SELECTING HYDRAULIC REACTION TURBINES

UNITED STATES DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
As the Nation's principal conservation agency, the Department of the Interior has responsibility for most of our nationally owned public lands and natural resources. This includes fostering the wisest use of our land and water resources, protecting our fish and wildlife, preserving the environmental and cultural values of our national parks and historical places, and providing for the enjoyment of life through outdoor recreation. The Department assesses our energy and mineral resources and works to assure that their development is in the best interests of all our people. The Department also has a major responsibility for American Indian reservation communities and for people who live in Island Territories under U.S. administration.

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Preface

"Selecting Hydraulic Reaction Turbines," first issued in 1954, was originally prepared by R. E. Krueger under the supervision of Ireal A. Winter, then Chief, Hydraulic Machinery Branch, of the Bureau of Reclamation's Division of Design at the Denver Engineering Center. Revised editions were issued in 1959 and 1966.

This revision reflects modern trends in the selection of hydraulic reaction turbines. Most of the figures have been revised to include recent experience. International Metric System (SI) units as well as U.S. customary units are used. The revised figures were prepared by Richard N. Walters, mechanical engineer in the Mechanical Branch, Division of Design, Engineering and Research Center, Denver. The text was revised by Walters and Carlos G. Bates, Head, Hydraulic Machinery Section. Personnel of the Technical Services and Publications Branch, Division of Engineering Support, provided editing services. Printing was accomplished through the Bureau's Publications Branch, Division of General Services, Washington, D.C.
Letter Symbols and Quantities

$A = \text{Dimension of } p \text{ at } 0^\circ$

$a = \text{Radial cross-sectional area of semispiral case}$

$b = \text{Distance from } D_2 \text{ to } \Phi \text{ of distributor}$

$D_1 = \text{Entrance diameter of runner on distributor } \Phi$

$D_2 = \text{Minimum opening diameter of runner}$

$D_3 = \text{Discharge diameter of runner}$

$D_4 = \text{Design diameter of draft tube}$

$D_M = \text{Maximum outside diameter of runner}$

$F_H = \text{Hydraulic thrust}$

$FY_B = \text{Servomotor blade capacity}$

$FY_M = \text{Servomotor gate capacity}$

$ft = \text{foot}$

$g = \text{Gravitational constant (acceleration)}$

$H_a = \text{Atmospheric pressure}$

$H_{av} = \text{Weighted average head}$

$H_b = \text{Atmospheric pressure minus vapor pressure}$

$H_e = \text{Gross head}$

$H_{max} = \text{Maximum head}$

$H_{min} = \text{Minimum head}$

$H_n = \text{Net head}$

$H_s = \text{Static draft head, distance from } D_2 \text{ to minimum tailwater}$

$H_v = \text{Vapor pressure of water}$

$Hz = \text{A-c frequency}$

$h_{cr} = \text{Critical head}$

$h_d = \text{Design head}$

$h_r = \text{Rated head}$

$hp = \text{Horsepower}$

$K = \text{Constant} = (A - (\rho \text{ at } \theta))/\theta^2$

$K = \text{Factor} = T_w/T_t$, speed rise factor

$kVA = \text{Kilovolt-amperes capacity of generator}$

$kW = \text{Kilowatt power of generator}$

$L = \text{Length of penstock}$

$m = \text{Metre}$

$n = \text{Rotational speed, design}$

$n' = \text{Trial rotational speed}$

$n_{max} = \text{Runaway speed at } H_{max}$

$n_r = \text{Runaway speed}$

$n_s = \text{Design specific speed}$

$n'_s = \text{Trial specific speed}$

$P.F. = \text{Generator power factor}$

$P_d = \text{Turbine full-gate capacity at } h_d$

$P_t = \text{Turbine full-gate capacity at } h_r$

$P_{min} = \text{Servomotor minimum rated oil pressure}$

$Q_{cr} = \text{Turbine full-gate discharge at } h_{cr}$

$Q_a = \text{Turbine full-gate discharge at } h_a$

$Q_{max} = \text{Turbine full-gate discharge at } H_{max}$

$Q_r = \text{Turbine full-gate discharge at } h_r$

$q = \text{Quantity of flow, semispiral case}$

$R = \text{Radius}$

$s = \text{Second}$

$r/min = \text{Revolutions per minute}$

$S_R = \text{Speed rise}$

$S'_R = \text{Speed rise, effects of water hammer}$

$T_t = \text{Servomotor minimum closing time}$

$T_k = \text{Full closing time of governor}$

$T_m = \text{Mechanical startup time}$

$T_w = \text{Water startup time}$

$V_a = \text{Semispiral case uniform angular velocity}$

$V_c = \text{Spiral case water velocity}$

$V_r = \text{Conduit water velocity for full gate at } h_r$

$V_s = \text{Servomotor net volume}$

$WR^2 = \text{Product of weight of revolving parts and the square of the radius of gyration}$

$Z = \text{Total draft head, } \Phi \text{ distributor to minimum tailwater}$

$\gamma = \text{Gamma} = \text{specific weight of water}$

$\eta_d = \text{Eta}_d = \text{turbine design efficiency}$

$\eta_s = \text{Eta}_s = \text{generator efficiency}$

$\eta_t = \text{Eta}_t = \text{turbine efficiency}$

$\theta = \text{Theta} = \text{angle of turn expressed in radians}$

$\pi = \text{Pi} = 3.14159 \ldots$

$\rho = \text{Rho} = \text{distance from } \Phi \text{ of unit to neatline of the large semispiral}$

$\Sigma = \text{Sigma} = \text{summation of}$

$\sigma = \text{Sigma} = \text{cavitation coefficient}$

$\phi_3 = \text{Phi}_3 = n\pi D_3/(60\sqrt{2gH_d})$, velocity ratio
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Introduction

Multipurpose projects in the Western United States usually generate power incidental to release of water for other purposes. This is especially true of the large Bureau of Reclamation projects where water is impounded and released according to irrigation requirements, and the power generated during that release is used primarily for the benefit of landowners.

The water level in reservoirs for irrigation storage is subject to large fluctuations, with consequent extremes of head variation on the turbine. In selecting hydraulic turbines to serve under such conditions, the designer must consider the effect of this head variation upon power capability, water release capability, efficiency, and cost of maintenance. These considerations must be added to the economic problems faced by the designer whose primary concerns are revenue and the costs normally associated with power developments. This computation outline applies to reaction turbines only. It was prepared to permit rapid selection of the proper unit, estimation of its major dimensions, and prediction of its performance.

The outline is a series of computations that proceed logically from given basic data to estimated dimensions and performance characteristics for the turbines. Each step is based on experience records plotted as curves having basic parameters which permit visual comparison of the characteristics of the selected unit, or one proposed by a manufacturer, with existing installations of similar character. Adjustments or deviations from the normal installation can be noted and evaluated.

The experience data are drawn mainly from the records of powerplants built and operated by the Bureau of Reclamation. These curves have been placed in a logical sequence, resulting in a selection without trial and error and without excessive readjustment and, at the same time, giving due consideration to the many factors affecting the selection of size and type.

Development of Hydraulic Turbines

In its present state of development, the hydraulic turbine is an admirable prime mover for central station power generating service. It is simple, efficient, easily controlled, and long lived. Its ability to act as a standby unit is outstanding, for it can be started cold and assume full load
in a matter of minutes, can follow load variations with little attention—it can go from speed-no-load to full load in 4 to 10 seconds—and it can drop load instantly without damage. Because of its simplicity, the hydraulic turbine can be made fully automatic and can be designed to operate with little attention. Hydroelectric plants may be combined with steam generating plants in the same system so that the steamplants carry the block load and the hydroplants follow the swings of the system. Where sufficient water power is available the hydraulic turbine has therefore become the backbone of large power distribution systems.

Power has been developed from flowing water by many types of water motors. The major classifications of motors are displacement, gravity, impulse, and reaction. Displacement motors are illustrated on figure 1, the simplest being the hoist or elevator, and the more complex the pumping motor which had widespread usage a number of years ago. The motor was actuated by hard water from the city mains and the pump supplied household needs with soft rainwater from a cistern.

In the gravity type of water motor, shown on figure 2, the weight of water in buckets on one side of a wheel or belt creates an unbalance, causing the wheel to turn. The overshot wheel operated grist mills for centuries and is a good example of the type. The breast wheel is more efficient, and the chain- or belt-driven motor is merely an elongated breast wheel, used where the drop is so great that a single wheel would be too cumbersome. Though simple in construction, these wheels are seriously limited in power and efficiency.

The impulse or velocity wheel, of which several varieties are illustrated on figure 3, is of great antiquity. Chinese artisans fabricated paddle wheels with attached buckets to lift water from flowing streams to adjacent rice fields more than 2,000 years ago, and it is known that much of the water supply of London in 1580 came from the Thames, lifted by similar water wheels suspended from the arches of London Bridge. The wheels served until the great fire of London in 1666.

The undershot paddle wheel also illustrated is a refinement of the earlier design, and has higher efficiency than the simple wheel.

Although the Atkins wheel was never developed commercially, its principle is sound hydraulically.
High-velocity water flowing over a curved surface loses little velocity, and a semicircular bucket attached to the curved surface, moving at half the velocity of the jet of water, would discharge the water at near zero velocity, producing maximum power. Buckets on the Atkins wheel were so placed as to discharge water radially inward toward the hub of the wheel.

The Pelton wheel, an adaptation of the Atkins principle, has been applied to power generation for a number of years. The buckets of a Pelton wheel are divided, splitting the jet, and discharging the water outward from the side of the bucket.

All of the foregoing wheels and turbines operate in the open air, or with only a splash cover over the wheel. Reaction turbines, on the other hand, all operate with the wheel submerged. This remaining category of water motor is illustrated on figure 4.

The Barker mill is an adaptation of Hero's engine to hydraulic use. The central reservoir is under water pressure, and the reaction of water escaping from ports at the ends of the arms causes them to rotate. The most familiar application of the principle is the rotating lawn sprinkler.

The Fourneyron turbine was designed to receive its driving force from the reaction of water flowing outward from the central pipe impinging on the blades of a water wheel, rotating a shaft passing upward through the elbow. The changing direction of the radial flow, as influenced by the impeller blades, provides the necessary reaction to operate the turbine. The Fourneyron turbine design was the result of a prize competition in France. The first turbines at Niagara Falls were of this type.

The Francis turbine is actuated in a similar manner, differing from the Fourneyron turbine only in that the inflowing water enters the blades of the runner from the periphery and is discharged inward and down. This modification of the Fourneyron turbine was improved by James Bichens Francis during his tests carried out at the Holyoke Testing Flume, where he developed his famous weir formula. The Francis turbine operates efficiently over a wide range of medium and high heads, as shown by the examples of Flatiron Powerplant (fig. 27), at a rated head of 1,055 feet (322 metres) and Grand Coulee Third Powerplant (fig. 28) at a rated head of 285 feet (87 metres). Both installations are discussed in this monograph.

The remaining variety of reaction turbine, the screw or propeller type, is a later development. The adjustable blade propeller has its best applications where the head on the turbine varies considerably. Propeller turbines operate at higher speeds than do Francis turbines and are more suitable than the Francis type for low-head installations.

The regions of application for the various types of turbines are shown on figure 5.
FIGURE 5.—Application diagram for types of hydraulic turbines.
General Requirements

A hydraulic turbine should be selected and designed to suit the specific range of conditions under which it is to operate if it is to attain the high order of efficiency expected in present-day installations. The possible combinations of head and capacity, even when speeds are restricted by the synchronous speed requirement, are so numerous that only seldom can identical units be used at more than one site. Consequently, the selection of a so-called standard, off-the-shelf unit may result in low plant efficiency and high cost of operation and maintenance without material reduction in first cost.

Continued development of water power in the United States requires consideration of sites which are less practicable than those previously developed. In fact, where serious power shortages exist, sites formerly considered infeasible may become feasible if only as locations for peaking plants or periodic block load units during power-deficient periods. These plants must be carefully designed to develop the full potential, and accurate selection of a suitable turbine becomes a matter of prime importance.

Careful studies of streamflow and reservoir operation and accurate field data are a necessity if proper turbine selection is to be made. Power demand and electrical characteristics have little effect on the type of turbine or ultimate plant capacity, especially when the prime purpose of the dam is irrigation storage and flood control rather than power.

If reservoir and power studies are not complete enough to reflect the limitations of the turbine, or if an improper selection of a turbine is made, the result may be a reduction of revenue that will impair the earnings and repayment ability of the plant. Improper selection may also result in a unit that will have excessive operating costs and be difficult to control.

Required Field Information

Rainfall and streamflow records combined with reservoir operation studies are used to estimate the amount of power available at a given site. The proportion of the available potential that can be economically developed depends upon the limitations of the turbines and associated equipment. Provision should always be made in the basic design for ultimate development of a site to its full economic potential, rather than to provide for a partial development to satisfy immediate demands.
The determination of the amount of installed capacity should take into account the head range, and the loss of efficiency and increased maintenance due to part-gate operation at the higher heads. Therefore, the power studies should be carefully reviewed by the plant designer to be certain that the expected revenue can be obtained from the units selected.

The project planning studies, from which the number and size of units are determined, should include:

1. Reservoir operation curves or tables:
   Detailed study by months, using available streamflow records to compute reservoir water elevations, power, spillway, and outlet releases, and potential power output.

2. Head and water levels:
   a. Maximum, weighted average, and minimum water elevations.
   b. Spillway crest elevation or maximum water elevation without spillway discharge.
   c. Maximum capacity of spillway and corresponding water elevation.
   d. Any additional significant elevations that may affect turbine operation.

3. Tailwater: Curve of tailwater elevation at the site showing elevations for all flows from zero to average yearly flood given to the nearest half-foot. This should be extended to the maximum spillway capacity. The important portion of the curve for the turbine is from zero to average yearly flood.

4. Power: Statement of expected power requirements including any limiting conditions such as capacity desired at minimum head, type of operation, or character of load demand.

5. General arrangement:
   a. Sufficient information to approximate the penstock size, length, and profile.
   b. Location of plant with respect to dam on topographical map of area.

### Definitions of various heads

Operating heads for reaction turbines are shown on figure 6.

**Gross head** ($H_g$) is the difference in elevation between the water levels of the forebay and the tailrace.

**Net head** ($H_n$) is the gross head less all hydraulic losses except those chargeable to the turbine. Net head is the head available for doing work on the turbine. The intake and penstock losses are not included in net head, but the spiral case and draft tube losses are considered chargeable to the turbine and are included in net head. For penstock lengths less than three times the maximum head, the total hydraulic loss ordinarily should not exceed 1 percent of the rated head, with the trashrack, intake, and bend losses accounting for approximately half the total loss. Longer penstocks may have losses of approximately 3 to 10 percent of the rated head. Penstock velocities should be based on economic studies, but should not exceed 30 ft/s (9 m/s).

**Maximum head** ($H_{max}$) is the gross head resulting from the difference in elevations between the maximum forebay level without surcharge and the tailrace level without spillway discharge, and with one unit operating at speed-no-load (turbine discharge of approximately 5 percent of rated flow). Under this condition, hydraulic losses are negligible and may be disregarded.

**Minimum head** ($H_{min}$) is the net head resulting from the difference in elevation between the minimum forebay level and the tailrace level minus losses with all turbines operating at full gate.

**Weighted average head** ($H_{ave}$) is the net head determined from reservoir operation calculations which will produce the same amount of energy in kilowatt-hours between that head and maximum head as is developed between that same head and minimum head.

**Design head** ($h_d$) is the net head at which peak efficiency is desired. This head should preferably approximate the weighted average head, but must be so selected that the maximum and minimum heads are not beyond the permissible operating range of the turbine. This is the head which determines the basic dimensions of the turbine and therefore of the powerplant.

**Rated head** ($h_r$) is the net head at which the full-gate output of the turbine produces the generator...
rated output in kilowatts. The turbine nameplate rating usually is given at this head. Selection of this head requires foresight and deliberation. The selection for Shasta Powerplant shown on figure 26 is an example of a poor selection as viewed through hindsight. Note these units were rated 75,000 kVA at a low head of 330 feet (101 metres) providing turbines with a tremendous overload capacity. The generators have been overloaded for more than 30 years, causing premature failure of the windings. They have been rewound for a rating of 95,000 kVA and consideration now is being given to a rating of 125,000 kVA with the original turbines.

In anticipation that the manufacturers would provide 5 percent overcapacity, the turbines for Grand Coulee Third Powerplant were rated at 285 feet (87 metres), so that turbines would develop rated generator capacity at weighted average head at best efficiency gate, and also would develop the generator overload capacity at full gate. Operation at heads above 325 feet (99 metres) will be extremely rare. See figure 28.

Critical head \( (h_{cr}) \) is the net head at which the full-gate output of the turbine produces the permissible overload on the generator at unity power factor (usually 115 percent of the generator kVA rating). This head will produce the maximum discharge through the turbine.

Permissible range of head

The peripheral speed of the turbine runner at the entering edge of the runner blades in relation to the spouting velocity of the water affects the efficiency and the cavitation characteristics. The permissible departure from the design head, at which this relation is optimum, has been found through experience to be as follows (fig. 7):

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<th>Maximum head (percent)</th>
<th>Minimum head (percent)</th>
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![Figure 6. Operating heads for reaction turbines.](106-D-339)
SELECTING HYDRAULIC REACTION TURBINES

Plant Rating

Reservoir operation studies determine the ultimate plant capacity and indicate the head at which that capacity should be developed. The studies should be checked to assure that capacity limitations, due to generator size and variation of turbine capacity with head, are allowed for in the kilowatt and kilowatt-hour revenue estimates. Average turbine performance curves suitable for reservoir operation studies are shown on figure 7.

\[
\text{Plant kilowatts} = 0.0846 \cdot \left[ \frac{\text{discharge}}{\text{in ft}^3/\text{s}} \right] \cdot \left[ \frac{\text{rated head}}{\text{in feet}} \right] \cdot \left[ \frac{\text{plant efficiency}}{} \right]
\]

\[
= 9.804 \cdot \left[ \frac{\text{discharge}}{\text{in m}^3/\text{s}} \right] \cdot \left[ \frac{\text{rated head}}{\text{in metres}} \right] \cdot \left[ \frac{\text{plant efficiency}}{} \right]
\]

Note: 1 megawatt = 1,000 kilowatts, and kilowatts = kilovolt-amperes • power factor

1 horsepower (U.S.) = 550 foot-pounds per second
= 0.7457 kilowatt

1 horsepower (metric) = 1.014 horsepower (U.S.).

The plant factor is the ratio of the average generation load on the plant (for the time period under study) to the aggregate rating of all the generation equipment installed in the plant. The result is stated as a ratio of energy:

\[
\text{Plant factor} = \frac{\text{average plant output}}{\text{plant capacity}}
\]

Number of Units

The capital cost per kilowatt for a hydroelectric powerplant of a given capacity generally decreases with a fewer number of units.

Multiunit plants can efficiently meet large variations of load by varying the number of units in service to assure operation in the high-efficiency range. However, modern systems with many powerplants can usually adjust the load distribution between plants and avoid low-efficiency operation of any single plant, thus making even a single-unit plant an efficient producer.

Turbine full-gate capacity varies approximately as the head to the three-halves power, as shown by the performance curves on figure 7. A Francis-type unit operating at 65 percent of its design head will develop approximately 45 percent of its design power.

Generally, a two- to four-unit plant is sufficiently adaptable to the usual load and flow variations, and the first cost is very close to a minimum. The service bay and plant equipment such as cranes, air compressors, oil handling equipment, etc., are smaller and can be used more efficiently. Single-unit plants have lower operating and maintenance costs because there are fewer machines to service, but the service equipment must be larger and is more expensive.

The number of units can best be determined by a careful weighing of the foregoing limitations and criteria, rather than by following a fixed rule.

Unit Capacity

Generally, the cost per kilowatt for generator, turbine, governor, and transformers decreases with an increase in unit size. However, this may be offset by smaller service equipment and cheaper foundation for the small units. The result is a tendency toward a fewer number of units for a given plant.

A runner with a maximum overall diameter of 18 feet (5.5 metres) is about the largest that can be shipped by rail across this country. Large installations require construction in segments for field assembly.

Operation at small gate openings results in low efficiency and accelerated damage from cavitation. Operation is preferably restricted to a power range having an efficiency of 80 percent or more. Figure 8 shows the change in shape of the efficiency curve with specific speed, from a unit of \( n_s = 35 \) (156 metric), and a minimum capacity of 25 percent at an efficiency of 80 percent, to a unit of \( n_s = 143 \) (636 metric), with a minimum capacity of 75 percent.

The typical Francis turbine full-gate performance characteristics when operating at constant speed such as in hydroelectric service are shown
GENERAL REQUIREMENTS

ADJUSTABLE PROPELLER TURBINE—$n_b = 70$ TO $225$ (300 TO 1000 METRIC)
Design head = 10 to 200 feet (3 to 60 metres)

FIXED PROPELLER TURBINE—$n_b = 70$ TO $225$ (300 TO 1000 METRIC)
Design head = 10 to 200 feet (3 to 60 metres)

FRANCIS TURBINE—$n_b = 15$ TO $100$ (65 TO 445 METRIC)
Design head = 30 to 2000 feet (10 to 600 metres)

Figure 7.—Turbine performance curves at full gate—maximum head range.
on figure 9. The full-gate power varies with the head to the three-halves power and the discharge varies with the square root of the head above the pumping head of the runner. This pumping head is the back pressure developed by the submerged spinning runner when the unit is motoring. This ranges from 25 percent at the lower heads to 40 percent of the design head at the high heads. The power required to motor the unit at rotational speed varies from 15 percent of the full-gate power at design head for the low head units to 5 percent for the high head units. Within the permissible range of head from 65 to 125 percent of the design head, the available power of a typical Francis unit will vary from approximately 45 percent to 140 percent of the design power unless limited by the capacity of the attached generator.

Full-gate capacity usually is selected by the turbine designer at a point slightly less than the position where additional gate opening produces no further increase in power, or where unstable operation starts. This normally results in a full-gate efficiency of approximately 87.5 percent. The peak efficiency, where the flow velocity and direction are ideal with respect to the runner, usually is found at some point less than full gate. On figure 10, this point is at 80 percent of full-gate power at design head.

![Hydraulic turbine characteristic efficiency curves.](image)
Figure 9.—Typical Francis turbine performance—constant speed and full gate.

Figure 10.—Efficiency curves—Francis turbine.
Unit Selection

Bureau specifications require that the manufacturer be responsible for the mechanical design and hydraulic efficiency of the turbine. The Bureau objective in preparing designs and specifications is to obtain a turbine that will result in the most economical combination of turbine, related water passages, and structures. Since Bureau purchases are made under competitive bidding, the least expensive turbine that will meet specification requirements is furnished. In evaluating the efficiency of a proposed turbine, the performance is estimated on the basis of experience rather than theoretical turbine design.

To develop a given power at a specified head for the lowest possible first cost, the turbine and generator unit should have the highest speed practicable. However, the speed may be limited by mechanical design, cavitation tendency, vibration, drop in peak efficiency, or loss of overall efficiency because the best efficiency range of the power efficiency curve is narrowed. In addition greater speed requires the turbine to be placed lower with respect to the tailwater, which generally increases excavation and structural costs. The greater speed also reduces the head range under which the turbine will satisfactorily operate.

The selection of speed and setting described in the following section is satisfactory for conditions normally found at most sites and will usually result in a balance of factors that will produce power at the least cost.

Speed

The specific speed of a turbine is the speed in revolutions per minute at which the given turbine would rotate, if reduced homologically in size, so that it would develop 1 horsepower under 1 foot of head at full gate. Low specific speeds are associated with high heads and high specific speeds are associated with low heads. Moreover, there is a wide range of specific speeds which may be suitable for a given head.

The specific speed range for the selected design head is shown on figure 11. Selection of a high specific speed for a given head will result in a smaller turbine and generator, with savings in capital cost. However, the turbine will have to be placed lower, for which the cost may offset the savings. Also, lower efficiency may be expected (fig. 8). The values of electrical energy, plant factor, interest rate, and period of analysis enter into the selection of an economic specific speed.
SELECTING HYDRAULIC REACTION TURBINES

1. Trial specific speed, \( n' \):
Select from figure 11 or from economic analysis. Except for unusual circumstances, the Bureau of Reclamation is currently selecting specific speeds near 950/\( \sqrt{h_d} \) (2334/\( \sqrt{h_d} \) metric).

2. Trial speed, \( n' \):

\[
n' = \frac{n'_d (h_d)^{5/4}}{(P_d)^{1/2}} \quad \text{or} \quad \frac{n'_d h_d}{(P_d)^{1/2} (h_d)^{1/2}}
\]

where

\( n' \) = trial rotational speed,
\( n'_d \) = trial specific speed,
\( h_d \) = design head, and
\( P_d \) = turbine full-gate capacity at \( h_d \).

3. Rotational speed or design speed, \( n \):
The rotational speed nearest the design speed is selected subject to the following considerations:

a. A multiple of four poles is preferred, but standard generators are available in some multiples of two poles.
b. If the head is expected to vary less than 10 percent from design head, the next greater speed may be chosen. A head varying in excess of 10 percent from design head suggests the next lower speed.

Rotational speed, \( n \) = \( \frac{120 \cdot \text{frequency}}{\text{number of poles}} \)

\( n = \frac{7200}{\text{number of poles}} \) at 60 Hz.

4. Design specific speed, \( n_d \):

\[n_d = \frac{n(P_d)^{1/2}}{(h_d)^{5/4}} \quad \text{or} \quad \frac{n (P_d)^{1/2} h_d}{(h_d)^{1/2}}\]

The design specific speed is the basic parameter to which most other factors of this selection are plotted.

**Turbine Runner Size**

The actual prototype runner size is determined by the manufacturer in accordance with model tests and design criteria. Turbines by different manufacturers vary slightly in discharge diameters for a given power even when the speed is fixed. For estimating purposes and for preliminary layouts, a diameter slightly greater than the average diameters of units already installed, as determined from the experience curve (fig. 12), should prove satisfactory.

\[
D_1 = \text{entrance diameter of runner on distributor centerline}
\]

\[
D_2 = \text{minimum opening diameter of runner}
\]

\[
D_3 = \text{discharge diameter of runner}
\]

\( n_d \) = design specific speed,
\( h_d \) = design head,
\( n \) = rotational speed,
\( \phi_s \) = velocity ratio at \( D_3 \).

Francis turbine:
\[\phi_s = 0.057 \ (n_d)^{2/3} \quad \text{(U.S.)} = 0.0211 \ (n_d)^{2/3} \quad \text{(metric)}\]

Propeller turbine:
\[\phi_s = 0.063 \ (n_d)^{2/3} \quad \text{(U.S.)} = 0.0233 \ (n_d)^{2/3} \quad \text{(metric)}\]

The discharge diameter of either type is given by:
\[
D_3 = \frac{153\phi_s (h_d)^{1/2}}{n} \quad \text{(U.S.)} = \frac{84.47\phi_s (h_d)^{1/2}}{n} \quad \text{(metric)}
\]

The average velocity of the water through \( D_3 \) at full gate may range from 20 feet per second (6 m/s) to 32 feet per second (9.8 m/s).

The hub of of the propeller turbine will be approximately 35 percent of the throat diameter, or about 12 percent of the gross area.

**Turbine Weight**

Large turbine runners are made of either stainless cast steel or carbon steel with stainless overlay to withstand cavitation. Small sizes are usually of cast bronze or cast aluminum bronze to resist cavitation. Replaceable wearing rings are provided at the band and crown. Estimation of approximate weights of Francis and propeller runners are shown on the experience curves of figure 13.

The approximate weight of a Francis turbine can be estimated by the experience curve shown on figure 14.

**Shaft Size**

Turbine main shafts are made of forged carbon or alloy steel which has been properly heat treated. They may be a single forging or they may be of multiple forged components, which are provided with flanged couplings. Shafts more than 15 inches
APPROXIMATE TURBINE SPEED

ROTATIONAL TURBINE SPEED

DESIGN SPECIFIC SPEED

Trial speed \( n' = \frac{n_s h_d^{5/4}}{P_d^{1/2}} \)

Number of poles in Generator \( = 120 \) (A.C. Frequency)

Adjust number of poles to be an even number, and for large machines, a number divisible by 4. For 50 Hz frequency avoid selection of 54 and 108 poles.

Rotational speed \( n = \frac{120 \text{ (A.C. Frequency)}}{P_d^{1/2}} \)

Adjusted number of poles

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<tr>
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<th>DATA SOURCE</th>
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LETTER SYMBOLS

- \( h_d \) : Design head, ft(m)
- \( n' \) : Trial speed, r/min
- \( n_s \) : Trial specific speed
- \( h_r \) : Rated head, ft(m)
- \( n' \) : Rotational speed, r/min
- \( n_s \) : Design specific speed
- \( P_d \) : Turbine full-gate capacity at design head, hp
- \( P_r \) : Turbine full-gate capacity at rated head, hp

FIGURE 11.—Turbine characteristics—experience curves for determination of turbine speed.
SELECTING HYDRAULIC REACTION TURBINES

LETTER SYMBOLS

- \( D_1 \): Entrance diameter of runner, ft (m)
- \( D_2 \): Minimum diameter of runner, ft (m)
- \( D_m \): Maximum outside diameter of runner, ft (m)
- \( H_d \): Design head, ft (m)
- \( n \): Rotational speed, r/min
- \( n_s \): Specific speed
- \( \beta_r \): Velocity ratio
- \( g \): Gravitational constant (acceleration), ft/s² (m/s²)

NOTE:

\[ \beta_r = \frac{n s D_m}{60 \sqrt{2 g h_d}} \]

Figure 12.—Hydraulic turbines—basic runner proportions.
Figure 13.—Hydraulic turbine runner weight—experience curve.
SELECTING HYDRAULIC REACTION TURBINES

TURBINE WEIGHT - W

KILOGRAMS

POUNDS

RUNNER MAXIMUM DIAMETER - D_max

FEET

METRES

FIGURE 14. — Hydraulic turbine weight—experience curve.
(0.4 m) in diameter are hollow bored. The size of a forging is limited by the capacity of available equipment for heating, handling, and forging.

The diameter of the shaft can be estimated from the formula:

\[ \text{Shaft diameter} = \left( \frac{70 \, P_d}{n} \right)^{1/3} \text{ inches} \]

where

- \( P_d \) = turbine full-gate capacity at \( h_d \), and
- \( n \) = rotational speed, design.

The flange dimensions can be calculated from the approximate proportions:

- Flange diameter = 1.75 \times \text{shaft diameter},
- and for shafts not subject to bending at the coupling
- Flange thickness = 0.20 \times \text{shaft diameter}.

**Turbine Spiral Case**

The spiral case is used for units with heads exceeding 100 feet (30 m). The manufacturer selects the detail dimensions in accordance with his own design criteria and with specification restrictions on spiral case velocity or penstock size. The curves on figure 15 give dimensions seldom exceeded. The preliminary dimensions should be checked against the following conditions:

a. The water velocity in the casing, when measured tangentially, should be 22 percent of spouting velocity at the design head, but in no case be more than 35 feet per second (10.7 m/s).

\[ V_e = 0.22 \left(2g h_d\right)^{1/2} \]

b. That the entrance diameter preferably be less than but may equal the penstock diameter.

c. That the tangential velocity in successive sections remain constant.

A turbine with an outside gate-operating ring, common in small units, will require a larger case in plan than shown by the curves.

**Semispiral Case**

Medium and large sized units with heads of less than 100 feet (30 m), in plants adjacent to and forming part of the dam, will usually prove less costly if provided with a semispiral case formed in concrete. The general overall size and shape of this case directly affects the physical size and general arrangement of the plant, and can be approximated in advance by using figure 16. Detailed dimensions of the final design are provided by the turbine manufacturer as the shape has a direct effect on turbine efficiency.

The basic criteria governing the design of the case on figure 16 are as follows:

1. The velocity (V) at the entrance to the semispiral case, just upstream from the stay-vane foundation cone, should be 14 percent of the spouting velocity at design head, but in no case should \( V \) be less than 5 feet per second (1.5 m/s).

2. Water passage sections should approach a square to minimize friction. The height of the entrance section should be approximately one-third of the width of the intake. All interior corners should have 12-inch (0.3-m) fillets to minimize spiral eddies.

3. The baffle vane should be in a downstream quadrant approximately 45° from the transverse centerline. To induce equal distribution of flow in the entrance, the turbine center is offset from the centerline of the intake passage. This offset will place one-third of the stay-ring intake opening in one-third of the area of the entrance. The remaining two-thirds of the flow enters the stay ring directly and through the large semispiral passage.

4. The large semispiral is designed for uniform angular velocity (\( V_a \)) of flow around the spiral.

\[ V_a = q/a \] and is a constant in which \( q \) is diminished in proportion to the remaining stay-ring arc and \( a \) is the radial cross-sectional area.

5. The entrance wall upstream from the large semispiral section should lie in the same vertical plane with the corresponding sidewall of the draft tube for structural economy.

6. The intake is laid out as a single rectangular intake without intermediate piers, with
I. Dimensions Dq, K, M, N, and O are based on experience with approximately 60 turbines purchased between 1925 and 1970.

2. Spiral case dimensions are computed using $n_s = 950 \sqrt{H_d}$ for heads above 90 feet and $n_s = 1100 \sqrt{H_d}$ for heads below 90 feet and are based on the full-gate velocity at design head being 0.22 $\sqrt{gH_d}$ but not exceeding 35 feet per second.

3. Dimensions derived from this chart are considered adequate for preliminary plant layout and estimating purposes. Except where necessary to predetermine offset dimension A and/or inlet diameter B, controlling dimensions should be obtained from potential suppliers prior to any final layout.

**Figure 15.** Hydraulic turbine spiral case dimensions—basic proportions.
UNIT SELECTION

FLOW

Plane of
Trash racks
Metal nose and tail piece as required

Pier tells warped inward to provide equal discharge areas

Gate slots located as required

START slope at intersection of cone with roof and floor

HEAT AT OPENING LINE OF ENTRANCE 12.00

UNIFORM HEIGHT 6150

OUTER limits of draft tube

TRANVERSE UNIT 8202

INLET Angle at start, bottom 45 degrees

INNER Angle at end, bottom 10 degrees

Diameter cone

Note line development along outside of large spiral, fillets not shown

RADIAL SECTION 0° TO 130°
LARGE SPIRAL

SECTION 0° TO 130°
SMALL SPIRAL

ENLARGED DETAIL OF BAFTEL VANE

These dimensions are approximate. May be subject to minor changes.

SECTIONS A-A

Select R to form curves tangent to surfaces of baffle zone

120°

Notes:
All dimensions are proportional to outside dimensions of stay vane.
Stay vane 0° approximately 1.67 x 0.3.
115° R at O equals one half draft tube width, which is assumed to be 3,500 ft.
Roof and floor of spirals are horizontal on radial sections and symmetrical about the centerline of distributor.

Figure 16.—Hydraulic turbine semispiral case—typical proportions.
a 0.7 coefficient of contraction from intake opening to casing entrance. The change of area is similar to the change found in a jet issuing from an orifice. The intake opening should have a minimum depth of water above it equivalent to 0.3 the height of the opening to avoid entraining air and to assure uniform vertical distribution of flow in the entrance section of the case.

7. Intermediate piers, which may be necessary as buttresses or to reduce gate or bulk-head spans, are positioned and shaped to give identical entrance and exit areas to the resultant water passages, thus inducing equally divided flow.

8. The intermediate piers should stop short of contact with the stay ring to avoid excessive side thrust on the runner. This side thrust would occur when the shear pins of the wicket gates have failed or the gates have otherwise become inoperative, and it becomes necessary to lower the head gates one at a time in order to stop the unit. The trailing edges of the piers should be placed sufficiently upstream to provide interpassage access and complete drainage through one drain opening.

Figure 16 shows a basic layout of a semispiral case based on these criteria, with dimensions proportioned to the stay-ring outside diameter. This diameter may vary in its proportion to the runner discharge diameter in accordance with curve 0 on figure 15. The design of figure 16 is based on a stay-vane outside diameter 167 percent of the runner discharge diameter. This semispiral case design, combined with the basic draft tube illustrated on figure 18, produces the radius of 1.115 (expressed in terms of the outside diameter of stay vanes) at 0° of the large semispiral. Any change of these proportions will change the 0° dimension and, therefore, the constants of the equation.

\[ p = A - K \theta^2 \]

where

\[ p = \text{distance from center of unit to neat line of the large semispiral,} \]
\[ A = \rho \text{ at } 0^\circ \text{ or } 1.115 \text{ in basic layout,} \]
\[ K = \frac{A - (\rho \text{ at } \theta)}{\theta^2} \]
\[ \theta = \text{angle of turn expressed in radians, at } 135^\circ \text{ in the basic layout; } \rho = 0.5, \text{ and} \]
\[ \theta = 2.356, \text{ giving} \]
\[ K = 0.1108 \]

The spiral is modified in the vicinity of the baffle to form a tangent to the surface of the baffle vane in the stay ring.

**Recommended Draft Head**

The recommended draft head \( Z \) established by the experience curve (fig. 17) of the cavitation factor \( \sigma \) has been plotted to place the turbine 1 foot (0.3 m) lower than the elevation at which cavitation damage and loss of performance have approached unacceptable values. The 1-foot (0.3 m) margin allows for variation of atmospheric pressure and minor variations in runner characteristics. See also figure 18.

\[ H_b = H_s - H_v = \text{atmospheric pressure minus vapor pressure in feet (metres) of water.} \]
\[ h_{cr} = \text{maximum head in feet (metres) at which turbine may be operated at full gate considering overload capability of generator.} \]
\[ \sigma = \text{cavitation factor (Thoma)—from curve or from model if available.} \]
\[ b = \text{vertical distance from centerline of distributor or case, to} \]
\[ D_2 \text{ or minimum runner diameter in feet (metres).} \]
\[ H_s = \text{static draft head in feet (metres).} \]
\[ Z = \text{total draft head in feet (metres).} \]
\[ \sigma = \frac{(n_0)^{1.64}}{4325} \text{(U.S.)} \]
\[ \sigma = \frac{(n_0)^{1.64}}{50 \times 327} \text{(metric)} \]
\[ H_s = H_b - v h_{cr} \]
\[ Z = H_s + b \]

Elevation of centerline of distributor or case is \( Z \) feet above the tailwater elevation. For a one- or two-unit plant, the tailwater elevation considered should be the minimum likely to exist with one unit operating at full gate. For a plant having
UNIT SELECTION

ATMOSPHERIC PRESSURE

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<tr>
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<tr>
<td>4500</td>
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$H_a$ = Atmospheric pressure for altitude, ft (m).

$H_w$ = Vapor pressure of water, use highest expected temperature, ft (m).

$H_b$ = $H_a - H_w$. Atmospheric pressure minus vapor pressure, ft (m).

WATER PROPERTIES

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<th>$H_a$</th>
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FRANCIS - PROPELLER

$D_a = \text{at shroud}=\text{Least diameter through shroud, ft (m).}$

$D_o = \text{Discharge diameter of runner, ft (m).}$

$b = \text{Distance from } D_o \text{ to } E \text{ of distributor, ft (m).}$

(estimated from curve % $D_o$ vs $n_s$).

$n_s = \text{Specific speed of turbine.}$

MINIMUM TAILWATER SURFACE

$D_b = \text{Distributor} - \text{Minimum tailwater} = H_5 + b$ (Total draft head), ft (m).

Note: Place % of distributor at next lowest full foot elevation, (0.30 m).

FIGURE 17.—Recommended total draft head—reaction turbines.
several units, it is not likely that only one unit would be operating overloaded while all other units are shut down. So, the tailwater most likely to exist is that which corresponds to nearly all units operating before any one unit is overloaded.

**Draft Tube Outline**

The manufacturer considers the draft tube as part of the turbine when determining an efficiency warranty, as it is difficult to measure the net effective head or pressure at the discharge diameter of the runner (Dₙ). Therefore, the manufacturer furnishes the draft tube shape and dimensions within the limitations of the specifications and existing structure, if any. For medium or large units, the specifications usually require a well-designed elbow tube because it is more efficient than other types and requires less excavation. For preliminary plant layouts, the proportions shown on figure 18 usually will be acceptable for overall depth, width, and length, but may require revision in other details. The piers are for structural purposes only.

The discharge diameter (Dₙ) of a runner for a low specific speed Francis turbine may be slightly larger than the minimum throat diameter of the water outlet passage from the runner. In this monograph, it is assumed that Dₙ is located at the minimum draft tube throat diameter and that the dimension N on figure 18 represents the distance from the centerline of the distributor to the location of Dₙ. This, then, also becomes the location of the beginning of the draft tube.

The plate steel draft tube liner extends from the runner discharge at Dₙ to the following points:

1. To the point at which the draft tube area is twice the runner discharge area (at Dₙ) for all propeller-type turbines and for Francis turbines if the design head is less than 350 feet (107 m).
2. To include the full elbow with pier nose if the design head exceeds 350 feet (107 m).

Pier noses are armored by plate steel, and the superimposed load is carried by the concrete where the span is small and the load light, but steel noses are used to carry the superimposed loads when necessary.

To meet construction schedules or to permit construction of the substructure before delivery of the turbines, the manufacturer may be required to design the elbow to fit the horizontal flare portion of the draft tube. Detail dimensions of tubes found to provide satisfactory performance are given on figures 19, 20, and 21.

**Governor Capacity**

The size, type, and cost of governors vary with their capacity to perform work which is measured in foot-pounds (metre-kilograms). Governors having a capacity of more than 60,000 foot-pounds (8295 m·kg) usually are of the actuator or cabinet-actuator type. Those having a capacity less than 50,000 foot-pounds (6913 m·kg) usually are of the gate-shaft type. Between 50,000 and 60,000 foot-pounds capacity, they may be of either type as specified.

The capacity is the product of the following factors: turbine gates servomotor area, governor minimum rated oil pressure, and turbine gates servomotor stroke. For gate shaft governors, the turbine gates servomotor area is the net area obtained by subtracting the piston rod area from the gross piston area. For governors controlling two servomotors mounted directly on the turbine, the effective area is the sum of the net area of the two servomotors.

Servomotor capacities are plotted on the experience curve, figure 22, and can be estimated by the formulas:

1. Wicket gates servomotor capacity.
   
   \[ \text{FY}_M = 4.23 \left( \frac{h_{w_h} \cdot D_g \cdot M}{1.14} \right) \] (U.S.)
   \[ \text{FY}_M = 34 \left( \frac{h_{w_h} \cdot D_g \cdot M}{1.14} \right) \] (metric)

   where
   
   \( M \) — wicket gate height,
   \( h_{w_h} \) — maximum head, including water hammer, and
   \( D_g \) — wicket gate circle diameter.

2. Blade servomotor capacity (adjustable blade propeller turbine).—The blade servomotor capacity also varies among manufacturers. This can be roughly estimated by the formula:

   \[ \text{FY}_b = \frac{16.47}{(H_{max})^{1/2}} \left( \frac{P_{max}}{(n_s)^{1/4}} \right) \] (U.S.)
   \[ \text{FY}_b = \frac{6.17}{(H_{max})^{1/2}} \left( \frac{P_{max}}{(n_s)^{1/4}} \right) \] (metric)

   where
   
   \( H_{max} \) — maximum head,
   \( n_s \) — design specific speed, and
   \( P_{max} \) — turbine full-gate capacity at \( H_{max} \).
UNIT SELECTION

Section beyond pier nose may have a maximum floor slope of 10%, while maintaining similar areas.

PROJECTED PLAN

Section beyond pier nose may have a maximum floor slope of 10%, while maintaining similar areas.

PROJECTED PLAN

PROJECTED PLAN

PROJECTED PLAN

PROJECTED PLAN

CONICAL TUBE PROPORTIONS

Preferred: \( R_0 = D_3 \cdot L = 4 \cdot D_3 \)

\( D_3 \cdot V_e = 35 \text{ ft/s}(10.5 \text{ m/s}) \cdot R_0 \cdot V_e = 7.5 \text{ ft/s}(2.3 \text{ m/s}) \)

L and \( R_0 \) may vary as required by any specific installation.

ELEVATION

Beyond this point passage may have upward slope of a maximum of 10:1.

CONICAL DRAFT TUBE

All dimensions of discharge passage proportional to \( D_3 \).

\( D_3 \cdot 2 \text{ ft}(0.61 \text{ m}) \) if not less; if draft tube requires excess excavations, if not \( D_3 \cdot 4 \text{ ft}(1.22 \text{ m}) \) or less.

\( R_0 \) discharge radius of conical tube.

Figure 18.—Turbine draft tubes—preliminary basic outlines.
NOTES (To contractor)
Contractor shall interpolate between sections as necessary for intermediate rib outlines.
Contractor shall check rib assembly before applying sheathing, using spline, and adjust where required to produce smooth continuous surface.

DEVELOPED HALF PLAN ON % OF TUBE

DESIGN NOTES
All dimensions given as ratio of \( D_3 \).
Maximum upward slope of floor of draft tube is 1 in 10.
Pier nose to be made of metal and capable of carrying loads imposed by the structure.

SECTION ON % OF TUBE

PREFERRED RATIOS

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area at top of tube</td>
<td>0.381</td>
</tr>
<tr>
<td>Net area at pier nose</td>
<td>0.660 (max)</td>
</tr>
<tr>
<td>Net area at discharge</td>
<td>4.769</td>
</tr>
<tr>
<td>Net width at pier nose</td>
<td>1.400</td>
</tr>
<tr>
<td>Height of pier nose</td>
<td>2.500 (min)</td>
</tr>
<tr>
<td>Height of tube</td>
<td>1.400</td>
</tr>
<tr>
<td>Dia. at top of tube</td>
<td>1.400</td>
</tr>
<tr>
<td>Dia. at top of tube</td>
<td>1.400</td>
</tr>
<tr>
<td>Length along % of tube (top to pier nose)</td>
<td>2.967 (min)</td>
</tr>
<tr>
<td>Dia. at top of tube</td>
<td>1.400</td>
</tr>
<tr>
<td>Horizontal distance % of turbine to discharge</td>
<td>3.80 (min)</td>
</tr>
<tr>
<td>Dia. at top of tube</td>
<td>1.400</td>
</tr>
<tr>
<td>Developed Length ( D_3 ) to discharge</td>
<td>125 ft (38 m) (max)</td>
</tr>
</tbody>
</table>

Figure 19.—Turbine draft tubes—two piers—optimum proportions.
UNIT SELECTION

NOTES (To contractor)
Contractor shall interpolate between sections as necessary for intermediate rib outline.
Contractor shall check rib assembly before applying sheathing, using spline, and adjust where required to produce smooth continuous surface.

DEVELOPED HALF PLAN ON % OF TUBE

DESIGN NOTES
All dimensions given as ratio of D3.
Maximum upward slope of floor of draft tube is 1 in 10.
Pier nose to have metal protector.

SECTION ON % OF TUBE

METHOD OF EXTENDING LENGTH OF HORIZONTAL DIFFUSER. USE SAME RATE OF EXPANSION.
FULL GATE DISCHARGE

PREFERRED RATIOS

| Area at top of tube | 0.36 |
| Net area at pier nose | 0.660 (\text{max}) |
| Net area at discharge | 4.769 |
| Net width at pier nose | 2.500 |
| Height of pier nose | 1.400 |
| Height of tube | 2.967 (\text{min}) |
| Dia. at top of tube | 2.967 (\text{min}) |
| Dia. at top of tube | 3.80 (\text{min}) |
| Dia. at top of tube | 125 ft (38 m) (\text{max}) |

FIGURE 20. — Turbine draft tubes—one pier—optimum proportions.
3. Total servomotor capacity.—The total foot-pound (metre-kilogram) capacity required by an adjustable blade propeller turbine is the sum of the capacities of the gate ($F_{YM}$) and blade ($F_{Yb}$) servomotors, assuming that both act simultaneously.

4. Governor time—closing and opening.—Governor closing time is the minimum time in seconds for the governor system to move the gates from 100-percent gate opening down to speed-no-load gate.

Governor opening time is the minimum time in seconds for the governor system to move the gates from speed-no-load gate to 100-percent gate.

The closing and opening times may be the same or they may be different, depending on hydraulic conditions and other factors. Where closing or opening rate is tapered in approach to limits to avoid damage to the mechanism, the straight-line portion of the gate-motion curve projected to 100-percent gate and to speed-no-load gate is used for determining closing time or opening time.

5. Hydraulic thrust.—Figure 23 can be used to estimate the hydraulic thrust for both Francis and propeller turbines. The expressions on figure 23 show that the thrust force ($F_h$) increases logarithmically with the products of head, square of the diameter, and the square root of specific speed.

Figure 21.—Turbine draft tubes—one pier—underground plant design.
**LETTER SYMBOLS**

- **M** = Wicket gate height, ft (m)
- **hwh** = Maximum head, including water hammer, ft (m)
- **Dg** = Wicket gate circle diameter, ft (m)
- **FY_M** = Servomotor capacity, ft*lb (m*kg)
- **P_min** = Servomotor minimum rated oil pressure, lb/in² (kg/m²)
- **V_s** = Servomotor net volume, in³ (m³)

**NOTE:**

\[
V_s = \frac{12FY_M}{P_{min} \text{ in}^3} \quad \text{or} \quad \frac{FY_M}{P_{min} \text{ m}^3}
\]

**REACTION HYDRAULIC TURBINES**

**WICKET GATE SERVOMOTOR CAPACITY**

**FIGURE 22.**—Hydraulic turbine servomotor capacity—experience curve.
SELECTING HYDRAULIC REACTION TURBINES

THE HYDRAULIC THRUST DOES NOT INCLUDE THE WEIGHT OF ROTATING PARTS

TURBINE HYDRAULIC THRUST — \( F_h \)

<table>
<thead>
<tr>
<th>SYSTEM</th>
<th>UNITS</th>
<th>FRANCIS</th>
<th>PROPELLER</th>
</tr>
</thead>
<tbody>
<tr>
<td>U.S.</td>
<td>POUNDS</td>
<td>( 1.32 \pi n_s^{1/2} (D_s)^2 H_{\text{MAX}} )</td>
<td>( 3.80 \pi n_s^{1/2} (D_s)^2 H_{\text{MAX}} )</td>
</tr>
<tr>
<td>METRIC</td>
<td>KILOGRAMS</td>
<td>( 10.02 \pi n_s^{1/2} (D_s)^2 H_{\text{MAX}} )</td>
<td>( 28.85 \pi n_s^{1/2} (D_s)^2 H_{\text{MAX}} )</td>
</tr>
</tbody>
</table>

**LETTER SYMBOLS**

- \( D_s \) = Discharge diameter of runner, ft (m)
- \( F_h \) = Hydraulic thrust, lb (kg)
- \( H_{\text{MAX}} \) = Maximum head, ft (m)

**Figure 23.**—Hydraulic thrust—experience curve.
The high efficiency of hydroelectric units and their ability to follow rapid load changes depend not only on the design of the units, but also on the design of the associated water passages from forebay through tailrace. The overall head loss in the water passages can greatly affect the earning ability or power production. Another source of revenue loss is a nonuniform distribution of flow to the runner, which causes a loss in unit efficiency. This disturbance is usually caused by a sharp bend, by a butterfly valve immediately upstream from the entrance to the spiral casing, or by poor layout of the entrance water passages in the case of the concrete semispiral case.

Regulating Characteristics

The regulating characteristics of a unit are basically a function of the flywheel effect and the water column inertia. The flywheel effect is a stabilizing influence, while the water column is an unstabilizing influence on speed regulation.

Flywheel effect is most conveniently expressed as starting-up time of the unit,

\[ T_m = \frac{WR^2 n^2}{6.7 \cdot 10^4 P} \quad \text{(metric).} \]

where

\[ n = \text{rotational speed}, \]
\[ P = \text{turbine full-gate capacity in horsepower for the condition under consideration, and} \]
\[ WR^2 = \text{product of weight of revolving parts and the square of the radius of gyration}. \]

This is the time in seconds for torque to accelerate the rotating mass from zero to rotational speed. Together, the turbine runner in water and the generator develop the \( WR^2 \), which may be estimated from the formulas:

Turbine \( WR^2 = 23,800 \left( \frac{P_d}{n^{3/2}} \right)^{5/4} \quad \text{(U.S.)} \]

\[ = 24 \ 213 \left( \frac{P_d}{n^{3/2}} \right)^{5/4} \quad \text{(metric)} \]

Normal generator \( WR^2 = 356,000 \left( \frac{kVA}{n^{3/2}} \right)^{5/4} \quad \text{(U.S.)} \]

\[ = 15000 \left( \frac{kVA}{n^{3/2}} \right)^{5/4} \quad \text{(metric)} \]

Additional generator \( WR^2 \) may be specified if required.
SELECTING HYDRAULIC REACTION TURBINES

Water column inertia is conveniently expressed as starting-up time of water column, 
\[ T_w = \frac{\sum L V}{g h} \]

This is the time in seconds for head (h) to accelerate the flow from zero to maximum velocity (V). The products of length and velocity of every component from forebay or surge tank to tailrace should be added to obtain \( \sum L V \).

Units for which \( T_w > 2 \) (\( T_w^2 \)) can be expected to have good regulating capacity. This test should be applied over the entire operating head range. For further information on regulating characteristics, see “Governor Characteristics for Large Hydraulic Turbines” by F. R. Schleif, USBR Report REC-ERC-71-14, February 1971.

Plants in which more than one turbine are served from one penstock should be analyzed to determine proper governor settings and appropriate operating practices. Such plants may be unable to contribute to system transient speed regulation, but adverse effects upon the system may be avoided by specifying the number of units which may be allowed to operate on free governor (unblocked) at any one time.

**Speed Rise**

Speed rise upon load rejection may require special attention. Speed rise is the increase in speed from the rated speed if the generator, while operating at rated speed, is suddenly and completely disconnected from the load while the turbine is operating under governor control.

The turbine and generator are designed to withstand runaway speed, but at excessive speed severe vibrations sometimes develop which snap the shear pins of the gate mechanism. To minimize vibration, a speed rise not to exceed 60 percent can be permitted in contrast to the 35 to 45 percent desired for satisfactory regulation of independently operated units.

The speed rise may be calculated from formulas on figure 24. However, note that the purpose of figure 24 is to determine whether a surge tank, more WR, or a larger penstock would be required and, therefore, only rated conditions are considered. Speed rise also should be computed for full gate at critical head and at minimum head and for part gate at maximum head. Also, when more than one unit operates off a common penstock, the actual speed rise should be computed using the water startup time (\( T_w \)) for all units.

**Runaway Speed**

Runaway speed is the speed attained by a unit at full gate when the generator is disconnected from the system and the governor is inoperative.

Runaway speed differs among manufacturers because of variations in the design of turbines and generators. Figure 25 shows runaway speed based on model tests for turbines of varying specific speed. Field tests indicate that runaway speed is less than usually predicted, but it can be expected not to exceed the following:

\[ \frac{n_r}{n} = 0.85 (n_s)^{1/5} \]  (U.S.)
\[ \frac{n_r}{n} = 0.63 (n_s)^{1/5} \]  (metric)

\[ n_{max} = n_r \left( \frac{h_{max}}{h_d} \right)^{1/2} \]

where
- \( h_d \) = best efficiency head (design head),
- \( h_{max} \) = maximum head,
- \( n \) = rotational speed,
- \( n_{max} \) = runaway speed at maximum head,
- \( n_r \) = runaway speed at best efficiency head and full gate, and
- \( n_s \) = specific speed based on full-gate output at best efficiency head.

The adjustable blade propeller turbine has a theoretical runaway speed approaching infinity at the closed or flat blade position, but the friction and windage of the connected generator normally will limit the runaway speed to 275 percent of normal. The specifications usually include a requirement for an adjustable stop on the blade rotation which will limit the runaway speed at 275 percent to avoid excessive stresses in the generator.

Maximum speed for which the unit must be designed will be encountered at maximum head. Runaway speed should also be computed for minimum head since this value may be less than speed rise at maximum head. These data are necessary to select proper speed switch settings.
BASIS FOR DETERMINING WHEN SURGE TANKS WILL BE REQUIRED ON TURBINE PENSTOCK INSTALLATION FOR SPEED REGULATION

1. Surge tanks shall be provided where the resulting reduction in waterhammer pressures will provide a more economical turbine penstock installation.

2. A surge tank may also be provided for the condition where the computed percent of speed rise, based on the rejection of the entire rated load for a unit operating independently, cannot be reduced to about 45 percent by other practical methods.

3. For a unit which is one of several units and common penstock header system, the permissible percent of speed rise will be computed on the basis of one unit operating alone.

4. If the turbine is equipped with a water-saving type of pressure regulator, the critical regulation occurs during load additions when the pressure regulator does not act. Therefore, the permissible percent of speed rise shall be computed on the basis that the pressure regulator is inoperative.

5. For preliminary studies, speed rise may be computed by the following illustrated method. Final design may require a more accurate step-by-step process.

METHOD FOR COMPUTING SPEED RISE

Notation:
- Tr = Servomotor minimum closing time, sec.
- Pr = Turbine full-gate capacity at hr., hp
- h' = Rated head, ft
- n = Rotational speed, rpm
- Wm = Design specific speed
- WR = Flywheel effect of revolving parts, lb-ft/sec
- L = Equivalent length of water conduit, ft
- A = Equivalent area of water conduit, ft²
- g = Gravitational constant (acceleration), ft/sec²
- Qr = 62.4 hp x 0.875 = Turbine full-gate discharge, ft³/sec
- Vr = Qr
- A = Conduct water velocity for full-gate at hr., ft/sec

To obtain the speed rise for full-load rejection, determine the following values:
- (a) Tm = 0.25 + Tr, full-closing time of servomotor(s)
- (b) Tm = \( \frac{WR^2}{L} \), mechanical startup time.

Example:
- Given: Tr = 5 sec, Pr = 40,000 hp, h' = 80 ft,
- h' = 94.7 ft, WR = 53,240,000 lb-ft/sec, Vr = 13,779 ft/sec, L = 540 ft
- Tm = 5.25 + 5.25 = 10.5 sec
- Tm = 7.46 + 0.704 = 8.16 sec
- (d) Wm² = 94.7 (40,000)² = 79.2
- (e) SR = 28.1 percent from chart A.
- (f) Tw = (13,779) (53,240,000)
- (g) K = Tm = 1.80 = 0.364
- (h) SR = (28.1) (0.364) = 10.06 percent.

References:
- Memorandum by John Parmakian dated March 25, 1949, titled "A Proposed Basis for Determining When Surge Tanks Will Be Required for Speed Regulation of Turbine Penstock Installations".

Note:
The above example is for one set of variables. Calculations should also be made at other conditions to establish speed rise magnitude, i.e., for heads at hmin and hmax.
SELECTING HYDRAULIC REACTION TURBINES

\[ \frac{n_r}{n} = 0.85 n_s^{0.20} \text{ (U.S.)} \]

\[ \frac{n_r}{n} = 0.63 n_s^{0.20} \text{ (METRIC)} \]

\[ n_{\text{max}} = n_r \left( \frac{h_{\text{max}}}{h_d} \right)^{0.50} \]

SPECIFIC SPEED \(- n_s\)

\[ h_d = \text{Best efficiency head (design head), ft (m)} \]
\[ h_{\text{max}} = \text{Maximum head, ft (m)} \]
\[ n = \text{Rotational speed, r/min} \]
\[ n_{\text{max}} = \text{Runaway speed at maximum head, r/min} \]
\[ n_r = \text{Runaway speed at best efficiency head and full gate.} \]
\[ n_s = \text{Specific speed based on full gate output at best efficiency head.} \]

The curve is based on model test data.

Figure 25.—Hydraulic turbine runaway speed.
Predicted Performance Curves

Predicted performance of a unit operating over a wide range of head can best be shown by a curve of the type on figure 26. Reservoir power studies may assume operation of a unit at full load for the percent of time necessary to pass the water rather than part-load operation 100 percent of the time. This procedure reflects somewhat less than maximum efficiency and more closely approximates operating conditions.

The hydraulic turbine data sheet of Grand Coulee Third Powerplant (fig. 28) is an example of the type of curve prepared for use in detailed power studies and design of such auxiliaries as the penstock, forebay, and tailrace. The preliminary curve prepared for reservoir operation studies before award of turbine contract should have the efficiency curve shifted bodily to peak at 90 percent turbine efficiency with a corresponding shift in the discharge curve. The curves shown on figure 28 were prepared after award of contract from the manufacturer’s predicted performance.

Curves of this type should check at all heads with curves computed by the following formulas in U.S. customary units:

\[ P = 0.0846 \, Q \, H \eta_g \eta_t, \text{ kilowatts}, \]

\[ P = 0.1134 \, Q \, H \eta_t, \text{ horsepower}, \]

where

- \( P \) = generator output and turbine output, respectively,
- \( Q \) = discharge at \( H \),
- \( H \) = head,
- \( \eta_g \) = efficiency of generator, and
- \( \eta_t \) = efficiency of turbine.

The performance curves for a Francis turbine can be closely approximated by use of the curves on figures 31 to 37, which represent the average of many manufacturers’ curves. Actually, the slope of the power output curve varies with the specific
SELECTING HYDRAULIC REACTION TURBINES

Figure 27.—Hydraulic turbine data—Flatiron Powerplant.

Figure 28.—Hydraulic turbine data—Grand Coulee Third Powerplant.
speed, as indicated by figure 30, but the curves can be used without correction for all but the most detailed studies.

The adjustable blade propeller turbine performance curves, figures 38 and 39, are plotted for the most efficient setting of gate and blade and are therefore plotted without an indication of the gate opening.

The fixed blade propeller unit has a much narrower efficiency peak, as shown on figure 8, and is usually restricted to operation above 75-percent gate.

Preliminary turbine data sheets similar to figures 27, 28, and 29, prepared from performance curve data or from predicted performance data furnished by the manufacturer, and dimensional data furnished by the unit selection portion of this outline, are useful in making detailed reservoir operation and preliminary power production studies and in the preparation of plant layout drawings for specification purposes.

**Hydraulic Similarity**

Hydraulic machinery is considered to be homologous when the ratio of the dimensions in all directions is the same, or when the corresponding characteristic angles are the same.

Homologous hydraulic machines also have hydraulic similarity. If the discharge, power, speed, and efficiency of a turbine runner of a given diameter are known for a given head, the discharge, power, and speed for a homologous runner of a different diameter, under a different head, for the same efficiency, may be calculated directly from the following equations:

1. Homologous equations:

For constant diameter: \[ Q_2 = \left( \frac{H_2}{H_1} \right)^{1/2} Q_1 \]
\[ P_2 = \left( \frac{H_2}{H_1} \right)^{3/2} P_1 \]
\[ n_2 = \left( \frac{H_1}{H_2} \right)^{1/2} n_1 \]

For constant head:
\[ Q_2 = \left( \frac{D_2}{D_1} \right)^2 Q_1 \]
\[ P_2 = \left( \frac{D_2}{D_1} \right)^2 P_1 \]
\[ n_2 = \frac{D_1}{D_2} n_1 \]

where, for different conditions:

- \( Q_1 \) and \( Q_2 \) = turbine discharge,
- \( P_1 \) and \( P_2 \) = turbine power output,
- \( n_1 \) and \( n_2 \) = rotational speed,
- \( D_1 \) and \( D_2 \) = runner diameter, and
- \( H_1 \) and \( H_2 \) = head.
The above equations are accurate for most purposes. However, the assumption that the efficiency will be the same for different sized machines is not completely correct. Differences in friction losses due to surface roughness and length of the water passages, and slight variations from true geometric similarity, would require adjustment in efficiency between otherwise apparently homologous machines. The larger machine will have the higher efficiency.
Figure 31.—Francis turbine performance—head vs. power—\( n_s = 22 \) (98 metric).
FIGURE 32.—Francis turbine performance—head vs. power—\( n_r = 48 \) to 75 (214 to 334 metric).
FIGURE 33.—Francis turbine performance—head vs. discharge—n, = 48 to 75 (314 to 334 metric).
Figure 34.—Francis turbine performance—head vs. power—n = 40 to 48 (178 to 214 metric).
Figure 35.—Francis turbine performance—head vs. discharge—$n_s = 40$ to $48$ (178 to 214 metric).
Figure 36.—Francis turbine performance—head vs. power—n_s = 25 to 40 (111 to 178 metric).
Figure 37.—Francis turbine performance—head vs. discharge—$n_r = 25$ to 40 (111 to 178 metric).
Figure 38.—Adjustable blade turbine—head vs. power—$n_s = 142$ (632 metric).
Figure 39.—Adjustable blade turbine—head vs. discharge—$n_r = 142$ (632 metric).
2. Efficiency step-up.—The efficiency step-up is illustrated on Figure 40. A generally accepted formula for adjusting the efficiency between two different sized homologous Francis turbines, such as a prototype and a model turbine, is:

\[
\frac{1 - \eta_p}{1 - \eta_m} = \left( \frac{D_m}{D_p} \right)^{1/5} \quad \text{(Moody formula)}
\]

or

\[
\Delta \eta = (1 - \eta_m) \left[ 1 - \left( \frac{D_m}{D_p} \right)^{1/5} \right]
\]

where

- \( \eta_p \) = prototype turbine efficiency,
- \( \eta_m \) = model turbine efficiency,
- \( D_p \) = prototype turbine runner diameter,
- \( D_m \) = model turbine runner diameter, and
- \( \Delta \eta \) = increase in efficiency.

It should be noted that the step-up in efficiency for scale effect, as adjusted by the Moody formula above, is strictly applicable to the point of best efficiency. For practical purposes, it is generally assumed that the \( \Delta \eta \) calculated for the point of best efficiency is applicable at all heads and wicket gate openings.
Units of Power Measurement

When U.S. customary units are used:
\[ P_d = 0.1134 \cdot Q_d \cdot h_d \cdot \eta_d \]

where
- \( P_d \) = turbine output, horsepower (1 hp = 550 ft\( \cdot \)lb/s),
- \( Q_d \) = full-gate discharge (at \( h_d \)), ft\(^3\)/s,
- \( h_d \) = design head, feet, and
- \( \eta_d \) = design efficiency, percent.

When metric units are used, power will be expressed in metric horsepower or kilowatts, and:
\[ P_d = 13.35 \cdot Q_d \cdot h_d \cdot \eta_d \]

where
- \( P_d \) = turbine output, horsepower (1 hp = 75 m\( \cdot \)kg/s),
- \( Q_d \) = full-gate discharge (at \( h_d \)), m\(^3\)/s,
- \( h_d \) = design head, metres, and
- \( \eta_d \) = design efficiency, percent.

Note:
- \( n_s \), metric hp units = 4.45 \( n_s \), U.S. customary hp units
- \( n_s \), kilowatt units = 3.81 \( n_s \), U.S. customary hp units
- \( n_s \), kilowatt units = 0.86 \( n_s \), metric hp units

\[ n_s = \frac{n(P_d)^{1/2}}{(h_d)^{5/4}} \]

\( n \) = rotational speed, r/min

\( \eta_d \) = design efficiency, percent