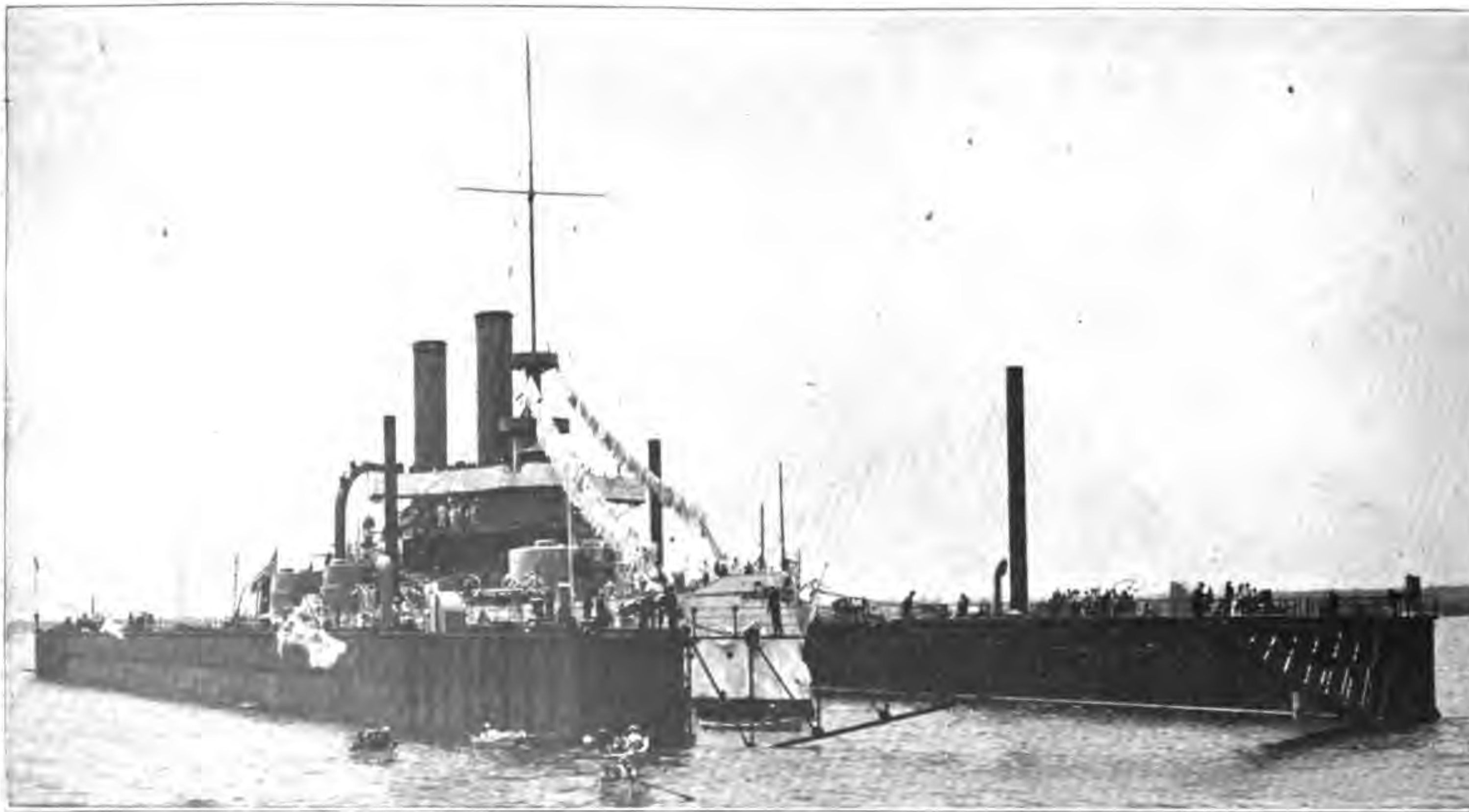


Fig. 63. U. S. Naval Floating Dry Dock "Dewey" Lifting Battleship "Iowa" out of the Water.

The "Dewey" is the largest floating dry dock ever built. Lifting capacity, 18,500 tons with a freeboard of 2 feet. Draught ranges from more than 50 feet to a little over 6 feet. Over 11,000 tons of steel and 2,000,000 rivets used in construction. Cost, over $1,125,000. Safely towed in 1906 from Chesapeake Bay on the Atlantic Coast to the naval station at Olongapo in the Philippines, a journey of more than 11,000 miles.

DREDGING

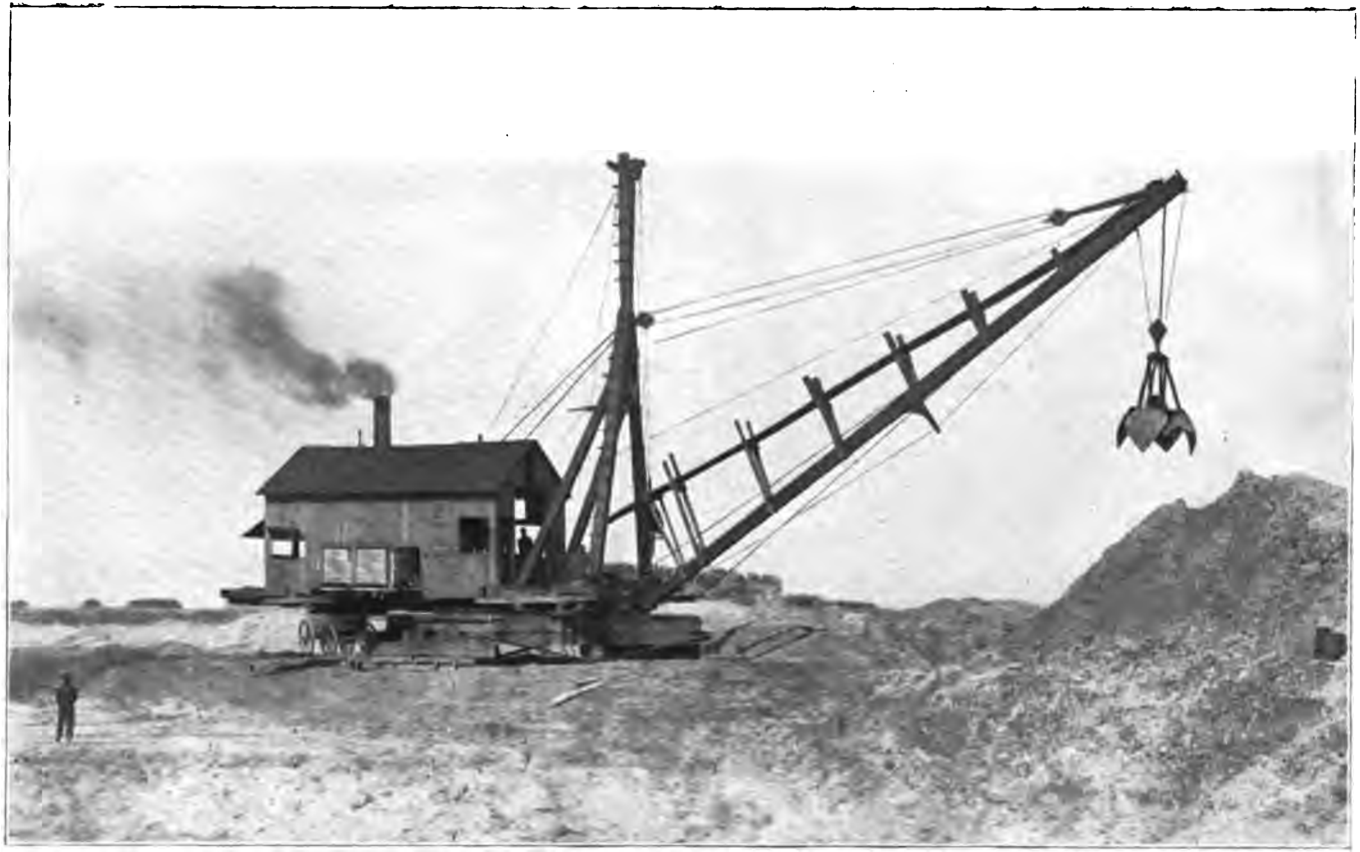
The dredging of the channel leading to a harbor is usually a necessary undertaking, and is performed for the purpose of removing material which the ebb-tide or current is unable to carry away.

In its simplest form, the improvement of a channel consists merely in loosening or breaking up any such material and allowing the current to carry it away. In this connection, jets of water have been applied in various ways to remove soft material; but as they are not particularly satisfactory, usually other means are employed.

Fig. 64. Type of Floating Dry Dock Used on Puget Sound. When the caissons are air-filled, the dock rises, lifting the vessel out of the water.

The more general methods are: (1) To discharge the soft material in an excess of water, through long pipes, to the mainland, where the silt is deposited to form "fill," and the water drains away; (2) to excavate the material by means of dredges, and discharge it into scows which are emptied at sea; or to discharge the material into the dredge itself, which, when full, is taken out to sea and unloaded. Dredges are of various types, and will remove material ranging from the liquid state to a consistency approaching that of hard rock. The types may be divided into *bucket-ladder* dredges, *clamshell* dredges, *orange-peel* dredges, *dipper* dredges, *pump* or *suction* dredges, and *rock-breaking* dredges.

The *bucket-ladder* or *elevator* dredge (Fig. 65) may be again divided into two classes—those in which there are single ladders, and those in which the ladders are double. It is maintained that the latter type work to better advantage where there is a large amount of material to handle, and when this material is heavy in character. The former, on the other hand, can discharge on either



TURNTABLE DREDGE WITH ORANGE-PEEL BUCKET
On line of Illinois and Mississippi Canal.

side of the vessel; while, with the double ladders, each set of buckets can discharge only on its respective side, and consequently this type requires more room.

In these dredges a series of buckets attached to an endless chain pass over revolving trunnions. The lower end of the arm to which the trunnions are fastened is lowered into the water; and the buckets, being made to revolve, excavate the material. The excavated ma-

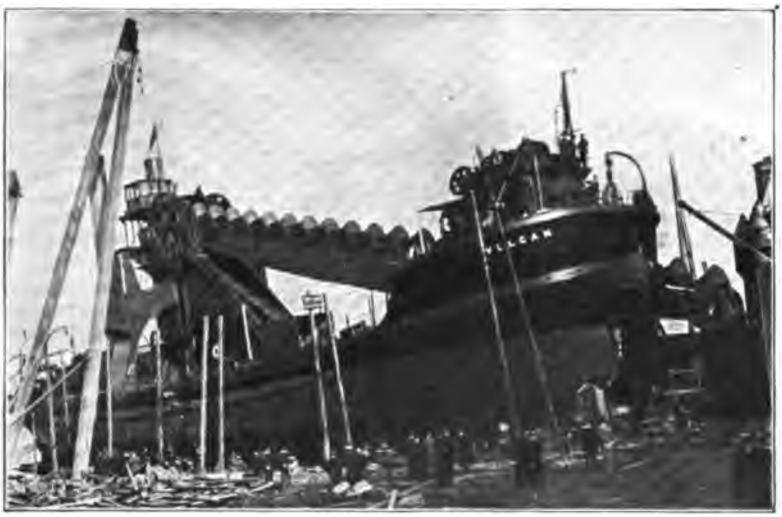


Fig. 65. Dredge of "Bucket-Ladder" Type.
Built for dredging very hard material in harbor of Liverpool, England. Capacity, 2,000,000 lbs. of material per hour. Works at maximum depth of 60 feet.

terial is carried in the buckets by the chain over the upper trunnion, where, as the buckets are inverted, it is discharged into a hopper or some other receptacle, from which it is conveyed to a scow. The buckets have perforations in the bottom to allow the water to flow out. Usually the bucket-ladder is situated on the center line of the dredge.

These dredges will handle the hardest clay, hardpan, or soft rock; and the buckets will lift stones weighing from 500 pounds to two tons. A small dredge of this character, 75 feet long, 17 feet beam, and 8 feet deep, with 45-horse-power engines, can raise about 50 tons of clay per hour; while one of 250 horse-power can handle 800 tons per hour.

One of the largest dredges of this class measures 200 feet in length, 37 feet in breadth, and 12½ feet in depth. It has 47 buckets attached to the ladder, each of a capacity of 22 cubic feet; will raise 600 tons per hour from a depth of 40 feet, and has engines of 750 horse-power.

The *clamshell dredge* (Figs. 66 and 67) has a sort of hemispherical bucket suspended by a chain from a jib, the two halves of the bucket being kept open while it is descending, so as to dig into the material to be excavated. By tightening one of the chains, the

Fig. 66. Clamshell Dredge Discharging into Scow.

bucket is closed, and then hoisted by the other to the surface. The load is dumped by slacking one chain and holding the other, whereupon the bucket opens again as in the first position. Frequently a hemispherical bucket is used, divided into four parts; in this case the dredge is called an *orange-peel* or *grapple* dredge. Clamshell and orange-peel dredges are more serviceable for general use than other forms.

These dredges will operate very satisfactorily in soft or loose material; and when hard material is encountered, teeth may be used so as more readily to penetrate it. In hard clay, the ordinary clamshell possesses little value.

Clamshell and orange-peel buckets vary in capacity from 500 pounds to two tons, and are particularly advantageous in confined places and in almost any depth of water. They may be worked by either a single or a double chain, the action being as follows: On the bucket being released, the chain runs over the pulley at the jib head, and the bucket descends through the water into the material to be lifted, with its cutting edges open, and the faces or teeth falling perpendicularly. The weight of the bucket forces them into the material. The chain is then drawn in by the drum of the winch, closing the grab and raising it.

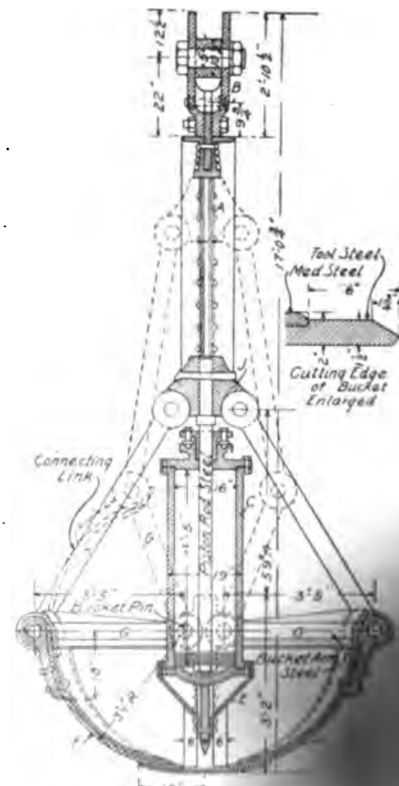


Fig. 67. Vertical Section of Pa[illegible] Clamshell Bucket.

The *dipper dredge* (Fig. 68) consists of a massive steel dipper or bucket, which is moved by means of an arm, in a vertical plane, and which may be revolved on a table through a considerable arc of a circle. It is similar to the steam shovel in every respect. The process is to scrape the dipper, at the end of its arm, over the bottom of the channel, until the dipper is full, when it is raised above the water by the arm, moved horizontally through an arc of a circle by the turning of the table, the bottom opened, and the contents discharged into a scow. The scow is held in position by means of timbers driven into the bed of the harbor.

The dippers are made of sizes from one-half a cubic yard to eight cubic yards in capacity, and will operate in water up to 40 feet in depth. They have the disadvantage of requiring considerable space in which to operate, and in this respect are inferior in confined situations to the clamshell type. They will however excavate almost any material except solid rock.

Pump or *suction dredges* are principally used for raising sand cr mud, the material being pumped up through suction-tubes, and discharged either into a hopper or through pipes or troughs onto the shore. They can also be made to remove clay by having a cutter placed at the end of a shaft working in the suction-tube; the cutter breaks up the material sufficiently small for detached pieces to be drawn up the suction-tube. When working in heavy sand, the material left in the hopper represents about 50 per cent of the water and sand raised; in clayey sand, about 40 per cent remains; and in sticky clay, about 10 per cent.

Fig. 68. Dipper Dredge Discharging into Scow.

The advantage of these dredges is that they can work in any ordinary weather, and at times when it would be quite impossible for those of other types to do so; and the material can be delivered directly for distances of from 200 to 300 yards by the force of the pump, so that silt raised from the bed of a river or harbor may be delivered directly at the back of training walls, or used for raising land for harbor purposes. The pumps used will raise and pass through them stones or other hard substances of any size that will pass through the suction-pipes.

One of the more recently constructed dredges of this class is 150 feet long, 50 feet wide, and has engines of 400 horse-power. When dredging in clay, the suction-pipe is placed in a vertical position, and is made to travel around the end of the vessel. Attached to the pipe is a vertical shaft with horizontal cutters at the bottom. These are hooded over; and as the material is broken up, it is drawn into the suction-pipe by the action of the pumps. The suction-pipe, made in three telescopic lengths, is 21 inches in diameter, and is capable of being used in 26 feet of water. The centrifugal pump will raise 16,000 cubic feet of water and sludge a minute. Sometimes bricks, stones, iron, etc., weighing from 6 to 8 pounds are drawn up through the suction-pipes without injury to the pumps. The material raised is discharged through iron pipes varying in length from 1,000 to 4,000 feet, onto low swampy ground which is thus being raised.

To moor these dredges, piles fitted with heavy iron shoes are suspended vertically from frames on the deck, the lower end being kept in place by guide-rings. When the vessel is to be moored, the piles are set free, and drop onto and penetrate the ground at the bottom of the channel. When the vessel requires warping forward, the piles are lifted to their former position by chains passing over winches worked by the engines.

In the improvements carried out for deepening the channel leading to the harbor of New York, nearly 5,000,000 cubic yards of material were removed, at an average cost of 26.4 cents per cubic yard. The material had to be conveyed $10\frac{1}{2}$ miles to sea. The quantity on which the contractor was paid was the quantity in the barges; but this was about 27 per cent less than the actual quantity removed, the remainder being carried away in suspension by the current. The material had to be raised from a depth varying from 24 to 35 feet under water, the total lift being 36 to 46 feet. With every cubic yard of solid matter thus transported by the barges and paid for, a large additional quantity had to be raised. The material consisted of mud, clay, and sand; and part of this being nearly of the same specific gravity as the water when it became incorporated with it, only a certain quantity settled in the barges, the remainder going overboard with the water and being carried away by the current. The plant provided for doing the principal part of the work consisted of three seagoing dredging steamers, four large barges, and four steam tugs. These dredges varied in length from 132 to 157 feet, 31 to 37 feet beam, and 8 to 16 feet depth, their carrying capacity varying from 275 to 650 cubic yards. Each dredge was provided with two pumping outfits independently arranged, and each pump was capable of lifting 4,200 gallons per minute. The suction-pipes 15 to 18 inches in diameter, and 60 feet long, were located one on each side of the steamer amidships. To render them flexible so as to accommodate them to the rolling and pitching motion, about 12 feet consisted of rubber supported by chains against the vertical strain caused by the weight of the remainder of the pipes. The pipes were provided with scrapers at the bottom for loosening the material. The hoppers were divided into compartments surrounded by longitudinal sluiceways extending either way from a central receiving hopper. These sluices were provided along their source with a series of adjustable bottom and side gates, to regulate the discharge of the material into the different compartments, and prevent the listing of the vessels. The steamers worked in all but the roughest weather, and were kept under headway from the time they left their berths in the morning until they returned to them at night. On arriving at the station, the pipes were lowered, and the vessel kept constantly under steering headway until the hoppers were full, when the pipes were hoisted, and the vessel under full steam proceeded to the discharging grounds.

Rock-Breaking Dredges. Heretofore, where it was desired to remove rock from beneath the surface of the water, various means of applying powder or dynamite to break up the material first were employed, after which some sort of dredge would remove the broken pieces. This has been replaced to some extent—at least where the rock is soft—by using strong steel claws fastened to the buckets; or every other bucket has been replaced by them.

In the case of very hard rock, heavy chisels have been employed, which by their weight falling upon the material crush it, when it may be removed. The height of the fall varies from 10 to 20 feet; and the weight of the chisel or ram, from 4 to 10 tons.

Several machines like this were used for removing a shoal containing about 3,000,000 tons of hard limestone rock in the Suez Canal. The first dredge was 180 feet long, 40 feet beam, and 12 feet draught. It was fitted with ten heavy chisel-pointed rams or rock-cutters, each 42 feet long and weighing 4 tons. Five of these were fitted on each side of the well through which the broken rock was lifted by a ladder and buckets, the bag of the bucket-chain being supported by a guide-wheel specially designed to relieve the strain on the bearings, bucket-links, and pins. These rams were raised by hydraulic power, and allowed to fall from 10 to 20 feet on the rock, delivering from 200 to 300 blows an hour. The bucket-chains were driven by a four-cylinder compound engine of 200 horse-power. When at work, the machine was moved over the surface in a series of arcs by winch motion arranged by swinging the vessel from side to side, pivoting on a steel mooring pile three feet in diameter, which passed down through the hull in the after part of the vessel and rested on the rock. There were two of these piles manœuvered by hydraulic power, enabling the vessel to advance a given distance after each swing, and preventing her losing her position.

Dredges somewhat similar to the above are being used at present on the Panama Canal construction by the United States Government.

HARBOR IMPROVEMENTS ON THE ATLANTIC COAST

The improvement of harbors on that part of the Atlantic coast south of New England involves almost entirely such protective or corrective works as are necessary for a sandy coastline. In making a study of these works, it is necessary to consider forces which cause the movement of sand; for almost the whole object of river and harbor improvements on a coastline of this character is the control of the movements, or the retention in place, of large masses of sand that would otherwise be prejudicially shifted by winds and waves or tidal currents. Wherever dry sands are found, they are always shifted by prevailing winds; indeed sand is drifted about under the action of winds precisely like snow. Captain Black, in his paper on the improvement of harbors on the south Atlantic coast of the United States, asserts that on the Florida coast sand moves like drifting snow with a depth of 4 to 8 inches, while a properly constructed sand-catch will form a dune 10 feet high in a single year. It is thus evident that, inasmuch as the sand will be moved in the direction of the wind, if that direction is across a channel the result will be deposition of sand in that channel, with a possible obstruction of the latter so far as the passage of vessels of considerable size is concerned. Similar results from the same wind forces are constantly produced on those portions of the Pacific Coast which are sandy.

The most troublesome forces to control in connection with the littoral movements of sand, are those of waves and currents—particularly tidal currents, which are largely shifted in direction by bars and shoals of their own creation, though the winds, also, frequently create difficulty. The study of wave motion has been largely developed by Scott-Russell, Hagen, the Brothers Weber, Rankin, Froude, and others, but it is not necessary to consider in this connection the analytic basis of such work. It is only requisite to observe that wave motion is a combination of rotation and translation. The orbit which a molecule of water describes in wave production has been taken to be both elliptic and circular. It is the successive descriptions of these orbits by molecules of water in series that produce what is called a wave. The velocity of translation of wave motion is, therefore, an entirely different thing from the velocity of movement of molecules of water in their orbits, the successive descriptions of which produce the wave. Stevenson, in his work on rivers and harbors, states that the height of the highest point of a wave crest above the normal level of the water in which it is produced is $\frac{2}{3}$ of the wave height from hollow to crest, and that the velocity of wave translation in the Atlantic Ocean may reach as high as 115 feet per second, while ordinary storm waves travel at a rate of 50 to 60 feet per second.

It is a matter of common observation that waves break as they roll in toward the shore in shallow water; and Stevenson states that the mean depth at which they will commence to break is from once to twice a wave height, but that they break earlier when the water shoals rapidly than when the slope of the bottom is more gradual. In all the preceding, it has been supposed that the features of wave motion have not been directly affected by the action of the wind. It is of course perfectly evident that the wind will have great influence upon the velocity of wave translation, as well as upon the shape of the wave and the circumstances under which its crest will break. A sufficiently high wind will cause almost any wave to break at its crest, and will cause each molecule to describe a very different orbit from that assumed in mathematical analysis.

It is evident that the crest of a wave will be at right angles to the direction of the wind, and that its motion will be along that direction; but adjacent to the shore line, if the wind is blowing ob-

OLD-STYLE BUCKET DREDGE
Cape Cod Canal.

SUCTION DREDGE "FRANCIS T. SIMMONS."

Engaged in hydraulic filling operations for the park system of Chicago. Capacity, 1,500 cubic yards an hour.

liquely to it, one end of the crest will be retarded and deflected by the shore, while the other will be in deep water. Consequently the crest will swing around more or less toward the shore line, and produce complicated motions in the water. As a wave rolls in shore, the water is piled up on the beach for a moment and then flows back; and inasmuch as successive waves are rolling in along the water surface, its back-flow will result in an under-current along the bottom at a constantly decreasing rate in receding from that part of the beach at the low-water line. The orbital motion of the molecules of water in waves will produce great disturbances of the bottom in shallows; in fact it is constantly picking up sand or sediment which will be carried in shore by inshore winds and by the motion of translation of the waves, until eventually the material is piled upon the beach. The back-flow or under-current on the bottom will, on the other hand, tend to carry material deposited on the beach out to deep water, but with very much less activity than the inshore forces just mentioned. As a consequence, the beach constantly gains material; and inasmuch as the back-flow or under-current is relatively weak, the finer portions only will be transported back towards deeper water. This accounts for the clean and relatively granular character of most sand beaches.

A current of water which is too weak to erode a bottom will transport very considerable material in suspension or in semi-suspension if it can be once placed in that current. Various authorities name currents from 6 inches to 15 inches per second as sufficient to begin a scour upon a sandy bottom of various degrees of coarseness. The liability to scour, however, of any sandy bottom, depends to a considerable extent upon the degree of solidity with which the sand is packed. Captain W. M. Black, in the paper already cited, states that he has seen a bank of packed sand stand without erosion under a current of 3 feet per second. With a relatively high current, considerable material will, of course, be transported, while a slower current will support considerably less material. Hence, if the back-flow or under-current on a beach is relatively strong, considerable material will be carried into the water from the beach, and the resulting bottom slope will be much more gradual than if the back-flow is weak and able to transport but little material, and that of the finest.

If the shore is eroded by a littoral current, the sand which is transported by the latter will be deposited with the decrease of velocity which takes place as the current flows into deeper water, and a spit or shoal point will thus be formed by deposition. Again, if these currents are formed and directed by the winds, this deposition of sand may become very variable and form hooks or loops, and thus produce a shoal bottom of eccentric character; at the same time, the depositions may be spread so as to form extended bars or shoals across a river entrance or harbor.

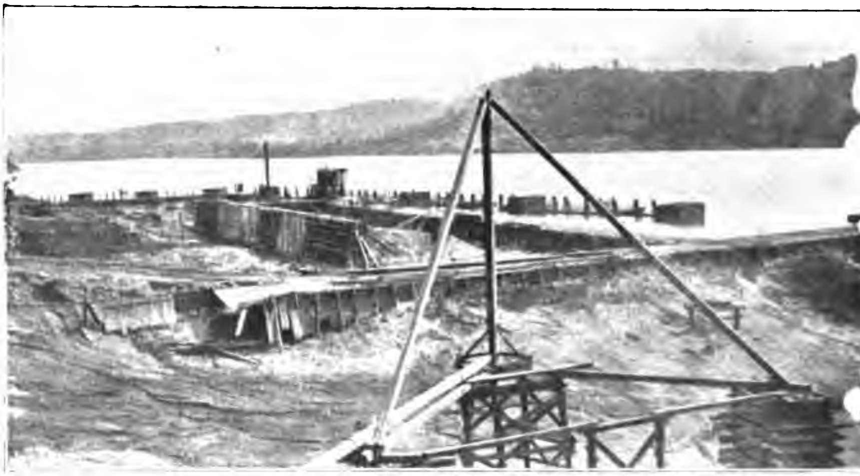
The force of waves on the Atlantic coast of the United States is somewhat less than that of similar action upon some foreign shores, although they are seen frequently to exert destructive action upon protection works. The following accounts are taken from the Report of the Chief of Engineers, U. S. A., of 1890, on the Improvement of St. Augustine Harbor, as well as from the paper of Captain Black already cited:

"A wave may act on the jetty directly, by a blow or a push, or a blow and a push combined; and indirectly, by a pull, by compressing the air in the voids of the masonry, by upward pressure due to the difference of head produced on the two sides of the jetty, or by a combination of these actions. The direct action measured on the dynamometers had effects equal to pressures varying between 190 and 753 pounds per square foot. This action took place when a wave broke directly on or in advance of the jetty; this also compressed the air in the voids of the jetty. . . . . . Jets of water and sand were sometimes projected up from the cracks in the jetty to some height. The maximum height of any wave observed striking the work was 6 feet. . . . . Up to a height of about 2 feet above mean low water, riprap weighing 40 to 50 pounds was but little disturbed. Above this limit, to the height of 10 feet, the highest point observed, riprap varying in weight from 40 to 200 pounds, could not be held at any slope. An isolated piece of concrete weighing 350 pounds, and resting on its flat base 1.7 square feet in area, with its center of gravity 7 feet above mean low water, was moved several feet by breakers whose crests were about 7½ feet above mean low water. These breakers measured 3½ feet from hollow to crest. . . . . . All that portion of a mound or wing composed of riprap (varying in weight from 40 to 220 pounds) tightly chinked with oyster shells, lying between 4 and 6 feet above mean low water, no matter what side slopes the riprap was given, would be carried away in a single tide whenever breakers greater than 4 feet in height struck it fairly. . . . . A block of concrete weighing 527 pounds was elevated 1.3 feet by the action of a single breaker. During the same tide, it was moved 23 feet inshore. A dynamometer within 8 feet of its original position recorded a maximum pressure of 575 pounds per square foot during this tide. A piece of concrete weighing 200 pounds was lifted vertically to a higher level than that of the water surface, by a wave which broke just in front of it. Another block of concrete weighing 1,600 pounds was lifted from its bed vertically at least 14 inches, and then moved several yards.

"Later, a concrete block 10 by 6 by 2½ feet and weighing dry 21,000 pounds, lying about at the mean low-water line of the beach, was lifted vertically 3 inches, and there caught and held fast. The maximum wave-height and dynamometer readings during that gale were 5.5 feet and 633 pounds respectively."

It is believed from many observations, that sand is much more in motion or "alive" during flood-tides than during the ebb, although some perfectly reliable observations to the contrary have occasionally been made. It is known that a new beach formed during a gale will at first be comparatively soft, and eventually become very hard, under the pounding action of breakers; in fact, some beaches become almost as hard as asphalt pavements, so that they will ring under a blow from the foot of a horse.

The coast from Cape Hatteras to the southern extremity of Florida is low and sandy, broken by many openings into rivers, bays, and inlets. The highest point in the entire line is Mount Cornelia, at the mouth of St. John's River, and it is only 60 feet high. There is a general motion toward the south, of the sand on this entire stretch of coast, in consequence of the prevailing winds having a direction which is in the main southerly. The slope of the country bordering the coast is invariably very slight; consequently all the rivers emptying into the ocean through it are tidal streams, the effect of the tides being felt for a considerable number of miles back from their mouths. As a result, the material brought down by the different streams does not reach the ocean, but is deposited in the basins immediately back of their mouths, except in a very few cases where small amounts may reach the ocean. The slope of the beach along low-water line is also very slight, being but one in 45 off Cape Hatteras, and as low as one in 1,000 at Sapelo Inlet just south of Savannah, from which point it increases southward until it reaches one in 55 at Cape Canaveral, about midway down the coast of Florida. The decrease in bottom slope in the vicinity of Florida is largely due to the bay-like shape of the shore between Cape Hatteras and Cape Florida. As the tides move into this "Southern Bay," as it is called, they become concentrated in their effects, and rise in height as the point of the bay in the vicinity of Savannah is reached. The material moved by the tides is therefore concentrated in the same vicinity, and a steeper slope of bottom results.



Partial View of Lock and Dam at Fernbank, Ohio, under Construction.

Cofferdam Used in Construction of the Dam at Fernbank.

Main Guide-Wall of Fernbank Dam under Construction.

Fig. 69. Improvement of the Ohio River. Views of the construction at Fernbank, a few miles below Cincinnati, which will give that city a permanent harbor of 9-foot stage, and make the Ohio River navigable throughout the year for fifty miles above the dam.

NEW WOODEN PIER AT LA BOCA, ISTHMUS OF PANAMA
Pier 800 feet long.

In the vicinity of Cape Hatteras, the prevailing winds are from the north and northeast, and the same is true for the vicinity of Charleston, while on the coast of Florida the reverse condition takes place. Nevertheless, in all cases, the heaviest storms and highest winds are from a northerly direction, which produce at all points along the coast the southerly movement of the coast sand, which has already been mentioned. It is important to regard this fact in the designing and construction of correction works at the mouths of the more important rivers; and it will be noticed in many cases that this tendency has deflected the mouths of many of the rivers in a southerly direction. The general conditions prevailing on the portion of the coast under discussion may be summarized thus:

First—The shores are low, fringed by a beach of light, shifting sand; the foreshore has a very gentle slope, giving comparative protection from wave action.

Second—The heaviest and most prolonged gales are from the northeast, with occasional severe gales from the southeast.

Third—The mean tidal range is from 1.5 to 7 feet, with two tides daily.

Fourth—There are no continuous, strong littoral currents, and no strong tidal currents excepting in the immediate vicinity of the openings on the coastline.

Fifth—The coast storm-drift of sand is large in volume, with a resultant movement to the south.

Sixth—This drift forms fan-shaped shoals across all openings on the coastline.

Seventh—There is little or no fresh-water sediment brought to the coastline.

Eighth—The channels across the shoals are generally shifting in character; and the navigable channels are maintained by the ebb outflow from the interior tidal basins, reinforced by the fresh-water discharge from the interior.

Improvement at Mouth of Cape Fear River. The City of Wilmington, North Carolina, is situated nearly 28 miles from the mouth of Cape Fear River; and for about one-half the distance, the latter is roughly parallel to the sea beach, from which it is separated in the main by a narrow stretch or tongue of sand in some places but a few hundred feet wide, but, near its mouth, by the accretion of sediment called "Smith's Island," which is two miles wide and intersected by numerous threads of water. It is probable that in the early age of the coast, Cape Fear River emptied into the ocean at a point nearly opposite Wilmington, and that the southern motion of the coast sand gradually deflected its mouth by slow stages to its present position. There are historical reasons for believing that Cape Fear River had a channel capable of passing vessels of 14 feet draught through Bald Head channel to the City of Wilmington, up to 1761; but in that year a violent equinoctial storm of four days' duration caused the sea to break through the narrow sand beach at what is now known as New Inlet, some 8 miles up the river from the former entrance. This new channel was reported at different times to have a depth of 2½ feet to 18 feet; but the effect of this shortened route to the sea for the river water was a scouring from a certain depth of six feet at low water in 1797 to 10 feet at low water in 1839, with a corresponding shallowing by bar formation in both the Western and Bald Head channels, although up to 1839 the latter was the main entrance to the river. From 1839 to 1872, that channel was discontinued, while New Inlet and the Western channel were used as the main entrances to the river. Inasmuch as New Inlet was vastly more exposed to the prevailing northeast winds than the natural or Bald Head channel, making both the entrance for vessels and the preservation of correction works much more expensive at the former point, it was determined in 1872 to close New Inlet and reopen and maintain Bald Head channel. In consequence of the decreased velocity in the old channel, due to the efflux of water through New Inlet, there were produced horseshoe shoals and a fan-like bar formation at the outer extremity of Bald Head channel. The shoaling influence at different points was also considerably complicated by the tidal impulse up the river through Bald Head channel not being felt at New Inlet until considerably later than on the sea-front.

The New Inlet dike was constructed between 1874 and 1881. It was begun by constructing a cribwork pier 500 feet long out from Federal Point for the usual pier purposes for such work. The same scouring effect around the advancing end of the cribwork was experienced which gave the same kind of trouble in the construction of the Zekes Island dike, and resulted in deepening the water from six feet to twelve feet at a point 200 feet from the shore end of the pier. The pier was completed in the month of November, and the end crib was sunk in 19 feet of water, where the original depth was but 6 feet. The settlement of these cribs and the extreme difficulty in building upon them fast enough to keep them above low water, caused their abandonment for all subsequent work. In 1875 it was therefore determined to close the remainder of the New Inlet opening by first sinking an apron of log mattresses covered with a layer of brush and loaded with stone, in which the logs were laid normal to the axis of the jetty. The scour in front of the advancing end of the apron did not exceed three feet, and there was no settlement after the mattresses were placed in position.

The jetty was completed by dumping upon the apron already in position, one-man riprap stone. The slope of the finished work on the sea side was about 1 in 2, and on the land side 1 in 1½. On the top of this mass of riprap and above low water, there was carefully laid a capping of undressed granite blocks weighing from half a ton to four tons, which has a slope of 1 in 2 on the sea face, and a somewhat less slope on the land side. While this system of construction was an improvement upon the crib plan, it was found that the logs did not close the bottom of the jetty in a perfectly water-tight manner; hence material was gradually scoured out between the logs, causing them to settle. No settlement, however, was found where the original depth was greater than 13 feet below low water; but in some places there was a scour of 8 to 12 feet along the edges of the foundation, due to the flow of water over the top of the dike. In consequence of the irregularity of the tide effects at this point, the difference in head of water on the two sides of the dike was as much as two feet at some stages of the tide. This difference in head, combined with the flow over the top of the dike, caused the scour between the logs.

The crest of the completed dike is one foot above high water of ordinary spring tides, and the total length of the entire work is 4,752 feet; it contains nearly 183,000 cubic yards of material; and it cost complete about \$2.75 per cubic yard, or \$105 per linear foot.

This completed work joined the Zekes Island dam at its northernmost point, and completely closed the opening between Federal Point and the northern extremity of Smith's Island.

The only remaining jetty work for the improvement of this river mouth, is the dike across the shoals at the mouth of Bald Head channel, which has a length of 5,200 feet, and was built between 1883 and 1888. In consequence of the experience with the log foundation of the mattresses already described, it was determined to employ a different plan for this dike of mattresses.

The foundation consisted of a brush mattress apron, part in sections and part continuous, in 5 to 12 feet of water. This mattress was broad enough to extend beyond the side slopes of the dike, and thus to act as an apron for overflow. It consisted of three layers of brush, of which the lowest was transverse to the axis of the jetty, the next longitudinal, and the third transverse. These were compressed and held by fascine copes, or binders, 5 to 8 inches in diameter, strongly bound with spun yarn. The binders were parallel to the axes of the jetty, and were spaced 3½ to 4 feet apart in sets of two, one above and one beneath the mattress. The brush was compressed between the binders by a lever to the full strength of the 9 to 18-thread ratlin which tied the upper and lower binders together. The average thickness of the mattresses after compression was 9 inches. They are described as strong and pliable. A portion of the foundation was made of separate mattresses, constructed on a tilting table, 90 by 30 feet, and towed to the site of the work and sunk. Five thousand two hundred feet of the foundation was made of continuous mattress work in two parts, 3,100 and 2,100 feet long, on a scow 80 by 28 feet, provided with an inclined table 60 by 36 feet. A section having been launched, its binders were spliced to binders on the scow; another section was made, and the scow run from beneath on the line of the axis as before. The sinking of the mattress by stone followed a short distance behind the scow. In the small mattresses made on the tilting table, it was found that by practice an average of 25 feet in length of fascine binders was made each hour by one man. After the fascines were ready and the binders spliced, the mattresses were made and fully bound at the rate of 7½ square yards for each hour's labor. The cost of labor on mattresses and binders was 3½ cents per square yard of mattress. The cost of labor on the continuous mattresses made on the scow was about 5 cents per square yard. The cost of each style in place was about the same. The cost of the mattresses in place, including materials and plant, was 66.2 cents per square yard, or, allowing 46 per cent for compression, about $2.65 per cubic yard. The cost of the stone in place, including handling, superintendence, etc., was about $2.30 per cubic yard, assuming 1.12 cubic yards to the gross ton. The stone used for sinking the mattresses was a very light shell stone.

The dike was finished in 1888. The total amount of stone used in the construction was 76,000 cubic yards. Its crest is 6 feet wide, and is at the level of high water of ordinary spring tides. The side slopes are 1 on 1½. The crests and slopes to low water are capped with heavy stone, hand-laid to a smooth face, with an average thickness of 9 to 12 inches. The cost of the labor of placing, including wages of foreman but not superintendence, was 28.5 cents per superficial yard. The average subsidence of the dam due to settling in the mud, compression of mattresses, and consolidation of riprap, was about two feet.

It was necessary to dredge channels both through Horseshoe Shoal and the short shoal about one and one-half miles to the north of it, as well as through the outer bar along Bald Head channel. The dredge used for this work was of the hydraulic type. She was a small propeller of 145 tons burden, and was able to work very satisfactorily on the outer bar, except in stormy weather, when she was usually operated on the shoals inside the river mouth. The excavation was made with a nine-inch centrifugal pump driven by a double oscillating engine with ten-inch cylinders and with two suction-pipes of wrought iron, of 6¾ inches internal diameter. She was able to dredge in 6 to 14 feet depth of water, and frequently raised 45 cubic yards, her bin capacity, in 25 minutes; and the average day's work was 515 cubic yards. The operation of dredging, including repairs and an allowance for deterioration, cost between 11¾ and 14.8 cents per cubic yard. The contract price for similar work in the smooth water of the river since that time, has varied from 12 to 16.7 cents per cubic yard.

The experience gained in this work all shows in a marked manner the inadvisability of attempting to force a dike or jetty of full depth across a channel with a sandy and shifting bottom, in water in which there is an appreciable current. An increasing depth of scour will invariably take place in front of the advancing point of the work, and the difficulties caused both by construction and maintenance are very largely increased. The proper plan to follow, then, is that which has been almost invariably adopted by the later and similar works, by first sinking a pliable foundation which can be quickly put in place without greatly disturbing the flow. In this manner a bottom or foundation is secured for the superstructure which will not destructively yield. It is found, however, that water would even find its way under the Bald Head channel dike to some extent, and it produced a slight settlement. Some engineers have even recommended that such foundations be formed of riprap which settles into and forms a part of the soft or yielding bottom; but, on the whole, brush fascines have been found to be quite satisfactory. The result of all this Cape Fear River improvement work has been a channel with a low-water depth of seventeen feet through the outer bar in Bald Head channel, and sixteen feet through Horseshoe Shoal, the dredged channels having a width of 200 feet, with the depths named.

REVIEW QUESTIONS.

PRACTICAL TEST QUESTIONS.

In the foregoing sections of this Cyclopedia numerous illustrative examples are worked out in detail in order to show the application of the various methods and principles. Accompanying these are examples for practice which will aid the reader in fixing the principles in mind.

In the following pages are given a large number of test questions and problems which afford a valuable means of testing the reader's knowledge of the subjects treated. They will be found excellent practice for those preparing for Civil Service Examinations. In some cases numerical answers are given as a further aid in this work.

REVIEW QUESTIONS

ON THE SUBJECT OF

HYDRAULICS

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1. What will be the exact weight of 1 U. S. gallon of distilled water at a temperature of 160° F.?

2. How much higher will a water barometer stand at a place 500 feet above sea level than it will when 4,000 feet above sea level?

3. Assuming that it is practicable to lift water by suction a distance equal to three-fourths the theoretical height as shown by the water barometer, how high is a suction lift practicable at a place located 6,000 feet above sea level?

4. What will be the pressure per sq. inch on the bottom or side of a vessel containing water, at a point 20 feet below the water surface?

5. In Fig. 2, how heavy a weight W will be supported by a weight P equal to 100 pounds, the area of W being 20 sq. in. and that of P. being 2 sq. in.?

6. What is the total pressure on the side of a vessel 10 in. wide and filled with water to a depth of 8 inches?

7. What is the pressure on the face of a plate 10 × 10-in. in area, inclined at an angle of 10° from the vertical, and submerged so that the center of the plate is 30 in. below the surface?

8. What are the horizontal and vertical components of the pressure in question 7?

9. What is the stress per sq. in. in a pipe 10 in. internal diameter and ½ in. thick, under a pressure head of water of 100 lb. per sq. inch?

10. If the safe stress on cast iron is 5,000 lb. per sq. in., what is the necessary thickness of a cast iron pipe 12 in. in diam-

eter to resist a bursting pressure due to a head of water of 150 feet?

11. What is the total pressure on the vertical face of a dam one foot long and having a depth of water in front of it of 25 feet?

12. At what depth is the center of pressure in question 11?

13. What will be the buoyant effect exerted upon a submerged body having a volume of 2,000 cu. in.?

14. Is the buoyant effect upon a body different when the body is placed at different depths in the water, being wholly submerged in all cases?

15. A body of a specific gravity of 1.5 is submerged. What will be its weight in air per cubic foot of volume? What will be its weight in water?

16. How should orifices be made to be reliable for measuring water?

17. What is meant by the "coefficient of contraction"? The "coefficient of discharge?"

18. If the coefficient of discharge is .6 what will be the discharge through an orifice 2 in. in diameter, under a head of 2 feet?

19. If the coefficient of contraction is .65 in question 18, what is the actual area of the jet at its reduced section?

20. Using Tables 3 and 4 what is the discharge from a round and a square orifice under a head of 10 feet, the orifices being respectively 6 in. in diameter and 6 in. × 6 in. square?

21. What is the discharge of a square orifice of the same actual *area* as the round orifice of question 20, and under the same head?

22. Using a coefficient of discharge of .60 what is the necessary diameter of a round orifice to discharge 2 cu. ft. per second under a head of 20 ft.?

23. How should a weir be arranged to give reliable results in the measurement of water?

24. What is meant by the "velocity of approach"?

25. Using Table No. 7, what is the discharge of a weir 5 ft. long under a head of 12 inches?

26. Using formula 25, what will be the answer to question 25? What is the difference between the two methods in per cent?

REVIEW QUESTIONS

ON THE SUBJECT OF

WATER-POWER DEVELOPMENT

PART I

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1. Define the terms *work*, *power*, and *energy*. What are their units of measurement?

2. Illustrate the relation between *pressure-head*, *velocity-head* and *gravity-head*.

3. Define the terms *absolute velocity* and *relative velocity*.

4. Distinguish *impulse* and *reaction*; *static pressure* and *dynamic pressure*. To give an impulse of 120 lbs., what must be the velocity of a jet of water issuing from a nozzle 1.25 inches in diameter?

5. Explain how the dynamic pressure of a jet of water can be measured.

6. What, in general, is the difference between a water-wheel and a turbine?

7. Discuss the requirements for high efficiency in hydraulic motors. To what causes are losses of energy due?

8. Compare the respective advantages or disadvantages of, overshot, breast, and undershot wheels.

9. Illustrate by diagrams, and discuss the significance of, different shapes of buckets on overshot wheels.

10. What conditions should be observed in installing overshot, breast, and undershot wheels?

11. Distinguish several different types of impulse wheels.

12. Describe the essential features of the downward-flow impulse wheel.

13. Compare the respective advantages of inward and outward flow in the case of impulse wheel

REVIEW QUESTIONS

ON THE SUBJECT OF

WATER-POWER DEVELOPMENT

PART II

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1. Draw a diagram showing the essential parts of a turbine, and giving their notation.

2. Describe various methods of measuring the water delivered to a turbine.

3. In installing a water-power plant, what general precautions should be observed to avoid loss of head outside the motor?

4. How is the effective head on a turbine determined?

5. Explain the distinction between the various general classes into which turbines are divided according to method of admitting water and passing it through them.

6. What, briefly, are the distinguishing characteristics of *impulse, reaction*, and *limit* turbines. What is the difference between an impulse turbine and an impulse water-wheel?

7. What are the relative advantages, in general, of radial, axial, and mixed flow in inward-flow and outward-flow reaction turbines? To which type does the Fourneyron turbine belong? the Francis turbine? the Jonval turbine? What general type is most commonly used in America?

8. Describe the usual method of measuring the effective work of a hydraulic motor.

9. Describe the mechanical features and working principles of the Holyoke testing-flume.

10. What important general principles were demonstrated by Mr. Francis's historic test of the Tremont turbine?

11. What factors must be taken into consideration in determining the work and efficiency of reaction and impulse turbines?

REVIEW QUESTIONS

ON THE SUBJECT OF

WATER-POWER DEVELOPMENT

PART III

1. Name the principal parts of a typical hydro-electric power plant.

2. What devices are used for preventing floating ice and debris from entering turbine-chambers?

3. What precautions are to be observed in the construction of head-races and tail-races?

4. How would you attack the problem raised by the head-race water carrying considerable quantities of sand in suspension? Is the removal of the sand in all cases advisable?

5. What rules should be observed in determining the size of conductors for conveying water to and from turbines?

6. What mechanical devices are employed for regulating the speed of water in penstocks?

7. Is the length of the penstock ever an unimportant factor? Explain.

8. Name, and describe briefly, the different types of head-gates for turbine plants, and the mechanism for operating them.

9. Enumerate the principal losses of head of water which occur between the head-race and the tail-race. Indicate briefly how these losses may be minimized.

10. Under what conditions is high speed of water in penstock permissible?

11. Under what conditions is low speed of water in penstock advisable?

12. Of what materials are penstocks composed? Compare their relative advantages.

REVIEW QUESTIONS

ON THE SUBJECT OF

RIVER AND HARBOR IMPROVEMENT

1. Define the terms *cut-off*, *artificial outlet*, *levee*, *banquette*, *revetment*, *crevasse*, *fascine*, *mattress*, *spur-dike*, *longitudinal dike*, *submerged dike*.

2. What three general methods may be adopted for protecting bottom lands along rivers subject to overflow?

3. Discuss the conditions under which flood control is feasible by construction of cut-offs or by diversion.

4. Describe fully the functions of the Great Lakes as storage reservoirs. How far may nature be imitated in reservoir construction for flood protection?

5. State the rules to be observed in the construction of levees or dikes.

6. Describe the methods of river-bank protection as developed along the lower Mississippi.

7. Compare as to their power to resist erosion the different materials used in levee construction.

8. How are harbors classified? Name at least one of each class.

9. Discuss the operation of the natural phenomena that affect harbor works.

10. Classify breakwaters, and describe the methods used in their construction.

11. Describe briefly the most important artificial aids to navigation installed in harbors and their approaches and along seaboards.

12. Distinguish between dry docks, wet docks, and slips, and indicate briefly their requirements.

13. Describe the principal types of dredges. For what conditions is each best adapted?

14. What conditions determine the type of breakwater to be built?

GENERAL INDEX

GENERAL INDEX

In this index the *Volume number* appears in Roman numerals—thus, I, II, III, IV, etc.; and the *Page number* in Arabic numerals—thus, 1, 2, 3, 4, etc. For example, Volume IV, Page 327, is written IV, 327.

In addition to this "General Index," the reader is referred to the more exhaustive analytical index to be found in each volume.

The page numbers of each volume will be found at the bottom of the pages; the numbers at the top refer only to the section.

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