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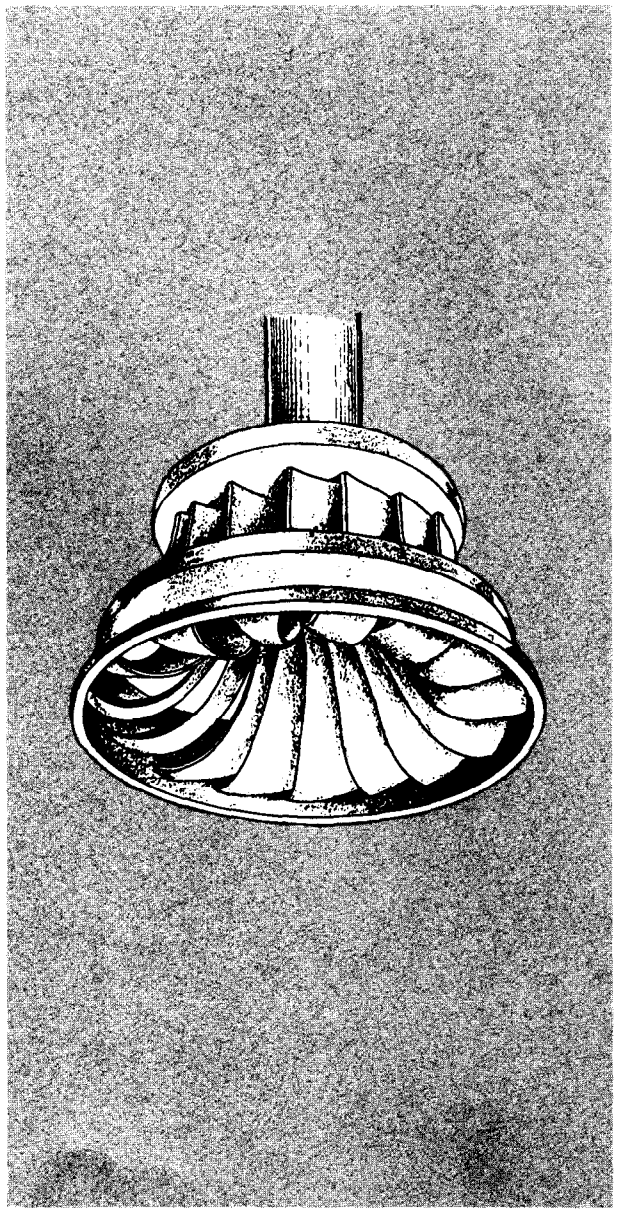

SELECTING HYDRAULIC REACTION TURBINES

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.

ENGINEERING MONOGRAPHS are published in limited editions for the technical staff of the Bureau of Reclamation and interested technical circles in Government and private agencies. Their purpose is to record developments, innovations, and progress in the engineering and scientific techniques and practices that are employed in the planning, design, construction, and operation of Reclamation structures and equipment. Copies may be obtained from the Bureau of Reclamation, Denver Federal Center, Denver, Colorado, and Washington, D.C.

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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=Dimension of ρ at 0°
a=Radial cross-sectional area of semispiral case
b=Distance from D_2 to ℄ of distributor
D_1=Entrance diameter of runner on distributor ℄
D_2=Minimum opening diameter of runner
D_3=Discharge diameter of runner
D_4=Design diameter of draft tube
D_M=Maximum outside diameter of runner
F_h=Hydraulic thrust
FY_b=Servomotor blade capacity
FY_M=Servomotor gate capacity
ft=foot
g=Gravitational constant (acceleration)
H_a=Atmospheric pressure
H_{ave}=Weighted average head
H_b=Atmospheric pressure minus vapor pressure
H_g=Gross head
H_{max}=Maximum head
H_{min}=Minimum head
H_n=Net head
H_s=Static draft head, distance from D_2 to minimum tailwater
H_v=Vapor pressure of water
Hz=A-c frequency
h_{cr}=Critical head
h_d=Design head
h_r=Rated head
hp=Horsepower
K=Constant=$(A-(\rho \text{ at } \theta))/\theta^2$
K=Factor=T_w/T_t, speed rise factor
kVA=Kilovolt-amperes capacity of generator
kW=Kilowatt power of generator
L=Length of penstock
m=Metre
n=Rotational speed, design
n'=Trial rotational speed
n_{max}=Runaway speed at H_{max}
n_r=Runaway speed
n_s=Design specific speed
n_s'=Trial specific speed
P.F.=Generator power factor
P_d=Turbine full-gate capacity at h_d
P_r=Turbine full-gate capacity at h_r
P_{min}=Servomotor minimum rated oil pressure
Q_{cr}=Turbine full-gate discharge at h_{cr}
Q_d=Turbine full-gate discharge at h_d
Q_{max}=Turbine full-gate discharge at H_{max}
Q_r=Turbine full-gate discharge at h_r
q=Quantity of flow, semispiral case
R=Radius
s=Second
r/min=Revolutions per minute
S_R=Speed rise
S_R'=Speed rise, effects of water hammer
T_t=Servomotor minimum closing time
T_k=Full closing time of governor
T_m=Mechanical startup time
T_w=Water startup time
V_a=Semispiral case uniform angular velocity
V_c=Spiral case water velocity
V_r=Conduit water velocity for full gate at h_r
V_s=Servomotor net volume
WR^2=Product of weight of revolving parts and the square of the radius of gyration
Z=Total draft head, ℄ distributor to minimum tailwater
γ=Gamma=specific weight of water
η_d=Eta$_d$=turbine design efficiency
η_g=Eta$_g$=generator efficiency
η_t=Eta$_t$=turbine efficiency
θ=Theta=angle of turn expressed in radians
π=Pi=3.14159 . . .
ρ=Rho=distance from ℄ of unit to neatline of the large semispiral
Σ=Sigma=summation of
σ=Sigma=cavitation coefficient
ϕ_3=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

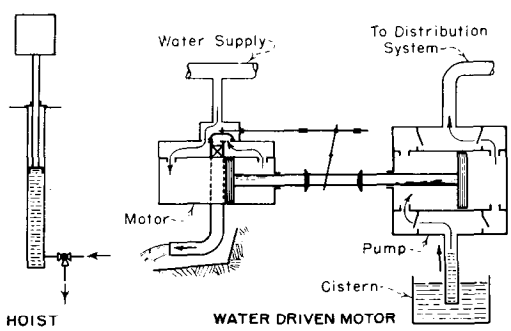

FIGURE 1.—*Displacement-type water motors.*

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.

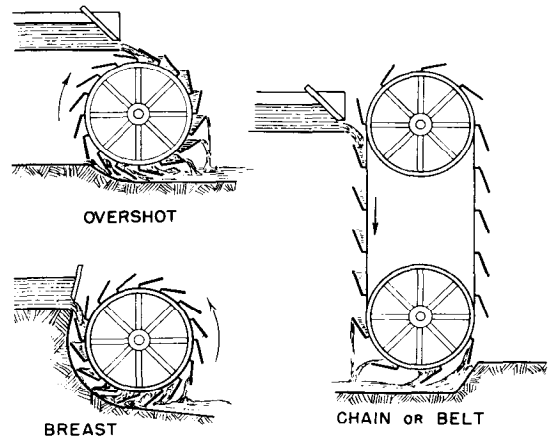

FIGURE 2.—*Gravity-type water motors.*

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.

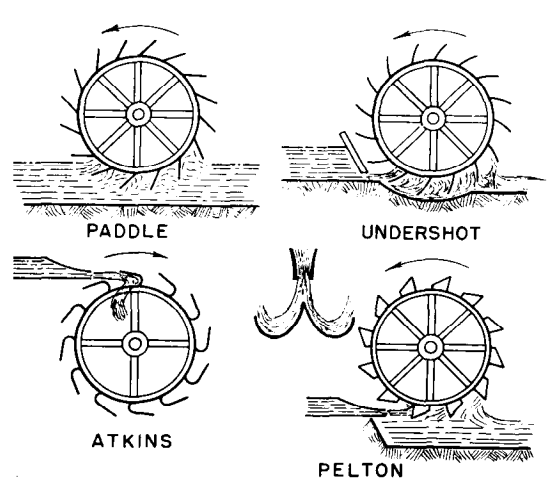

FIGURE 3.—*Impulse-type water motors.*

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

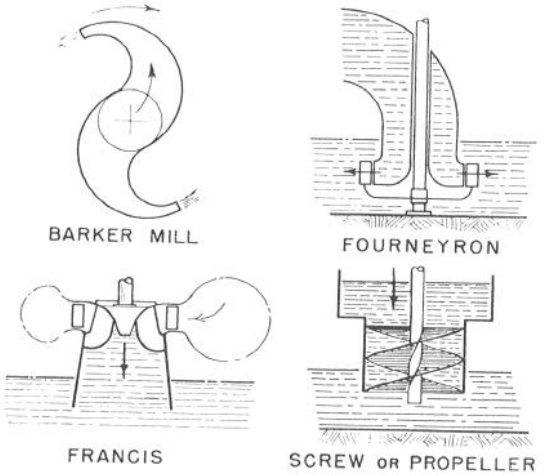

FIGURE 4.—*Reaction-type water motors.*

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.

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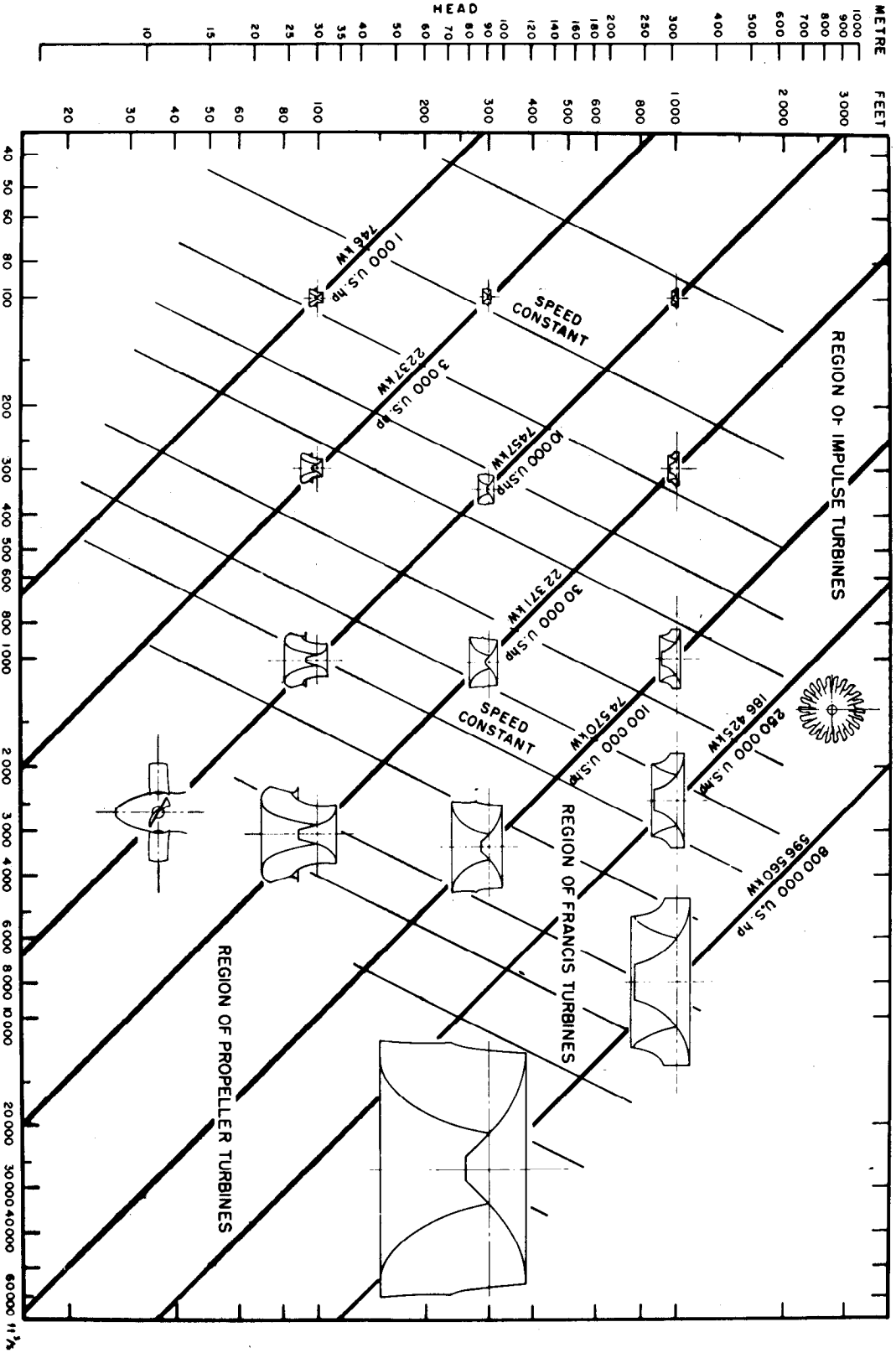


FIGURE 5.—*Application diagram for types of hydraulic turbines.*

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.

Head

A turbine operating at constant speed produces a maximum peak efficiency of 90 to 95 percent under one head and at one power output. At any other head or power output, the unit will operate at reduced efficiency.

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):

Type of turbine	Maximum head (percent)	Minimum head (percent)
Francis or reaction	125	65
Propeller—fixed blade	110	90
Propeller—adjustable blade	125	65

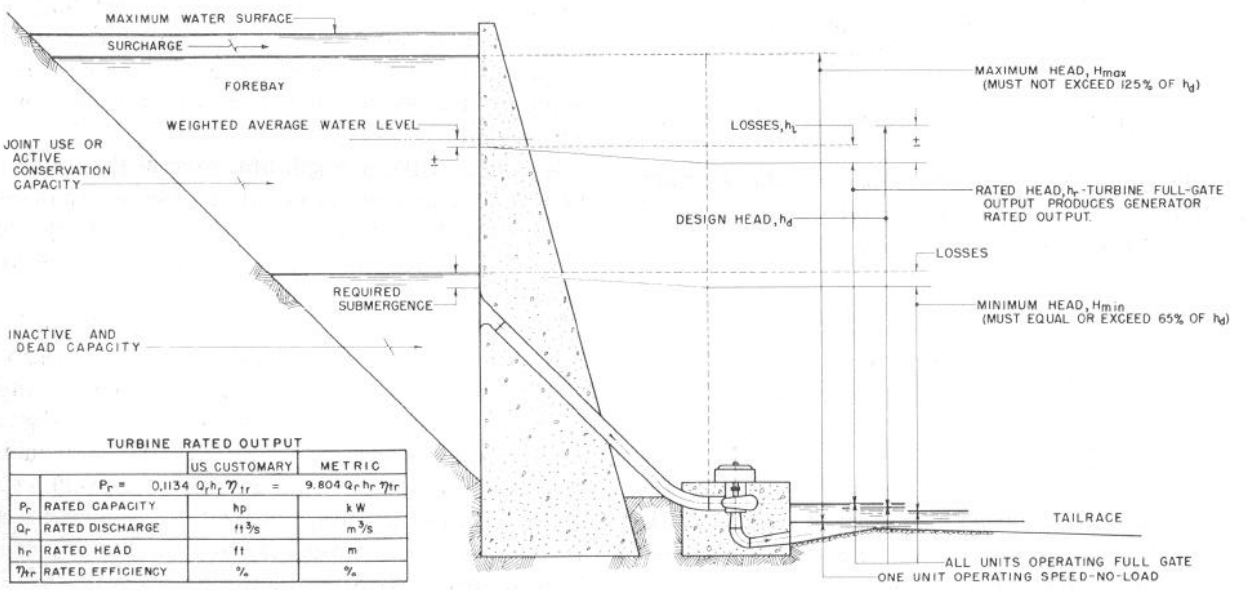


FIGURE 6.—*Operating heads for reaction turbines.*

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.

$$\text{Plant kilowatts} = 0.0846 \cdot \begin{bmatrix} \text{rated} \\ \text{discharge} \\ \text{in ft}^3/\text{s} \end{bmatrix} \cdot \begin{bmatrix} \text{rated head} \\ \text{in feet} \end{bmatrix} \cdot \begin{bmatrix} \text{plant} \\ \text{efficiency} \end{bmatrix}$$

$$= 9.804 \cdot \begin{bmatrix} \text{rated} \\ \text{discharge} \\ \text{in m}^3/\text{s} \end{bmatrix} \cdot \begin{bmatrix} \text{rated head} \\ \text{in metres} \end{bmatrix} \cdot \begin{bmatrix} \text{plant} \\ \text{efficiency} \end{bmatrix}$$

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.

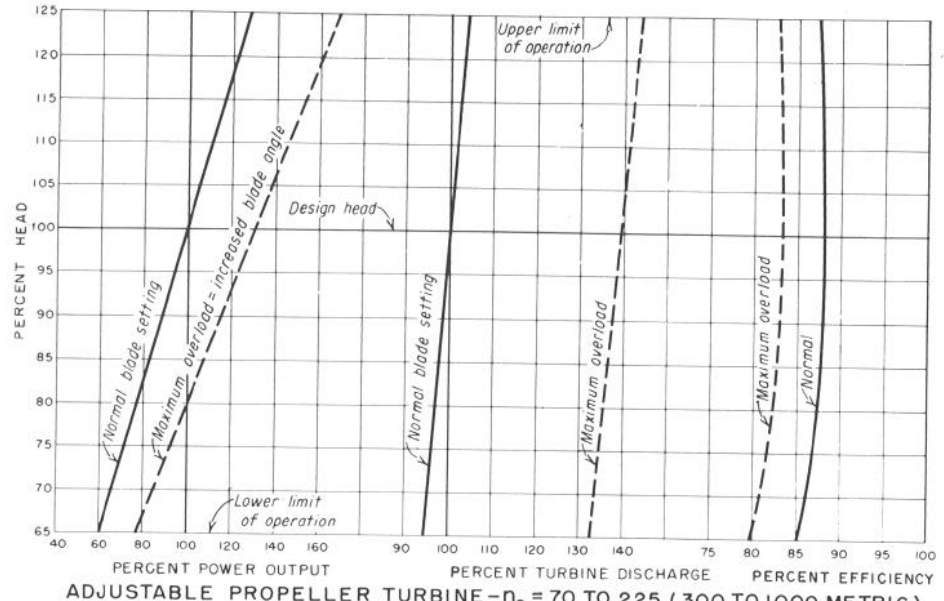
Unit Capacity

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.

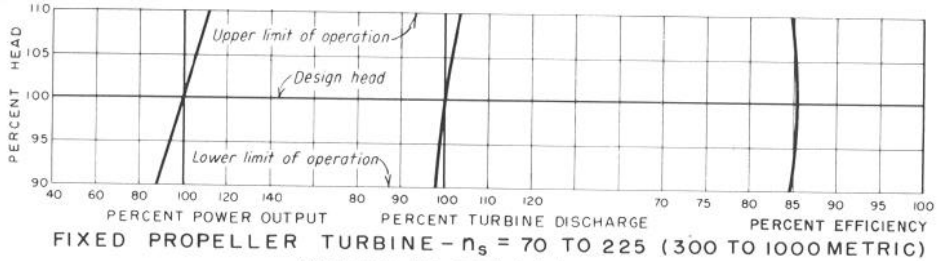
Operation at small gate openings results in low efficiency and accelerated damage from cavitation. Operation is preferably restricted to a power range having an efficiency of 80 percent or more. Figure 8 shows the change in shape of the efficiency curve with specific speed, from a unit of $n_s=35$ (156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



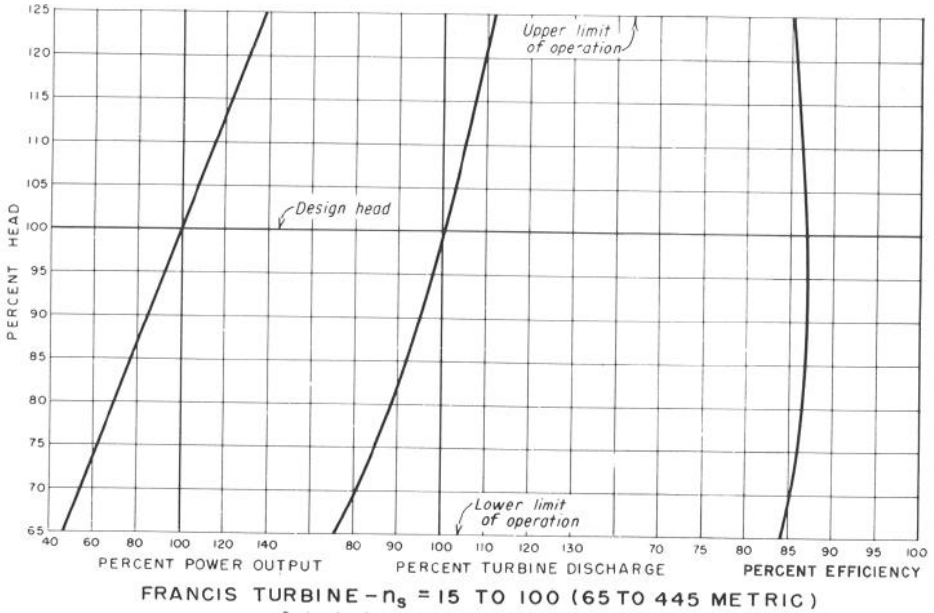
ADJUSTABLE PROPELLER TURBINE – n_s = 70 TO 225 (300 TO 1000 METRIC)

Design head = 10 to 200 feet (3 to 60 metres)



FIXED PROPELLER TURBINE – n_s = 70 TO 225 (300 TO 1000 METRIC)

Design head = 10 to 200 feet (3 to 60 metres)



FRANCIS TURBINE – n_s = 15 TO 100 (65 TO 445 METRIC)

Design head = 30 to 2000 feet (10 to 600 metres)

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REVISED SEPT. 1, 1970
REVISED MAY 27, 1953

106-D-136

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.

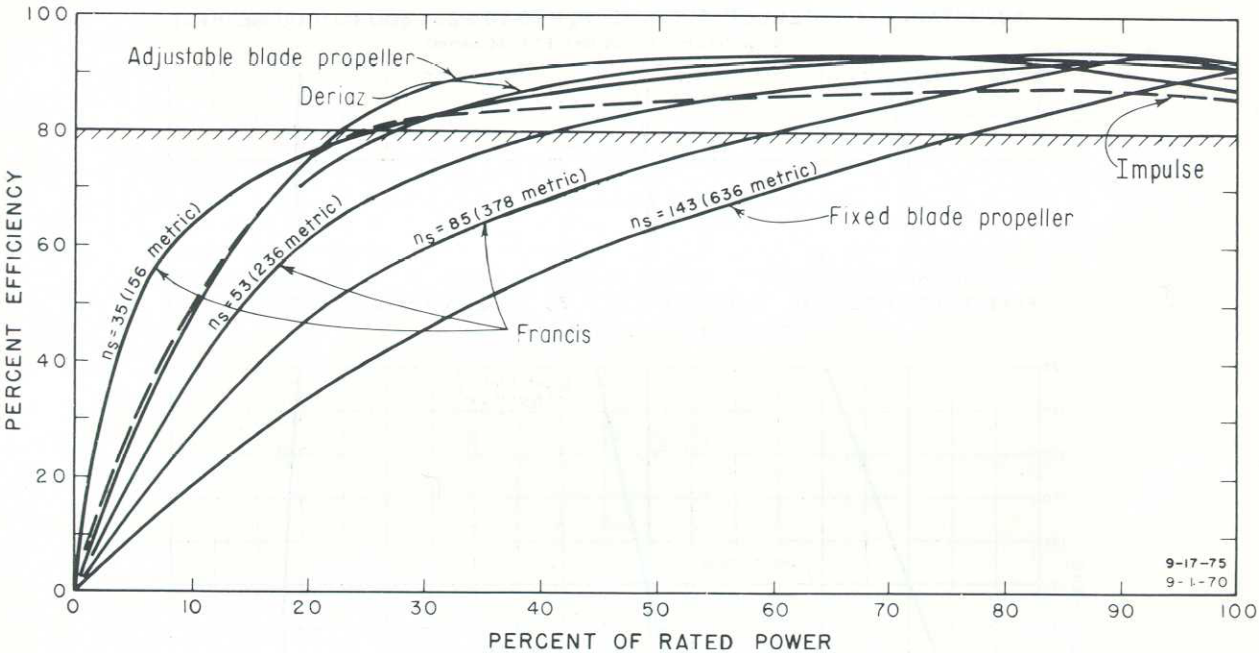
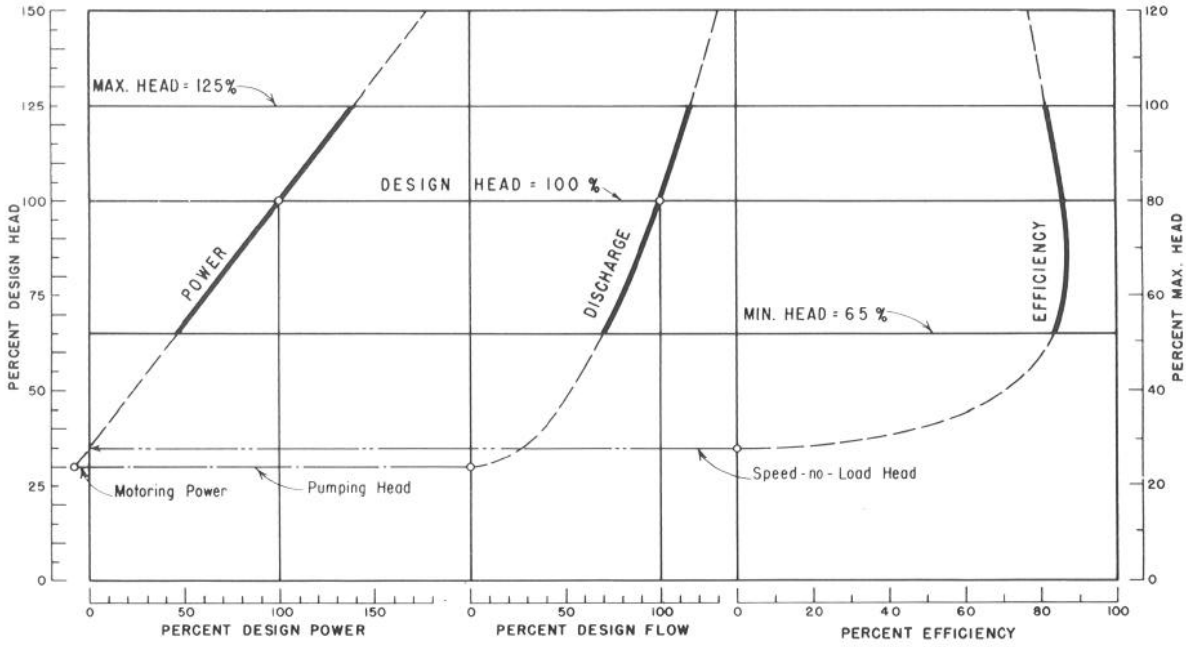


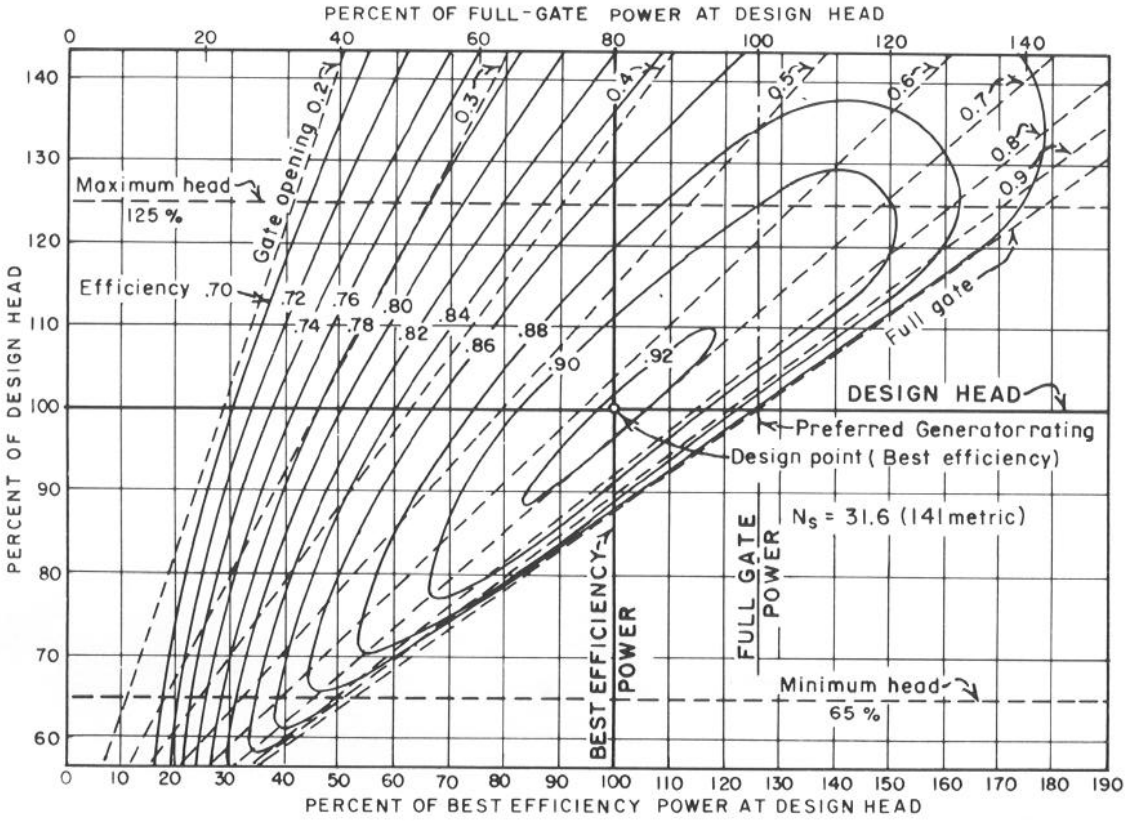
FIGURE 8.—*Hydraulic turbine characteristic efficiency curves.*



9-17-75
6-30-73
9-1-50

106-D-353

FIGURE 9.—*Typical Francis turbine performance—constant speed and full gate.*



REV. 9-17-75

106-D-277

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.

1. Trial specific speed, n_s':

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':

$$n'=\frac{n_s'\,(h_d)^{5/4}}{(P_d)^{1/2}} \text{ or } \frac{n_s'\,h_d}{\left(\frac{P_d}{h_d^{1/2}}\right)^{1/2}}$$

where

n'=trial rotational speed,
n_s'=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.

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

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

4. Design specific speed, n_s:

$$n_s=\frac{n(P_d)^{1/2}}{(h_d)^{5/4}} \text{ or } \frac{n\left(\frac{P_d}{h_d^{1/2}}\right)^{1/2}}{h_d}.$$

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=entrance diameter of runner on distributor centerline,
D_2=minimum opening diameter of runner,
D_3=discharge diameter of runner,
n_s=design specific speed,
h_d=design head,
n=rotational speed,
ϕ_3=velocity ratio at D_3.

Francis turbine:
$\phi_3=0.057\ (n_s)^{2/3}$ (U.S.)$=0.0211\ (n_s)^{2/3}$ (metric).

Propeller turbine:
$\phi_3=0.063\ (n_s)^{2/3}$ (U.S.)$=0.0233\ (n_s)^{2/3}$ (metric).

The discharge diameter of either type is given by:

$$D_3=\frac{153\phi_3(h_d)^{1/2}}{n} \text{ (U.S.)}=\frac{84.47\phi_3(h_d)^{1/2}}{n} \text{ (metric).}$$

The average velocity of the water through D_2 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.

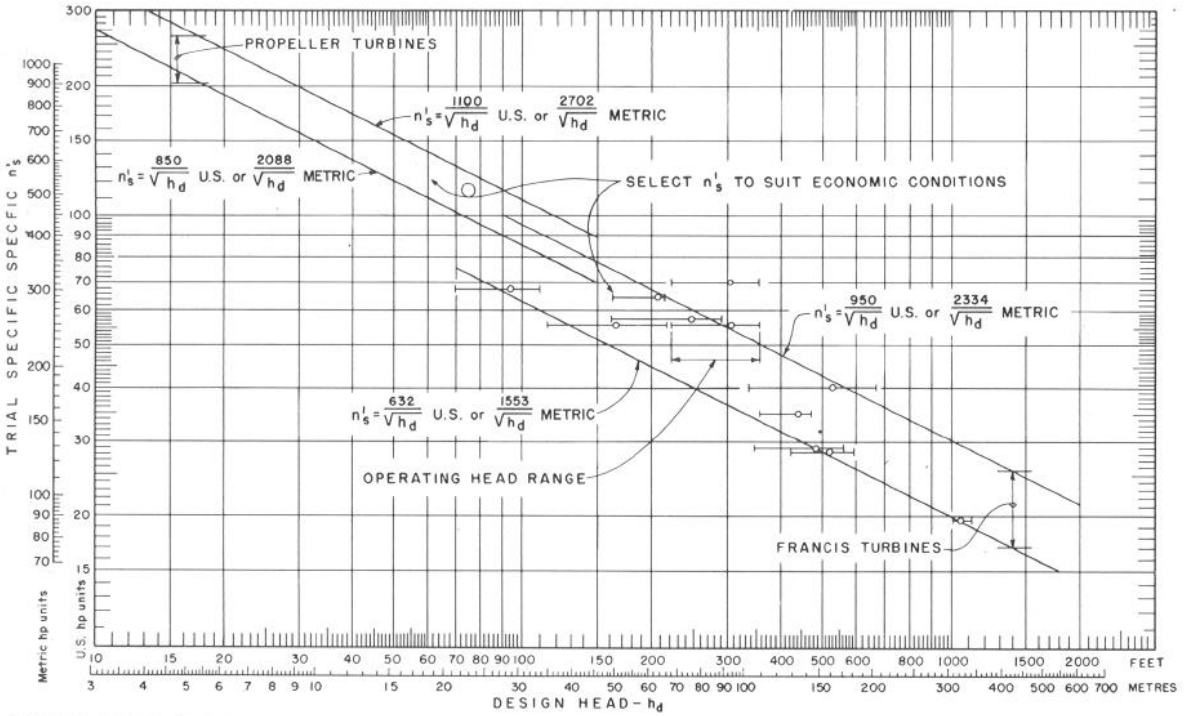
Turbine Weight

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

Trial speed $n' = \frac{n'_s h_d^{5/4}}{P_d^{1/2}}$

ROTATIONAL TURBINE SPEED

Number of poles in Generator = $\frac{120\ (\text{A.C. Frequency})}{n'}$

Adjust number of poles to be an even number, and for large machines, a number divisible by 4. For 60 Hz frequency avoid selection of 54 and 108 poles.

Rotational speed $n = \frac{120\ (\text{A.C. Frequency})}{\text{Adjusted number of poles}}$

DESIGN SPECIFIC SPEED

Specific speed $n_s = \frac{n P_d^{1/2}}{h_d^{5/4}}$

PLANT	TURBINE RATING					DESIGN CONDITIONS						DATA SOURCE	
	HEAD h_r		CAPACITY P_r		SPEED n	HEAD h_d		CAPACITY P_d		SPECIFIC SPEED n_s		DATE	
	ft	m	U.S.	Metric	r/min	ft	m	U.S.	Metric	U.S.	Metric		
FLATIRON	1055	322	48 000	48 672	514	1055	322	52 250	52 982	19.5	87	12-51	MODEL TEST
GLEN CANYON	450	137	155 500	157 677	150	485	148	188 200	190 835	28.6	127	8-60	MODEL TEST
FONTENELLE	87	26	14 100	14 297	150	94	29	17 400	17 644	67.5	300	8-63	MODEL TEST
YELLOWTAIL	440	134	87 500	88 725	225	440	134	96 400	97 750	34.7	154	8-63	MODEL TEST
SEMINOE	166	51	20 800	21 091	225	166	51	21 850	22 156	55.8	248	4-64	MODEL TEST
HOOVER N1-N4	480	146	115 000	116 610	180	520	158	152 000	154 128	28.3	126	8-66	MODEL TEST
GRAND COULEE G19,G20,G21	285	87	820 000	831 480	72	305	93	976 000	989 664	55.8	248	11-69	MODEL TEST
GRAND COULEE G22,G23,G24	285	87	960 000	973 000	85.7	305	93	1085 000	1100 000	70.0	312	10-73	MODEL TEST
TETON	229	70	13 800	13 993	450	248	76	15 600	15 818	57.1	254	9-72	MODEL TEST
CRYSTAL	207	63	39 000	39 546	257	207	63	39 000	39 546	64.6	288	1-75	MODEL TEST
AUBURN	467	142	206 000	208 884	200	531	162	257 000	260 598	39.8	177		SPECIFICATIONS

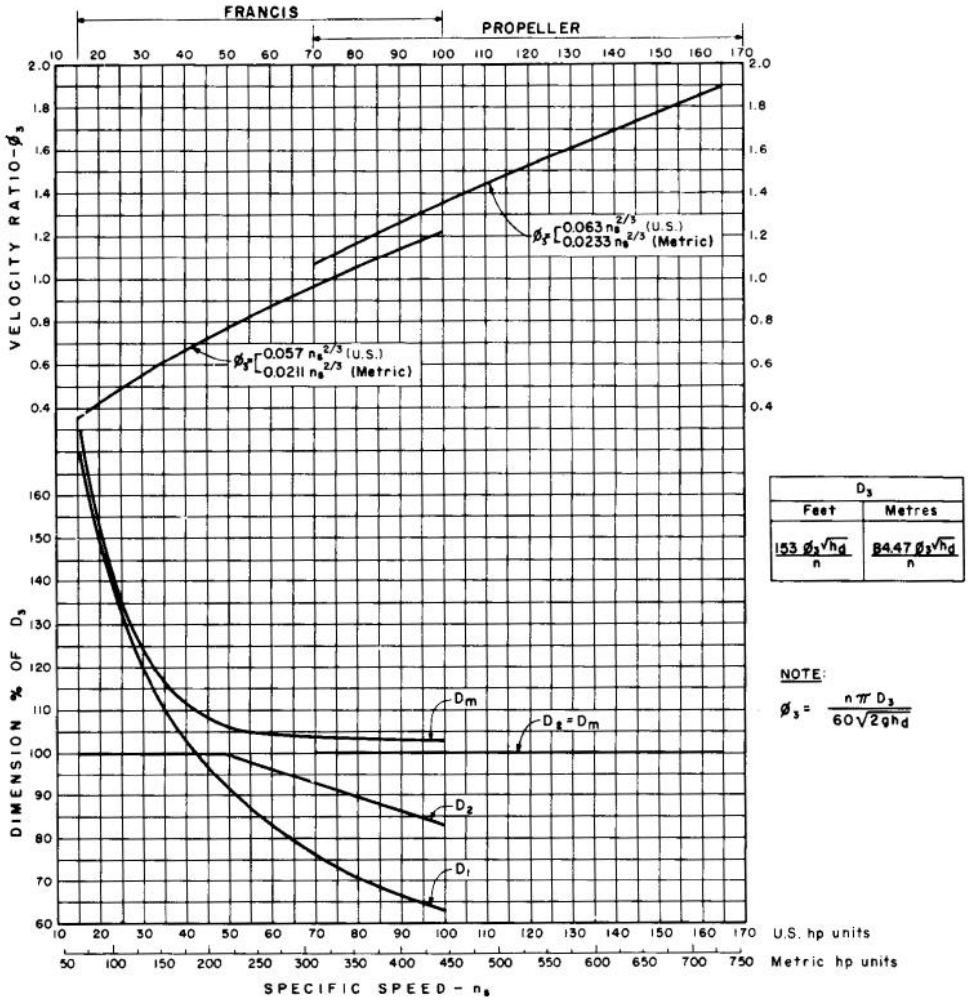
LETTER SYMBOLS

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

9-17-75
9-17-74
11-10-71
9-1-70

106-D-135

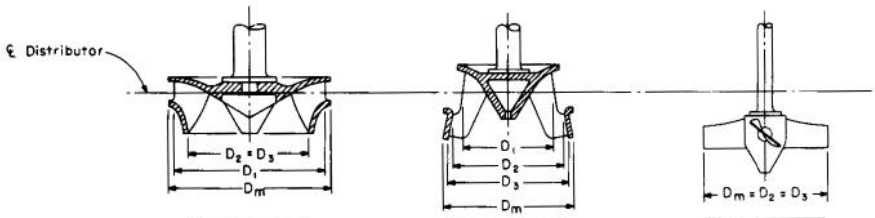
FIGURE 11.—*Turbine characteristics—experience curves for determination of turbine speed.*



D_3	
Feet	Metres
$\frac{153\, \phi_3 \sqrt{h_d}}{n}$	$\frac{84.47\, \phi_3 \sqrt{h_d}}{n}$

NOTE:

$$\phi_3 = \frac{n \pi D_3}{60 \sqrt{2 g h_d}}$$



n_s	
(U.S.)	25
Metric	111

n_s	
(U.S.)	85
Metric	378

n_s	
(U.S.)	145
Metric	645

LETTER SYMBOLS

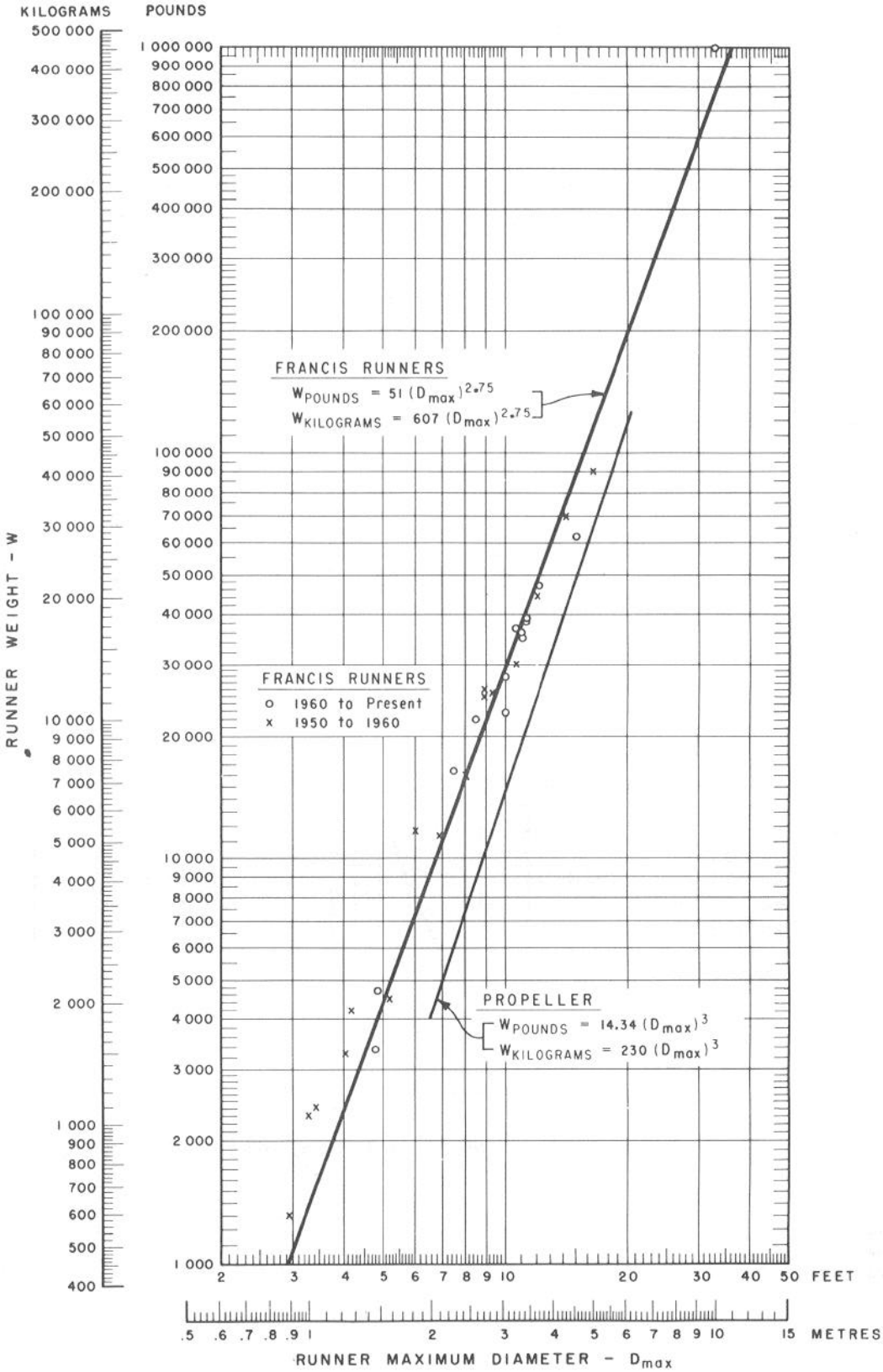
D_1 = Entrance diameter of runner, ft (m)
D_2 = Minimum diameter of runner, ft (m)
D_3 = Discharge diameter of runner, ft (m)
D_m = Maximum outside diameter of runner, ft (m)
h_d = Design head, ft (m)

n = Rotational speed, r/min
n_s = Specific speed
π = 3.14159 . . .
ϕ_3 = Velocity ratio
g = Gravitational constant (acceleration), ft/s² (m/s²)

9-17-75
6-30-73
9-1-70
4-28-63

106-D-233

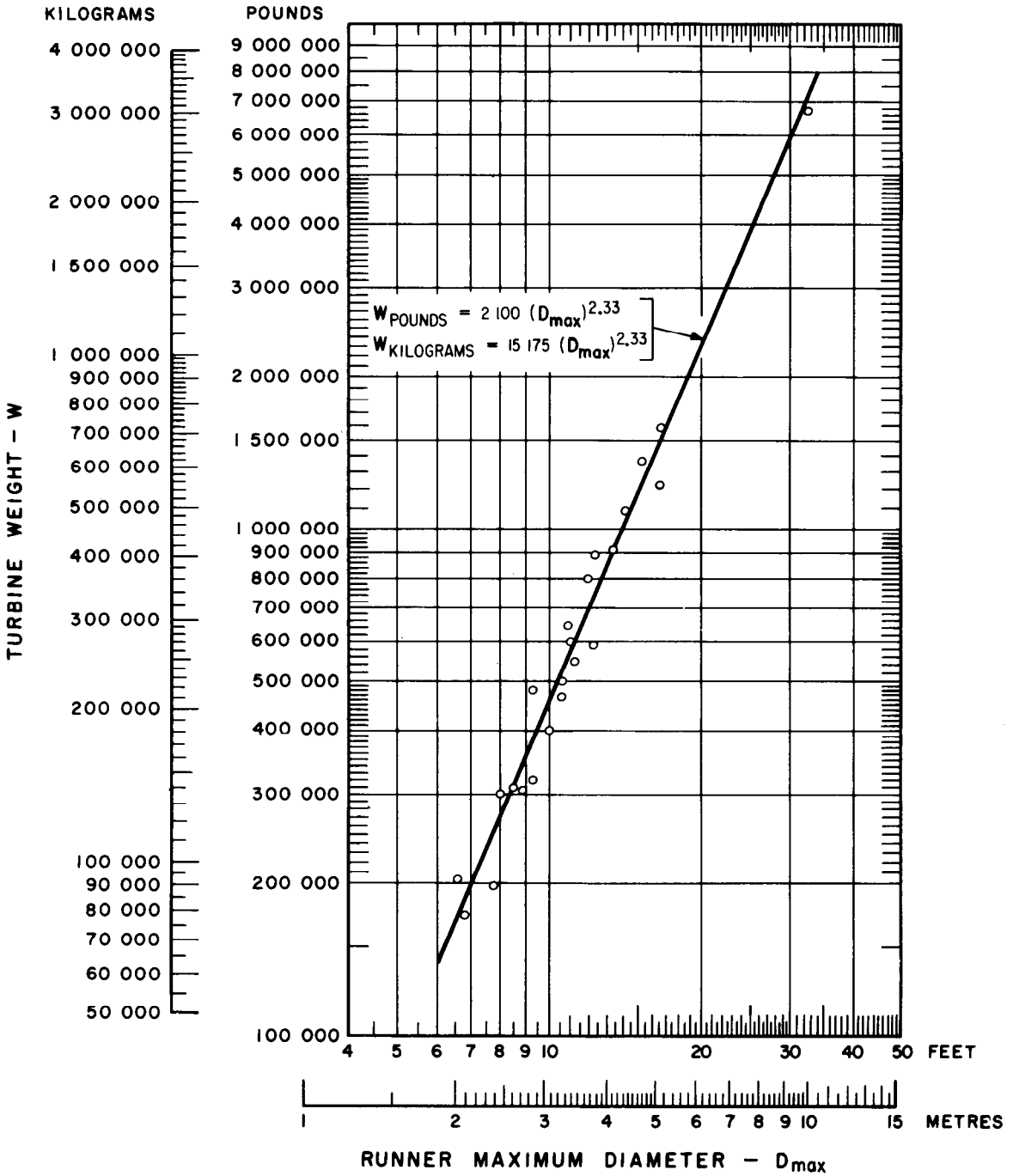
FIGURE 12.—*Hydraulic turbines—basic runner proportions.*



9-17-75
6-30-73

106-D-23

FIGURE 13.—*Hydraulic turbine runner weight—experience curve.*



9-17-75
6-30-73
9-1-70

106-D-349

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:

$$\text{Shaft diameter} = (70\ P_d/n)^{1/3}, \text{ inches}$$

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:

$$\text{Flange diameter} = 1.75 \cdot \text{shaft diameter},$$

and for shafts not subject to bending at the coupling

$$\text{Flange thickness} = 0.20 \cdot \text{shaft diameter}.$$

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).

$$V_e = 0.22\ (2g\ h_d)^{1/2}$$

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

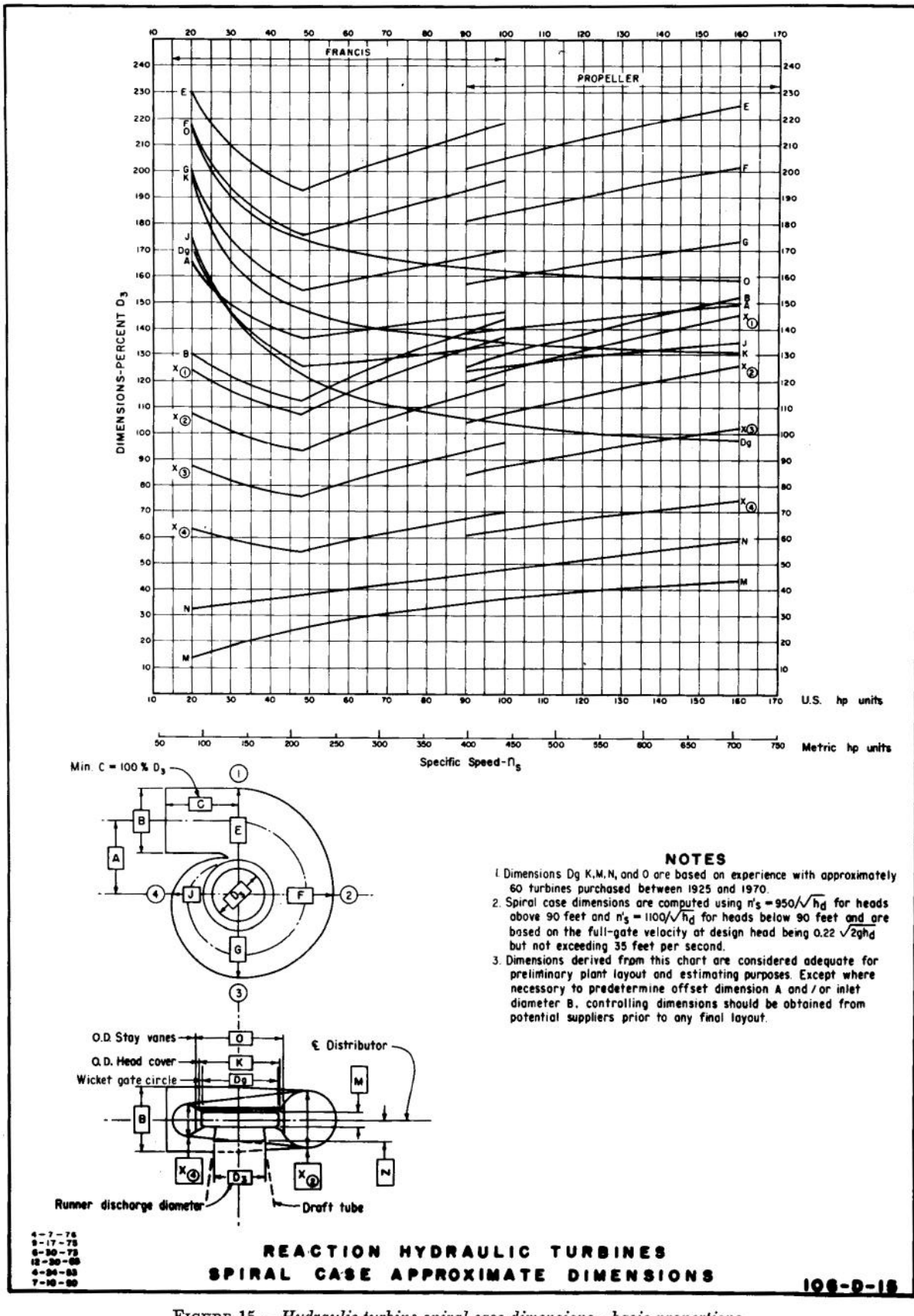


FIGURE 15.—*Hydraulic turbine spiral case dimensions—basic proportions.*

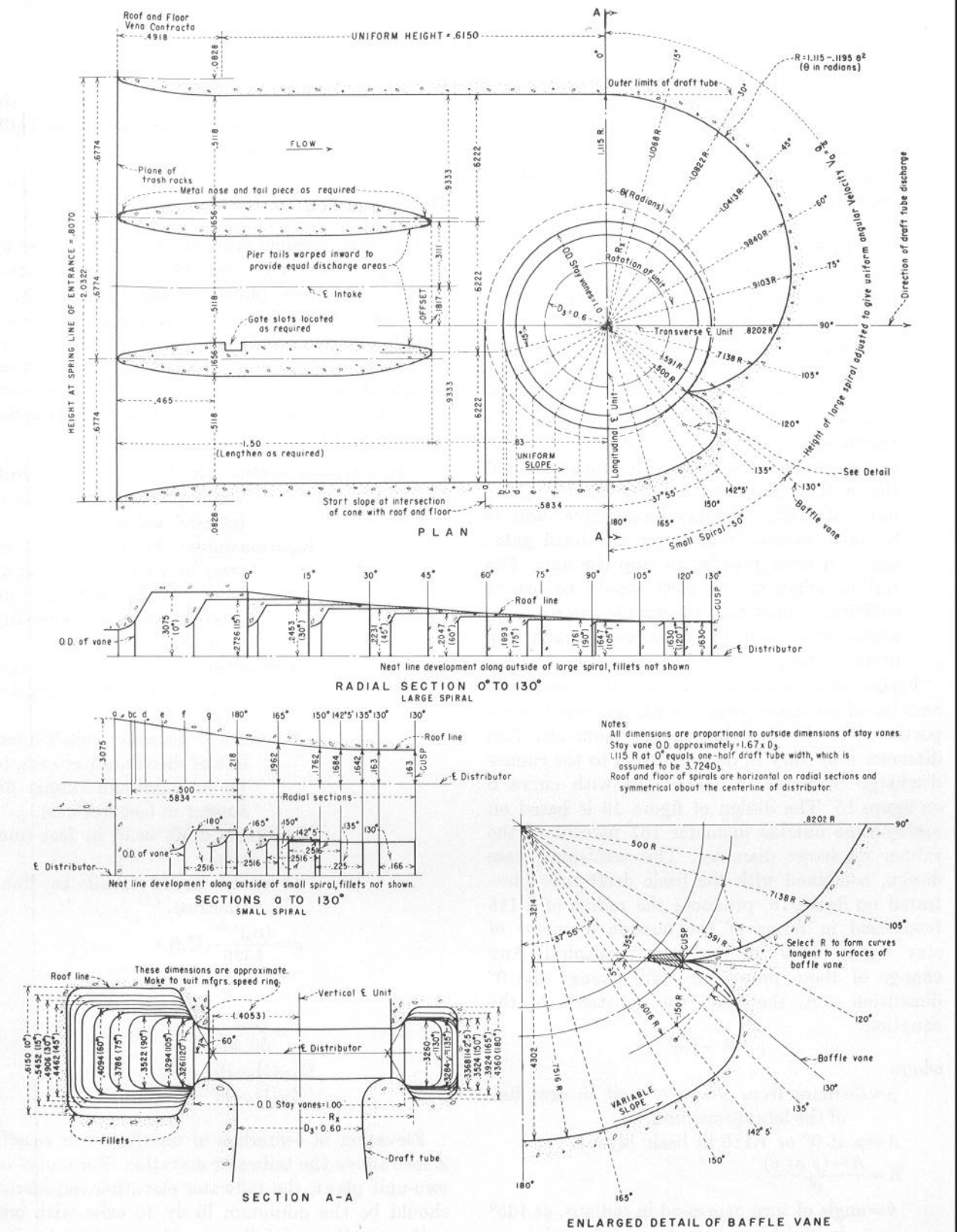


FIGURE 16.—*Hydraulic turbine semispiral case—typical proportions.*

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 bulkhead 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.

$$\rho = A - K\theta^2$$

where

ρ = distance from center of unit to neat line of the large semispiral,

$A = \rho$ at 0° or 1.115 in basic layout,

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

θ = angle of turn expressed in radians, at 135° in the basic layout; $\rho = 0.5$, and

$\theta = 2.356$, 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 (σ) 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.

$H_b = H_a - H_v$ = atmospheric pressure minus vapor pressure in feet (metres) of water.

h_{cr} = maximum head in feet (metres) at which turbine may be operated at full gate considering overload capability of generator.

σ = cavitation factor (Thoma)—from curve or from model if available.

b = vertical distance from centerline of distributor or case, to D_2 or minimum runner diameter in feet (metres).

H_s = static draft head in feet (metres).

Z = total draft head in feet (metres).

$$\sigma = \frac{(n_s)^{1.64}}{4325} \text{ (U.S.)}$$

or

$$= \frac{(n_s)^{1.64}}{50\ 327} \text{ (metric)}$$

$$H_s = H_b - \sigma h_{cr}$$

$$Z = H_s + b$$

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

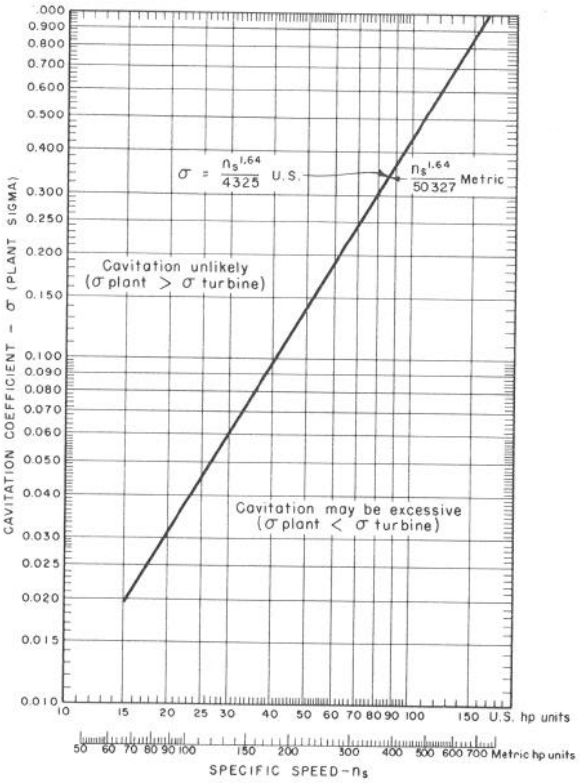
ATMOSPHERIC PRESSURE					
ALTITUDE FEET	H_a PSI	H_a Ft of H_2O	ALTITUDE METRES	H_a mm of Hg	H_a M of H_2O
0	14.696	33.959	0	760.00	10.351
500	14.43	33.35	500	715.99	9.751
1 000	14.17	32.75	1 000	674.07	9.180
1 500	13.92	32.16	1 500	634.16	8.637
2 000	13.66	31.57	2 000	596.18	8.120
2 500	13.42	31.00	2 500	560.07	7.628
3 000	13.17	30.43	3 000	525.75	7.160
3 500	12.93	29.88	3 500	493.15	6.716
4 000	12.69	29.33	4 000	462.21	6.295
4 500	12.46	28.79			
5 000	12.23	28.25			
5 500	12.00	27.73			
6 000	11.78	27.21			
6 500	11.56	26.70			
7 000	11.34	26.20			
7 500	11.12	25.71			
8 000	10.91	25.22			
8 500	10.71	24.74			
9 000	10.50	24.27			
9 500	10.30	23.81			
10 000	10.10	23.35			

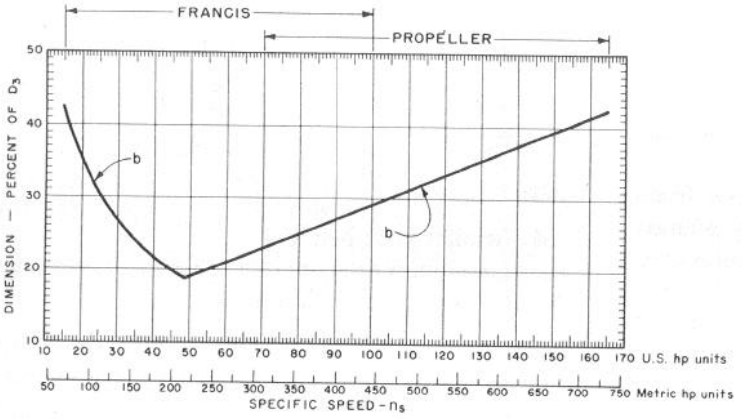
WATER PROPERTIES			
TEMP °F	H_v FEET	TEMP °C	H_v METRE
40	0.28	5	0.089
50	0.41	10	0.125
60	0.59	15	0.174
70	0.84	20	0.239
80	0.17	25	0.324

H_a = Atmospheric pressure for altitude, ft (m).

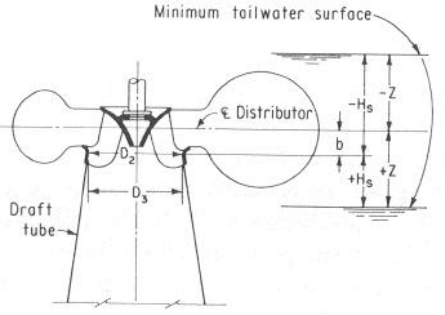
H_v = Vapor pressure of water, use highest expected temperature, ft (m).

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





D_2 at shrouding = Least diameter through shroud, ft (m).
D_3 = Discharge diameter of runner, ft (m)
b = Distance from D_2 to ℄ of distributor, ft (m).
(estimated from curve % D_3 vs n_s).
n_s = Specific speed of turbine



h_{cr} = Critical head, ft (m).
H_s = Distance from D_2 to minimum tailwater (one unit operating at full gate), ft (m).
$H_s = H_b - \sigma h_{cr}$ or $\sigma = (H_b - H_s) / h_{cr}$, ft (m).

Z = ℄ Distributor to minimum tailwater = H_s + b (Total draft head), ft (m).

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

9-17-75
11-19-74
6-30-73
11-3-64
2-10-48

106-D-141

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 (D_3). 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 (D_3) 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 D_3 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 D_3. 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 D_3 to the following points:

1. To the point at which the draft tube area is twice the runner discharge area (at D_3) 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.

$$FY_M = 4.23\ (h_{wh} \cdot D_g \cdot M)^{1.14}\ \text{(U.S.)}$$

$$FY_M = 34\ (h_{wh} \cdot D_g \cdot M)^{1.14}\ \text{(metric)}$$

where

M = wicket gate height,
h_{wh} = 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:

$$FY_b = \frac{16.47\ P_{max}(n_s)^{1/4}}{(H_{max})^{1/2}}\ \text{(U.S.)}$$

$$FY_b = \frac{6.17\ P_{max}(n_s)^{1/4}}{(H_{max})^{1/2}}\ \text{(metric)}$$

where

H_{max} = maximum head,
n_s = design specific speed, and
P_{max} = turbine full-gate capacity at H_{max}.

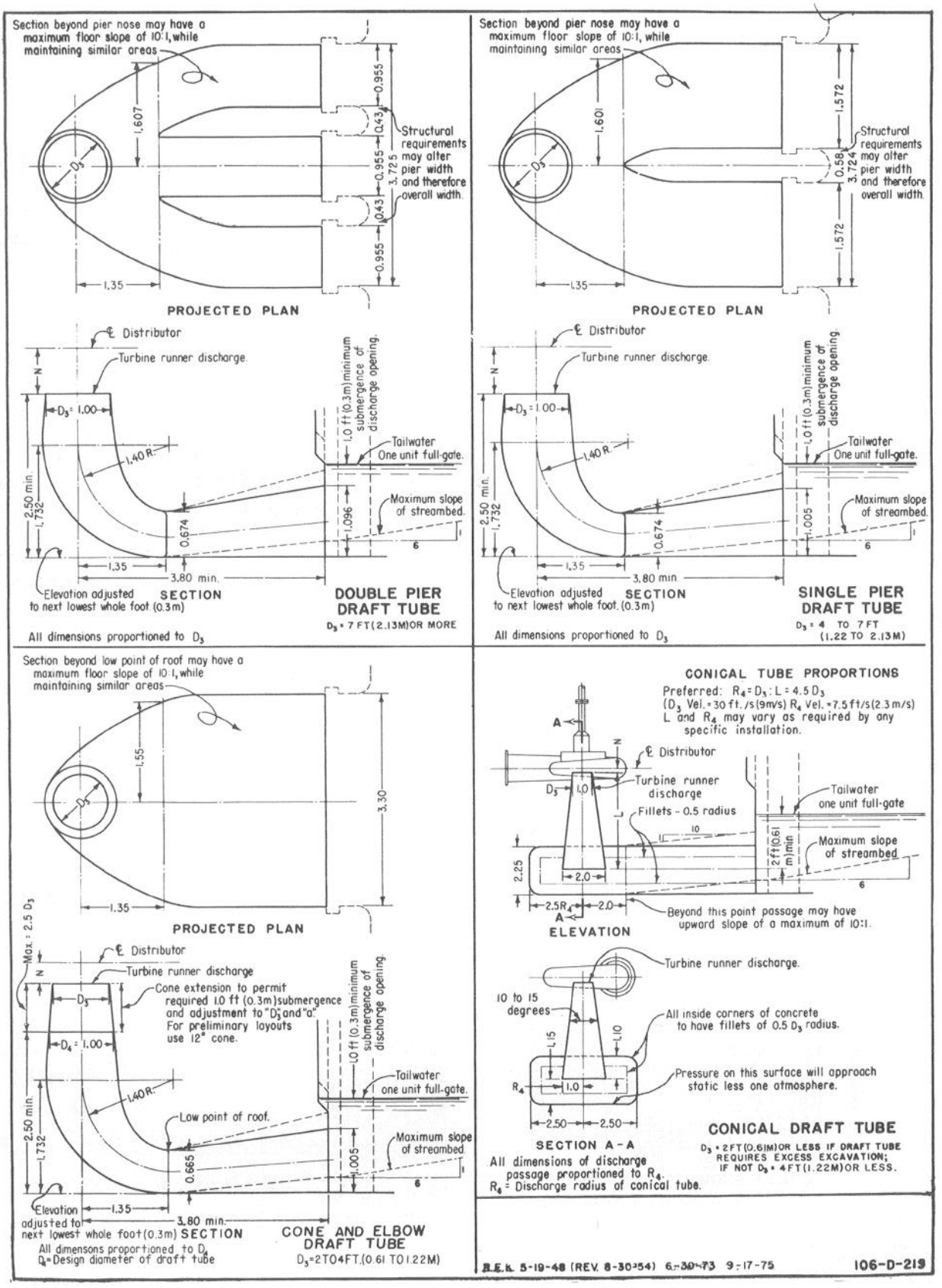
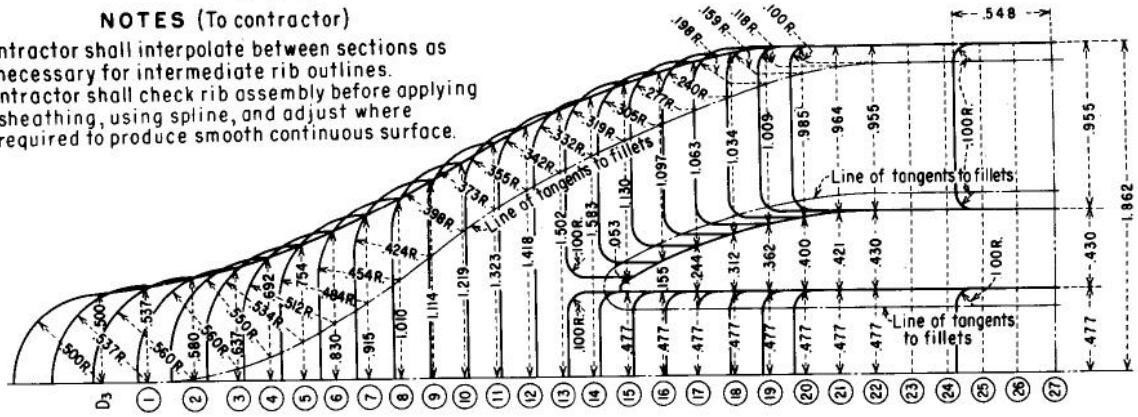


FIGURE 18.—*Turbine draft tubes—preliminary basic outlines.*

NOTES (To contractor)

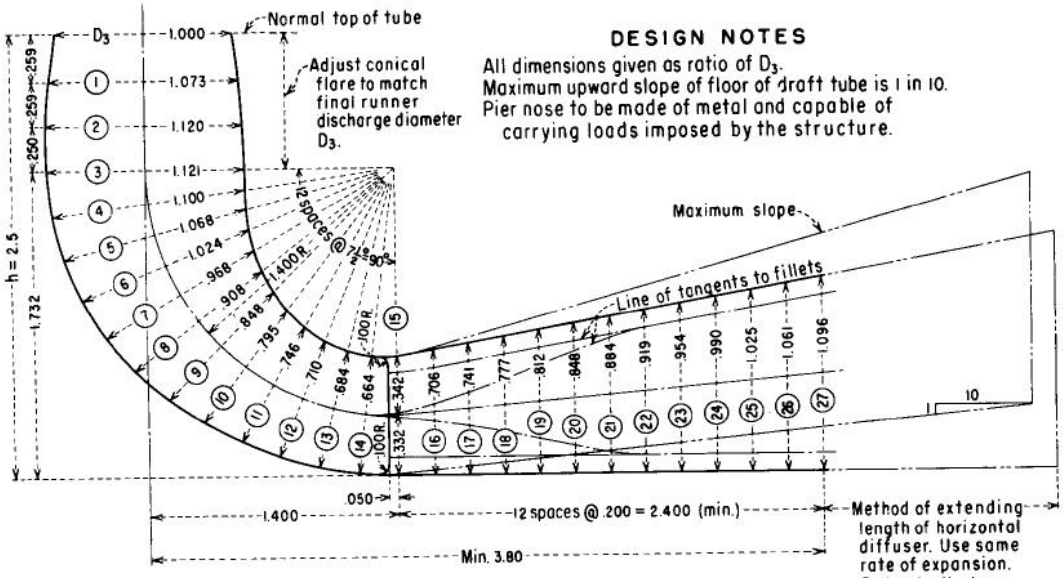
Contractor shall interpolate between sections as necessary for intermediate rib outlines.
Contractor shall check rib assembly before applying sheathing, using spline, and adjust where required to produce smooth continuous surface.



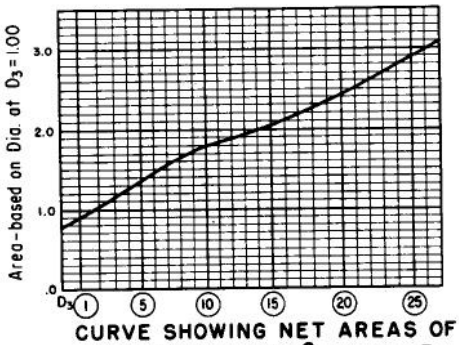
DEVELOPED HALF PLAN ON ℄ OF TUBE

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



CURVE SHOWING NET AREAS OF SECTIONS ALONG ℄ OF TUBE

PREFERRED RATIOS

Area at top of tube / Net area at pier nose	0.381
Net area at pier nose / Net area at discharge	0.660 (max.)
Net width at pier nose / Height of pier nose	4.769
Height of tube / Dia. at top of tube	2.500 (min.)
Radius to ℄ elbow / Dia. at top of tube	1.400
Length along ℄ tube (top to pier nose) / Dia. at top of tube	2.967 (min.)
Horizontal distance ℄ turbine to discharge / Dia. at top of tube	3.80 (min.)
Developed Length (D_3 to discharge)	125 ft (38 m) (max.)

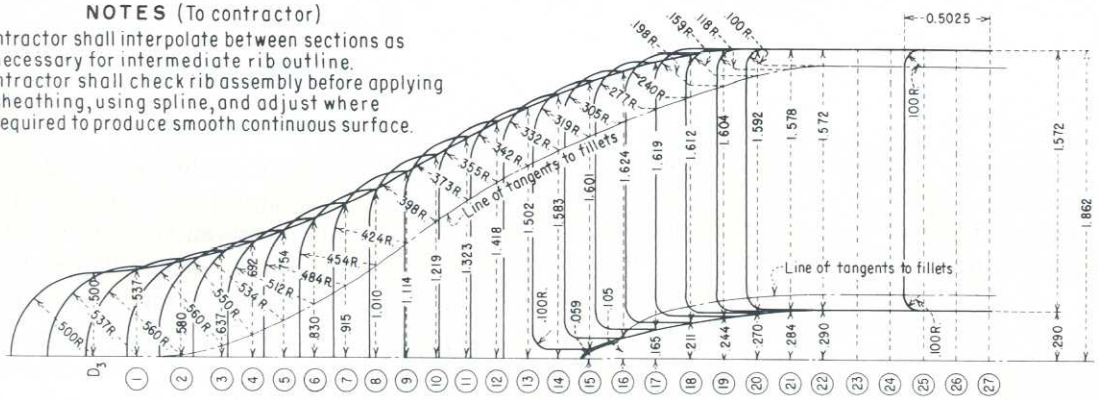
REV. 9-17-75
REV. 6-30-73
REV. 12-15-54
RETR. 9-25-53
MARCH 20, 1945

106-D-158

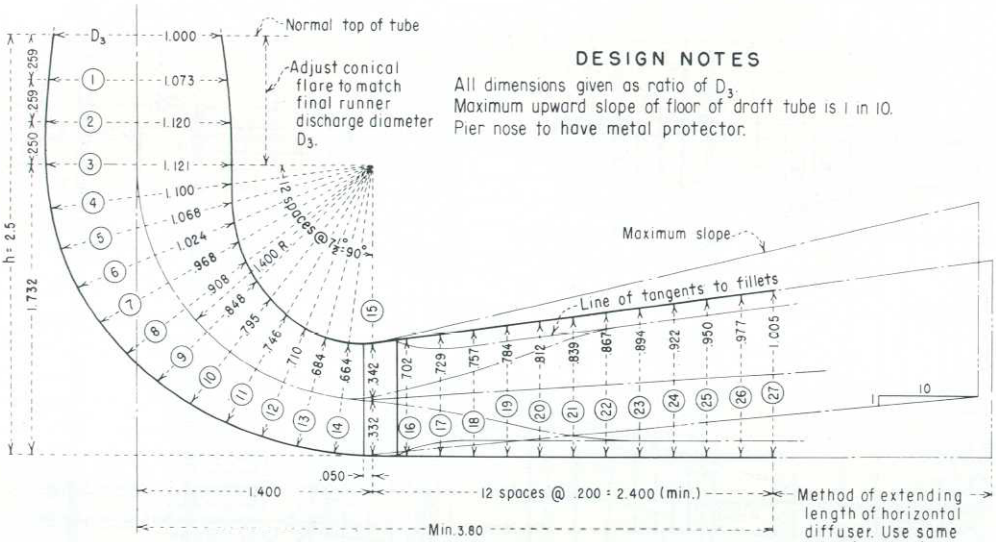
FIGURE 19.—*Turbine draft tubes—two piers—optimum proportions.*

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.



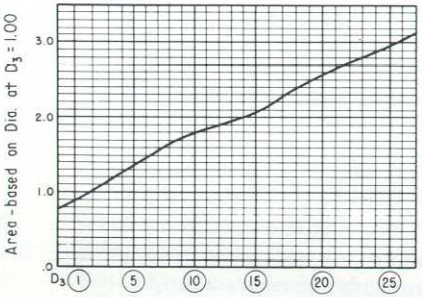
DEVELOPED HALF PLAN ON ℄ OF TUBE



DESIGN NOTES

All dimensions given as ratio of D_3.
Maximum upward slope of floor of draft tube is 1 in 10.
Pier nose to have metal protector.

SECTION ON ℄ OF TUBE



CURVE SHOWING NET AREAS OF SECTIONS ALONG ℄ OF TUBE

PREFERRED RATIOS

Area at top of tube / Net area at pier nose	0.381
Net area at pier nose / Net area at discharge	0.660 (max.)
Net width at pier nose / Height of pier nose	4.769
Height of tube / Dia. at top of tube	2.500
Radius to ℄ elbow / Dia. at top of tube	1.400
Length along ℄ tube (top to pier nose) / Dia. at top of tube	2.967 (min.)
Horizontal distance ℄ turbine to discharge / Dia. at top of tube	3.80 (min.)
Developed Length (D_3 to discharge)	125 ft (38m) (max.)

REV. 9-17-75
REV. 6-30-73
REV. 12-15-54
REV. 10-1-53
FORMERLY X-D-3830

106-D-217

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 (FY_M) and blade (FY_b) 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.

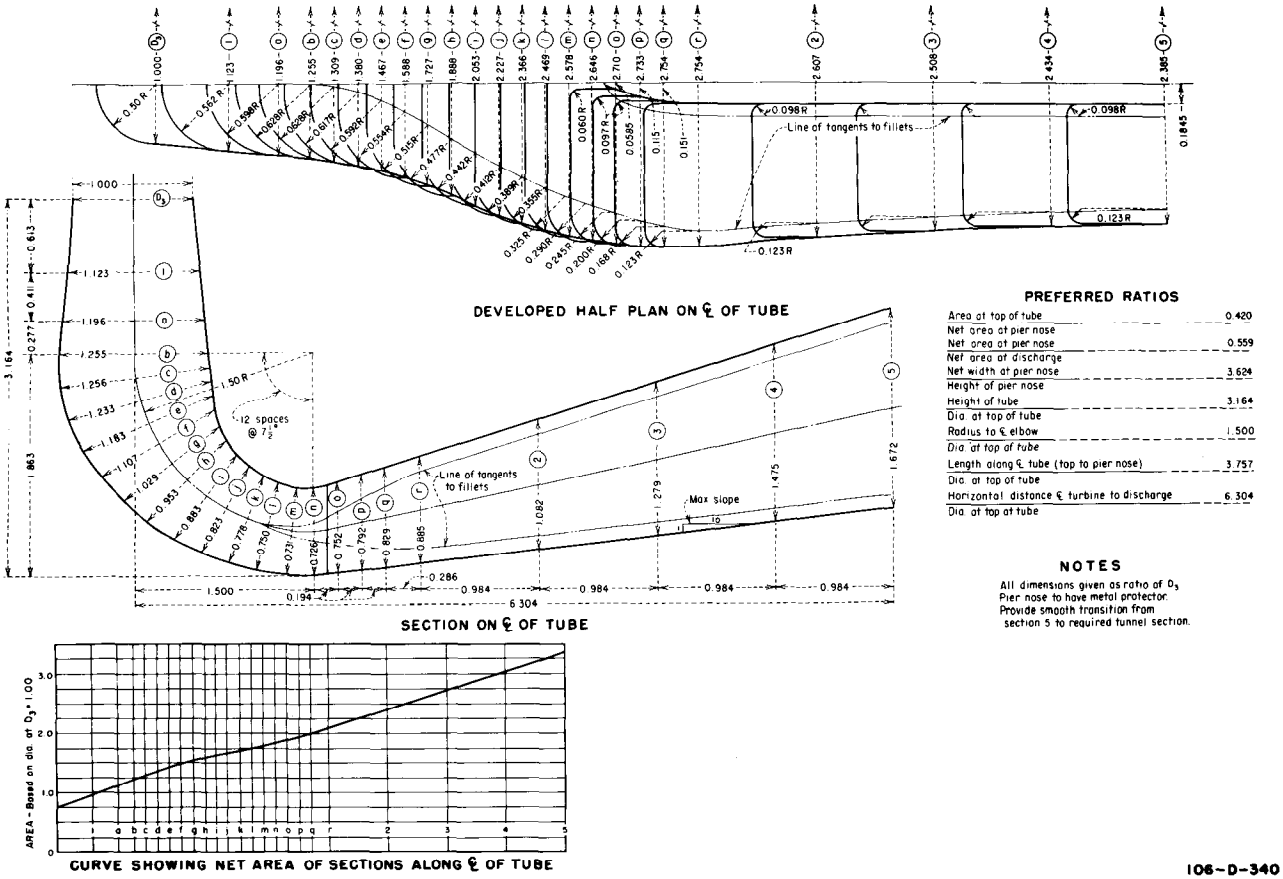
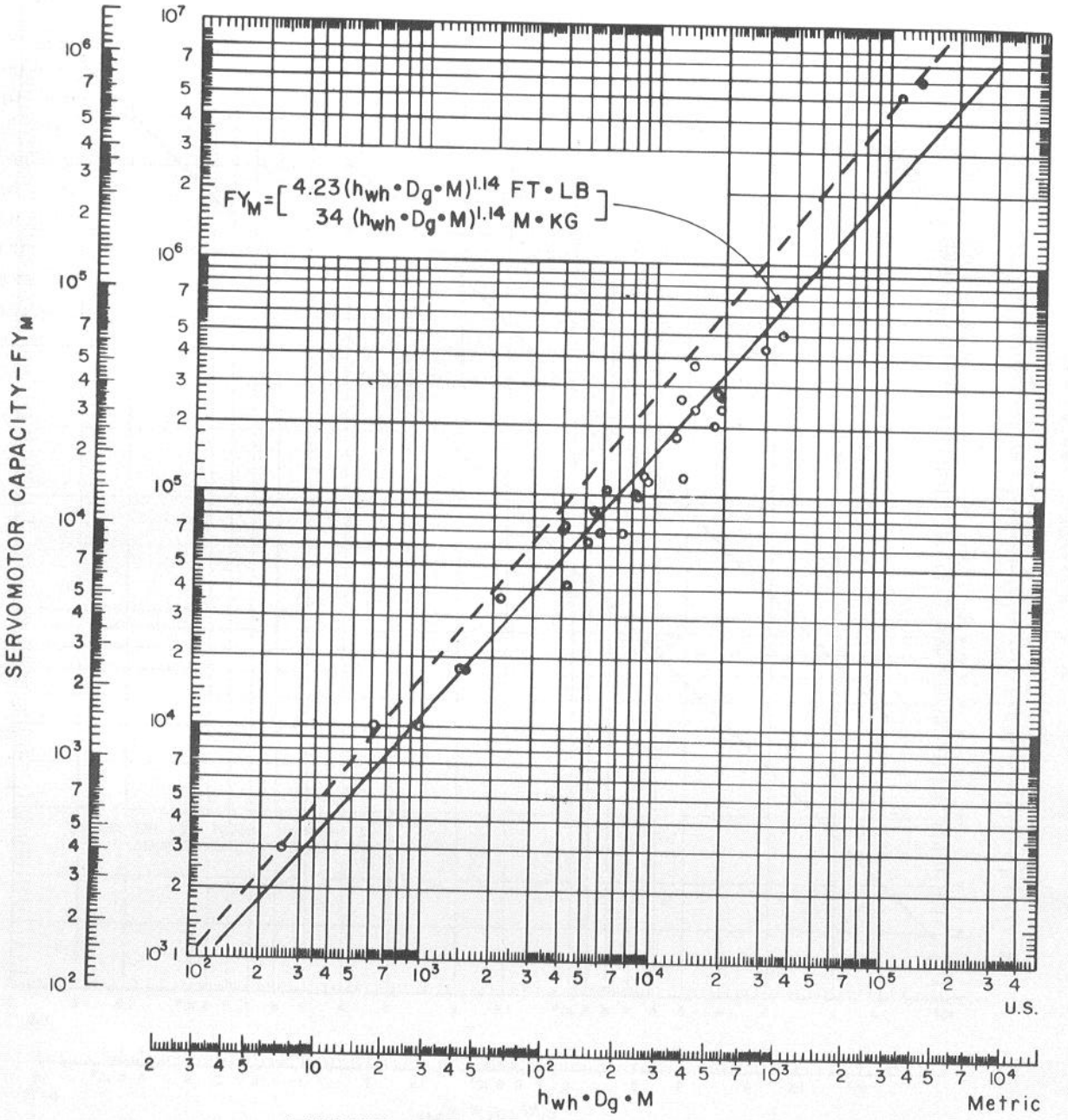


FIGURE 21.—*Turbine draft tubes—one pier—underground plant design.*



LETTER SYMBOLS

M = Wicket gate height, ft (m)
h_{wh} = Maximum head, including water hammer, ft (m)
D_g = Wicket gate circle diameter, ft (m)
FY_M = Servomotor capacity, ft • lb (m • kg)
P_{min} = Servomotor minimum rated oil pressure, lb/in^2(kg/m^2)
V_s = Servomotor net volume, in^3 (m^3)

NOTE:

$$V_s = \left[\begin{array}{l} \dfrac{12 FY_M}{P_{min}}, \text{ in}^3 \\ \dfrac{FY_M}{P_{min}}, \text{ m}^3 \end{array} \right.$$

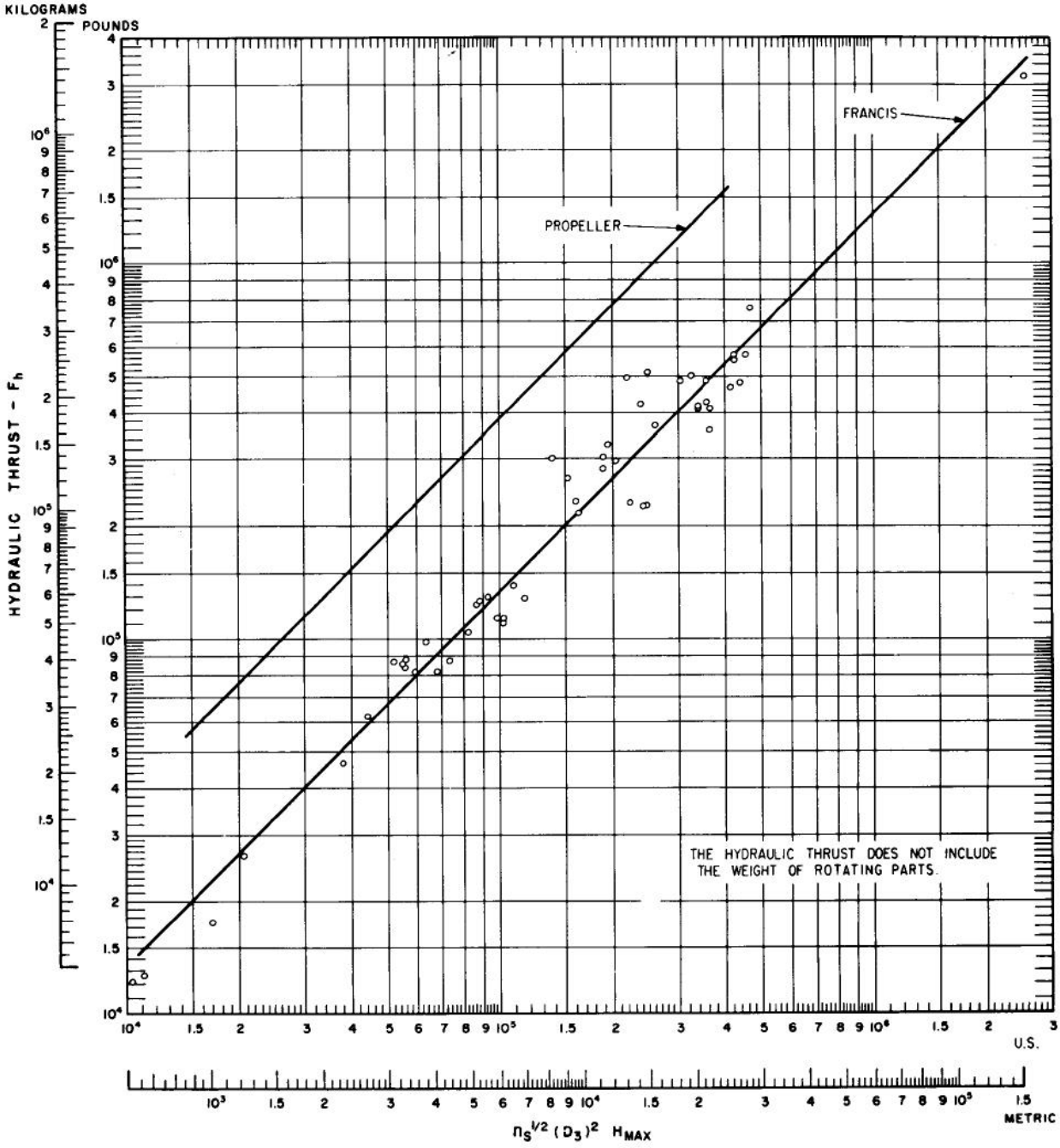
REACTION HYDRAULIC TURBINES

WICKET GATE SERVOMOTOR CAPACITY

3-19-76
9-17-75
6-30-73

106-D-22

FIGURE 22.—*Hydraulic turbine servomotor capacity—experience curve.*



TURBINE HYDRAULIC THRUST — F_h

SYSTEM	UNITS	FRANCIS	PROPELLER
U.S.	POUNDS	$1.32\ n_s^{1/2}(D_3)^2\ H_{MAX}$	$3.80\ n_s^{1/2}(D_3)^2\ H_{MAX}$
METRIC	KILOGRAMS	$10.02\ n_s^{1/2}(D_3)^2\ H_{MAX}$	$28.85\ n_s^{1/2}(D_3)^2\ H_{MAX}$

LETTER SYMBOLS

D_3 = Discharge diameter of runner, ft (m)
F_h = Hydraulic thrust, lb (kg)
H_{MAX} = Maximum head, ft (m)

6-30-73
4-7-50

106-D-24

FIGURE 23.—*Hydraulic thrust—experience curve.*

Turbine Performance

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,

$$T_m = \frac{WR^2\, n^2}{1.6 \cdot 10^6\, P} \text{ (U.S.)}.$$

$$= \frac{WR^2\, n^2}{6.7 \cdot 10^4\, P} \text{ (metric)}.$$

where

n = rotational speed,

P = turbine full-gate capacity in horsepower for the condition under consideration, and

WR^2 = 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:

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

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

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

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

Additional generator WR^2 may be specified if required.

Water column inertia is conveniently expressed as starting-up time of water column, $T_w = \frac{\Sigma LV}{gh}$. 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 ΣLV.

Units for which $T_m \geq 2\ (T_w)^2$ 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^2, 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:

$$\frac{n_r}{n} = 0.85(n_s)^{1/5} \quad \text{(U.S.)}$$

$$\frac{n_r}{n} = 0.63(n_s)^{1/5} \quad \text{(metric)}$$

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

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 ecomical 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 on a 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 watersaving 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:

T_f = Servomotor minimum closing time, sec.
P_r = Turbine full-gate capacity at h_r, hp
h_r = Rated head, ft
n = Rotational speed; design, r/min
n_s = Design specific speed
WR^2 = Flywheel effect of revolving parts, lb·ft²
L = Equivalent length of water conduit, ft
A = Equivalent area of water conduit, ft²
g = Gravitational constant (acceleration), ft/s²
$Q_r = \frac{550 \cdot P_r}{62.4 \cdot h_r \cdot 0.875}$ = Turbine full-gate discharge, ft³/s
$V_r = \frac{Q_r}{A}$ = Conduit water velocity for full-gate at h_r, ft/s

To obtain the speed rise for full-load rejection, determine the following values:

(a) $T_k = 0.25 + T_f$, full closing time of servomotor(s)

(b) $T_m = \frac{WR^2 n^2}{1\,600\,000 \cdot P_r}$, mechanical startup time.

(c) $\frac{T_k}{T_m}$

(d) $n_{sr} = \frac{n(P_r)^{1/2}}{(h_r)^{5/4}}$ At rated condition, hp units

(e) Determine S_R from chart A using n_s and $\frac{T_k}{T_m}$ where:
S_R is speed rise in percent of rotational speed n, for full-gate load rejection to zero, excluding effects of waterhammer

(f) $T_w = \frac{\Sigma L V_r}{g h_r}$ Water startup time

(g) $K = \frac{T_w}{T_f}$

(h) $S_R' = S_R(1+K)$, speed rise in percent of rotational speed n, for full-gate load rejection to zero, including effects of waterhammer

Example:

Given: $T_f = 5$ sec, $P_r = 40\,000$ hp, $h_r = 80$ ft, $n_{sr} = 94.7$, $WR^2 = 53\,240\,000$ lb·ft², $V_r = 13.779$ ft/s, $L = 340$ ft

(a) $T_k = 0.25 + 5 = 5.25$

(b) $T_m = \frac{(53\,240\,000)(94.7)^2}{(1\,600\,000)(40\,000)} = 7.46$

(c) $\frac{T_k}{T_m} = \frac{5.25}{7.46} = 0.704$

(d) $n_{sr} = \frac{(94.7)(40\,000)^{1/2}}{(80)^{5/4}} = 79.2$

(e) S_R = 28.1 percent from chart A.

(f) $T_w = \frac{(340)(13.779)}{(32.174)(80)} = 1.820$

(g) $K = \frac{T_w}{T_f} = \frac{1.820}{5} = 0.364$

(h) $S_R' = (28.1)(1+0.364) = 38.9$ percent.

References:

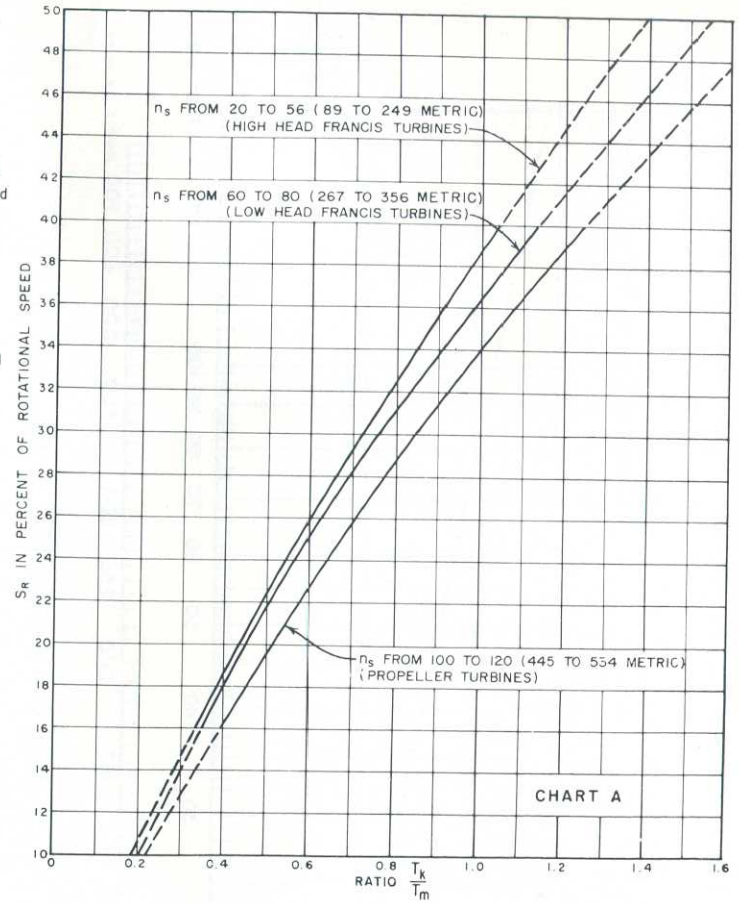
Memorandum by John Parmakian dated March 25, 1949, titled "A Proposed Basis for Determining When Surge Tanks Will be Required for Speed Regulation at Turbine Penstock Installations."

A.S.M.E. paper No. 48-A-37 "Speed Regulation Computations for Hydraulic Turbines," Robert Lowy, 1949.

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 h_{cr}, H_{min}, and H_{max}.

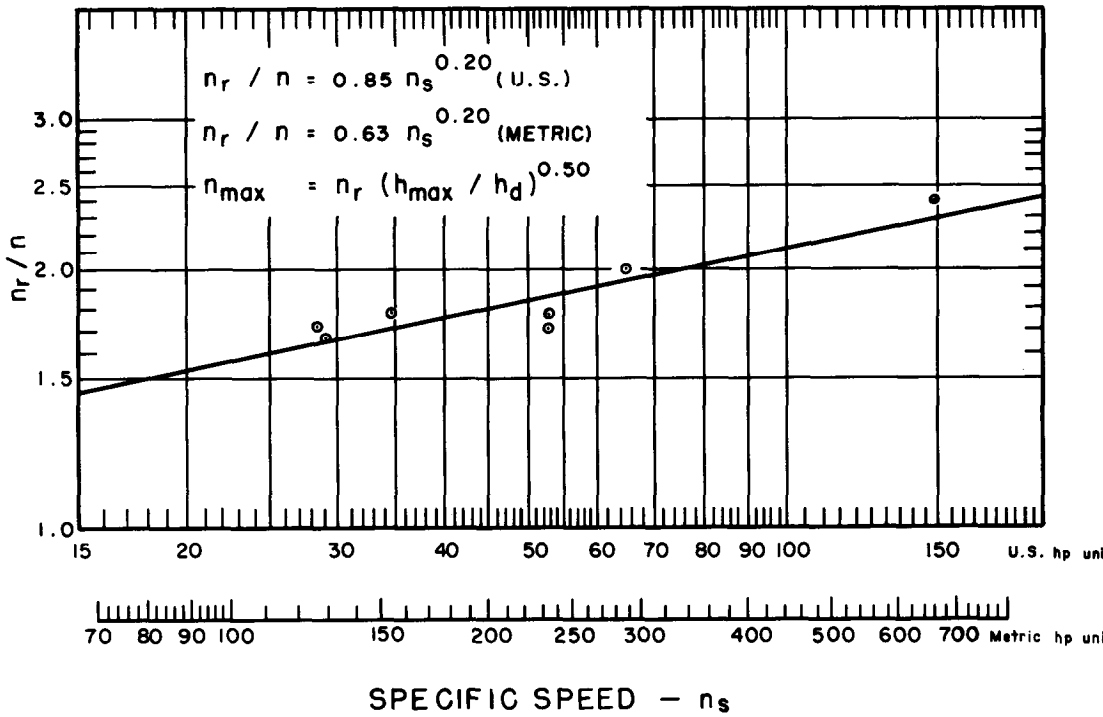
9-17-75
OCT. 1949 - JOHN PARMAKIAN



HYDRAULIC TURBINE SPEED REGULATION UTILIZATION OF SURGE TANKS

106-D-68

FIGURE 24.—*Hydraulic turbine speed regulation—utilization of surge tank.*



h_d = Best efficiency head (design head), ft (m)
h_{max} = Maximum head, ft (m)
n = Rotational speed, r/min
n_{max} = Runaway speed at maximum head, r/min
n_r = Runaway speed at best efficiency head and full gate.
n_s = Specific speed based on full gate output at best efficiency head.
The curve is based on model test data.

9-17-75
6-30-74
6-27-56

106-D-232

FIGURE 25.—*Hydraulic turbine runaway speed.*

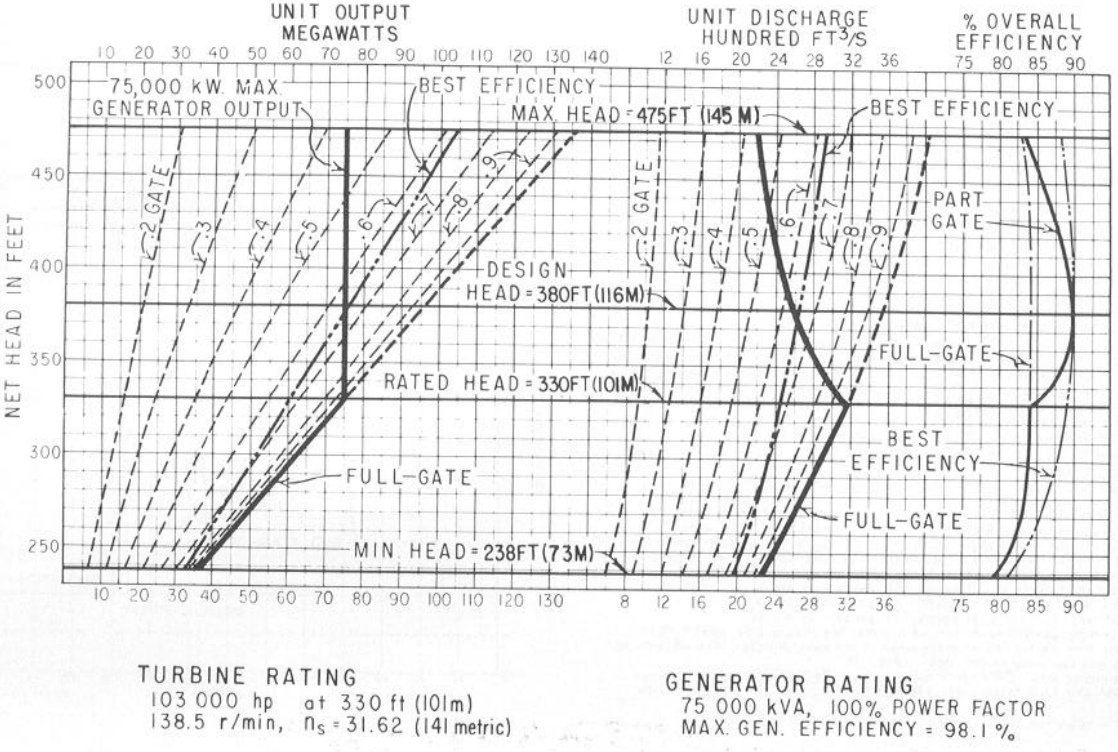


FIGURE 26.—*Performance curve—Shasta Powerplant.*

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:

$$P=0.0846\ QH\eta_g\eta_t, \text{ kilowatts,}$$

$$P=0.1134\ QH\eta_t, \text{ horsepower,}$$

where

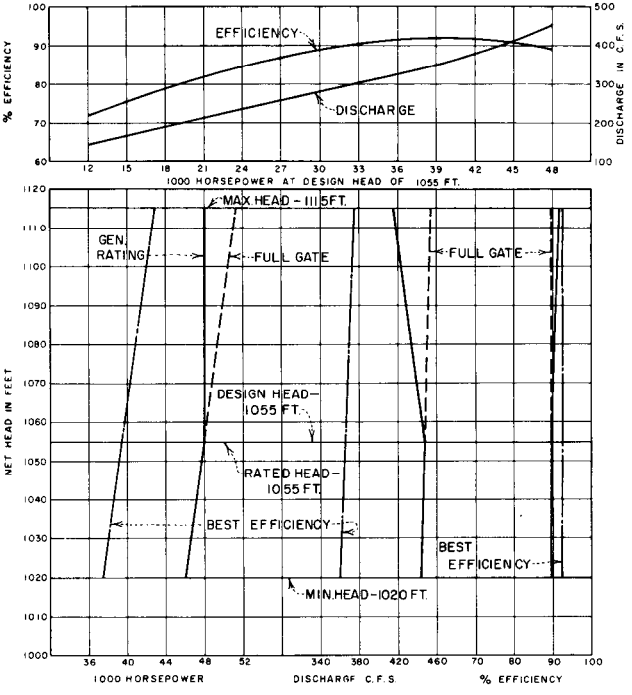
P=generator output and turbine output, respectively,
Q=discharge at H,
H=head,
η_g=efficiency of generator, and
η_t=efficiency of turbine.

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

FLATIRON POWER PLANT COLORADO-BIG THOMPSON PROJECT.
SPECIFICATIONS NO. 2996 & DS-3251 UNITS 1 AND 2 DATE April 17, 1950
TURBINE NAMEPLATE RATING: H.P. 48,000 ; HEAD 1055 FT; SPEED 514 R.P.M.
GENERATOR RATING IN KV-A 35,000 POWER FACTOR .90 PERCENT.

Turbine mfr. Pelton Water Wheel Co. Type Francis
Cost per unit f.o.b. factory $305,000 Weight *196,500 (EST.) lbs.
Cost per h.p. $6.36 Weight per h.p. 4.09 lbs.
Type of scroll case Spiral-Cast Steel Weight heaviest part 26,000 lbs.
Type of draft tube Elbow with plate steel liner-One pier
Weight of runner 10,500 lbs. Weight of rotating parts 25,500 lbs.
Weight of turbine parts including hydraulic thrust to be carried by generator thrust bearing 135,000 lbs. New; lbs. Worn rings.
Governor capacity in foot-lbs. 70,000 Pipe size 3 inches.
Gov. mfr. Woodward Governor Co. Time element 5 seconds.
Cost per unit f.o.b. factory $26,440 Weight 18,000 lbs.
Generator mfr. Westinghouse-Unit 1
Generator WR^2 1,792,000-Unit 1 lbs. at one foot radius.
Turbine WR^2 94,000 lbs. at one foot radius.
Regulating constant of unit ($R.P.M.^2 \times WR^2 \div$ Design H.P.) 10,380,000
n_s of runner 17.0 at 1055 ft. design head when delivering 39,400 h.p. (Best eff. gate).
N_s of runner 18.7 at 1055 ft. design head when delivering 48,000 h.p. (Full gate).
H.P. at 1055 ft (Design head) 48,000 ; at 100.0 percent of design head; 447 c.f.s.
H.P. at 1115 ft (Max. head) 51,200 ; at 105.8 percent of design head; 452 c.f.s.
H.P. at 1020 ft. (Min. head) 46,000 ; at 96.7 percent of design head; 442 c.f.s.
H.P. at 1055 ft (Mfrs. Rated Hd.) 48,000 ; at 100.0 percent of design head; 447 c.f.s.
H.P. at best efficiency equals 82.1 percent of h.p. at full gate.
Runaway speed at 1055 ft. hd. 950 r.p.m. equals 184.6 percent of normal speed.

DIMENSIONS OF TURBINE:
Unit spacing 41.25 ft. Dia. of shaft 19.0 inches.
Max. dia. of runner 9.92 ft. Dia. of cover plate 10.0 ft.
Dia. of gate circle 7.835 ft. Number of wicket gates 20
Height of distributor case 0.667 ft. Number of stay vanes 10
Dia. of scroll case inlet flange 4.5 ft. Dia. at top of draft tube 4.625 Ft=D_3.
Ouside radii of stay vanes 5.67 to 6.00 ft. Distributor ℄ Elev. 5457.0 Ft.
Distance from center line of distributor to top of draft tube 1.927 ft.
Depth of draft tube 12.17 ft. equals 263 percent of dia. D_3.
Length of draft tube 33.29 ft. equals 720 percent of dia. D_3.
Width of draft tube 18.92 ft. equals 409 percent of dia. D_3.
Distance from center line of turbine to center line of scroll case inlet 8.00 ft.
Distance from center line of distributor to minimum tailwater, (Elev. 5462.0 ft.) (One unit operating at full load) 5.0 ft.
Pressure regulator mfr. Pelton W.W. Co. Type Pelton Size 44 inches.
Cost per unit f.o.b. factory Included with turbine Weight 38,500 lbs.
REMARKS:
Placed in operation Unit 1-January 20, 1954; Unit 2-May 12, 1954
*Less relief valve and enery absorber weights



REV. 10-31-63

FIGURE 27.—*Hydraulic turbine data—Flatiron Powerplant.*

GRAND COULEE THIRD POWER PLANT COLUMBIA BASIN PROJECT.
SPECIFICATIONS NO. DS-6607 UNITS 19, 20 AND 21 DATE March 15, 1968
TURBINE NAMEPLATE RATING: H.P. 820,000 ; HEAD 285 FT; SPEED 72 R.P.M.
GENERATOR RATING IN KV-A 615,385 POWER FACTOR 97.5 PERCENT.

Turbine mfr. Willamette Iron and Steel Co. Type Francis
Cost per unit f.o.b. factory $ 6,512,414 Weight 6,700,000 lbs.
Cost per h.p. $ 6.67 Weight per h.p. 6.86 lbs.
Type of scroll case Welded T-1 Pl. Steel Spiral Weight heaviest part 560 TONS.
Type of draft tube Elbow - Two pier Stay Rings
Weight of runner 1,100,000 lbs. Weight of rotating parts 1,700,000 lbs.
Weight of turbine parts including hydraulic thrust to be carried by generator thrust bearing 4,100,000 lbs. New; 4,410,000 lbs. Worn rings.
Governor capacity in foot-lbs. 9,100,000 Pipe size 8 inches.
Gov. mfr. Baldwin - Lima - Hamilton Time element 8 seconds.
Cost per unit f.o.b. factory $ 184,277 Weight 130,000 lbs.
Generator mfr. Westinghouse Electric Corp.
Generator WR^2 2,542,000,000 lbs. at one foot radius.
Turbine WR^2 200,000,000 lbs. at one foot radius.
Regulating constant of unit ($R.P.M.^2 \times WR^2 \div$ Design H.P.) 14,550,000
n_s of runner 50.0 at 305 ft. design head when delivering 785,000 h.p. (Best eff. gate).
N_s of runner 55.8 at 305 ft. design head when delivering 976,000 h.p. (Full gate).
H.P. at 305 ft. (Design head) 976,000 ; at 100 percent of design head; 31,800 c.f.s.
H.P. at 355 ft. (Max. head) 1,217,500 ; at 116.4 percent of design head; 34,700 c.f.s.
H.P. at 220 ft. (Min. head) 568,000 ; at 72.1 percent of design head; 26,500 c.f.s.
H.P. at 272 ft. (Mfrs. Rated Hd.) 820,000 ; at 89.2 percent of design head; 29,600 c.f.s.
H.P. at best efficiency equals 80.1 percent of h.p. at full gate.
Runaway speed at 355 ft. hd. 138 r.p.m. equals 191.7 percent of normal speed.

DIMENSIONS OF TURBINE:
Unit spacing 119.0 ft. Dia. of shaft 96 inches.
Max. dia. of runner 32.58 ft. Dia. of cover plate 39.17 ft.
Dia. of gate circle 35.00 ft. Number of wicket gates 32
Height of distributor case 7.56 ft. Number of stay vanes 32
Dia. of scroll case inlet flange 34.67 ft. Dia. at top of draft tube 31.11 Ft=D_3.
Ouside radii of stay vanes to 22.46 ft. Distributor ℄ Elev. 946.00 Ft.
Distance from center line of distributor to top of draft tube 10.99 ft.
Depth of draft tube 80.05 ft. equals 257 percent of dia. D_3.
Length of draft tube 203.0 ft. equals 652 percent of dia. D_3.
Width of draft tube 99.0 ft. equals 318 percent of dia. D_3.
Distance from center line of turbine to center line of scroll case inlet 37.63 ft.
Distance from center line of distributor to minimum tailwater, (Elev. 958.0 ft.) (One unit operating at full load) 12.0 ft.
Pressure regulator mfr. None Type Size inches.
Cost per unit f.o.b. factory Weight lbs.
REMARKS:
Placed in operation

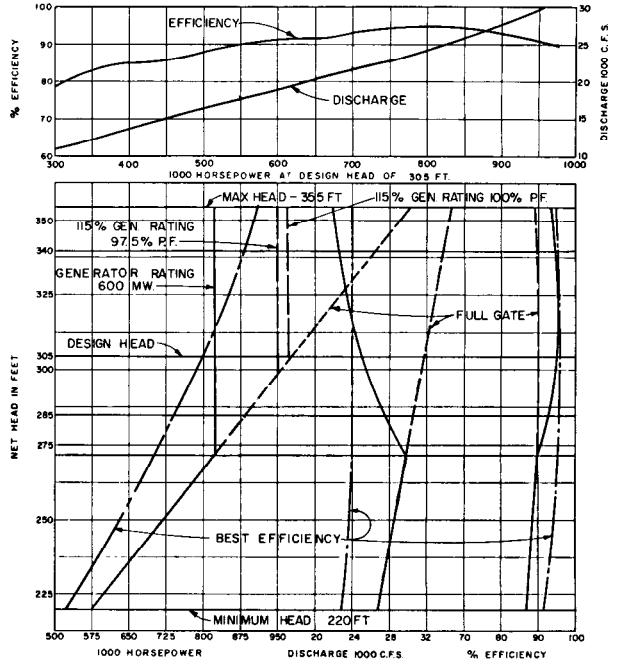


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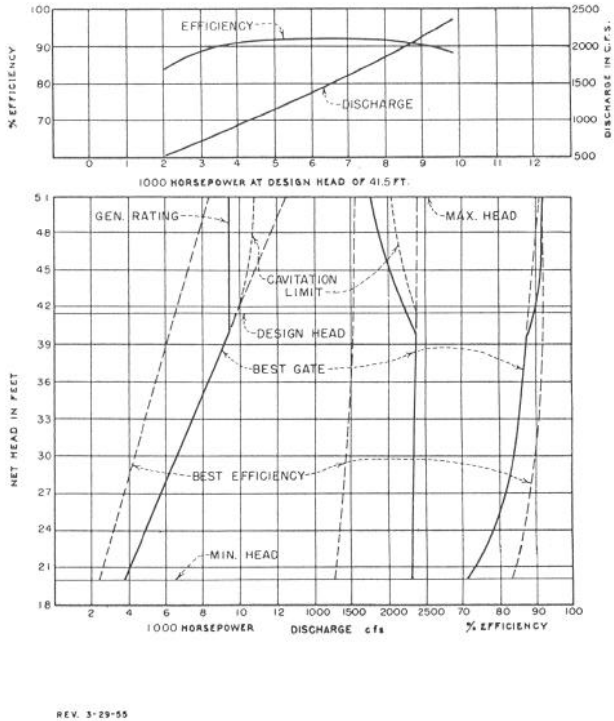
FIGURE 28.—*Hydraulic turbine data—Grand Coulee Third Powerplant.*



NIMBUS Power Plant AMERICAN RIVER DIV. - CENTRAL VALLEY PROJECT.
SPECIFICATIONS NO. DS-3697 UNITS 1 & 2 DATE MAY 1, 1952
TURBINE RATING IN H.P. 9400 RATED HEAD 41.5 FT. SPEED 150 R.P.M.
Generator rating in kv-a. 7500 Power factor 90 percent.
Turbine mfgr. S. Morgan Smith Co. Type Adj. blade propeller
Cost per unit f.o.b. factory $ 244,267.50 Weight 328,850 lbs.
Cost per h.p. $ 26.00 Weight per h.p. 33.4 lbs.
Type of scroll case Concrete - semi-spiral
Type of draft tube Elbow (two pier)
Weight of runner 31,368 lbs.
Weight of turbine parts including hydraulic thrust to be carried by generator thrust bearing 275,000 lbs.
Governor capacity in foot-lbs. 213,500 ft. Pipe size 2½ inches.
Gov. mfgr. Pelton Water Wheel Co. Time element 5 seconds.
Cost per unit f.o.b. factory $ 37,500 Weight 25,500 lbs.
Generator mfgr. Elliot Co.
Generator WR² 2,200,000 lbs. at one foot radius.
Turbine WR² 160,000 lbs. at one foot radius.
Regulating constant of unit (R.P.M.² x WR² ÷ H.P.) 5,650,000
n_s of runner 115.8 at 41.5 foot head when delivering 6600 h.p. (Best eff.)
N_s of runner 141.4 at 41.5 foot head when delivering 9850 h.p. (Best gate).
H.P. at 41.5 ft. (Design head) 9850 ; at 100.0 percent of design head.
H.P. at 51 ft. (Max. head) 12,600 ; at 122.8 percent of design head.
H.P. at 20.2 ft. (Min. head) 3750 ; at 48.2 percent of design head.
H.P. at best efficiency equals 67 percent of h.p. at best gate.
Discharge (Best gate) at 41.5 ft. hd 2360 c.f.s., at 51 ft. hd 2410 c.f.s., at 20 ft. hd. 2280 c.f.s.
Runaway speed at 47 ft. hd. 412 r.p.m. equals 275 percent of normal speed.
Dimensions of turbine:
Unit spacing 45 ft. Dia. of shaft 17 inches.
Max. dia. of runner 10.75 ft. Dia. of cover plate 11.17 ft.
Dia. of gate circle 13.42 ft. Number of wicket gates 16
Height of distributor case 4.42 ft. Number of stay vanes 16
Dia. of scroll case inlet — ft. Dia. at top of draft tube 10.92 ft. = D_3.
Outside radii of stay vanes to 8.835 ft.
Distance from center line of distributor to top of draft tube 6.75 ft.
Depth of draft tube 34.75 ft. equals 308 percent of dia. D_3.
Length of draft tube 43.00 ft. equals 394 percent of dia. D_3.
Width of draft tube 39.50 ft. equals 362 percent of dia. D_3.
Distance from center line of turbine to center line of scroll case inlet 4.08 ft.
Distance from center line of distributor to minimum tailwater elevation (One unit operating at full load) 4.00 ft.
Pressure regulator mfgr. None Type Size inches.
Cost per unit f.o.b. factory Weight lbs.
Remarks:
Placed in operation Unit 1 - May, 1955; Unit 2 - June, 1955
☆ 126,000 FT. LBS. for Gate servomotor 87,500 FT. LBS. For Blade servomotor
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FIGURE 29.—*Hydraulic turbine data—Nimbus 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:

$$\frac{Q_2}{Q_1}=\left(\frac{H_2}{H_1}\right)^{1/2}$$

$$\frac{P_2}{P_1}=\left(\frac{H_2}{H_1}\right)^{3/2}$$

$$\frac{n_2}{n_1}=\left(\frac{H_2}{H_1}\right)^{1/2}$$

For constant head:

$$\frac{Q_2}{Q_1}=\left(\frac{D_2}{D_1}\right)^{2}$$

$$\frac{P_2}{P_1}=\left(\frac{D_2}{D_1}\right)^{2}$$

$$\frac{n_2}{n_1}=\frac{D_1}{D_2}$$

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.

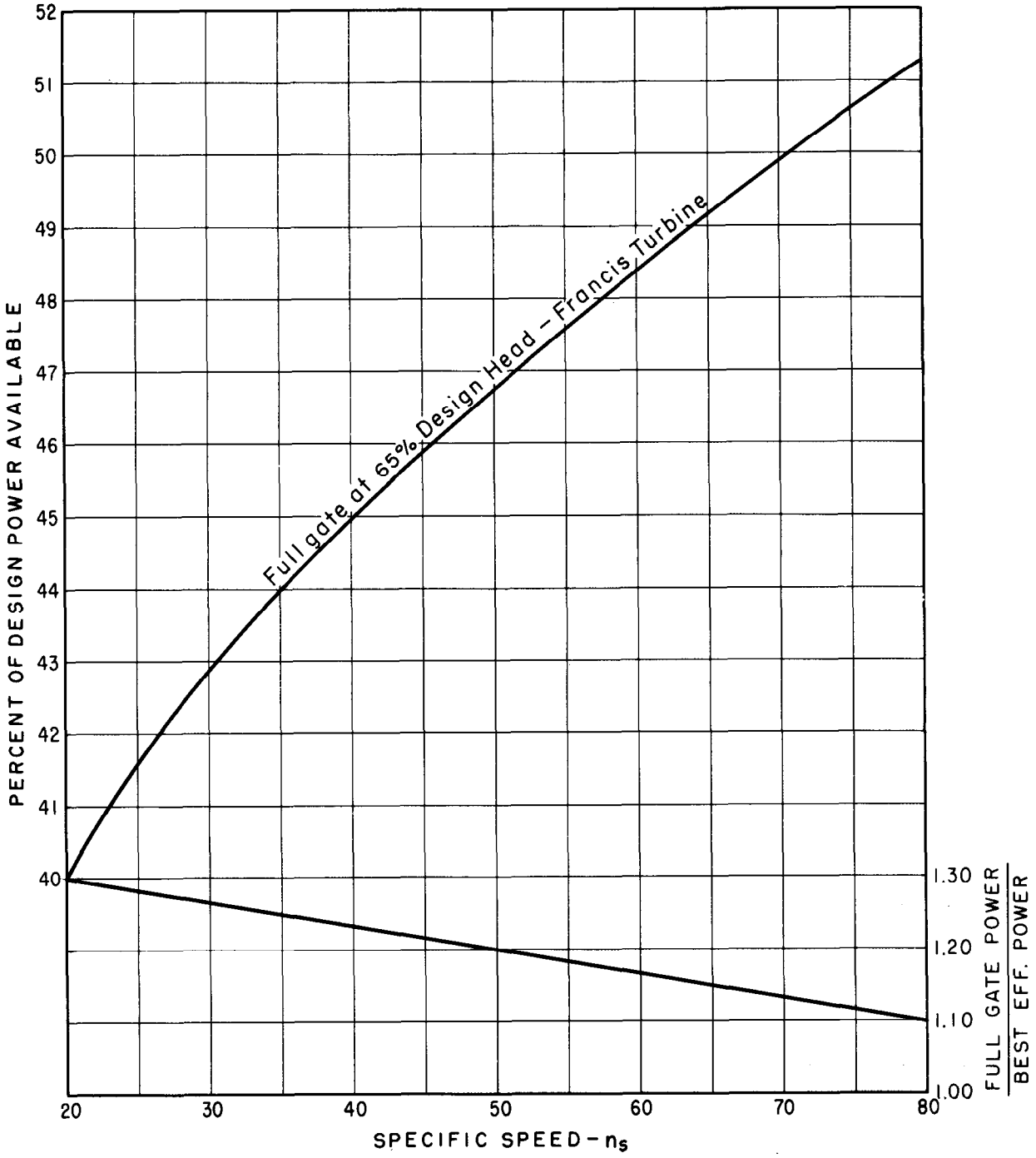

FIGURE 30.—*Turbine characteristics—Francis turbine—minimum head power.*

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.

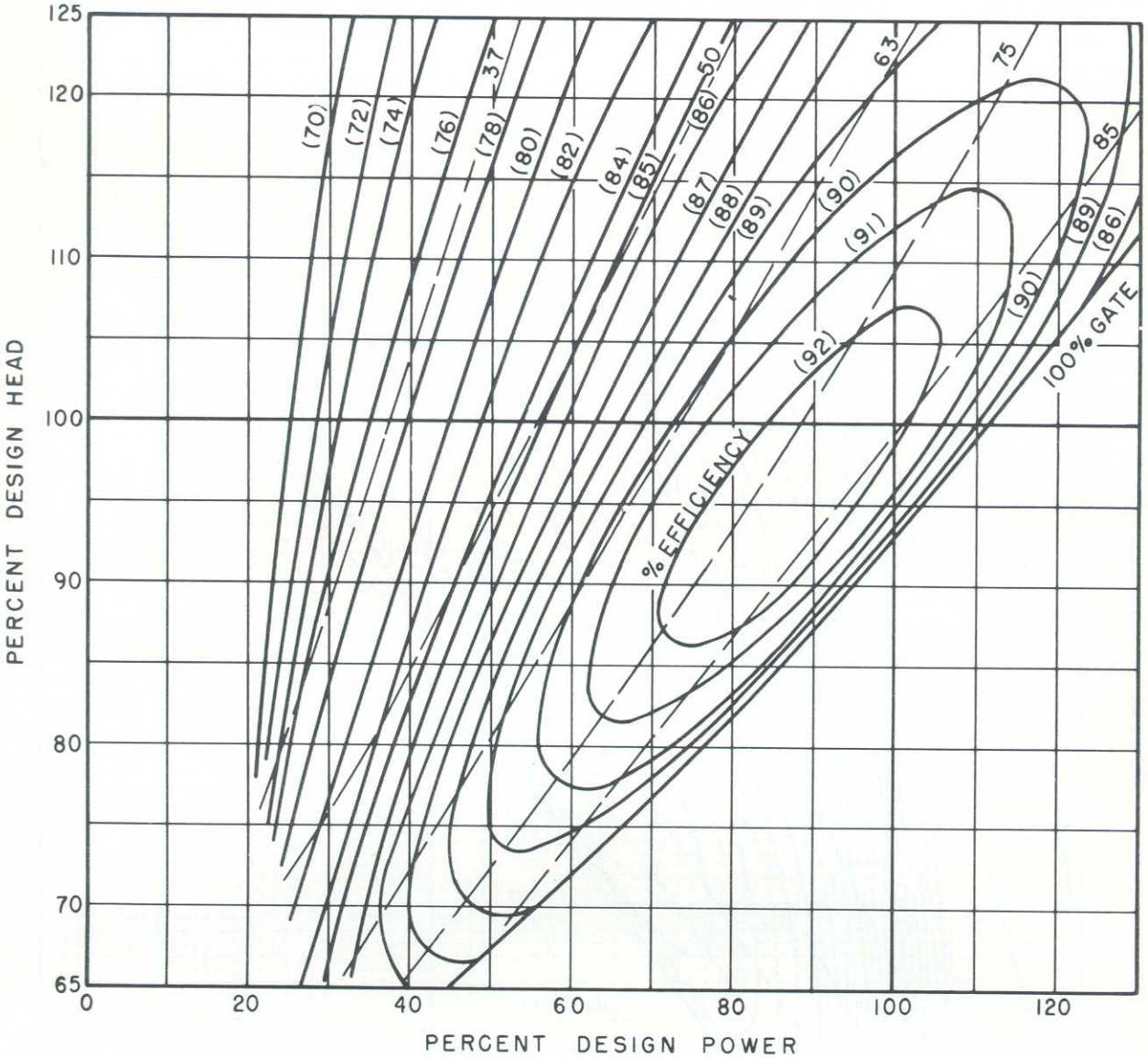

FIGURE 31.—*Francis turbine performance—head vs. power—*$n_s = 22$ *(98 metric).*

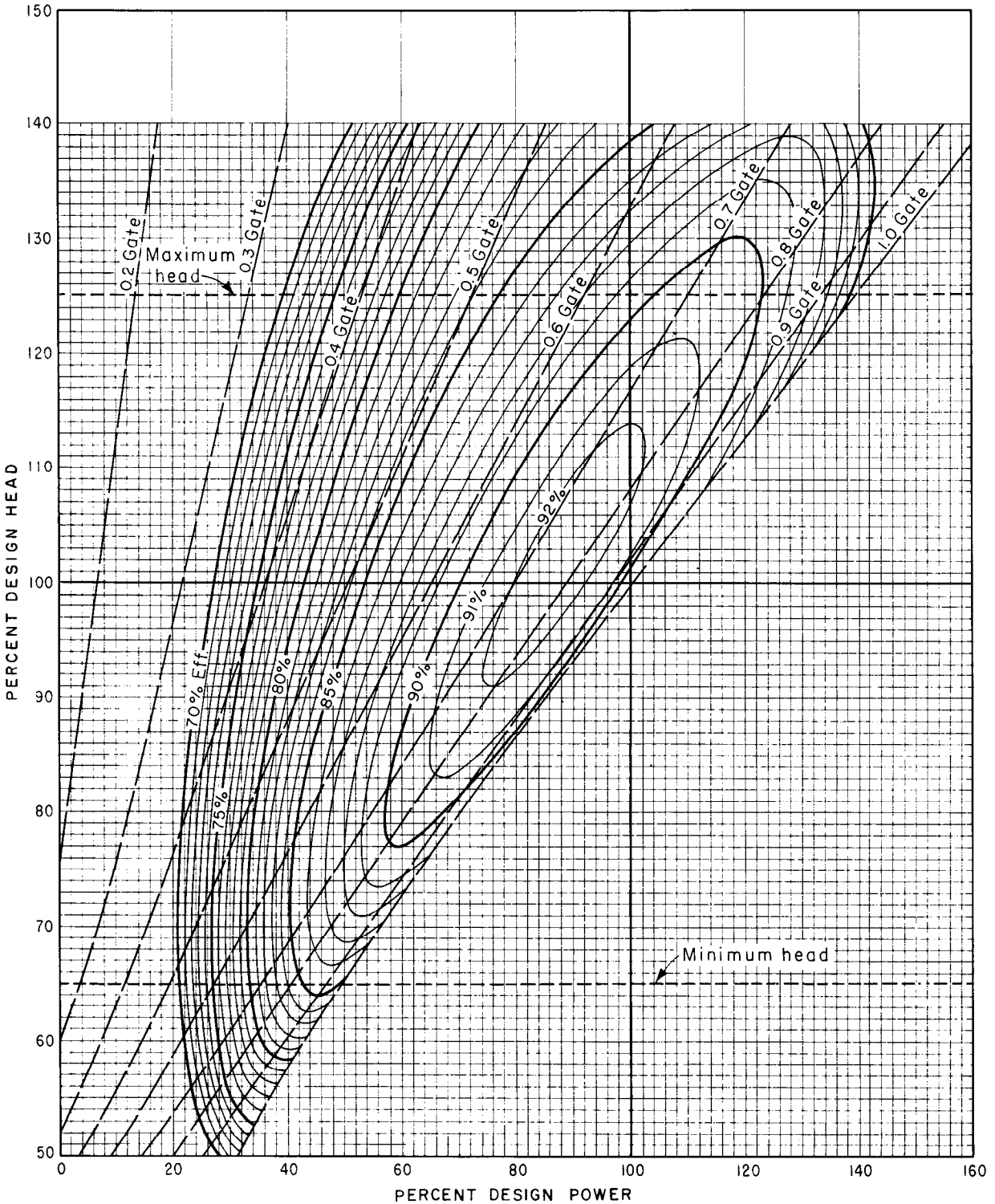

FIGURE 32.—*Francis turbine performance—head vs. power—*n_s*=48 to 75 (214 to 334 metric).*

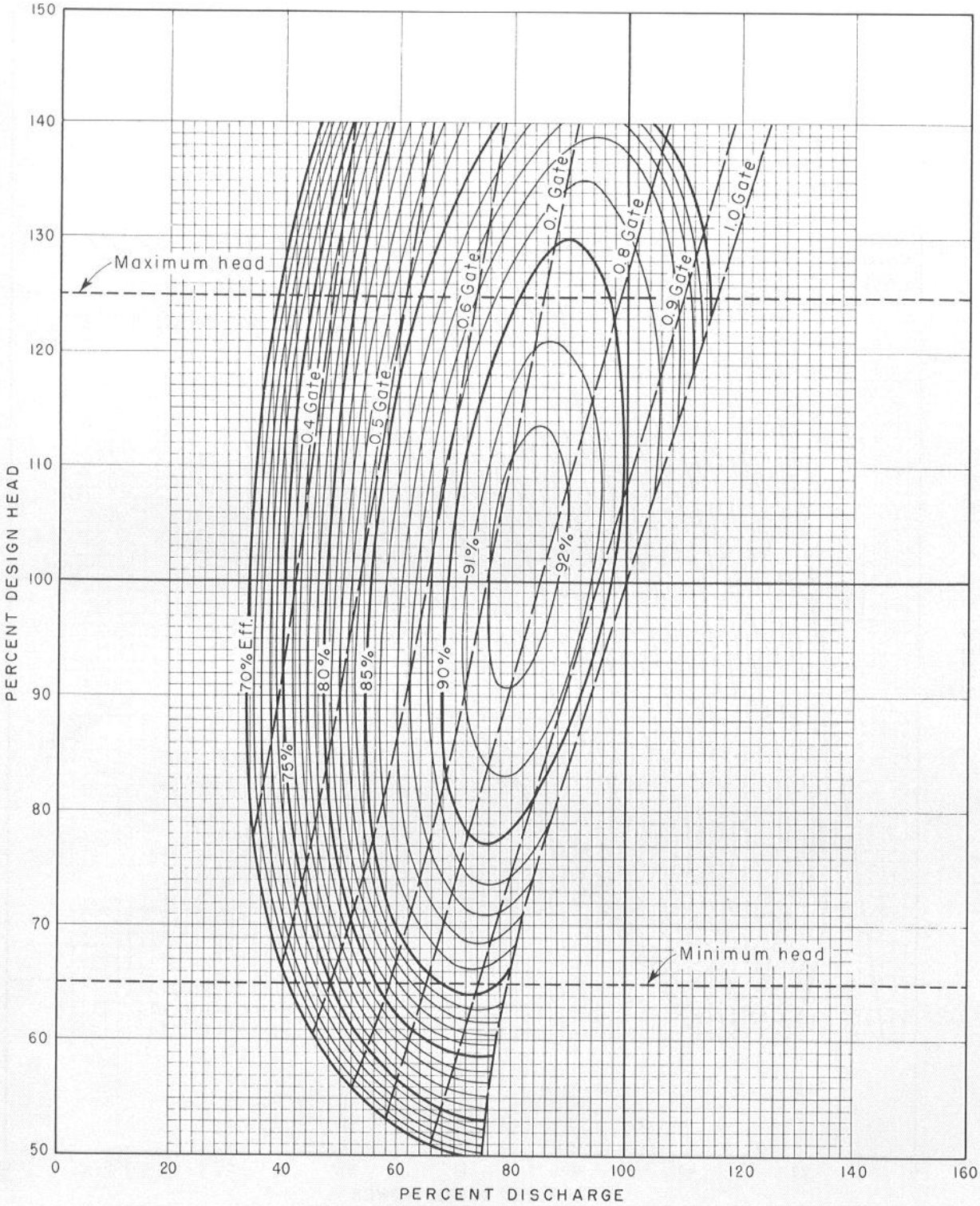

FIGURE 33.—*Francis turbine performance—head vs. discharge—*$n_s = 48$ *to 75 (214 to 334 metric).*

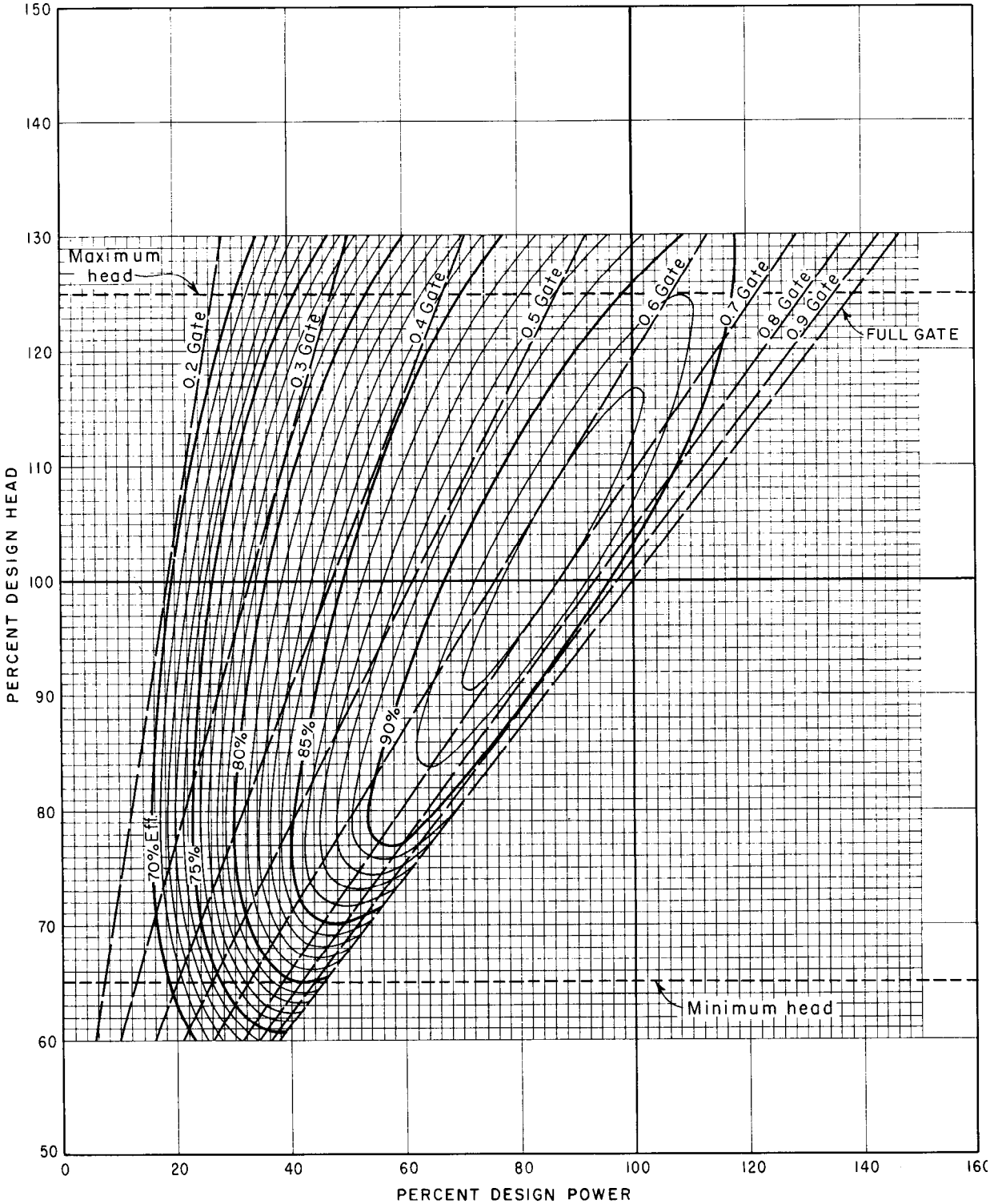

FIGURE 34.—*Francis turbine performance—head vs. power—*$n_s = 40$ *to* 48 *(178 to 214 metric).*

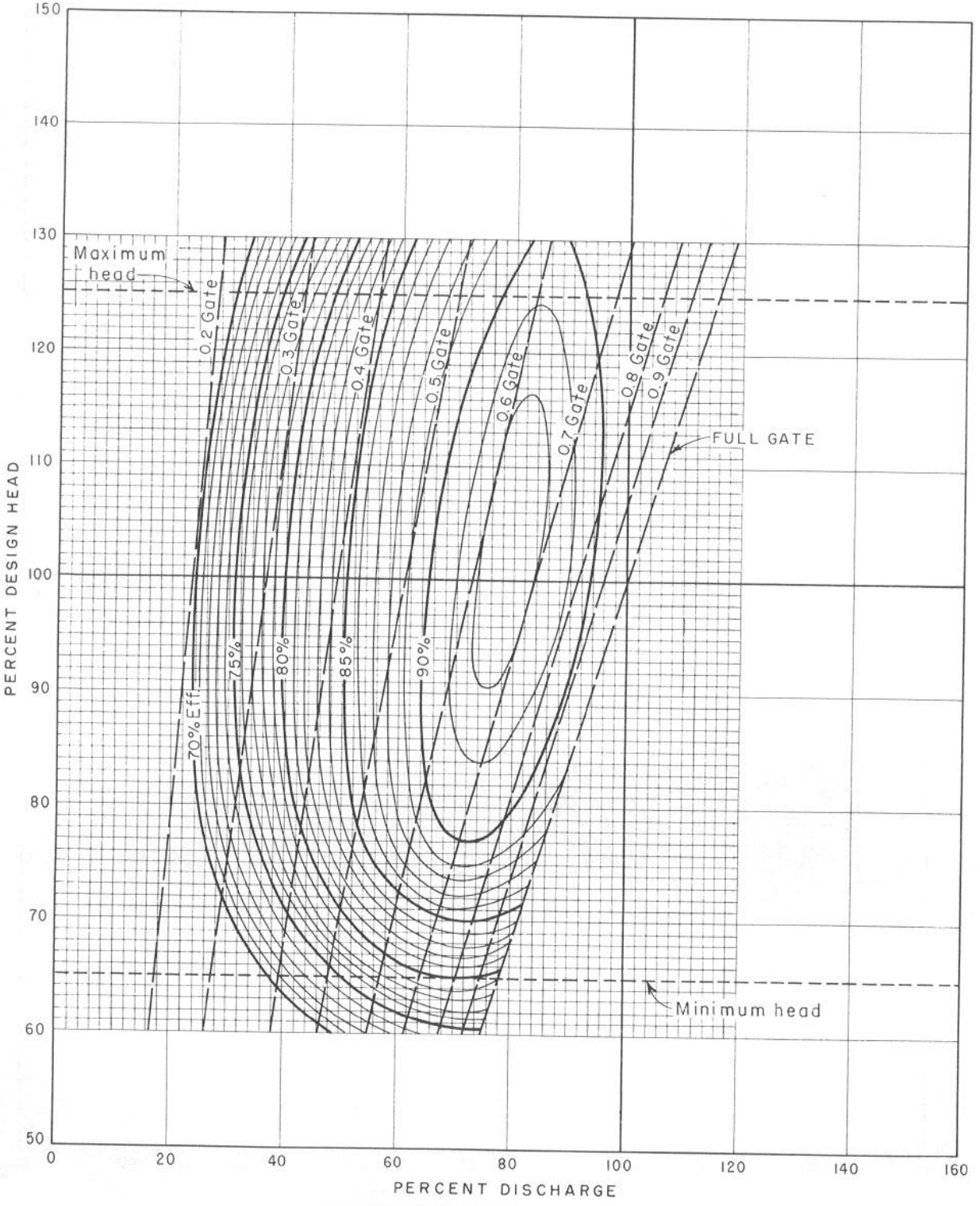

FIGURE 35.—*Francis turbine performance—head vs. discharge—*$n_s = 40$ *to* 48 *(178 to 214 metric).*

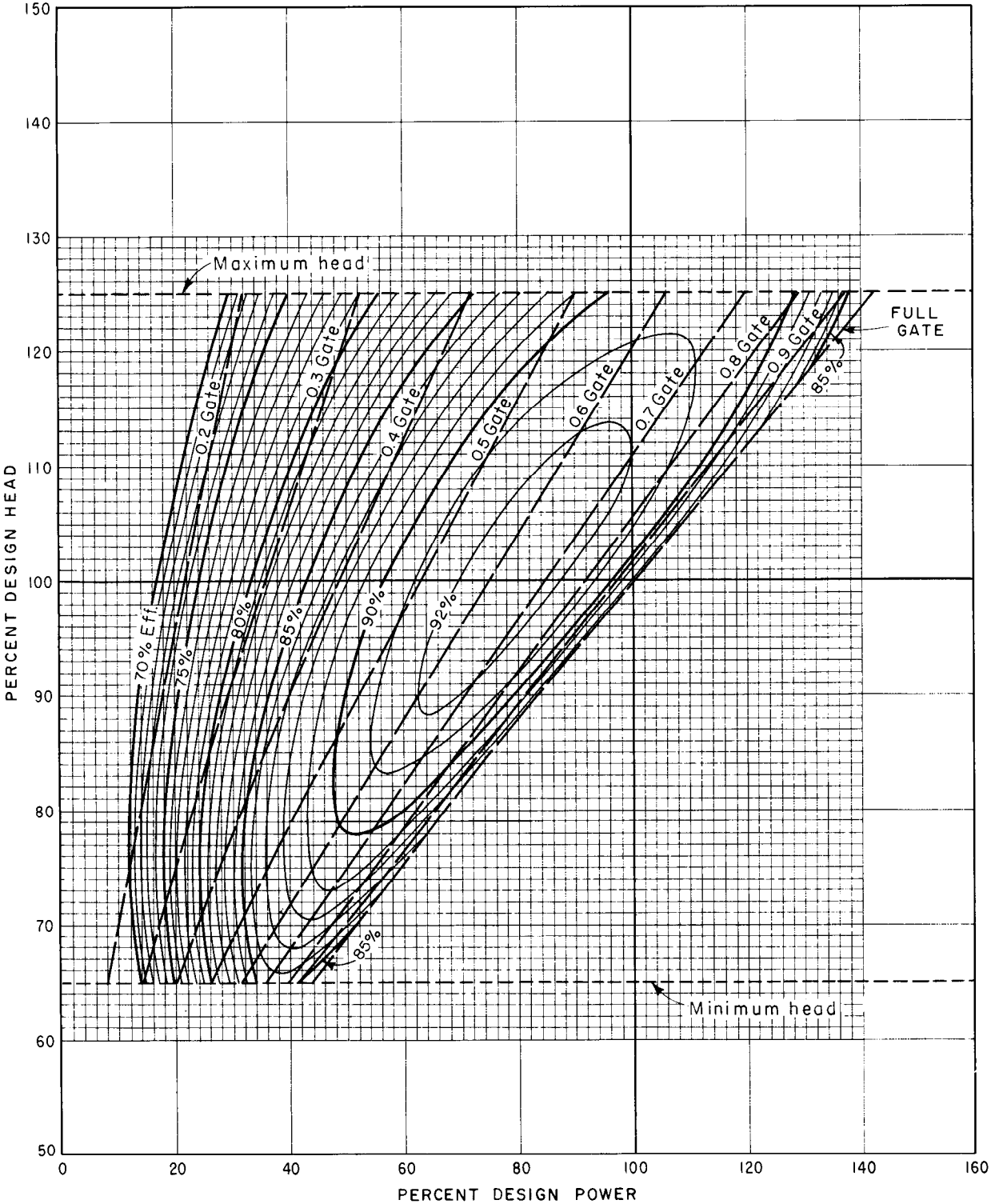

FIGURE 36.—*Francis turbine performance—head vs. power—*$n_s = 25$ *to 40 (111 to 178 metric).*

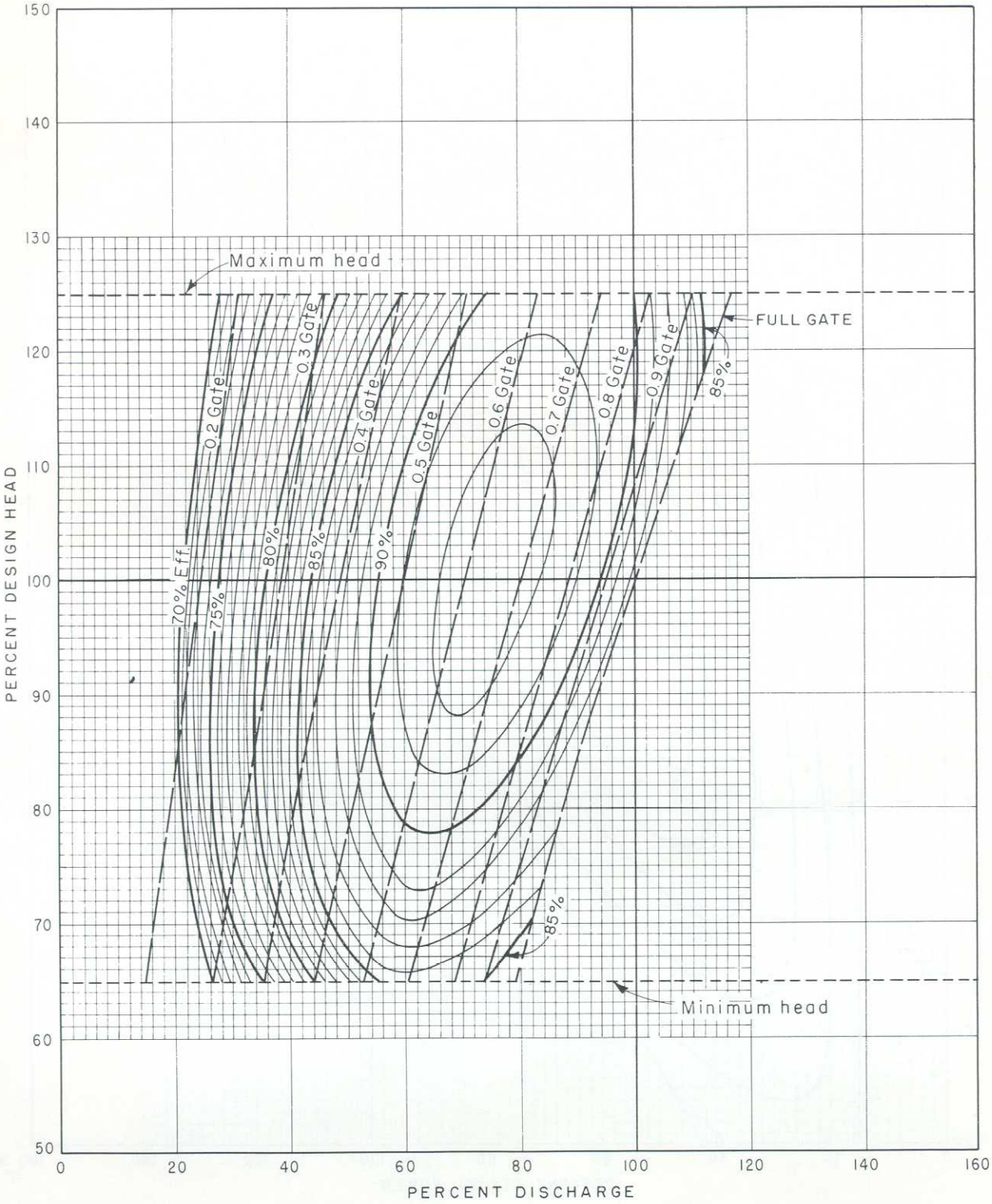

FIGURE 37.—*Francis turbine performance—head vs. discharge—*$n_s = 25$ *to 40 (111 to 178 metric).*

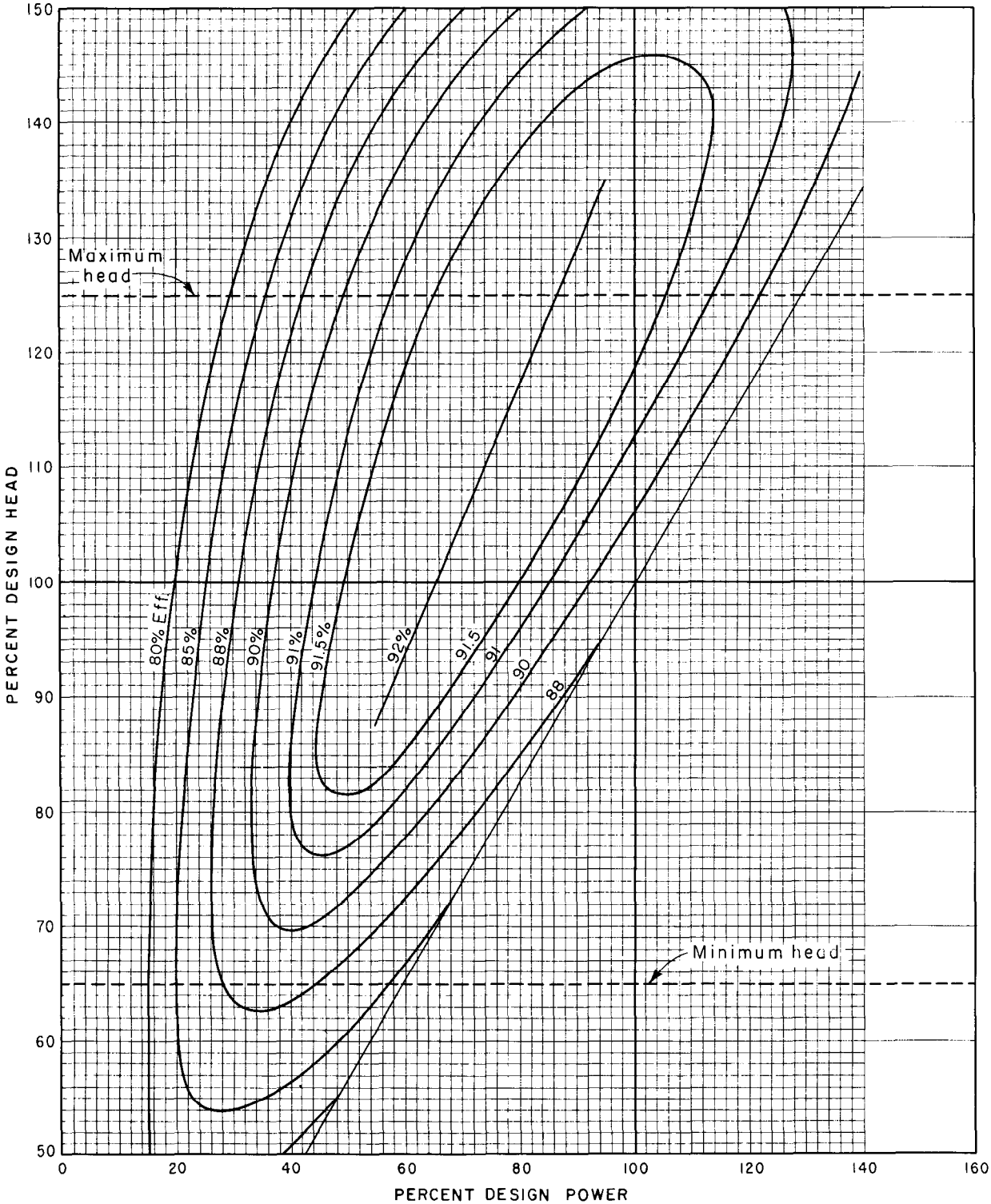

FIGURE 38.—*Adjustable blade turbine—head vs. power—*$n_s = 142$ *(632 metric).*

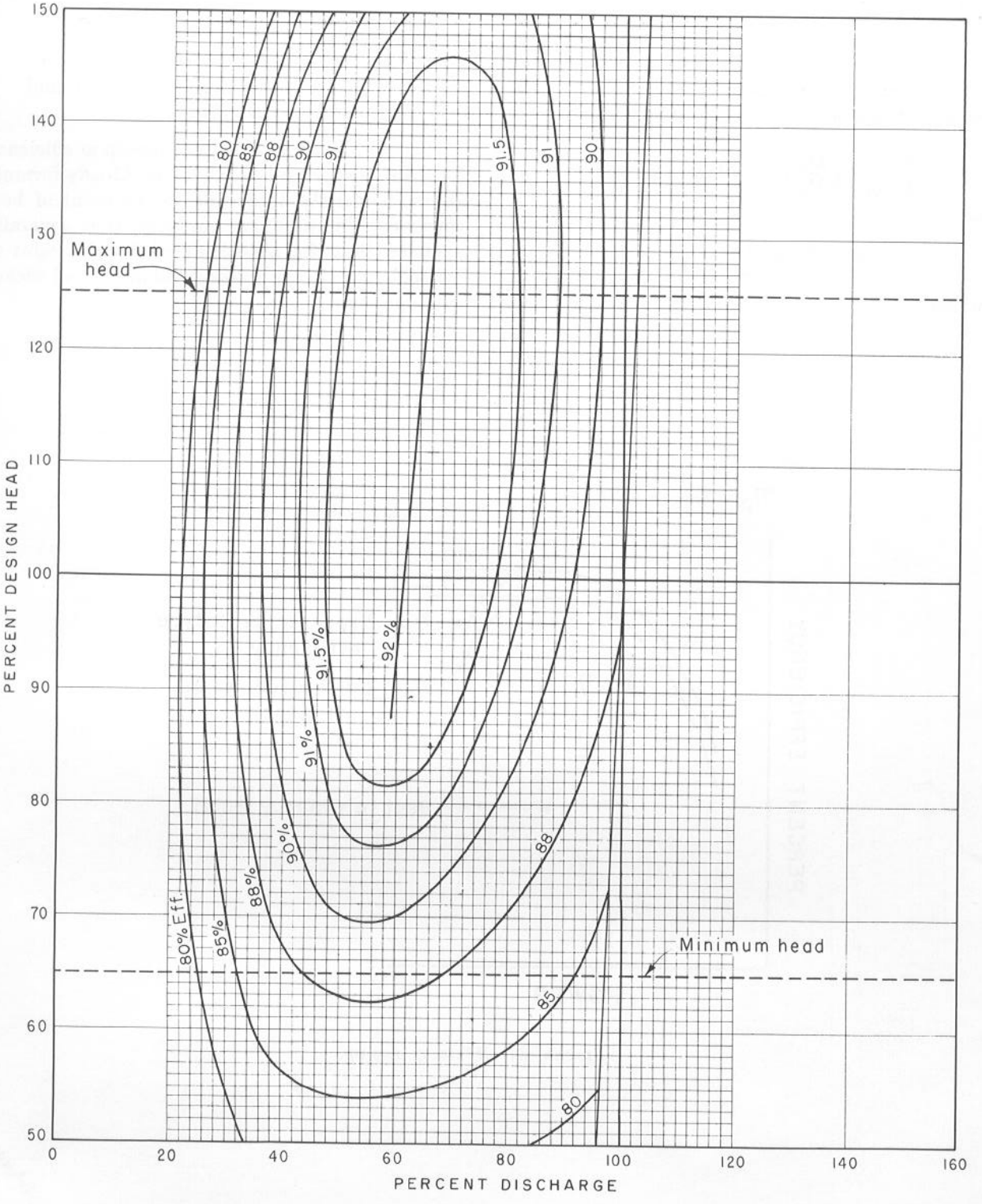

FIGURE 39.—*Adjustable blade turbine—head vs. discharge*—$n_s = 142$ (*632 metric*).

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

η_p=prototype turbine efficiency,
η_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.

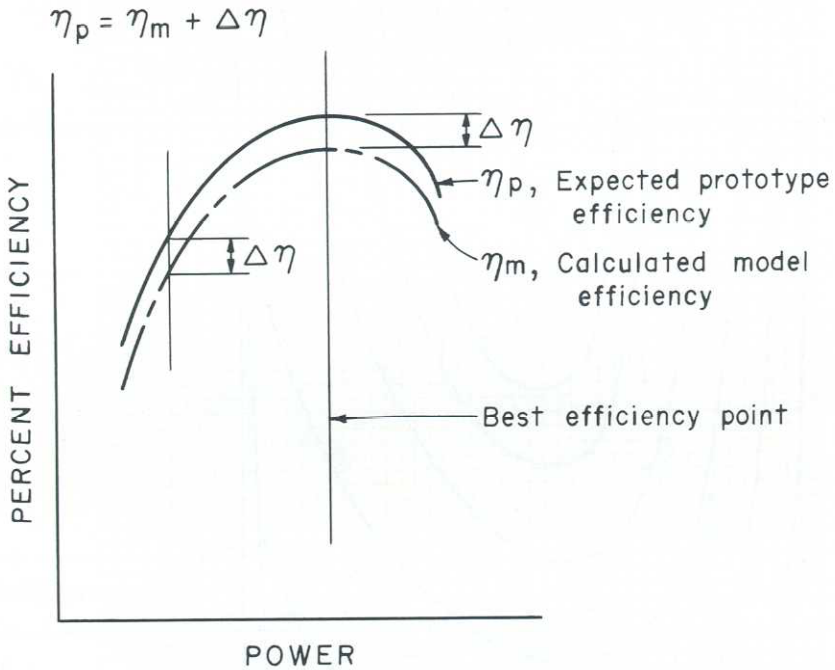

FIGURE 40.—*Efficiency step-up.*

Units of Power Measurement

When U.S. customary units are used:

$$P_d = 0.1134\ Q_d h_d \eta_d$$

where

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

When metric units are used, power will be expressed in metric horsepower or kilowatts, and:

$$P_d = 13.35\ Q_d h_d \eta_d$$

where

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

or

$$P_d = 9.804\ Q_d h_d \eta_d$$

where

P_d = turbine output, kilowatts (1 kw = 101.971 m•kg/s),
Q_d = full-gate discharge (at h_d), m^3/s,
h_d = design head, metres, and
η_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

η_d = design efficiency, percent

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