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No. 1110-2-317 15 December 1988

Engineering and Design
SELECTING REACTION-TYPE HYDRAULIC TURBINES AND PUMP TURBINES
AND
HYDROELECTRIC GENERATORS AND GENERATOR-MOTORS

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Engineering Technical
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ENGINEERING AND DESIGN
SELECTING REACTION-TYPE HYDRAULIC TURBINES AND PUMP TURBINES
AND
HYDROELECTRIC GENERATORS AND GENERATOR-MOTORS

1. Purpose. This letter provides advance criteria for selection of Reaction-Type Hydraulic Turbines and Pump Turbines and Generators and Generator motors. This criteria is to be used pending incorporation into an Engineering Manual.

2. Applicability. This letter applies to all HQUSACE/OCE elements and field operating activities having civil works hydroelectric design responsibilities.

3. Discussion. Development of this criteria has been in progress for several years and is published to insure that the experience and expertise of the several authors is not lost to the Corps with the retirement of these people. The criteria provides guidance on all of the factors pertaining to the selection, setting, and characteristics which must be understood in design of a conventional hydroelectric generating or pump-turbine plants. Criteria covering unconventional and small hydroelectric plants will be published at a later date. Emphasis is placed on the fact that manufactures recommendation and proposals must be sought and obtained in the equipment selection process, however, guidance contained herein will provide a basis for accepting manufacturers recommendation.

FOR THE DIRECTOR OF ENGINEERING AND CONSTRUCTION:

Encl

HERBERT H. KENNON
Chief, Engineering Division
Directorate of Engineering and
Construction

This ETL reissued to include previously missing pages (Appendices A-E).

CEEC-EE

DEPARTMENT OF THE ARMY
U.S. Army Corps of Engineers
Washington D.C. 20314

Engineering and Design

SELECTING REACTION-TYPE HYDRAULIC TURBINES AND PUMP TURBINES and HYDROELECTRIC GENERATORS AND GENERATOR-MOTORS

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CHAPTER 1

INTRODUCTION

1-1. PURPOSE. This manual has been prepared for use in planning and design leading to the selection and preparation of technical specifications for reaction turbines and pump-turbines, generators and generator-motors. The information included in this manual is not intended to eliminate the necessity or desirability of consulting with equipment manufacturers.

1-2. APPLICABILITY. This manual is applicable to all field operating activities having hydroelectric civil works design responsibilities.

1-3. REFERENCES.

a. CE 2201.01 HYDRAULIC TURBINES - FRANCIS TYPE

b. CE 2201.02 HYDRAULIC PUMP - TURBINES - FRANCIS TYPE

c. CE 2201.03 HYDRAULIC TURBINES - KAPLAN TYPE

d. CE 2202.01 HYDRAULIC TURBINE DRIVEN - ALTERNATING CURRENT GENERATORS

e. ANSI/IEEE Std 421.1-1986, "IEEE Standard Definitions for Excitations Systems for Synchronous Machines," available from IEEE, 345 East 47th St., NY, NY 10017.

1-4. DISCUSSION.

a. Corps of Engineers hydroelectric power plants are part of multi-purpose projects which develop power incidental to their major purposes of flood control and/or navigation. Corps projects may concurrently serve irrigation, recreation and water supply purposes.

b. Hydraulic turbines and pump-turbines are not off-the-shelf items and must be designed to suit the specific range of conditions under which they will operate. Selection of the most suitable hydraulic and electrical equipment requires careful study and investigation.

c. This manual includes procedures to be followed, model test data of reaction turbines and pump-turbines and other material useful in selecting the equipment and preparing the performance data to be included in technical specifications.

1-5. PROJECT PLANNING AND FIELD SURVEY STUDIES. These studies establish the following data:

a. Power capacity - dependable and rated.

b. Energy output.

c. Reservoir capacity and headwater curves.

d. Tailwater and afterbay capacity curves.

e. Minimum flow requirements.

f. Other use requirements.

g. Pumping requirements.

h. Preliminary selection of type and number of Units.

i. Heads - maximum, minimum and average.

j. Foundation conditions.

k. Special conditions under which the plant must operate.

1-6. GENERAL PRINCIPLES.

a. The function of a hydroelectric power plant is the conversion of potential energy (water falling over a distance or head) into mechanical energy (rotation of the turbine or pump-turbine shaft). This shaft in turn is connected to the shaft of a generator or generator-motor to convert mechanical energy into electrical energy.

b. In the pumping mode, the generator-motor drives the pump-turbine to pump water to a higher elevation so that it will be available when needed to operate the pump-turbine in the generating mode to produce electrical energy.

c. The quantity of water available for the production of power in the foot-pound-system is measured in cubic feet per second (cfs) and designated as Q. The vertical distance available is measured in feet and designated as H. The theoretical horsepower available or water horsepower (WHP) due to a quantity of water (Q) falling H feet is WQH foot pounds per second, where W is the specific weight of water in pounds per cubic feet, and the available water horsepower is:

$$WHP = \frac{w\,Q\,H}{550}$$

d. The amount of power that can be produced under practical working conditions is less than the theoretical amount. This is due to losses in the conveyance of water (including the tailrace), and the losses in the conversion equipment.

e. Conveyance losses show up as the difference between the gross head (H_g) on a plant and the net or effective head (H_e). For Francis and propeller type turbines, the net head is the difference in level between headwater and tailwater minus all frictional losses and minus the velocity head of the water in the tailrace. Friction losses occur as the water passes through the trash racks, intake, penstock and tunnel including bends, branching pipes, transitions and valves up to the entrance of the spiral case. The power delivered by the turbine to the generator is measured in horsepower (HP).

$$HP = \frac{w\,Q\,He\,Ep}{550}$$

where E_P is the efficiency of the prototype turbine. The kilowatt output of the generator is 0.746 E_G times the horsepower delivered to the generator shaft, where E_G is the efficiency of the generator.

f. In pumping, the head is the total head from suction pool to the discharge of the spiral case plus the conveyance losses, except that for Tube or Slant Axis turbines and low head vertical units with short intakes it is the pool-to-pool head.

1-7. <u>SIZE AND NUMBER OF UNITS.</u>

a. The capital cost per kilowatt of a hydroelectric plant of a given total capacity generally decreases as the number of units decreases. A minimum of two units is usually preferred, but in special cases one unit may be acceptable.

b. Size alone is not a determining factor in selecting the number of units to be installed. The most economical size and number of units can only be determined by a careful analysis of limitations and conditions. The following limitations, requirements and conditions must be carefully considered.

(1) Single unit plants have lower operating and maintenance costs, but service equipment, cranes, etc. will be more expensive.

(2) A new unit of larger size than any other in the system may necessitate additional system capacity.

(3) Character of the load that the plant is expected to supply and the flexibility of operation required.

(4) Requirement to supply an isolated load.

(5) Requirement to supply low flow releases.

(6) The need to install units of unequal size.

(7) Requirement for future units.

(8) Even or odd number of units. Electrical connections may dictate an even number of units.

(9) Shipping limitations (one piece runner, etc.).

(10) Foundation conditions.

(11) Requirements of the pumping cycle.

1-8. <u>TYPES OF TURBINES.</u> Modern hydraulic turbines may be classified as reaction turbines or impulse turbines.

a. Types of Reaction Turbines include:

(1) Francis.

(2) Fixed Blade Propeller.

(3) Adjustable Blade Propeller (Kaplan, Tube or Slant Axis, and Bulb).

(4) Fixed Blade Mixed Flow.

(5) Adjustable Blade Mixed Flow (Deriaz).

Water passages are enclosed and completely filled with water. The energy transfer from the water to the turbine runner is due to the pressure and change in direction of the water. Reaction turbines operate at heads up to 1600 feet or more. The setting, while usually vertical, may be horizontal or inclined.

b. Impulse turbines are suitable for operation at heads as high as 6000 feet. The water is open to atmosphere at all points beyond the nozzle and the transfer of energy from the water to the runner is due to the turning of the jet nearly 180 degrees by the buckets which are arranged around the periphery of the runner. The setting may be horizontal or vertical.

1-9. TYPES OF PUMP-TURBINES.

a. Pump-turbines are similar to reaction turbines, except that they operate in one direction of rotation as pumps and in the opposite direction as turbines. They consist of three principal types:

(1) Radial flow or Francis type.

(2) Mixed flow or diagonal flow.

(3) Axial flow or propeller type.

b. The mixed flow and Axial flow types include both fixed-blade and adjustable-blade machines.

1-10. MODEL TEST.

a. Hydraulic turbines and pump-turbines are not off-the-shelf items of equipment. They are designed to suit the head, power and pumping requirements of a particular site.

b. While manufacturers have models developed to cover a range in heads and capacities, modifications to an existing model or development of a new model design may be necessary to determine performance at a specified condition.

c. Model testing is necessary if the state of the art is to advance. New improved designs which permit more economical speeds, improved efficiencies and settings with relationship to tailwater need to be developed. The Corps of Engineers requires model tests to develop a runner most suitable to the requirements of the project and to confirm that the guaranteed performance will be met. As more model and prototype tests become available, more accurate results of model changes and model development can be predicted.

d. In some cases field tests are not possible to check guaranteed values of performance. In these cases, model tests are accepted as the guaranteed tests.

e. The allowable specific speed for a turbine or pump-turbine under given head conditions is dependent upon the setting with respect to tailwater, atmospheric pressure, water density and the vapor pressure of water. A manufacturer may have a family of curves for heads up to 1500 feet or more (depending on the type of turbine and the setting above or below tailwater). However, prototype tests should be made to validate his design and model tests.

f. The size of the model runner tested may vary with different manufacturers and the test results may be based on inlet, throat or discharge diameters. Corps specifications require the model runner throat diameter to be not less than 10 inches and further requires that the guaranteed model efficiency be based on a model with a runner throat diameter of 12 inches.

g. Efficiency of prototype turbines should be higher than that of models. The amount of increase to be expected will vary depending on the manufacturer and his experience. Therefore, an exact comparison of performance of two models by different manufacturers cannot be made.

1-11. EVALUATION OF EFFICIENCY.

a. The increase in efficiency to be expected from tests of identical models in different laboratories has been known to vary two percent.

b. The surface finish on runner and gates on a model has been known to vary the efficiency by as much as 1/2 percent.

c. Model test values can be repeated closer than 1/4 percent.

d. Manufacturing tolerances may result in a step-up between model and prototype that is different than expected.

e. Two units of the same design and identical within manufacturing tolerances, installed in the same plant have given test results differing by more than the probable error of testing.

f. For many years, european test codes as well as the International Test Code acknowledged the probability of error in instrumentation by a tolerance for output and efficiency of ± two percent. Guarantees of 92 percent are considered to be met if final computations of field test results showed 90 to 94 percent. The United States Test Code (ASME) does not provide any tolerance on guarantees and this has resulted in the lowering of guaranteed values by american manufacturers. These guaranteed values, depending on the size of the unit, have varied from 90 to 92 percent.

g. In a known case where high efficiencies were guaranteed with large penalties for not meeting guarantees included in the contract, a two percent decrease in efficiency was equivalent to more than half of the contract price.

h. Specifying too high an efficiency can result in excessively high bid prices.

i. Model tests are used as a means of providing the unit best suited for a particular project. The Corps specifies minimum efficiencies to be met for both model and prototype at specified outputs when field tests are to be made and only model efficiencies when field tests are not to be made.

j. Efficiencies are not evaluated in the comparison of bids. However, penalties for failure of the prototype to meet guarantees are included in the specifications.

1-12. DATA ON UNITS INSTALLED IN CORPS OF ENGINEERS PLANTS. Data on units of equipment installed in Corps of Engineer's Hydroelectric Plants is included in Appendix B. This data will be of assistance in selecting equipment, but it must be recognized that considerable improvement in design and performance has been made on some units in recent years and that foundation conditions may have imposed restrictions on selecting the speed and size of a unit, and the depth of the draft setting.

CHAPTER 2

TURBINE AND PUMP-TURBINE CHARACTERISTICS

2-1. SPECIFIC SPEEDS.

a. The basis for comparison of the characteristics of hydraulic turbines is the specific speed (N_{st}). This is defined as the speed in revolutions per minute (N) at which a turbine of homologous design would operate if the runner was reduced in size to that which would develop one horsepower under one foot of head.

b. The specific speed varies directly as the square root of the horsepower (HP) and inversely with five-quarters power of the head (H) in feet.

$$N_{s_t} = \frac{(N)\,(HP)^{1/2}}{H^{5/4}}$$

c. In the metric system (N_{st}) is the speed of a homologous turbine of a size to develop one metric horsepower under one meter head. The metric specific speed is equal to 4.446 times the specific speed in the foot-pounds system.

d. In general, for a given head and horsepower, the higher the specific speed, the higher the speed of the unit and the lower the overall cost of the installation. But there are limits on the specific speed of a runner for a given head and output. Too high a specific speed would reduce the dimensions of the runner to values that would cause excessively high velocities for the water discharge through the throat of the runner and draft tube. Too high a specific speed could reduce the runner structural dimensions and the rotating parts of the generator to such small dimensions that high stresses would make it uneconomical, if not impracticable, to design. Too low a specific speed would unduly increase the size and cost of the generator in order to maintain the WR^2 of the unit. Obviously, there are practical limitations to the range in specific speeds for any head.

e. For Francis turbines, the specific speed is indicative of the type and shape of the runner. A low specific speed runner (high head) has an inlet diameter greater than the discharge diameter while the reverse is true for a high specific speed runner (low head).

f. For propeller turbines, higher specific speeds for higher heads require an increase in the number of blades.

g. Normal N_{st} is defined as the specific speed for best efficiency and rated N_{st} is defined as the specific speed at rated capacity or guaranteed horsepower under the head for which the turbine is designed.

h. Pumping specific speed (N_{sp}) is the speed at which the runner would rotate if reduced geometrically to such a size that it would deliver one U.S. gallon per minute under one foot of head.

$$N_{s_p} = \frac{(N)\,(Q)^{1/2}}{H^{3/4}}$$

2-2. PERIPHERAL COEFFICIENTS (ϕ).

a. The peripheral coefficient, a dimensionless number used for convenience in plotting model performance curves, is the ratio of the peripheral velocity of the runner blades to the spouting velocity of the water.

$$\phi = \frac{\text{Peripheral speed of the runner (fps)}}{\text{Spouting velocity of water (fps)}}$$

At the runner throat:

$$\phi_{TH} = \frac{\pi \left(\frac{N}{60}\right)\left(\frac{D_{TH}}{12}\right)}{\sqrt{2g\,He}} = \frac{N\,D_{TH}}{1838\,H^{1/2}}$$

where

ϕ_{TH} = Peripheral coefficient at runner throat

N = Runner speed in revolutions per minute

D_{TH} = throat diameter of the runner in inches.

Note: While D may denote any representative dimension of the runner such as inlet, throat and discharge diameters, it is Corps practice to use throat diameter.

g = Acceleration due to gravity = 32.17 ft./sec^2

H_e = Effective head in feet.

b. The runner speed must be selected to match a synchronous speed for the generator (see Appendix A, Page A-6, "GENERATOR SPEED VS NUMBER OF POLES").

$$N = \frac{120\ Hz}{n}$$

where Hz = frequency in cycles per second and n = number of poles of the generators.

c. While, in general, higher runner speeds for a specified horsepower at a specified head should result in a lower first cost for a turbine, the speed may be limited by the cavitation tendency of the runner, the drop in peak efficiency over the normal range of operation, vibration, and by mechanical design of the turbine or generator. Higher speeds require a lower setting of the runner with respect to tailwater and are accompanied by increased excavation and structural costs. Higher speeds also reduce the head range under which the turbine will operate satisfactorily.

d. Pump-turbines are more subject to cavitation in the pumping mode than in the generating mode. Therefore, the pumping mode determines the setting of the runners.

2-3. SETTING OF TURBINE AND PUMP-TURBINE.

a. The setting of the turbine or pump-turbine is very important. Too low a setting would result in unnecessary excavation and structure costs. Too high a setting could result in excessive cavitation of the runner buckets or blades with a resulting loss in efficiency and increased operating and maintenance costs.

b. The setting of a turbine or pump-turbine can best be determined by the consideration of the Thoma cavitation coefficient Sigma (σ).

$$\sigma = \frac{H_b - H_v - H_s}{H_e}$$

where H_b= Barometric pressure head at elevation of the runner above mean sea level

H_v= Vapor pressure head in feet at water temperature

H_s= For Francis runners is the distance from the lowest point on the runner vanes to tailwater in a vertical shaft unit, and the distance from the highest elevation of the runner band to tailwater for horizontal units.

H_s= For fixed blade and Kaplan runners is the distance from the center line of the blade trunnion to tailwater for vertical units and from the highest elevation of the blades to tailwater for horizontal and inclined units.

H_s= For diagonal flow runners is the distance from the bottom of the gate to tailwater for vertical units and from the highest elevation of the blades to tailwater for other units.

H_e= Net or Effective head on the turbine in feet.

H_s may be positive or negative, depending on whether or not the referenced point on the runner is above or below tailwater. When the referenced point is below tailwater, it is negative. If "a" is the distance from the center line of the turbine distributor to the lowest point on the runner vanes for Francis units and to the centerline of the blades for propeller-type runners, then the distance from the centerline of the distributor to tailwater for H_s positive is $(a + H_s)$, and for H_s negative is $(a - H_s)$.

c. It is customary for manufacturers to add a safety allowance to the cavitation coefficient Sigma (σ).

$$\sigma = \frac{H_b - H_v - H_s - \text{Safety}}{He}$$

2-4. CRITICAL SIGMA (σ_c).

a. Over the years, since facilities for making cavitation tests have been available, there have been several methods proposed and used for determining critical sigma from model tests. There has been no fixed agreement on a standard method of determining critical sigma and in using manufacturers' critical sigma curves. It is important that the method used in determining critical sigma for a particular model be clearly established as a manufacturer may have used a different method of determining critical sigma, depending on the method in use at the time of the test.

b. In some model tests, the cavitation limits were considered to be at the points where power drops off and the discharge increases, thus decreasing the efficiency.

c. In other model tests, critical sigma was considered to be the value obtained at the point of intersection of the constant horizontal HP or Q (pump) curve with the slope of the line under cavitating conditions.

d. The International Code for Model Acceptance Tests of Hydraulic Turbines, IEEE Publication 193, gives the following three definitions of sigma:

(1) σ_0, the lowest value of sigma for which the efficiency remains unchanged as compared to non-cavitating conditions,

(2) σ_1, the lowest value of sigma for which a drop of one percent in efficiency is attained compared to a non-cavitating condition, and

(3) σ_s, Standard Sigma, the value sigma at the intersection of the constant efficiency line (non-cavitating) with the strongly dropping straight line along which measuring points align themselves for a high cavitating degree.

e. The Corps of Engineers Specifications defines "the Critical Sigma of the turbine or pump turbine for such desired turbine output or pump capacity and head shall be the sigma corresponding to the tailwater level of such tests which results in a one percent decrease in efficiency or turbine output, or pump power input which ever occurs first." (See CE 2201.01, .02 and .03, paragraph MT-4.5).

f. Because of the shape of some model sigma curves, considerable judgment is necessary in determining critical sigma.

g. Prototype experience is necessary to determine the factor of safety to include with H_s and also how much prewelding of the runner blades can be used as a trade-off against deeper submergence.

2-5. PERFORMANCE CURVES.

a. Turbine prototype performance curves are plots of efficiency and discharge versus horsepower for various heads and gate openings and are based on laboratory test data of a model homologous to the prototype with regards to runner and water passages.

b. The power is stepped up from the model by the formula:

$$HP_p = HP_m \left(\frac{D_p}{D_m}\right)^2 \left(\frac{H_p}{H_m}\right)^{3/2}$$

c. The turbine discharge, neglecting any step-up in efficiency, may be calculated by inserting the value of horsepower calculated from the formula under (b) above into the formula for turbine horsepower $HP = WQHE_m/550$. H is the net or effective head and E_m is the model efficiency.

d. The expected efficiency of the prototype turbine is the model efficiency plus not more than 2/3 of the step-up in efficiency $(E_p - E_m)$ as determined by the Moody formula where E_m is the maximum model turbine efficiency at best speed or phi ($\emptyset$). The allowable step-up in efficiency is added to all efficiency points to obtain the expected prototype corrected efficiencies, (E_c).

$$E_p = 100 - (100 - E_m)\left(\frac{D_m}{D_p}\right)^{1/5}$$

$$E_c = E_m + \left(\frac{2}{3}\right)(E_p - E_m)$$

e. No step-up in power is permitted by the guide specifications however the corrected efficiency is used in calculating prototype discharges.

$$Q = \frac{550\ HP}{w \cdot H_e\ E_c}$$

f. Pumping performance curves are plots of efficiency, head and horsepower versus discharge at various gate openings and are based on laboratory test data for a model homologous to the prototype with regards to runner and water passages. For pumping, unless otherwise stated, the head is the total head from the suction pool to the discharge of the spiral case. The prototype head and discharge capacity values are stepped up from the model by the affinity laws and the capacity values so determined should not be less than the guaranteed values. The pump corrected efficiencies are obtained by adding the allowable step-up in efficiency to all efficiencies points. The expected head-capacity curves are developed using corrected capacity values which are the values stepped up from the model multiplied by the ratio E_c/E_m. The expected efficiency capacity curves are developed using the corrected efficiency and capacity values and the horsepower values used in developing the expected horsepower-capacity curves are computed using the formula HP = $wQH/550E_c$, where Q, H, and E_c are taken directly from the expected performance curves. The maximum pump input horsepower determined from the curves should not exceed the maximum pump horsepower permitted by the specifications.

2-6. MODEL-PROTOTYPE RELATIONSHIPS. Affinity laws and model to prototype relationships for turbines and pump turbines are included in Appendix A.

2-7. GUARANTEES.

a. When available, previous model tests can be used as the basis for guarantees, the guaranteed efficiency values should be set 1/4 percent less than the indicated model efficiencies. See also Paragraph 1-11.

b. Likewise, horsepower guarantees should be set two percent less than the values shown on the expected Horsepower vs. Efficiency curves for the prototype.

CHAPTER 3

FRANCIS TURBINES

3-1. GENERAL USE. For many years, Francis turbines were used for low heads. Today they are in general use for heads from 75 feet up to 1600 feet while propeller type turbines have replaced Francis type turbines at the lower heads.

3-2. SPECIFIC SPEEDS.

a. Specific speeds for Francis turbines range from 20 to 90 and is obtained by changing the design proportions of the runner. A general discussion of specific speed is presented in 2-1.

b. A low specific speed Francis runner has a larger entrance diameter than discharge diameter. For a specific speed of approximately 42, the inlet diameter is approximately equal to the throat and discharge diameters. For higher specific speeds, the inlet diameter becomes smaller than the throat and discharge diameters. Also the discharge diameter is larger than the throat diameter.

c. The specific speed (N_s) will remain constant for any other size or head for the same design and the corresponding speed for another homologous runner.

d. Care must be taken when using specific speed values to insure that they are being correctly used. The best efficiency at rated head for a Francis turbine is matched at 85 - 90 percent of the generator KW rating. the normal KW rating of the generator, the horsepower equivalent of which is used in calculating the rated specific speed.

e. In the process of selecting a turbine for a specific installation, the specific speed should also be determined using the lowest head at which the maximum power must be developed (generator KW rating). This will give the highest N_s under which the unit must operate and may dictate the selection of the runner. When the lowest head is appreciably lower than the average operating head and when the power required is exceptionally high in comparison to the requirement under normal head, it may be necessary to install an oversize turbine to meet the low head capacity requirement. In this case the turbine shaft may be sized to meet the generator rating with the provision that the turbine gate openings be restricted when operating at heads where if the gate openings were not restricted, the generator rating would be exceeded. The same head and gate opening restrictions apply to turbines where increase in heads under flood conditions could cause a turbine output in excess of the generator rating.

f. For many years, hydraulic laboratories did not have the facilities for testing the cavitation characteristics of Francis runners. Therefore the cavitation characteristics were estimated on the basis of experience with installations of similar types. During this period a value of 632 was used for K in $N_s = K/H^{1/2}$. In 1951 the manufacturers of hydraulic turbines recommended a K value of 650 to be used in the above formula on the basis that the vertical Francis-type turbines could usually be set with the centerline of the distributor about eight feet above tailwater at sea level. More recent experience indicates an economic advantage for smaller, higher speed units with deeper settings consistent with a K value of approximately 700. This relationship is shown on Figure 1, Appendix C and is recommended for preliminary studies.

3-3. DEVELOPMENT OF PROTOTYPE PERFORMANCE CURVES FROM MODEL TESTS.

a. Model test curves covering a wide range of specific speeds are shown on Figures F1 through F8 in Section I, Appendix D. This method of representation is commonly referred to as "oak tree" or performance hill. The latter designation derives from the fact that the figure is three-dimensional, as each constant efficiency contour represents a coordinate point in the Z-direction perpendicular to the plane of the paper. All data has been reduced to unit values corresponding with D_{TH} = 12 inches (one foot) and head, H = one foot. The ordinate is unit horsepower, HP_1, and the abscissa is peripheral speed coefficient, ϕ_{TH}. All efficiencies are based on D_{TH} = 12 inches. The indicated specific speed is referred to the point of maximum efficiency. Some cavitation characteristics are shown on Figures S1 through S5, Section IV, Appendix D.

b. The significant characteristics of the eight designs are compared on Figure F9 of Appendix C. A number of other designs have been included to aid in the correlation of the data with respect to specific speed. The curves may be used for preliminary selection of the runner throat diameter, speed, design discharge and runner setting. The curve of critical runner sigma is based on a horsepower that is 15 percent greater than the horsepower at best efficiency. The curve must not be used for off-best phi conditions. For studies requiring more complete information regarding the turbine dimensions and performance, a selection must be made from one of the eight designs.

c. Pertinent dimensions of the turbine parts and water passages, expressed as a ratio to D_{TH}, are shown in Tables 1 and 2, and Figure 4 of Appendix C.

d. The following steps are made to select a turbine:

(1) Given the horsepower corresponding to 85 - 90 percent generator rating at rated head, a preliminary value for N_S can be selected from Figure 1, Appendix C. A design is selected from Figures F1 through F8 with a specific speed which most nearly approximates the preliminary value. The design specific speed is used in the ensuing calculations. Speed N is determined from $N = N_S H^{5/4}/HP^{1/2}$, but N must be adjusted to a synchronous speed. It is usually necessary to investigate three synchronous speeds in order to arrive at the most overall economic speed.

(2) Having selected a speed, a preliminary runner diameter may be determined using the selected design and the relationship of $\phi_{TH} = N D_{TH}/1838 H^{1/2}$ and D_{TH} may be adjusted to get ϕ_{TH} at rated head to be at or near best gate. Also, it may be necessary to change the runner diameter so the HP_1 picked off from the performance hill for the phi values corresponding to other head conditions will give the required prototype horsepower. This may necessitate a change in speed with a corresponding shift away from best phi at rated head conditions. This includes calculating ϕ_{TH} for the minimum head conditions at which the capacity value of the unit or units is based, and the horsepower output at this head. It may be necessary as stated above to readjust N and D_{TH} to get the necessary HP_1 when stepped up to give the required prototype horsepower at the minimum head condition. See also comments under 3-2(e) regarding the necessity of installing an oversize turbine to meet the low head capacity requirements.

(3) Performance curves may now be developed from the model tests using the appropriate model - prototype relationships included in Appendix A.

e. As previously pointed out, hydraulic turbines are not off-the-shelf items of equipment. Model tests previously made and prototypes of models test are indicative of the performance that can be expected and the turbine manufacturer can alter or design a model based on experience to meet the requirements specified for a particular procurement. This accounts for some of the scatter of points shown on Figure F9 of Appendix D.

f. For specific speeds N_S = 33 and below, increasing the vent (i.e. opening between buckets) will permit increasing the power and shifting the point of best efficiency to the right. Decreasing the vent will reduce the power and shift the point of best efficiency to the left. Increases or decrease in vent opening as much as 15 to 20 percent may be made. A small percentage increase in the inlet diameter may also be possible as a means of shifting the point of best efficiency and slightly reducing the power. Increasing the vent openings too much may result in a loss of efficiency.

g. For specific speeds greater than 33, the model may be changed by increasing or decreasing the vent opening up to approximately 10 percent. Small percentage increase and decrease in inlet diameter may be possible for runners with specific speeds at best gate up to N_s of approximately 60. For specific speeds above 60, a small increase in inlet diameter is usually permissible. These are the means by which models can be adjusted to give desired project performance.

h. Increasing the vent openings increases the power of the runner but may result in a drop in efficiency. Too large an increase in vent opening could cause the power and efficiency to drop off too sharply.

i. It should be noted that N_s for best gate is approximately 7-1/2 to 11 percent smaller than N_s for full gate at rated head.

j. The setting for the three synchronous speed runners can now be determined.

3-4. <u>SETTING OF RUNNER.</u>

a. Overall plant efficiency is dependent on the design of the water passages from forebay through the tailrace. However, the turbine manufacturer is only responsible for the design between the turbine casing inlet and the discharge of the draft tube. Therefore the following dimensions are necessary for inclusion in the turbine specifications as limiting dimensions:

(1) Elevation of center-line of distributor.

(2) Maximum elevation of low point of draft tube floor.

(3) Horizontal distance from center-line of unit to the end of the draft tube.

b. The setting of the runner can be established by calculating H_s (the distance from the lowest point on the runner buckets to tailwater corresponding to the Q for the prototype horsepower) and substituting the value of sigma obtained from the model tests curves for HP_1 corresponding to the prototype horsepower in the formula:

$$H_s = H_b - H_v - \text{safety} - \sigma H$$

Depending on the value of sigma, H_s may be positive or negative. The setting of the bottom of the runner blades may be above or below the elevation of the tailwater corresponding to the discharge Q for the prototype horsepower. The distance ratio from the centerline of the distributor to the bottom of the runner is listed in Tables 1 and 2, Appendix C. This ratio multiplied by the prototype runner diameter, D_{TH}, gives the dimension which when added to or subtracted from H_s yields the setting for the centerline of the distributor of the prototype.

c. In determining the setting of the runner, the possibility of lower future tailwater levels due to degradation of the river channel below the dam must be considered. Also, the time required for build-up of tailwater under low load factor operation conditions, if applicable, must be considered. Both factors dictate a lower setting. Foundation conditions at the site may make it economically desirable to set the unit higher by using a lower specific speed runner or to set the unit lower by using a higher specific speed runner. There is also the possibility of an economic "trade-off" between the maximum output of the runner at the lower heads, cost of excavation, a draft tube with a shorter vertical leg, and more stainless steel pre-welding of the runner to reduce pitting of the runner due to the higher setting.

3-5. SPIRAL CASE AND DRAFT TUBE.

a. While the turbine manufacturer is responsible for the design of the water passages from the turbine casing inlet to the discharge of the draft tube, there are limitations which are prudent to impose such as the velocity at inlet to the spiral case, the number and width of draft tube piers, the velocity at discharge of the draft tube and the elevation of the lowest point of the draft tube that will be permitted.

b. The diameter of the inlet to the spiral case may be the same as, or preferably less than, that of the penstock but the velocity at the inlet should not exceed 22 percent of $\sqrt{2gH}$. If the velocity is higher, a loss in efficiency and power result. There may be instances where it is desired to install, in an existing plant, a larger unit than the structure was designed to accommodate. In this case the increased head loss (H_L) between the net head measurement section and the runner is approximately 2/3 of the increase in velocity head. The reduced efficiency E and power HP can be calculated by the following:

$$E_r = \frac{E\,(H - H_L)}{H}$$

$$HP_r = HP\left(\frac{H - H_L}{H}\right)^{3/2}$$

$$H_L = 2/3\left(\frac{V_2^2}{2g} - \frac{V_1^2}{2g}\right) = \frac{Q^2}{3g}\left(\frac{1}{A_2^2} - \frac{1}{A_1^2}\right)$$

V_1 = Velocity at the inlet of the normal casing.
V_2 = Velocity of the smaller casing.
A_1 = Area of inlet of the normal casing.
A_2 = Area of the smaller casing.

c. Deviations from strictly homologous water passages may also affect runaway speed, thrust, critical sigma as well as design of moving parts.

d. While procedures based on model laws and model and prototype tests are necessary to the study and selection of equipment, they need to be augmented by skills and judgment acquired by experience.

3-6. RUNAWAY SPEED.

a. The runaway speed of the prototype turbine is determined from model tests by running the model at various gate opening for the full range of model RPM (N_1) or phi (ϕ) to maximum RPM or ϕ at minimum values of efficiency and power and extending the curves to zero. The corresponding value, $\phi_{max.}$, is shown on Figures F1 through F8 of Appendix D. Prototype maximum runaway speed is given by the following:

$$N_{max} = \phi_{max}\left(\frac{12}{D_{TH}}\right)\left(\frac{60}{\pi}\right)(2g\,H)^{1/2}$$

$$= \frac{1838\,\phi_{max}\,H^{1/2}}{D_{TH}}$$

b. It is difficult to design a generator to withstand the highest overspeed conditions. Therefore, it is sometimes necessary to limit the maximum gate opening of the prototype turbine in order to limit the overspeed.

c. While runaway speed is affected by sigma, for all practicable purposes, its effect, on a Francis turbine can be neglected.

d. With medium head Francis turbines the maximum overspeed occurs at full gate but for higher heads where the inlet diameter of the runner is somewhat greater than the discharge diameter, the maximum runaway speed may occur at less than full gate

3-7. DRAFT TUBE LINERS. Draft tube liners should extend a distance equal to at least one discharge diameter of the runner below the point of attachment to the bottom ring.

3-8. AIR ADMISSION.

a. When Francis units are operating at part gate, air must be admitted to the center of the runner cone or hub. An air valve, mechanically connected to the wicket gate mechanism controls the admission of air. If the tailwater can be higher than the elevation of the valve and also, if a tailwater depression system is used, a check valve must be installed. Depending on the specific speed of the turbine and its required submergence, it may be necessary for the runner to have alternate passages to admit air through the runner relief holes and to use a compressed air supply for air admission.

b. For a required horsepower at a given head, higher specific speeds will require deeper settings and increased air admission at part gate opening for stable operation. It will also be necessary in some cases to provide fins in the draft tube to reduce power swings to an acceptable level.

3-9. RUNNER SEAL CHAMBER DRAINS. When runner seal drains are required, the seal chamber pipe drain header should discharge in the vertical leg of the draft tube at a location furthermost away from the draft tube exit.

3-10. SAMPLE CALCULATIONS. The basic calculations for a typical installation are included in Section 1, Appendix E.

CHAPTER 4

PROPELLER TURBINES

4-1. GENERAL USE.

a. Propeller type units, operating at higher speeds and at heads less than 100 feet, have generally replaced Francis turbines. Fixed blade units generally operate over a head range of 6 to 120 feet while adjustable blade units operate up to 250 feet. They have fewer blades than the Francis runner has buckets and consequently do not require as close a spacing of trash rack bars.

b. Fixed blade propeller units are best suited to a narrow range of outputs due to peaked efficiency curves. Kaplan units have adjustable blades which can operate under reduced heads while maintaining good power outputs, have high part gate and overgate efficiencies and can be made responsive to changes in wicket gate opening.

c. Fixed blade propeller units are appropriate where operation will be at or near constant load with small variations in heads. Capital cost will be 25 percent less than adjustable blade units for operation under the same conditions.

d. While adjustable blade units meeting the same conditions could be of smaller diameter and possibly operate at a higher speed, they also have higher runaway speeds and require a lower setting or submergence of the blades.

4-2. SPECIFIC SPEEDS.

a. A general discussion of specific speed is presented in paragraph 2-1. The usual range in specific speeds is from 82 to 205 for fixed blade propeller type units and 90 to 220 for the adjustable blade propeller type (Kaplan). The number of blades will vary from four to eight depending on the range in head, specific speed, and setting.

b. Care must be taken when using specific speed values to insure that they are being correctly used. The best efficiency horsepower at rated head for a fixed blade propeller turbine is matched to 90 - 95 percent of the generator KW rating. The horsepower equivalent of the KW rating is used in calculating the rated specific speed, the blade angle or tilt of the blades being selected to best suit the project requirements. The rated output of a Kaplan turbine is usually matched with the KW rating of the generator at rated head near full gate horsepower at maximum blade angle. The horsepower equivalent of the rating is used in calculating the rated specific speed.

c. A number of existing propeller turbine installations have been examined to develop some general rules for the preliminary selection of specific speed with respect to head. This information has been summarized in the form, $N_s = K/H^{1/2}$, and is presented on Figure 3 of Appendix C. The normal range in heads and associated K values for four, five and six blade runners shown on the curve sheets are recommended for use in determining the first value of N_s. A preliminary value for speed (N) is then calculated from the formula:

$$N = \frac{N_s \, H^{5/4}}{HP^{1/2}}$$

4-3. MODEL TEST CURVES.

a. Typical performance hill curves developed from model tests, covering both fixed and adjustable blade propeller turbines are shown in Appendix D, Section III. These curves follow the same format as that adopted for the Francis turbine designs (refer to paragraph 3-3a). Pertinent dimensions of the turbine parts and water passages, expressed as a ratio to D_{TH} are shown in Tables 4 and 5, and Figure 5 of Appendix C.

b. In the fixed blade design the inclination of the blades or blade angle dictates the capacity of the unit. However, increases in the blade angle are accompanied by reduction in the peak efficiency. This generally dictates a compromise depending upon the requirements of the project. Model test curves for an adjustable blade turbine having the same number of blades and approximately the same pitch ratio can be helpful in evaluating the effect of change in blade angle on performance, bearing in mind that the smaller hub diameter of the fixed blade turbine will result in some increases in power and efficiency over an adjustable blade turbine of the same runner diameter and number of blades. The pitch ratio is the ratio of the blade length to blade pitch (L/T). This ratio is generally referred to the blade periphery where the blade pitch is equal to the circumference generated by the blade tip divided by the number of blades. Critical sigma can be greatly affected by blade design and blade area, and ample blade area is necessary to keep sigmas within acceptable limits.

4-4. PRELIMINARY DATA FOR FIXED BLADE TYPE.

a. The fixed blade hill curves shown in Appendix D are based on designs that were developed to satisfy specific requirements. Their respective specific speeds at the point of maximum efficiency are: 141 (Figure FB1, Appendix D.) 119 (Figure FB2) and 106 (Figure FB3). Referring to Figure 3 of Appendix C it may be noted that these designs are ideally suited for heads of 32, 57 and 88 feet, respectively. They may be used for other rated head conditions with the precaution that the calculated speeds and runner throat diameters will be at variance with normal Corps practice. In the lower head range this error tends to produce larger, slower speed units, whereas, in the upper head range it tends to produce smaller, higher speed units.

b. If the user is chiefly interested in the size and speed of the unit, the following approximation will produce results more consistent with normal Corps practice. For rated conditions, compute the speed using the method presented in paragraph 4-2c. The peripheral speed coefficient can be estimated from the relationship:

$$\phi_{TH} = 0.089\, N_s^{0.58}$$

The runner throat diameter is calculated through the equation:

$$D_{TH} = \frac{1838\, \phi_{TH}\, H^{1/2}}{N}$$

c. The following procedure is used to compute prototype data from the hill curves in Appendix D.

(1) Pick off HP_1 at desired ϕ_{TH} and efficiency. This point will generally coincide with the maximum efficiency. Determine D_{TH} from the equation:

$$HP = HP_1 \left(\frac{D_{TH}}{12}\right)^2 H^{3/2}$$

$$D_{TH} = \frac{12}{H^{3/4}} \left(\frac{HP}{HP_1}\right)^{1/2}$$

(2) Calculate N from the equation:

$$N = \frac{1838\ \phi_{TH}\ H^{1/2}}{D_{TH}}$$

(3) N must be adjusted to the nearest synchronous speed.

(4) Readjust ϕ_{TH} and repeat steps (1) - (3), if required.

(5) Check performance required at other heads. Computed performance full gate horsepowers should be at least 2 percent higher than the required horsepower to allow for governing and variations such as manufacturing tolerances.

(6) The next step is to determine the setting by computing HP_1 for the required horsepowers, picking off from the sigma curves the corresponding value of critical sigma and solving for H_s in the formula:

$$\sigma_c = \frac{H_b - H_v - H_s - \text{Safety}}{H}$$

(7) Usually it is necessary to investigate three synchronous speeds in order to arrive at the most overall economic speed.

4-5. PRELIMINARY DATA FOR ADJUSTABLE BLADE TYPE.

a. The adjustable blade hill diagrams shown in Appendix D are also based on designs that were developed to satisfy specific requirements. The precautions noted in paragraph 4-4a, also apply to these curves. One requirement that generally dictates this type of unit is a widely varying head. In most of these cases the maximum capacity of the units is required at a rated head considerably lower than maximum head. For this reason the rated ϕ_{TH} is picked to the right of optimum ϕ_{TH}. Since most designs are capable of sustaining good efficiencies up to about 32 degrees blade angle, the rated conditions are generally associated with the on-cam performance at this blade angle. However, other over-riding requirements such as restricted submergence or efficiency may dictate that the rated conditions be referred to other blade angles. As the associated point for rated conditions is moved to the right away from optimum ϕ_{TH} the on-cam HP_1 for fixed blade angles increases, which provides for a smaller, higher speed unit. This advantage is generally offset by slightly reduced efficiency and higher critical sigma.

b. The method described in paragraph 4-4b may be used for approximating the speed and runner throat diameter for the adjustable type by using the following empirical relationship for ϕ_{TH}:

$$\phi_{TH} = 0.049\ N_s^{0.695}$$

c. The step by step procedure for computing prototype data through the hill curves in Appendix D is identical to the procedure described in paragraph 4-4c (1)-(7) with the following exceptions. A preliminary value of HP_1 may be obtained at the intersection of 32 degree blade angle curve and the following ϕ_{TH} values: 2.1(4 blades), 1.7(5 blades) and 1.5(6 blades). The prototype horsepower to be associated with this value of HP_1 will generally correspond to the generator rating. These rules may be varied to suit the specific requirements of the user.

d. Foundation conditions may determine the setting, and require modifications in speed, diameter and vertical height of the draft tube.

e. The selection of the appropriate adjustable blade turbine is more complex than for other turbines and requires much more work in arriving at a satisfactory solution. The range in operating heads may require the preliminary selected value of ϕ_{TH} to be increased. The requirement for a higher efficiency at generator rating may require the selected ϕ_{TH} to be decreased, while an acceptable lower efficiency would permit the ϕ_{TH} to be increased. The horsepower requirements at minimum head may require a change in speed, runner throat diameter and ϕ_{TH}. The value of critical sigma may require a change in HP_1 which would affect the runner diameter and require a change in speed and/or ϕ_{TH}. The effect of all these ramifications on the cost of the turbine, generator and powerhouse structure must be fully considered in making the final selection.

4-6. SETTING OF RUNNER BLADES.

a. Overall plant efficiency is dependent on all portions of the water passages from forebay through the tailrace. The turbine manufacturer is generally responsible for design from the turbine casing inlet to the discharge of the draft tube subject to such limiting dimensions imposed by other considerations and made part of the turbine specifications.

b. The model test information included in Appendix C includes the principal model dimensions of the semi-spiral or spiral casing, draft tube and runner dimensions.

c. The following dimensions are necessary for inclusion in the turbine specifications as limiting dimensions:

(1) Elevation of the center line of distributor.

(2) Elevation of the low point of draft tube floor.

(3) Horizontal distance from center line of unit to end of the draft tube.

(4) Limiting dimensions and elevations of water passages.

d. The formula shown in 4-4 c. (6) is used to calculate the setting of the runner blades. The datum for defining H_s is described in 2-3 b. The value of critical sigma, σ_c, is obtained from the model test curves at the HP_1 corresponding to the rated output. Depending on the value of sigma, H_s may be negative or positive, although it is usually negative for propeller units. Refer to paragraph 2-3.

e. A safety factor must be added to the calculated values of H_s as previously discussed under Paragraph 2-3 c., and

$$H_s = H_b - H_v - \sigma_c H - \text{Safety}$$

f. When using manufacturer's model curves, the manufacturer's safety factor should be carefully considered in determining the setting of the runner. One manufacturer recommends a safety factor equal to 0.2 D_{TH} + 0.7 H, where D_{TH} and H are in feet. This safety factor does not take into consideration the pre-welding of stainless steel on the low pressure side of the blades to mitigate the removal of metal from the surface of the blade by cavitation. Considerable judgment is required in determining the setting of a turbine with consideration for the number of units to be installed, the method of operation and tailwater elevations for initial and ultimate conditions.

4-7. SEMI-SPIRAL AND SPIRAL CASING, AND DRAFT TUBES.

a. The turbine manufacturer is responsible for the design of the water passages from the entrance to the turbine casing inlet to the discharge of the draft tube. Design conditions and limitations, such as velocity at the inlet of the semi-spiral casing, velocity at the discharge of the draft tube and setting the width of the semi-spiral casing should be set by the Corps. These conditions may also include the exit dimensions of the draft tube including the width and number of piers, and lower than normal distances from the center line of the distributor to the bottom of the draft tube and from the center line of the distributor to the roof of the semi-spiral casing.

b. Deviations from strictly homologous water passages may also affect runaway speed, thrust, critical sigma as well as design of moving parts.

c. Procedures based on model laws and model and prototype tests are necessary to the study and selection of equipment, however, they need to be augmented by skills and judgment acquired by experience.

d. For comments regarding spiral casing see paragraph 3-5.

4-8. RUNAWAY SPEED.

a. The runaway speed of the prototype turbine is determined from model tests by running the model at the various gate openings and blade angles for the full range of model RPM (N_1) or ϕ_{TH} to maximum RPM or ϕ_{TH} at minimum values of efficiency and power and extending the curves to zero.

b. As runaway speed is affected by sigma it is also necessary to run sigma versus runaway N or ϕ_{TH} for a range of gate openings and blade angles.

c. Prototype maximum runaway speed is given by the following:

$$(1)\quad N_{max} = \frac{1838\,\phi_{TH}\,H^{1/2}}{D_{TH}},\quad \phi_{TH} = \text{max value}$$

or

$$(2)\quad N_{max} = N_1\left(\frac{D_m}{D_{TH}}\right)H^{1/2},\quad N_1 = \text{max value}$$

d. Restricting minimum blade angle and/or maximum gate opening is a means by which runaway speed can be reduced.

e. When the blade angle is restricted, the turbine will operate at reduced efficiency throughout the lower range of output.

f. When the blade angle is restricted, the outer edge or tip of the blade is required to be machined to the contours of the discharge ring with the blades locked in a position corresponding to the minimum angular position of from 14 to 20 degrees with 16.5 degrees being the usual minimum angular position specified. While restricting the blade reduces the flexibility of operation and the efficiency of the turbine at horsepowers below the blade angle restriction, it decreases the maximum runaway speed and improves the efficiencies at and above the blade angle restriction with the greatest increase being in the range of the lower heads. Restricting the blade angle has made it possible to design generators for installations where otherwise it would be impracticable to design generators to withstand the higher overspeeds. When units with restricted blade angles are operated as "spinning reserve" or motoring as synchronous condensers, the energy taken from the system is greater than it would be if the blade angle were not restricted; however, the economics are invariably in favor of restricting the blade angle.

4-9. DRAFT TUBE LINERS.

Draft tube liners should extend a distance equal to at least one discharge diameter of the runner below the point of attachment to the bottom ring.

4-10. AIR ADMISSION.

a. Fixed blade turbines require the installation of air valves connected to the wicket gate mechanism to control the air which must be admitted to the center of the runner cone or hub, the same as for Francis units. A check valve must be installed if the tailwater will be higher than the elevation of the valve or if a tailwater depression system is used.

b. Adjustable blade turbines require large automatic air inlet valves, fitted with dash pots to open on sudden load rejection in order to break the water column upon the gate closure. The air valve must also act as a check valve to prevent the outflow of air or water.

4-11. SLANT AXIS ADJUSTABLE-BLADE TURBINES.

a. The Corps of Engineers has used axial-flow adjustable-blade turbines of the slant (inclined) axis type in three low head projects. Engineering studies indicated a considerable savings in the first cost of these projects. Due to problems with these units, operation and maintenance costs have been high. Also, considerable down time has resulted from turbine problems. For these reasons, consideration of slant axis units should be limited to sites where small units are required and there is an economic advantage.

b. The first installations designed by the Corps of Engineers were for the Ozark and Webber's Falls Projects, both on the Arkansas River. (Rated head - 21 feet. Head range 17 to 34 feet). Each turbine was set at an angle of 12 degrees to the horizontal and drives through a 33,800 horsepower speed increaser a 20,000 KW generator. The size of the turbine was limited by the horsepower of the speed increaser.

c. Subsequent progress in design has permitted a direct connection between turbine and generator, thereby eliminating the need for a speed increaser. This design has been adopted for the Harry S. Truman Project (Kaysinger Bluff) pump turbines which have their shafts inclined at an angle of 24 degrees. Each unit is rated 42,400 horsepower when operating as a turbine at a net head of 42.5 feet and the range in heads is from 41 feet to 79 feet. These are adjustable five blade units and are capable of operating as a pump at a range in pumping heads from 44 to 55 feet. The size of the pump-turbine was limited by the physical size of the generator-motor that could be installed, maintaining the required concrete dimension between the generator housing and the top of the draft tube.

4-12. SAMPLE CALCULATIONS. The basic calculations for typical installations are included in Appendix E.

CHAPTER 5

PUMP TURBINES

5-1. GENERAL.

a. Pump turbines are dual purpose machines. They operate as a pump in one direction and a turbine in the reverse direction.

b. A pump will perform in reverse rotation as a good turbine. However, a turbine does not generally operate in the reverse rotation as a good pump. Consequently, the design of a pump-turbine impeller follows more closely pump design practice than turbine design practice.

c. There is a dependent relationship between the two modes of operation and a compromise can be made to favor one mode of operation over the other.

5-2. BASIC CLASSIFICATIONS.

There are three basic classifications of pump-turbines:

1. Radial flow - Francis type

2. Mixed flow or diagonal type - Fixed blade and adjustable blade (Deriaz)

3. Axial flow or propeller type - Fixed blade and adjustable blade (Kaplan)

5-3. RADIAL FLOW - FRANCIS TYPE.

a. Francis type pump-turbines have been installed for heads of 75 to 1500 feet.

b. The design of the impeller is basically that of a pump impeller rather than that of a turbine runner. The impeller has fewer and longer blades than does a turbine runner with a view to effecting an efficient deceleration of flow in the water passages. The overall diameter of a pump-turbine runner is of the order of 1.4 times larger than the conventional Francis turbine runner. This is due to the requirements for a larger discharge diameter than eye (throat) diameter in the pumping mode. A lower runaway speed results, due to to the choking action of the impeller on the flow at higher speeds. This characteristic affects the cost of the water passages and the cost of the rotating parts of the pump-turbine and motor-generator.

c. As the unit is designed as a pump, it may be preferable to establish the pumping capacity for a specific total head which fixes within narrow limits the turbine capacity. This is particularly true for a combined installation of pump-turbines and turbines. Following the selection of the pumping capacity of the pump-turbine unit to fit the desired program of operation, the turbine capabilities in the generating mode is determined and the rating of the conventional units fixed to give the desired generating capacity for the several specified head conditions.

d. If the installation is strictly a pump-storage scheme, then the selection of the unit would begin with the determination of the required generating capacity at minimum head and the number of units to provide this capacity. Establishing the generating capacity for a given type and specific speed determines within a narrow range the pumping capacity. Establishing the pumping capacity for a specified total head also establishes within narrow limits the generating capacity for the corresponding turbine.

e. It is customary to guarantee the discharge in the pumping mode of a pump-turbine only at rated head or near best efficiency.

f. Only if a suitable runner is available from existing tests is it practicable to specify very closely the requirements for both the pumping and generating cycles.

g. Economics generally favors the higher capacity units unless an excessive number of runner splits is required by machining or shipping limitations. The design of split runners becomes more difficult with higher specific speeds. Runaway speed for higher capacity units is a larger percentage of synchronous speed and the centrifugal force makes the design of the splits more difficult as the additional metal increases the centrifugal force and the stress level.

h. Head losses through the draft tube of a pump-turbine decrease the net available suction head and increase the runner's sensitivity to cavitation. Therefore, pump-turbine runners must be set deeper than turbine runners. In order to reduce the size and cost of pump-turbines, higher specific speeds are utilized for pump-turbines relatively to turbines and consequently require deeper settings. The unit is more subject to cavitation in the pumping mode than in the generating mode. Therefore, the pumping mode determines the setting of the runner.

i. While in general higher specific speeds for pump turbines may appear to be desirable for economic reasons; efficiency, cavitation characteristics, mechanical and hydraulic design must be evaluated to determine the most favorable specific speed. Cavitation increases with higher specific speeds. The use of metals more resistant to cavitation damage may allow the acceptance of higher cavitation levels.

j. Transient Behavior. A transient in hydro-power is the history of what occurs between two states of equilibrium. A study must be made of the transient condition in order to determine the minimum (WK^2) flywheel effect for the rotating parts of the entire unit. The study should include a power failure in the pumping mode and load rejection in the generating mode. The Corps has a computer program which should be utilized when making these studies.

k. Four Quadrant Synoptic Curves.

(1) The necessary information to analyze transient behavior is provided by model tests made of the model runner and furnished as Four Quadrant Synoptic Curves which show the possible combinations of Unit Discharge (Q_{11}): under one foot of head and one foot eye diameter runner versus Shaft Speed (N_{11}), under one foot of head for one foot eye diameter in both pumping and turbine directions and Torque (T_{11}), under one foot of head for one foot eye diameter versus Discharge (Q_{11}), under one foot head for one foot eye diameter for both pumping and turbine directions.

(2) These curves are prepared from test information obtained from all gate openings in the complete turbine and pump performance curves, plus information from two additional tests. The first one of these tests is identical to a normal pump test except that the sense of rotation of the impeller and of the torque applied to the shaft are opposite to that for a normal pump test with measurements being taken the same as during normal pump tests. The second test involves rotating the impeller in the normal pump direction with water being pumped through the model in the normal turbine direction by service pumps. (During the test the head, discharge, speed, and torque are measured for various gate openings and shaft speeds.)

(3) The Four Quadrant Synoptic Curves may also be supplied showing the relationship of horsepower and discharge to phi (ϕ) for the various gate openings. The curves show the possible combination of head, discharge, torque or power in the following modes of operation:

(a) Pumping operation

(b) Dissipation of energy with rotation in the pumping direction and flow in the generating direction.

(c) Turbine operation

(d) Dissipation of energy with rotation in the generating direction and flow in the pumping direction.

5-4. MIXED FLOW OR DIAGONAL FLOW (DERIAZ).

a. The mixed flow or diagonal flow type pump-turbines are used in the medium head range up to more than 250 feet.

b. While the mixed flow adjustable blade machine (Deriaz) previously manufactured by the English Electric Company are presently manufactured and preferred for use in Japan, the less costly Francis mixed flow types are preferred in the U.S.A.

5-5. AXIAL FLOW - PROPELLER TYPE - FIXED AND ADJUSTABLE BLADE.

a. Alternates for use in the low head range, below 75 feet, are the axial flow machines arranged with the shaft vertically, horizontally or inclined.

b. The Corps of Engineers Harry S. Truman (Kaysinger Bluff) project has the largest capacity slant axis axial flow pump storage machines under construction as of 1 January 1981.

5-6. SPECIFIC SPEEDS - SINGLE STAGE REVERSIBLE PUMP/TURBINES.

a. The turbine specific speed (N_{st}) of a pump turbine in the generating mode is defined as the speed in revolutions per minute (N) at which a pump turbine of homologous design would operate if the runner was reduced in size to that which would develop one horsepower under 1 foot of head.

b. The turbine specific speed is expressed as follows:

$$N_{s_t} = \frac{N\ HP^{1/2}}{H^{5/4}}$$

c. The turbine specific speed is somewhat higher than for a conventional turbine, being in the range of K from 800 to 1250, where K = N_{st} H 1/2. The range of K for a conventional Francis turbine is 700 to 850, based on N_{st} at best efficiency.

d. The pump specific speed (N_{sp}) of a pump turbine in the pumping mode is defined as the speed in revolutions per minute (N) at which the pump turbine runner of homologous design would operate if the runner were reduced geometrically to such a size that it would deliver one U.S. gallon per minute under one foot of head.

e. The pump specific speed is expressed as follows:

$$N_{s_p} = \frac{N\,Q^{1/2}}{H^{3/4}}$$

$$Q = \text{gpm}$$

f. A recommended relationship for selecting pump specific speeds for a range of pumping heads is shown on Figure 2, Appendix C.

g. In selecting the specific speed consideration must be given not only to pump-turbine and generator motor costs, powerhouse and auxiliary equipment costs, but to efficiency in both modes of operation, cavitation characteristics, mechanical, hydraulic design features and to any restrictions imposed by foundation and site conditions.

5-7. PRELIMINARY DATA FOR FRANCIS PUMP-TURBINES.

a. Model performance data for pump-turbines is shown in Appendix D. This data is based on model tests covering low, intermediate and high specific speed designs. The original test data has been reduced to the more convenient form shown on Figures PT1, PT2 and PT3, which are based on D_{TH} = 12" and H = 1 foot. The performance for the generating mode is presented in the same format as that adopted in the other Sections covering the conventional Francis and propeller turbine designs. The discharge, efficiency and critical sigma curves for the pumping mode represent the envelope performance for best efficiency from a number of fixed wicket gate tests. The curve of pumping specific speeds is derived from the other data. Pertinent dimensions of the pump-turbine and associated water passages, expressed as ratios to D_{TH}, are shown in Table 3 and Figure 4 of Appendix C.

b. Information established by the Project Planning and Field Survey Studies is listed in paragraph 1-5. For the pumping cycle this information includes the pumping requirements and maximum, minimum and average heads.

c. The results of planning studies will generally dictate the rated pumping conditions. This information should include the rated gpm discharge and the associated rated dynamic head. An appropriate specific speed for the rated head is obtained from Figure 2, Appendix C. A preliminary value for speed (N), is then calculated from the formula:

$$N = \frac{N_{s_p} H^{3/4}}{Q^{1/2}}$$

The calculated value is rounded to the nearest synchronous speed.

d. If the user is chiefly interested in the size of the prototype unit, the following empirical relationship may be used to approximate a value for ϕ_{TH}.

$$\phi_{TH} = 0.0015\, N_s^{0.785}$$

e. The runner throat diameter is calculated as follows:

$$D_{TH} = \frac{1838\, \phi_{TH}\, H^{1/2}}{N}$$

f. The following procedure is used to calculate prototype pumping performance from the model performance curves:

(1) From Figure 2 pick off the value of N_{sp} corresponding with the rated dynamic head.

(2) Inspect the model performance curves, noting the specific speed range of each, and select the design that best suits the value of N_{sp}.

(3) From the selected curves, note the value of Q_1 for maximum pumping efficiency. Also note that this value is in cfs units. The prototype runner throat diameter (eye diameter for pump impeller) is calculated from the following relationship, where the subscript 1 refers to the model and subscript 2 refers to the prototype:

$$\frac{Q_2}{Q_1} = \left(\frac{D_2}{D_1}\right)^2 \left(\frac{H_2}{H_1}\right)^{1/2}$$

D_1 = 12 inches and H_1 = 1 foot.

(4) At this same point note the value of ϕ_{TH}. The pump speed is calculated from the relationship:

$$N = \frac{1838\ \phi_{TH}\ H^{1/2}}{D_{TH}}$$

Round to the nearest synchronous speed.

(5) Readjust ϕ_{TH} for synchronous speed, pick off a new Q1 from the model curves and repeat steps (3) and (4), if required. Continue this process until the re-adjusted value of ϕ_{TH} and corresponding Q1 from the model curves produce a value of D_{TH} that results in a synchronous speed in step (4).

(6) With fixed values for D_{TH} and N, the ϕ_{TH} corresponding to other pumping heads is calculated from the following relationship:

$$\phi_{TH} = \frac{N\ D_{TH}}{1838\ H^{1/2}} = \frac{Kn}{H^{1/2}}$$

(7) For given head and corresponding value of ϕ_{TH} extract Q_1 from the model curves and calculate the uncorrected value of prototype discharge from the following relationship:

$$Q_2 = Q_1 \left(\frac{D_{TH}}{12}\right)^2 H^{1/2}$$

(8) The prototype efficiency of the pump-turbine is determined by the Moody formula in accordance with MT-4.5 of Guide Specifications CE-2201.02 HYDRAULIC PUMP-TURBINES - FRANCIS TYPE Paragraph MT-4.5. The efficiencies are increased by the same amount in both the pumping and generating modes of operation. The following formula is applied to the peak efficiency value:

$$E_2 = 100-(100-E_1)\left(\frac{D_m}{D_p}\right)^{0.2}$$

$E_2 - E_1$ = step-up in efficiency

Using two-thirds of the step-up, the model efficiencies (E_1) are increased by 2/3 ($E_2 - E_1$) to give the expected or prototype efficiencies, E_2.

(9) The values of discharge in step (7) are increased by the ratio (E_2/E_1) to account for the increased efficiency of the prototype. The increased values are used in plotting the expected head-cfs curve of the prototype.

(10) With Q, H and E being taken directly from the performance curves, the horsepower-cfs curve is completed using the formula:

$$HP = \frac{Q\,H\,w}{550\,E}$$

g. The following procedure is used to calculate prototype performance in the generating mode from the associated hill curves shown at the bottom of the model curves in Appendix D.

(1) ϕ_{TH} values for the range in net operating heads are calculated from the formula established in (6) above.

(2) Prototype horsepowers for each head are calculated by intercepting the fixed gate curves at the associated value of ϕ_{TH}, noting the corresponding HP_1 and substituting known values in the following formula:

$$HP_2 = HP_1 \left(\frac{D_2}{D_1}\right)^2 \left(\frac{H_2}{H_1}\right)^{3/2}$$

$$HP_2 = HP_1 \left(\frac{D_{TH}}{12}\right)^2 \left(\frac{H_2}{1}\right)^{3/2}$$

(3) The expected prototype efficiency (E_2) is calculated by adding the step-up determined in f(8) above to the associated model efficiency (E_1), at each fixed gate point.

(4) With HP, H and E being taken directly from the expected performance curves, the expected prototype discharge is calculated from the following formula:

$$Q = \frac{550\ HP}{w\ H\ E}$$

5-8. SETTING OF PUMP-TURBINE RUNNER - FRANCIS TYPE.

a. The setting, which is generally referred to the elevation of the distributor centerline, is determined by critical sigma in the pumping mode. The appropriate sigma value for each pumping head can be picked off directly at each corresponding ϕ_{TH} on the model curves, then substituted in the following formula to compute the submergence, H_s:

$$\sigma_c = \frac{H_b - H_v - H_s - \text{safety}}{H}$$

(1) Appropriate values for H_b and H_v are shown on Figure 6, Appendix C.

(2) A recommended safety margin is calculated by substituting in the following formula:

Safety margin = $0.2\ D_i + 0.4\ H^{1/2}$

where D_i is the prototype runner inlet diameter in feet. This diameter is expressed as a dimensionless ratio, D_1 in Table 3 and Figure 4. The actual diameter is the product of the ratio times D_{TH}, in feet.

b. The values of critical sigma from the model tests are referred to the bottom of the runner. Likewise, the submergence (H_s) computed above is referred to this same point. The distance, a, from the distributor centerline to the bottom of the runner is expressed as a dimensionless ratio, d, in Table 3 and Figure 4. The distance is the product of the ratio times D_{TH}, in feet. The setting is calculated as follows:

Elevation distributor centerline = Tailwater Elev. + H_s + a

5-9. <u>SPIRAL CASING AND DRAFT TUBE - FRANCIS TYPE.</u>

a. The turbine manufacturer is responsible for the design of the water passages from the upstream end of the turbine casing inlet to the discharge of the draft tube. Deviations from the model dimensions can affect performance.

b. Pumping considerations dictate the design of the spiral casing and draft tube, except for the number and width of draft tube piers which depend on structural requirements.

5-10. <u>DRAFT TUBE LINERS.</u> Draft tube liners for pump-turbines should extend to the pier noses.

5-11. RUNAWAY SPEEDS.

a. Runaway speed tests for Francis type or fixed-blade propeller type pump-turbines are conducted the same as those for the Francis type turbines.

b. The runaway speed tests for adjustable blade propeller type pump turbines are conducted the same as for the adjustable blade turbine.

c. The calculation of prototype maximum runaway speed is the same as that shown in paragraph 4-8c.

d. It is sometimes necessary to limit the gate opening of the prototype Francis pump-turbine to limit overspeed because of the difficulty of designing a generator to withstand the higher overspeed of high gate openings.

e. The effect of sigma on a Francis type pump-turbine can be neglected, however, sigma must be considered for propeller type pump-turbines.

5-12. AIR ADMISSION. An air and check valve (or valves) should be installed for Francis-type pump turbines to permit operating at gate openings below 50 percent.

5-13. RUNNER SEAL CHAMBER DRAINS. The seal chamber pipe drain header should discharge in the vertical leg of the draft tube on the side furthest away from the draft tube exit.

5-14. SAMPLE CALCULATIONS. A typical calculation for a pump turbine installation is included in Appendix E.

CHAPTER 6

GENERATORS AND GENERATOR - MOTORS

6-1. GENERAL.

a. The hydroelectric generators and generator-motors are synchronous machines. Both produce electric energy by the transformation of hydraulic power into electric power but the latter also acts in reverse rotation as a motor to drive the pump-turbine as a pump.

b. As hydraulic turbines and pump turbines must be designed to suit the specific range of conditions under which they will operate, each generator and generator-motor is unique in that the electrical and mechanical design must conform to the hydraulic characteristic of the site and to the specific requirements of the electrical system.

c. A major difference between a generator-motor and the conventional generator is the special design features incorporated in the former required for starting and operating the unit in the reverse direction as a motor.

d. Guide Specifications cover the electrical and mechanical characteristics and the structural details of generators and generator-motors.

6-2. THE SELECTION AND NUMBER OF UNITS.

a. Factors affecting the selection and number of units are outlined in Chapter 1, Paragraph 1-7.

b. The type of generator or generator-motor depends on the type of turbine or pump-turbine to which it is connected and also whether the center line of the shafts will be vertical, horizontal or inclined.

c. Vertical shaft generators connected to Francis and Fixed Blade Propeller Turbines have three basic designs:

1. Suspended Generator - a thrust bearing located on top of the generator with two guide bearings.

2. Umbrella Generator - a thrust bearing and one guide bearing located below the rotor.

3. Modified Umbrella Generator - a combination guide and thrust bearing located below the rotor and a second guide bearing located above the rotor.

d. Horizontal shaft generators usually require thrust bearings capable of taking thrust in both directions. The thrust required may not be the same in both directions.

e. While some vertical shaft generators connected to adjustable blade turbines have been of the umbrella type, the Corps requires all vertical shaft adjustable blade turbines and all vertical shaft pump-turbines to have two guide bearings and one thrust bearing located either above or below the rotor.

f. Generators and generator-motors connected to inclined axis turbines require thrust bearings capable of taking thrust in both directions (away from and towards the generator) and two guide bearings.

6-3. GENERATOR RATING. Generators are rated in kva (electrical output) with the power factor determined by consideration of anticipated loads and system characteristics to which the unit or units will be connected.

6-4. GENERATOR VOLTAGE AND FREQUENCY. Determination of generator voltage is based on economic factors which include the cost of generator leads, instrument transformers, surge protective equipment, circuit breakers, space limitations, the requirement to serve local loads and the generator costs for the various voltage levels available. The standard frequency in the U.S.A. is 60 Hz.

6-5. SPEED. The speed of the unit is established by the turbine selected speed which must take into consideration that some synchronous speeds have a number of poles which for the kva rating desired would not give an acceptable winding design. See Table of Generator Speeds in Appendix A.

6-6. POWER FACTOR. The power factor of the generator is determined by the transmission and distribution facilities involved in addition to probable loads and system characteristics. If the generator is connected to a long, high-voltage transmission line, it may be economic as well as desirable to install a generator capable of operating with leading power factors. If the project is located near a large load center it may be economical to install a generator with larger than normal reactive capability by using 0.80 or 0.90 power factor machines. For the majority of installations 0.95 power factors generators will be the economic ones to specify. In general, for best operation the power factor of the generators should match the power factor of the load.

6-7. FLYWHEEL EFFECT (WK^2).

a. While the moment of inertia (WK^2) of the rotating parts of a unit (generator plus turbine) is a factor affecting system stability, the use of high speed circuit breakers and relays has in most cases removed the need for higher than normal WK^2 from the standpoint of the electrical system. Then the need for higher than normal WK^2 depends on the WK^2 needed to keep the speed rise of the unit and the pressure rise in the water passages for the turbine or pump-turbine within acceptable limits.

b. Greater than normal WK^2 may be required for an isolated plant or a unit serving an isolated load.

6-8. GENERATOR - MOTOR RATING.

a. The generator-motor rating depends on the pump-turbine characteristics and the system to which it will be connected.

b. Having selected a pump-turbine to meet the required capacity head conditions, the motor rating can be determined. In pump-turbine installations there may be considerable variations in head and consequently a variation in the motor requirements. The motor requirements may dictate the maximum rating of the generator-motor but in no case should the maximum horsepower required in the pumping mode be more than 94 percent of the 100 percent, 75 degree Centigrade nameplate rating, the motor rating being in KW (shaft output) or its horsepower equivalent.

c. The power factor depends on the system voltage conditions and transformer impedance and usually results in selecting a power factor of 0.95 (over-excited). There may be cases, during the hours of pumping, in which the system voltage may be reduced to where voltage studies show it is necessary for the machine to operate in the pumping mode at a lower voltage than in the generating mode. The range may be such as to require a dual voltage rating. Usually a difference in ratings for generating and motoring of about three percent less can be furnished without undue cost or complication.

6-9. EXCITATION SYSTEMS.

a. While for many years the excitation for synchronous generators was provided by directly connected main and pilot exciters, the pilot exciter was later eliminated and a rotating and/or a static voltage regulator system was used in connection with the direct-connected main exciter. Currently, except for very small generators, a static excitation system is specified. A static excitation system is specified for all reversible units.

b. The static excitation system is a static potential-source-rectified type with power for the excitation circuit normally taken directly from the generator or generator-motor leads. The complete excitation equipment consists of the excitation transformer, rectifiers, a-c excitation power circuit breaker, a-c bus between the transformer and rectifier, silicon controlled rectifiers, d-c bus from the rectifier cubicle to the generator or generator-motor field brush terminals, and a static automatic voltage regulator which controls the firing of the SCR's. The system designed for a generator-motor includes the necessary provisions for both directions of operation.

c. Most of the recent excitation systems have been provided with power system stabilizing equipment which changes the generator field current in proportion to instantaneous deviations from normal frequency to help dampen oscillations. This equipment should usually be furnished with solid-state excitation systems.

d. The Institute of Electrical and Electronics Engineers "Standard Definitions for Excitation Systems for Synchronous Machines," ANSI/IEEE Std 421.1-1986 should be consulted for an understanding of the application of solid-state devices to excitation systems. This report includes the new and revised definitions for excitation systems as applied to synchronous machines and gives particular emphasis on solid-state devices. The report includes figures illustrating the essential elements of an automatic control system, the components commonly used and figures which show the actual configuration of principal excitation systems supplied by domestic manufacturers.

6-10. THRUST AND GUIDE BEARINGS.

a. The Guide Specifications require that the generator and or generator-motor be provided with a nonspherical, adjustable-shoe or self-equalizing Kingsbury type or a General Electric spring type thrust bearing and for a generator-motor be suitable for rotation in either direction. For vertical machines the thrust bearing may be above or below the rotor. Units with very high thrust bearing loads and areas where space below the rotor is limited generally have the thrust bearing located above the rotor. Locating the thrust bearing below the rotor can reduce costs. Effects of the bearing location on vibration should be considered.

b. The generator of a vertical machine may be provided with one guide bearing below the rotor if the generator is connected to a Francis or fixed blade propeller turbine. An upper guide bearing must be provided if its installation is deemed necessary by the manufacturer for satisfactory operation or if the thrust bearing provided is of the self-equalizing type.

c. The generator of a vertical machine connected to an adjustable blade propeller turbine must be provided with two guide bearings.

d. The generator-motor of a vertical machine must be provided with two guide bearings with provisions for adequate lubrication in both directions of rotation.

c. It is now common practice to require that thrust bearings be provided with an externally pressurized system for providing high pressure oil to the thrust bearing surfaces during the starting and stopping of the machine.

6-11. THRUST BEARING BELOW THE ROTOR.

a. When the thrust bearing is located below the rotor the shaft coupling must be located farther below the stator than when the thrust bearing is located above the rotor.

b. The thrust bearing block is required by specifications to be forged integrally with the shaft.

c. Thrust bearings located below the rotor must be larger in diameter (for the same load carrying ability) than thrust bearings located above the rotor, due to the larger diameter of the shaft at the bearing location.

d. Two methods of inspection and removal of thrust bearing parts are provided, depending on the manufacturer of the generator or generator-motor. One method provides removable bearing housing covers in a bridge type lower bearing bracket. With the covers removed the thrust bearing is exposed for inspection and can be removed through the space between the bracket arms without disturbing the main support. The second method involves the lowering of the thrust bearing by use of a specially designed lowering device into the turbine pit. This design requires that the shaft coupling be located at a greater distance below the stator.

6-12. THRUST BEARING ABOVE THE ROTOR.

a. This design permits access to the bearing parts by overhead crane and assures adequate working space regardless of machine size and thrust bearing capacity.

b. This location is the standard location for small and high speed machines where the space below the rotor is limited.

c. This location does not place a limitation on the size of shaft forging facilities available. Large structural members are required to support the thrust bearing located above the rotor. These members support the thrust bearing loads and must be stiff enough to minimize deflections. They have to bridge a larger span than the bearing bracket located below the rotor.

d. The location does not require as large a diameter of bearings as does the thrust bearing located below the rotor and consequently the bearing losses are lower.

e. The bearing may be split when required for installations, such as with Kaplan turbines. This not only facilitates handling, but permits removal of bearing and parts without dismantling the Kaplan piping.

6-13. THRUST BEARING NOT INCORPORATED IN THE GENERATOR OR GENERATOR-MOTOR.

a. Slant axis adjustable blade turbines and pump turbines require a thrust bearing design to take thrust in both directions and may or may not be purchased with the generator depending on its location.

b. It is the common practice in Europe to procure the thrust bearing with the turbine and to mount the thrust bearing on the turbine head cover. Some savings is claimed in certain installations by so doing as it omits the large thrust bearing supports.

c. This location presents problems in coordinating the procurement and installation of the bearing. Thrust bearings are customarily furnished by the generator manufacturers in the U.S. and installed by them in the generator.

d. For very large capacity slow speed units it may be necessary for economic reasons to locate the thrust bearing on the turbine head cover.

6-14. HIGH PRESSURE OIL SYSTEM. High pressure oil starting system in a thrust bearing is essential for the operation of a reversible unit and should be specified for any unit over 31,260 kva in rating. To start these units, the oil is pumped under high pressure through openings in the stationary segments, forcing lubricating oil between the stationary and rotating parts of the bearing before the unit is started. This reduces the friction and breakaway torque to very low values, and reducing wear. It is also in service during shut-down.

6-15. ELECTRICAL CHARACTERISTICS.

a. The electrical characteristics of a generator and generator-motor, in addition to determining its individual performance, will affect the performance of the power system to which it will be connected. These characteristics can be varied within limits to best suit overall performance. The values for these characteristics must be included in the procurement specifications.

b. Characteristics of a generator or generator-motor which have an important effect on the stability characteristics of the electrical system are Short Circuit Ratio, Transient Reactance, Exciter System Performance, and the electrical damping provided by the WK^2 of the rotating parts of the units.

c. A short circuit ratio (SCR) is the ratio of the field current required to produce rated voltage at no load to the field current required to circulate rated current or short circuit. With no saturation, it is the reciprocal of the synchronous impedance (Xd) and a convenient factor for comparing and specifying the relative steady-state characteristics of generators and generator-motors. The higher the ratio, the greater the inherent stability of the machine. A system stability study is necessary to determine whether a higher-than-normal short-circuit ratio is required. Increasing the ratio above normal increases the machine size (the machine being de-rated), the normal flywheel effect (WK^2) and the machine costs, and decreases the efficiency and the transient reactance of the machine.

d. Some electrical systems require a lower than normal transient reactance. However, when a higher than normal SCR is specified, the transient reactance will be less than normal. Either the lower than normal transient reactance or the higher than normal SCR, but not both, will increase the cost. The cost increase is determined by the more expensive option. Decreasing transient reactance increases fault currents.

e. An increase in the rotor inertia (WK^2) above normal increases the cost, size and weight of the machine and decreases the efficiency. In a pump-turbine installation increasing the WK^2 of the generator-motor increases the starting time in the pumping mode. See Paragraph 6-7.

f. Exciter System Performance (See IEEE 69TP154 - PWR).

(1) Modern voltage regulator systems have increased the dynamic as well as the steady state stability of electrical systems.

(2) The excitation system should be capable of reversing the excitation voltage of full negative voltage to rapidly reduce field current when required. Capability to reverse field current, however, is normally not required.

(3) It must be recognized that some systems, at times, require operation of the generating equipment at voltages below normal. It must also be recognized that the excitation system should be capable of achieving the required performance at a specified voltage available from the generator terminals. However, specifying a voltage materially below normal will increase the size of the excitation equipment, the space required to house it, and its cost.

h. Amortisseur Windings.

(1) While many hydroelectric generators have been provided with non-continuous amortisseur windings in the past, continuous amortisseur windings are now normally specified for all Corps of Engineers machines. Continuous amortisseur windings provide substantial benefits over non-connected windings regarding stability, supplying unbalanced loads, effects on hunting etc., as described below. These benefits are particularly advantageous because of the difficulty in prior determination of system conditions. Continuous, heavy-duty amortisseur windings are required for generator-motors which are to be started as induction motors.

(2) Amortisseur windings are designed for a calculated ratio of quadrature-axis subtransient reactance to direct-axis subtransient reactance not to exceed 1.35 for an open amortisseur winding and not to exceed 1.10 for a closed winding.

(3) The advantages of amortisseur windings are:

(a) They reduce and in some cases eliminate hunting or sustained pulsations in current and voltage that occur under certain conditions of operation of a synchronous generator.

(b) They are effective in reducing the overvoltages due to unbalanced loads and faults, which can be an important factor in the application of lightning arrestors and in the coordination of insulation levels.

(c) Because they are a material aid in damping out oscillation they are of benefit in improving the ability of the machines to ride through system disturbances.

(d) They provide additional stabilizing torque for generators which are automatically synchronized.

(e) They reduce circuit breaker recovery voltages but tend to increase the magnitude of the current required to be interrupted.

(f) They aid in protecting field windings against current surges caused by lightning or internal faults and in case of the latter are effective in reducing additional damage to the machine.

(g) They permit pump-turbine motor-generators to be started as induction motors.

(4) Amortisseur windings increase stresses in the machine and connected equipment due to the increase in short-circuit current. The additional winding in the rotor must be built to withstand the stresses at maximum overspeed conditions and the stresses in the rotor parts are also increased. A continuous amortisseur winding complicates the cooling and disassembly of the field coils.

(5) An amortisseur winding designed for a calculated ratio of quadrature-axis subtransient reactance to direct-axis subtransient reactance not to exceed 1.35 will increase the price of the machine by approximately one percent. For a ratio not to exceed 1.10, the price increase is approximately three percent. For a continuous winding suitable for use in starting, the price increase is five percent due to the increase in thermal capacity of the winding required.

6-16. METHODS AVAILABLE FOR STARTING PUMP-TURBINES IN THE PUMPING MODE OF OPERATION.

a. The methods for starting pump-turbines in the pumping mode of operation can be classified into three groups, namely: Group 1 which requires continuous amortisseur winding; Group 2 which does not require continuous amortisseur windings; and Group 3 which requires separate starting devices.

b. Group 1 Starting Methods: Four methods of starting the unit as an induction motor with full or reduced voltage applied to the generator-motor terminal as follows:

(1) Full voltage start.

(2) Reduced voltage start.

(3) Part winding start.

(4) Reduced frequency induction start. This method utilizes another generator or generator-motor which may be isolated for starting duty.

c. Group 2 Starting Methods:

(1) Synchronous start.

(2) Static converter start.

d. Group 3 Starting Methods:

(1) Wound rotor induction motor start - Pony motor.

(2) Shaft connected starting turbine.

(3) Exciter starting.

e. The first three methods under Group 1 are applicable only to moderately sized units. Of these methods, the full voltage method has the lowest cost in its applicable range because no extra switching equipment is required. It also has the shortest starting time of any method (approximately 20 to 30 seconds). Because of the large starting kva that the full voltage method requires (unless the unit will be connected to a very stiff system), the system voltage drop may be too great for the system to tolerate when a unit is being started. Methods

(2), (3) and (4) are applicable within the range of metal clad type breakers. The start up time will be of two to three minutes and the starting kva in the neighborhood of the machine rating. The part winding start, Method (3) of Group 1 requires a careful investigation in each case. The difficulties in design of the motor-generator may make this type of starting impractical. The reduced frequency induction start method, Group 1, Method (4), sometimes referred to as the semi-synchronous start method, is applicable to plants having conventional generating units as well as generator-motor units, where provision is made so that a generator unit and a generator-motor unit may be isolated for starting or where a remote unit and transmission line may be isolated to start or be started by an isolated unit in the plant. This method is the one that has been most applicable for the Corps pump-turbine installations. The amortisseur duty for this type of start is much less than for the other methods in Group 1. In this method of starting, the water in the pump-turbine draft tube is depressed, the generator unit is brought up to approximately 80 percent speed without excitation and then connected to the generator motor, field current is then applied to the generating unit and the generator-motor accelerates as an induction motor and the generating unit will decelerate until the same speed is reached usually 30 to 40 percent rated speed. At this point field current is applied to the motor synchronizing it to the generator, the turbine wicket gates opened and the units brought up to rated speed and synchronized to the system. The pumping load can now be transferred to the system at the rate desired and the generating unit shut down or used to start another pumping unit if so required. This method of starting and synchronizing to the system eliminates any sudden load change on the system. This method of starting is applicable to all ratings but requires a careful study of starting conditions be made. Usually the rating of the generating unit is about equal to that of the unit to be started, but its WK^2 can be as little as 50 percent of the WK^2 of the unit to be started.

f. The synchronous start method, Method (1) of Group 2 as in the semi-synchronous start method uses an isolated generating unit to provide the starting power. Water is depressed in the pump-turbine draft tube and both units are electrically connected at standstill. Excitation is applied to both machines at rest and they are then brought up to speed in synchronism by admitting water to the turbine. This system provides smooth and rapid starting in 1-5 minutes. A separate source of excitation is required for each unit. A continuous amortisseur winding is not required for this method of starting. It is applicable for all ratings. The units have the same rated speed and the starting unit can be as small as 15 percent of the rating of the maximum unit to be started. This system will permit start-up without the normal tailwater depression. As soon as the unit reaches a speed high enough to prime the pump-turbine, the gates of the pump turbine are opened. This avoids a rough pump-start condition at shut-off head and synchronous speed and provides smooth and rapid starting and transfer of load to the system.

g. The static converter start, Method (2) of Group 3 is applicable to all ratings but its cost limits its use to a plant having 3 or 4 units of large capacity. The static converter is connected to a starting bus which in turn is connected to the unit to be started. During starting, the converter supplies a variable frequency output to the motor. The generator-motor is connected to the starting bus with zero frequency, and the input frequency during acceleration up to synchronous speed is controlled by silicon-controlled rectifier thyristors. As soon as the pumping unit has been synchronized to the power systems, the static converter can be de-energized and the starting bus and converter connected for starting the next unit. The static converter can be used as a brake to reduce deceleration time when a unit is being removed from service to maintenance, or in an emergency. Its use will also reduce wear on the generator.

h. Wound rotor induction motor start (Pony motor), Method (1) of Group 3 is applicable to any size unit, but because of its high cost for small units, its use is usually limited to large units. The starting time depends on the motor capacity provided, but usually is of the order of ten minutes with an approximate starting kva of five percent. This is probably the most costly method because starting motors are required for each unit. The starting control also requires a large floor area. It does provide for a very smooth starting system, however, and permits balancing of the complete unit in both directions of rotation without watering the unit.

APPENDIX A

AFFINITY LAWS, MODEL RELATIONSHIPS AND GENERATOR SPEEDS VERSUS NUMBER OF POLES

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A1. PUMP AFFINITY LAWS

With Impeller Diameter Held Constant

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2$$

$$\frac{Bhp_1}{Bhp_2} = \left(\frac{N_1}{N_2}\right)^3$$

With Speed Held Constant

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2}$$

$$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2$$

$$\frac{Bhp_1}{Bhp_2} = \left(\frac{D_1}{D_2}\right)^3$$

A2. PUMP MODEL RELATIONSHIPS

$$\frac{N_1}{N_2} = \frac{D_2}{D_1} \times \left(\frac{H_1}{H_2}\right)^{1/2}$$

$$\frac{Q_1}{Q_2} = \left(\frac{D_1}{D_2}\right)^2 \times \left(\frac{H_1}{H_2}\right)^{1/2}$$

$$\frac{P_1}{P_2} = \left(\frac{H_1}{H_2}\right)^{3/2} \times \left(\frac{D_1}{D_2}\right)^2$$

$$\frac{H_1}{H_2} = \left(\frac{Q_1}{Q_2}\right)^{2/3} \times \left(\frac{N_1}{N_2}\right)^{4/3}$$

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \times \left(\frac{D_1}{D_2}\right)^3$$

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3 \times \left(\frac{D_1}{D_2}\right)^5$$

When $H_1 = H_2$

Then $$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2 \times \left(\frac{N_1}{N_2}\right)^2 = 1$$

Therefore $$\frac{D_1}{D_2} = \frac{N_2}{N_1}$$

Note: Subscript 1 refers to model pumps.
Subscript 2 refers to prototype pumps.

and

$$\frac{Q_2}{Q_1} = \left(\frac{D_2}{D_1}\right)^3 \times \frac{N_2}{N_1} = \left(\frac{D_2}{D_1}\right)^3 \times \frac{D_1}{D_2} = \left(\frac{D_2}{D_1}\right)^2$$

$$\frac{P_2}{P_1} = \left(\frac{D_2}{D_1}\right)^5 \times \left(\frac{N_2}{N_1}\right)^3 = \left(\frac{D_2}{D_2}\right)^5 \times \left(\frac{D_1}{D_2}\right)^3 = \left(\frac{D_2}{D_1}\right)^2$$

$$N_s = \frac{N \times Q^{1/2}}{H^{3/4}}$$

Where Q is in gpm, N is RPM, and H is in feet.

$$whp = \frac{W \times CFS \times H}{550} = \frac{S \times GPM \times H}{3960}$$

Where W is specific weight of water (lb/ft^3)

S is specific weight of liquid referred to water at 68^oF

H is turbine net head in feet

$$eff = \frac{whp}{bhp} \qquad \sigma = \frac{NPSH}{H}$$

A3. TURBINE AFFINITY LAWS

For Constant Diameter

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

$$\frac{N_1}{N_2} = \frac{Q_1}{Q_2} = \left(\frac{H_1}{H_2}\right)^{1/2}$$

For Constant Head

$$\frac{N_1}{N_2} = \frac{D_2}{D_1}$$

$$\frac{Q_1}{Q_2} = \left(\frac{D_1}{D_2}\right)^2 = \frac{P_1}{P_2}$$

A4. TURBINE MODEL RELATIONSHIPS

$$\frac{N_1}{N_2} = \frac{D_2}{D_1} \times \left(\frac{H_1}{H_2}\right)^{1/2}$$

N_R = Prototype Runaway Speed

$$N_R = \frac{1838 \times \phi \times (H_2)^{1/2}}{D_2}$$

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \times \left(\frac{D_1}{D_2}\right)^3$$

$$\frac{HP_1}{HP_2} = \left(\frac{H_1}{H_2}\right)^{3/2} \times \left(\frac{D_1}{D_2}\right)^2$$

$$\frac{Q_1}{Q_2} = \left(\frac{D_1}{D_2}\right)^2 \times \left(\frac{H_1}{H_2}\right)^{1/2}$$

$$\frac{HP_1}{HP_2} = \left(\frac{N_1}{N_2}\right)^3 \times \left(\frac{D_1}{D_2}\right)^5$$

$$\sigma = \frac{H_b - H_v - H_s}{H}$$

Roger's Curve: $\sigma = \dfrac{N_s^{\,2}}{16{,}000}$

$$\Phi = \frac{N \times D}{1838 \times H^{1/2}}$$

$$N_s = \frac{N \times P^{1/2}}{H^{5/4}} \qquad Q = \frac{HP \times 8.8}{H \times eff}$$

Turbine Shaft Diameters: $d = \dfrac{321{,}000 \times HP^{1/3}}{N \times \text{Stress}}$ (solid shafts)

Hollow Shafts: $s = \dfrac{321{,}000 \times HP \times d}{N\,(d^4 - d_1^{\,4})}$ (d_1 is inside diameter)

A5. GENERATOR SPEEDS vs NUMBER OF POLES

$$RPM = \frac{120 \times Hz}{n} \qquad \text{For 60 cycles: } RPM = \frac{7{,}200}{n}$$

Hz = Frequency in cycles per second

n = No. of Poles

60 Hz Synchronous Speeds

Poles	RPM	Poles	RPM	Poles	RPM
24	300.0	60	120.0	104	69.2
26	277.0	64	112.4	110	65.5
28	257.1	68	105.9	112	64.3
30	240.0	70	102.9	120	60.0
32	225.0	72	100.0	126	57.1
36	200.0	76	94.7	128	56.2
40	180.0	80	90.0	130	55.4
42	171.4	84	85.7	132	54.5
44	163.6	88	81.8	136	52.9
48	150.0	90	80.0		
50	144.0	96	75.0		
52	138.5	98	73.5		
56	128.6	100	72.0		

Note: Omit Poles 34, 38, 54, 58, 62, 66, 72, and 82.

APPENDIX B

DATA - CORPS OF ENGINEERS PLANTS

TABLE B-1

TURBINE DATA

POWER PLANT	STATE	UNIT NO.	TYPE	MFR.	RATING HP	RATING RPM	RATING HEAD (FT)	MAX. HEAD (FT)	MIN. HEAD (FT)	UNIT SPACING (FT)**	DRAFT TUBE DEPTH (FT)***	SETTING (FT)****	RUNNER THROAT (FT - IN)	TOTAL WEIGHT (TONS)
ALBENI FALLS	IDAHO	1-3	KAPLAN	BLH	19,600	54.5	22	33	8	78	65	-2.0	23-6	825
ALLATOONA	GEORGIA	1,2	FRANCIS	SMS	50,000	112.5	135	170	103.5	54	33	10.5	13-9	550.8
ALLATOONA	GEORGIA	4	FRANCIS	JL	2,900	450	135	170	103.5		18.5	4.5	3-3.2	25
AMISTAD	TEXAS	1,2	FRANCIS	H	42,300	200	176	234	115	45	27	-26	9-7.1	359.4
BARKLEY	KY. & TENN.	1-4	KAPLAN	NNS&D	58,000	65.5	44	72	9	86	69	7.0	25-2	800
BEAVER	ARKANSAS	1,2	FRANCIS	NNS&D	77,400	105.9	164	206	122	61	42	7.2	15-3.4	523.2
BIG BEND	S. DAKOTA	1-8	FIX-BL.PROP	EE&T	90,300	81.8	67	75	50	86	70	-2.0	23-2	609
BIG CLIFF	OREGON	1	KAPLAN	BLH	26,500	163.6	91	98	73	48.67	33.7	-2.0	12-4.2	
BLAKELY MTN.	ARKANSAS	1,2	FRANCIS	SMS	52,000	120	168	196	135	56	40.0	7.0	14-4	420
BONNEVILLE	OREGON	1,2	KAPLAN	SMS	66,000	75	50	69	25.7	82	59.6	5.5	23-4	900
BONNEVILLE	OREGON	3-10	KAPLAN	SMS	74,000	75	60	69	25.7	82	59.6	5.5	23-4	900
BONNEVILLE	OREGON	SS(0)	KAPLAN*	SMS	5,000	257	50	69	25.7	N/A			6-9	
BONNEVILLE	WASHINGTON	11-18	KAPLAN	A-C	105,000	69.2	52	70	32.5	94	72.12	-7.87	27-6	1,125
BONNEVILLE	WASHINGTON	F1,2	KAPLAN	EW	20,700	156.5	59	68	32.5	43	30.5	-20	11-8	175
BROKEN BOW	OLKAHOMA	1,2	FRANCIS	A-C	69,000	128.6	175	215	144	53	35	4.2	12-8.1	325
BUFORD	GEORGIA	1,2	FRANCIS	NNS&D	55,000	100	136	170	104	62	39	7	14-10.8	425
BUFORD	GEORGIA	3	FRANCIS	JL	8,400	277	136	170	108	62	16	7	5-7.1	74.5
BULL SHOALS	ARK. & MO.	1-4	FRANCIS	A-C	52,000	128.6	190	237	127	54	33	11.0	12-8.5	367.5
BULL SHOALS	ARK. & MO.	5-8	FRANCIS	NNS&D	62,200	128.6	190	237	127	54	33	10.7	12-8.1	400
CARTERS	GEORGIA	1,2	FRANCIS	NNS&D	172,000	163.6	345	427.5	320	63	28	-12.9	13-2	460
CENTER HILL	TENN.	1-3	FRANCIS	BLH	62,500	105.9	160	207	131	58	38	3.8	15-0.5	557.5
CHEATHAM	TENN.	1-3	KAPLAN	NNS&D	20,000	60	22	29	9	82	65	6.6	22-10	550
CHIEF JOSEPH	WASHINGTON	1-4,15,16	FRANCIS	SMS	100,000	100	165	178.5	148	70	42	-7	15-3	713
CHIEF JOSEPH	WASHINGTON	5-14	FRANCIS	NNS&D	100,000	100	165	182.5	148	70	42	-7	15-5.7	600
CHIEF JOSEPH	WASHINGTON	17-27	FRANCIS	HITACHI	136,000	112.5	163	196	130	70	43	-7	16-10	572
CHIEF JOSEPH	WASHINGTON	SS(2)	FRANCIS	PELTON	3,500	514	171	184	148	22	28.5	4	3-3.1	12

** Unit Spacing C to C ft.

*** C Dist. to Bottom of Draft Tube

**** C Dist. to Min. Tailwater with 1 Unit Operating
(-) indicates C Dist. below Min. Tailwater

NOTES:

1. Runners Replaced in 1987 by Voith

TABLE B-1 (cont)

TURBINE DATA

POWER PLANT	STATE	UNIT NO.	TYPE	MFR.	RATING HP	RATING RPM	RATING HEAD (FT)	MAX. HEAD (FT)	MIN. HEAD (FT)	UNIT SPACING (FT)**	DRAFT TUBE DEPTH (FT)***	SETTING (FT)****	RUNNER THROAT (FT - IN)	TOTAL WEIGHT (TONS)
CLARENCE CANNON	MISSOURI	1	KAPLAN	A-C	42,900	128.6	75	107	69	75	47.2	-9	15-5	520
CLARK HILL	GA. & S.C.	1-7	FRANCIS	NNS&D	55,000	100	136	151	112	62	39	8	14-10	480
CORDELL HULL	TENN.	1-3	KAPLAN	BLH	58,200	65.5	44	58	26	86	69	6.5	24-2	815
COUGAR	OREGON	1,2	FRANCIS	JL	17,250	400	321	448	257.2	34	15	5.5	5-4.2	79
DALE HOLLOW	TENN.	1-3	FRANCIS	SMS	25,000	163.6	140	154	115	42	24	7	9-5.5	275
DARDANELLE	ARKANSAS	1-4	KAPLAN	DEW	51,800	48	38	53	20	80	60	4.5	22-0	720
DEGRAY	ARKANSAS	1	FRANCIS	NNS&D	63,600	150	171	216	144	49	32	-1	12-0.2	333.8
DENISON	OKLA. & TEX.	1,2	FRANCIS	SMS	56,000	90	102.5	131	75.5	64	43	7.4	15-11.5'	501.3
DETROIT	OREGON	1,2	FRANCIS	BLH	70,000	163.6	285	375	225	54	32	5.0	10-10	437
DEXTER	OREGON	1	KAPLAN	NNS&D	20,700	128.6	51	58.3	51	51	36	-4.6	13-3	206.5
DWORSHAK	IDAHO	1,2	FRANCIS	A-C	142,000	200	560	631	455	45	36	-1	10-5	302
DWORSHAK	IDAHO	3	FRANCIS	A-C	346,000	128.6	560	629	457	65	75	1.5	16-0	860
EUFAULA	OKLAHOMA	1-3	FRANCIS	NNS&D	41,500	100	96	125	70	61	42	8.2	15-2	443.2
FORT GIBSON	OKLAHOMA	1-4	FRANCIS	BLH	16,000	100	59	90	56	53	34.25	8	12-7.2	212.5
FORT PECK (1st)	MONTANA	1	FRANCIS	SMS	50,000	128.6	170	216	120	46	32	9.45	12-1.6	400
FORT PECK (1st)	MONTANA	2	FRANCIS	SMS	20,000	163.6	140	216	120	46	32	9.45	8-8	177
FORT PECK (1st)	MONTANA	3	FRANCIS	SMS	50,000	128.6	170	216	120	46	32	9.45	12-1.6	400
FORT PECK (2nd)	MONTANA	4-5	FRANCIS	A-C	55,000	128.6	170	210	115	56	36	5	12-11	450
FORT RANDALL	S. DAKOTA	1-8	FRANCIS	A-C	57,000	85.7	112	145	75.5	70	42	9	15-6	560
FOSTER	OREGON	1,2	KAPLAN	BLH	13,800	257	101	115	80	40	24	-10	8-3	253
GARRISON	N. DAKOTA	1-3	FRANCIS	BLH	88,000	90	150	184	99	75	47	6	17-11.5	945
GARRISON	N. DAKOTA	4,5	FRANCIS	BLH	90,000	90	150	184	99	75	47	6	17-11.5	870
GAVINS POINT	NEB. & S.D.	1-3	KAPLAN	BLH	54,000	75	48	60	40	80	60	8	22-0	855.5
GREEN PETER	OREGON	1,2	FRANCIS	A-C	55,000	163.6	265	321	186	45	28	6	10-6	275
GREEN PETER	OREGON	F1	FRANCIS	JL	2,000	900	290	290	161	45	28	-15	1-11.4	16
GREERS FERRY	ARKANSAS	1,2	FRANCIS	BLH	66,300	120	175	214	156	56	37	8	13-8	376
HARTWELL	GA. & S.C.	1-4	FRANCIS	NNS&D	91,5000	100	170	187	144	68	42	7	14-10	
HARTWELL	GA. & S.C.	5	FRANCIS	V-A	126,000	112.5	170	187	-143	68	42	11	16-0	
HILLS CREEK	OREGON	1,2	FRANCIS	BLH	21,700	277	256	320	181.5	38	17	6.2	6-2	55
ICE HARBOR	WASH.	1-3	KAPLAN	SMS	143,000	90	89	105	78	86	70.67	-16.5	23-4	940
ICE HARBOR	WASH.	4-6	KAPLAN	A-C	174,000	85.7	89	102	77	86	70.67	-16.5	25-0	1,000
JIM WOODRUFF	FLORIDA	1-3	KAPLAN	NNS&D	14,000	75	26.5	33	14	65	48	8	17-4	357.5
JOHN DAY	ORE. & WASH.	1-16	KAPLAN	BLH	212,400	90	94	110	83.5	90	76	-40.5	26-0	1,075
JOHN H. KERR	N.C. & VA.	1	FRANCIS	BLH	17,000	138.5	90	108	60	60.33	30	5.9	10-1	205
JOHN H. KERR	N.C. & VA.	2-7	FRANCIS	NNS&D	45,000	85.7	90	108	60	70	43	6.5	16-9	600
JONES BLUFF	ALABAMA	1-4	FIX-BL. PROP.	NNS&D	23,480	72	28.2	41.7	10	73	59	17.2	21-4	500
J. PERCY PRIEST	TENN.	1	FIX-BL. PROP.	A-C	42,700	128.6	78	100	65		42	-1	15-0	380
KEYSTONE	OKLAHOMA	1-2	FIX-BL. PROP.	A-C	49,000	120	74	117	66	71	45.5	-2.9	16-3	500

TABLE B-1 (cont)

TURBINE DATA

POWER PLANT	STATE	UNIT NO.	TYPE	MFR.	RATING HP	RATING RPM	RATING HEAD (FT)	MAX. HEAD (FT)	MIN. HEAD (FT)	UNIT SPACING (FT)**	DRAFT TUBE DEPTH (FT)***	SETTING (FT)****	RUNNER THROAT (FT - IN)	TOTAL WEIGHT (TONS)
LAUREL	KENTUCKY	1	FRANCIS	BLH	98,000	144	237	255	214		33.5	0	13-3.1	488
LIBBY	MONTANA	1-4	FRANCIS	A-C	165,000	128.6	300	341	165	63	42	-1	14-10	502.5
LIBBY	MONTANA	5-8	FRANCIS	A-C	165,000	128.6	300	341	165	63	42	-1	14-10	502.5
LITTLE GOOSE	WASH.	1-3	KAPLAN	BLH	212,000	90	93	98	90.5	90	76	-39	26-0	1075
LITTLE GOOSE	WASH.	4-6	KAPLAN	A-C	212,000	90	93	98	90.5	90	76	-39	26-0	1250
LOOKOUT POINT	OREGON	1-3	FRANCIS	PELTON	52,500	128.6	185	233	126	60	33	11.2	13-0.8	362.5
LOST CREEK	OREGON	1-2	FRANCIS	A-C	33,800	240	275	323	194	36	24	-0.5	7-10	162.5
LOWER GRANITE	WASH.	1-3	KAPLAN	BLH	212,400	90	93	100	76	90	76	-41	26-0	1,075
LOWER GRANITE	WASH.	4-6	KAPLAN	A-C	212,400	90	93	100	76	90	76	-41	26-0	1,250
LOWER MONUMENTAL	WASH.	1-3	KAPLAN	BLH	212,400	90	94	100	87	90	76	-40.5	26-0	1,075
LOWER MONUMENTAL	WASH.	4-6	KAPLAN	A-C	212,400	90	94	106	87	90	76	-40.5	26-0	1,250
MCNARY	ORE. & WASH.	1-12	KAPLAN	SMS	111,300	65.7	80	92	62	86	59.5	-8.5	23-4	1,050
MCNARY	ORE. & WASH.	13,14	KAPLAN	SMS	111,300	65.7	80	92	62	66	59 .5	-8.5	23-4	1,050
MCNARY	ORE. & WASH.	SS(2)	FRANCIS	PELTON	4,500	277	80	92	62	27	38	-4	5-3.6	50
MILLERS FERRY	ALABAMA	1-3	FIX-BL.PROP.	NNS &D	34,000	69.3	35.5	47	25	60	60	11.5	22-3	550
NANPIL RIVER	POHNPEI	1	FRANCIS	BOV	882	1200	200.0	200.0	225.0	200.0			1-4.5	2.7
NANPIL RIVER	POHNPEI	2	FRANCIS	BOV	1536.8	900	200.0	200.0	225.0	200.0			1-10	7.3
NARROWS	ARKANSAS	1,2	FRANCIS	BLH	12,000	225	132	153.5	98	32	18	9	6-9	87.3
NARROWS	ARKANSAS	3	FRANCIS	BLH	12,000	225	132	153.5	98	32	18	9	6-9	87.3
NEW MELONES	CALIFORNIA	1,2	FRANCIS	A-C	205,000	163.6	460	585	302.5	54.5	38.83	7.5	12-8	
NORFORK	ARK. & MO.	1,2	FRANCIS	SMS	42,000	128.6	160	202	130	54	32	10	12-1.6	335.0
OAHE	N.D. & S.D.	1-7	FRANCIS	A-C	128,500	100	185	203	114	76	44	4	17-0	740.5
OLD HICKORY	TENN	1-4	KAPLAN	BLH	45,000	75	45	60	23	60	60	8	22-0	855.5
OZARK	ARKANSAS	1-5	ADJ.BL.INCL.AX.	A-C	27,900	60.2	21	35	17	65	39.78	-14.7	26-3	745
PHILIPOTT	VIRGINIA	1,2	FRANCIS	JL	9,400	277	152	176	108	32	16.5	3.7	5-5	70
RICHARD B. RUSSELL	GA AND S.C.	1-4	FRANCIS	V-A	104,000	120	144	162	134	71	43	12		
ROBERT S. KERR	OKLAHOMA	1-4	KAPLAN	A-C	38,000	75	29	47	20	87	67	7.5	24-2	675
SAINT MARYS (OLD)	MICHIGAN	10	KAPLAN	A-C	3,000	128.6	20	22.6	18.3	28	21.2	5	9-6	
SAINT MARYS (NEW)	MICHIGAN	1-3	FIX-BL.PROP.	A-C	6,975	80	21	22.6	18.3	54	36.33	3.23	15-4	
SAINT MARYS (NEW)	MICHIGAN	3A	KAPLAN	SMS	3,000	128.6	21	22.6	18.3	49.5	27.83	-4.27	9-2	
SAINT STEPHENS	S.C.		PROP	A-C	39,000	100	49	57.9	40.3		56	-10.4*	18-6	
SAM RAYBURN	TEXAS	1,2	KAPLAN	A-C	41,300	120	70	88	53	57	45.5	-3.5	15-0	293.8
SNETTISHAM	ALASKA	1,2	FRANCIS	M'BISHI	32,300	514	745	900	675	26	13	-2.0	4-3	146
SNETTISHAM	ALASKA	3	FRANCIS	DBS	47,000	600	945.5	990.5	788	26	13	-8	4-3	55
STOCKTON	MISSOURI	1	KAPLAN	NNS&D	71,800	75	81	108	46	N/A	70.67	-6.5	23-4.6	800
TABLE ROCK	ARK. & MO.	1-4	FRANCIS	EE&T	68,000	128.6	190	228	134	54	36	7.2	13-6	386
TENKILLER FERRY	OKLAHOMA	1,2	FRANICS	DMD	23,500	150	132	181	103.5	44	30	7.9	9-8.1	247.5
THE DALLES	OREGON	1-14	KAPLAN	BLH	123,800	85.7	81	90.5	60	66	70.7	-16	23-4	1,085

TABLE B-1 (cont)

TURBINE DATA

POWER PLANT	STATE	UNIT NO.	TYPE	MFR.	RATING HP	RATING RPM	RATING HEAD (FT)	MAX. HEAD (FT)	MIN. HEAD (FT)	UNIT SPACING (FT)**	DRAFT TUBE DEPTH (FT)***	SETTING (FT)****	RUNNER THROAT (FT - IN)	TOTAL WEIGHT (TONS)
THE DALLES	OREGON	15-22	KAPLAN	BLH	135,000	80	73	90	58	66	70.7	-14	25-0	589
THE DALLES	OREGON	F1,F2	KAPLAN	A-C	18,800	200	74	88	55	40	35.5	-14.5	10-0	167.5
THE DALLES	OREGON	SS(2)	FRANCIS	PELTON	4,500	277	81	90.5	60	27	50.3	-5.5	5-3.5	42.5
TOWN BLUFF	TEXAS	1-2	S" KAPLAN	DBS	4820	163.6	27.1	32	11.5	25	15.44	5	9-2.2	
WALTER F. GEORGE	GA. & ALA.	1-4	KAPLAN	NNS&D	45,500	112.5	70	88	37	67				
WAPPAPELLO	MISSOURI	1	FRANCIS	LEFFEL	177	257	15	30	5	N/A	45.5	-2.5	16-8	378
WEBBERS FALLS	OKLAHOMA	1-3	ADJ.BL.INCL.AX.	A-C	30,900	60.24	22	31	17	65	39.78	-14.8	26-3	745
WEST POINT	GEORGIA	1	FIX-BL.PROP.	JL	5,400	327	77	77	44	N/A	18	-2.0	6-0	77.5
WEST POINT	GEORGIA	2,3	FIX-BL.PROP.	NNS&D	51,500	100	58	72.4	44	77	52	-1.1	19-2	437.5
WHITNEY	TEXAS	1,2	FRANCIS	NNS&D	20,700	128.6	91.5	126	77.5	46	30	8.5	10-10.8	167.5
WOLF CREEK	KENTUCKY	1-6	FRANCIS	BLH	62,500	105.9	160	214	111	58	38	7	15-0.5	557.5

TABLE B-2

GENERATOR DATA

POWER PLANT	STATE	UNIT NOS.	MFR.	RPM	RATING KVA	RATING PF	RATING KW	RATING KV	S-C RATIO NLT	TRANSIENT REACTIVE NMT (PERCENT)	WK2 NLT (LB.-FT.2)	HOUSING DIAMETER (FT.-IN.)	TOTAL WEIGHT (TONS)
ALBENI FALLS	IDAHO	1-3	GE	54.5	15,778	0.90	14,200	13.8	1.10	55.0	55,000,000	48 - 6	444
ALLATOONA	GEORGIA	1,2	W'HSE	112.5	40,000	0.90	36,000	13.8	1.22	43.0	39,000,000	38 - 0	413
ALLATOONA	GEORGIA	4	W'HSE	450	2,500	0.80	2,000	2.4	1.00	NORM	89,000	12 - 6 (MAX)	28.3
AMISTAD	TEXAS	1-2	CGE	200	34,737	0.95	33,000	13.8	1.175	38.0	8,332,000	27 - 6	
BARKLEY	KY. & TENN.	1-4	GE	65.5	36,111	0.90	32,500	13.8	1.10	51.0	114,000,000	50 - 0	676.9
BEAVER	ARKANSAS	1,2	W'HSE	105.9	58,947	0.95	56,000	13.8	1.175	43.0	57,200,000	41 - 0	486.5
BIG BEND	S. DAKOTA	1-8	W'HSE	81.8	61,579	0.95	58,500	13.8	1.175	44.0	109,400,000	45 - 8	677.5
BIG CLIFF	OREGON	1	W'HSE	163.6	20,000	0.90	18,000	14.4	1.10	43.0	9,000,000	31 - 0	237.3
BLAKELY MTN.	ARKANSAS	1,2	W'HSE	120	41,667	0.90	37,500	13.8	1.10	42.0	44,000,000	40 - 0	435
BONNEVILLE	OREGON	1,2	GE	75	48,000	0.90	43,200	13.8	1.10	44.0	113,000,000	48 - 0	750
BONNEVILLE	OREGON	3-10	GE	75	60,000	0.90	54,000	13.8	1.30	44.0	113,000,000	48 - 0	1,000
BONNEVILLE	OREGON	SS(0)	SMS	257	5,000	0.80	4,000	4.16	1.04	0.30	884,000	20-0	56
BONNEVILLE	WASHINGTON	11-18	GE	69.2	70,000	0.95	66,500	13.8	1.175	42.0	164,960,000	52 - 2	820
BONNEVILLE	WASHINGTON	FW1&2	W'HSE	156.5	13,800	0.95	13,110	13.8	1.175	45.0	4,231,000	20'.0 x 30'.0	165
BROKEN BOW	OKLAHOMA	1,2	A-C	128.6	52,632	0.95	50,000	13.8	1.175	41.0	34,400,000	40 - 0	348.5
BUFORD	GEORGIA	1,2	W'HSE	100	44,444	0.90	40,000	13.8	1.10	43.0	80,000,000	43 - 4	587.5
BUFORD	GEORGIA	3	W'HSE	277	6,667	0.90	6,000	13.8	1.10	44.0	750,000	18 - 4(MAX)	91
BULL SHOALS	ARK. & MO.	1-4	A-C	128.6	42,100	0.95	40,000	13.8	1.17	42.0	34,000,000	40 - 0	360
BULL SHOALS	ARK. & MO.	5-8	A-C	128.6	47,368	0.95	45,000	13.8	1.175	41.0	34,000,000	40 - 0	362.8
CARTERS	GEORGIA	1,2	A-C	163.6	131,579	0.95	125,000	13.8	1.175	37.0	97,000,000	42 - 0	775
CENTER HILL	TENN.	1-3	GE	105.9	50,000	0.90	45,000	13.8	1.10	43.0	55,000,000	40 - 0	560
CHEATHAM	TENN.	1-3	W'HSE	60	13,333	0.90	12,000	13.8	1.10	54.0	47,500,000	44 - 2	431.6
CHIEF JOSEPH	WASHINGTON	1-4,15,16	W'HSE	100	67,368	0.95	64,000	13.8	1.17	43.0	84,000,000	42 - 6	680
CHIEF JOSEPH	WASHINGTON	5-14	W'HSE	100	67,368	0.95	64,000	13.8	1.17	43.0	84,000,000	42 - 6	680
CHIEF JOSEPH	WASHINGTON	17-27	GE	112.5	100,000	0.95	95,000	13.8	1.17	37.0	97,000,000	46 - 0	538
CHIEF JOSEPH	WASHINGTON	SS(2)	ELLIOTT	514	3,000	0.80	2,400	4.2	1.00	NORM	75,000	12 (MAX)	33.9
CLARENCE CANNON	MISSOURI	1	GE	128.6	28,421	0.95	27,000	13.8	1.175	43.0	15,500,000	35 - 0	265
CLARK HILL	GA. & S.C.	1-7	GE	100	44,444	0.90	40,000	13.8	1.10	43.0	55,000,000	41 - 0	569
CORDELL HULL	TENN.	1-3	GE	65.5	37,037	0.90	33,333	13.8	1.10	45.0	114,000,000	54 -(MAX)	584.5
COUGAR	OREGON	1,2	ELLIOTT	400	13,158	0.55	12,500	6.9	1.175	38.0	1,498,000	22 - 0	101.9
DALE HOLLOW	TENN.	1-3	W'HSE	163.6	20,000	0.90	18,000	13.8	1.10	42.0	7,500,000	28 - 4	195.8
DARDANELLE	ARKANSAS	1-4	W'HSE	75	37,632	0.95	31,000	13.8	1.175	48.0	54,500,000	42 - 8	460
DEGRAY	ARKANSAS	1	A-C	150	42,105	0.95	40,000	13.8	1.175	40.0	34,000,000	39 - 0	338
DENISON	OKLA. & TEX.	1,2	W'HSE	90	36,842	0.95	35,000	13.8	1.50	43.0	55,000,000	39 - 2	497
DETROIT	OREGON	1,2	W'HSE	163.6	55,555	0.90	50,000	13.8	1.76	31.0	29,000,000	38 - 3	590
DEXTER	OREGON	1,2	GE	128.6	15,000	1.00	15,000	13.8	1.25	46.0	9,000,000	31 - 8	262
DWORSHAK	IDAHO	1,2	A-C	200	94,737	0.95	90,000	13.8	1.175	36.0	36,200,000	36 -1	445

TABLE B-2 (cont)

GENERATOR DATA

POWER PLANT	STATE	UNIT NOS.	MFR.	RPM	RATING KVA	RATING PF	RATING KW	RATING KV	S-C RATIO NLT	TRANSIENT REACTIVE NMT (PERCENT)	WK2 NLT (LB.-FT.2)	HOUSING DIAMETER (FT.-IN.)	TOTAL WEIGHT (TONS)
DWORSHAK	IDAHO	3	A-C	128.6	231,579	0.95	220,000	13.8	1.175	39.0	200,000,000	48 - 3	921
EUFAULA	OKLAHOMA	1 - 3	W'HSE	100	31,579	0.95	30,000	13.8	1.175	44.0	31,900,000	39 - 2	345
FORT GIBSON	OKLAHOMA	1 - 4	A-C	100	12,579	0.90	11,250	13.8	1.10	49.0	13,500,000	28 - 0	226.5
FORT PECK (1ST)	MONTANA	1	A-C	128.6	38,889	0.90	35,000	13.8	1.10	39.0	30,000,000	36 -0	350
FORT PECK (1ST)	MONTANA	2	A-C	164	16,667	0.90	15,000	13.8	1.26	40.0	7,000,000	25 - 6	165
FORT PECK (1ST)	MONTANA	3	A-C	128.6	38,889	0.90	35,000	13.8	1.10	43.0	30,000,000	36 - 0	350
FORT PECK (2ND)	MONTANA	4 - 5	ELLIOTT	128.6	42,105	0.95	40,000	13.8	1.17	42.0	39,500,000	35 - 4	411.4
FORT RANDALL	S. DAKOTA	1 - 8	W'HSE	85.7	42,105	0.95	40,000	13.8	1.17	46.0	82,000,000	43 - 0	614
FOSTER	OREGON	1,2	A-C	257	10,526	0.95	10,000	4.2	1.175	42.0	2,000,000	17-1x22-10	115.5
GARRISON	N. DAKOTA	1 - 3	GE	90	84,210	0.95	80,000	13.8	1.96	34.0	150,000,000	50 - 6	1,122.5
GARRISON	N. DAKOTA	4,5	GE	90	84,210	0.95	80,000	13.8	1.175	43.0	126,000,000	50 - 6	790
GAVINS POINT	NEB. & S.D.	1 - 3	GE	75	35,100	0.95	33,345	13.8	1.17	48.0	80,200,000	47 - 10	631.4
GREEN PETER	OREGON	1,2	W'HSE	163.6	42,105	0.95	40,000	13.8	1.175	40.0	22,000,000	32 - 2	342.5
GREEN PETER	OREGON	F1	GE	900	1,500	0.95	1,425	480v	1.175	NORM	5,350	44 - 0	342.5
GREERS FERRY	ARKANSAS	1,2	ELLIOTT	120	50,326	0.95	48,000	13.8	1.96	41.0	40,000,000	38 - 0	424
HARTWELL	GA. & S.C.	1 - 4	W'HSE	100	73,333	0.95	65,000	13.8	1.175	43.0	96,000,000	44 - 0	615
HARTWELL	GA. & S.C.	5	CGE	112.5	84,210	0.95	80,000	13.8	1.17	43.0	77,614,000	44	
HILLS CREEK	OREGON	1,2	BB	277	15,789	0.95	15,000	6.9	1.175	40.0	1,600,000	21 - 8	113
ICE HARBOR	WASH.	1 - 3	GE	90	94,737	0.95	90,000	13.8	1.175	43.0	156,500,000	49 - 0	1,013
ICE HARBOR	WASH.	4 - 6	GE	85.7	116,800	0.95	110,960	13.8	1.175	32.4	205,208,000	54 -0	
JIM WOODRUFF	FLORIDA	1 - 3	W'HSE	75	11,111	0.90	10,000	13.8	1.10	53.0	20,000,000	37 - 10 (MAX)	267.5
JOHN DAY	ORE. & WASH.	1 - 16	GE	90	142,105	0.95	135,000	13.8	1.175	43.0	245,000,000	54 - 0	1,164.0
JOHN H. KERR	N.C. & VA.	1	W'HSE	138.5	13,333	0.90	12,000	13.8	1.175	45.0	7,500,000	31 - 8	170
JOHN H. KERR	N.C. & VA.	2 - 7	W'HSE	85.7	35,555	0.90	32,000	13.8	1.175	45.0	60,000,000	42 - 6	512.5
JONES BLUFF	ALABAMA	1 - 4	W'HSE	72	17,895	0.95	17,000	13.8	1.186	52.0	30,400,000	89 - 4	285
J. PERCY PRIEST	TENN.	5 - 8	W'HSE	128.7	31,111	0.90	28,000	13.8	1.10	42.0	28,000,000	38 - 6	310
KEYSTONE	OKLAHOMA	1 - 2	GE	120	36,842	0.95	35,000	13.8	1.175	43.0	30,000,000	38 - 2	401.5
LAUREL	KENTUCKY	1	GE	144	67,778	0.90	61,000	13.8	1.10	40.0	43,000,000	39 - 8	421
LIBBY	MONTANA	1 - 4	W'HSE	128.6	110,526	0.95	105,000	13.8		40.0	97,500,000	41 - 0	583
LIBBY	MONTANA	5 - 8	W'HSE	128.6	110,526	0.95	105,000	13.8	1.175	40.0	84,362,000	45 - 0	664.2
LITTLE GOOSE	WASH.	1 - 3	GE	90	142,105	0.95	135,000	13.8	1.175	43.0	245,000,000	54 - 0	1,161
LITTLE GOOSE	WASH.	4 - 6	GE	90	142,105	0.95	135,000	13.8	1.175	43.0	341,000,000	54 - 0	1,176
LOOKOUT POINT	OREGON	1 - 3	GE	128.6	47,222	0.95	40,111	13.8	1.10	42.0	26,450,000	37 - 0	454.8
LOST CREEK	OREGON	1 - 2	A-C	240	25,789	0.95	24,500	13.8	1.175	38.0	7,100,000	24-0x32-0	168.5
LOWER GRANITE	WASH.	1 - 3	GE	90	142,105	0.95	135,000	13.8	1.175	43.0	245,000,000	54 - 0	1,161
LOWER GRANITE	WASH.	4 - 6	GE	90	142,105	0.95	135,000	13.8	1.175	43.0	241,300,000	54 - 0	1,176
LOWER MONUMENTAL	WASH.	1-3	GE	90	142,105	0.95	135,000	13.8	1.175	43.0	245,000,000	54 - 0	1,161
LOWER MONUMENTAL	WASH.	4 - 6	GE	90	142,105	0.95	135,000	13.8	1.175	43.0	241,300,000	54 - 0	1,176

TABLE B-2 (cont)

GENERATOR DATA

POWER PLANT	STATE	UNIT NOS.	MFR.	RPM	RATING KVA	RATING PF	RATING KW	RATING KV	S-C RATIO NLT	TRANSIENT REACTIVE NMT (PERCENT)	WK2 NLT (LB.-FT.2)	HOUSING DIAMETER (FT.-IN.)	TOTAL WEIGHT (TONS)
MCNARY	ORE. & WASH.	1 - 12	GE	85.7	73,684	0.95	70,000	13.8	1.90	32.0	130,000,000	51 - 8	1,200
MCNARY	ORE. & WASH.	13,14	EE&T	85.7	73,684	0.95	70,000	13.8	1.90	32.0	168,000,000	51-8	1,165
MCNARY	ORE. & WASH.	SS(2)	EM	277	3,750	0.80	3,000	4.2	1.00	35.0	360,000	17-7(MAX)	50.5
MILLERS FERRY	ALABAMA	1-3	W'HSE	63.3	26,316	0.95	25,000	13.8	1.175	52.0	53,500,000	43-6	430
NANPIL RIVER	POHNPEI	1	A-J	1200	906	.8	725	.48			2,400		24,650.0
NANPIL RIVER	POHNPEI	2	A-J	900	1,666	.8	1,333	.48			4,500		33,075.0
NARROWS	ARKANSAS	1,2	ELLIOT	225	9,444	0.90	8,500	13.8	1.10	43.0	2,000,000	22-0	95
NARROWS	ARKANSAS	5	W'HSE	225	9,444	0.90	8,500	13.8	1.10	43.0	2,000,000	21-0	91
NEW MELONES	CALIFORNIA	1,2	GE	163.6	166,667	0.90	150,000	13.8	1.10	38.0	67,330,000	42-0	
NORFORK	ARK. & MO.	1,2	W'HSE	128.6	38,889	0.90	35,000	13.8	1.10	43.0	30,000,000	36-4	386.5
OAHE	N.D. & S.D.	1-7	GE	100	89,474	0.95	85,000	13.8	1.175	42.0	165,000,000	49-0	826.5
OLD HICKORY	TENNESSEE	1-4	GE	75	31,250	0.80	25,000	13.8	1.00	48.0	70,000,000	45-8	559.5
OZARK	ARKANSAS	1-5	GE	514	21,053	0.95	20,000	13.8	1.175	35.0	600,000	11-10	113.5
RICHARD B. RUSSELL	GEORGIA	1-4	S-A	120	78,947	0.95	75,000	13.8	1.175	37.0	63,400,000	23-3	
ROBERT S. KERR	OKLAHOMA	1-4	GE	75	28,947	0.95	27,500	13.8	1.175	49.0	61,500,000	44-8	451
SAINT MARYS (OLD)	MICHIGAN	10	GE	128.6	2,500	0.80	2,000	4.0	1.4	24-26	1,500,000	16-4	
SAINT MARYS (NEW)	MICHIGAN	1-3	GE	80	5,333	0.90	4,800	13.8	1.10	36.6	6,750,000	25-2	156.5
SAINT MARYS (NEW)	MICHIGAN	3a	GE	128.6	2,500	0.80	2,000	4.2	1.00	36.0	1,500,000	16-4	62.5
SAINT STEPHENS	S.C.												
SAM RAYBURN	TEXAS	1,2	W'HSE	120	27,368	0.95	26.000	13.8	1.175	44.0	20,700,000	32-8	303
SNETTISHAM	ALASKA	1,2	GE	514	26,200	0.90	23,580	13.8	1.0	34.0	1,300,000	26-0x16-6	
SNETTISHAM	ALASKA	3	SIEMENS	600	34,500	0.90	31,050	13.8	1.1	34.0	1,208,000	26-0x17-0	168
STOCKTON	MISSOURI	1	GE	75	47,579	0.95	45,200	13.8	1.175	47.0	86,500,000	48-0	716
TABLE ROCK	ARK. & MO.	1-4	W'HSE	128.6	52,632	0.95	50,000	13.8	1.17	41.0	40,000,000	38-0	438
TENKILLER FERRY	OKLAHOMA	1,2	ELLIOT	150	17,895	0.95	17,000	13.8	1.17	44.0	8,500,000	22-3	173.5
THE DALLES	OREGON	1-14	GE	85.7	82,105	0.95	78,000	13.8	1.175	44.0	138,000,000	50-8	949
THE DALLES	OREGON	15-22	GE	80	90,500	0.95	85,975	13.8	1.175	44.0	163,000,000	50-4	807.9
THE DALLES	OREGON	F1,F2	W'HSE	200	14,210	0.95	13,500	13.8	1.175	43.0	2,900,000	24-7	165
THE DALLES	OREGON	SS(2)	EM	277	3,750	0.80	3,000	4.2	1.00	NORM	360,000	16-0(MAX)	50.5
TOWN BLUFF	TEXAS	1-2	CGE	163.6	4,444	0.90	4,000	4.2	1.10	31.5	547,000	16-7x14-4	47.8
WALTER F. GEORGE	GA & ALA.	1-4	ELLIOT	112.5	36,111	0.90	32,500	13.8	1.10	43.0	40,000,000	36-0	464.5
WAPPAPELLO	MISSOURI	1											
WEBBERS FALLS	OKLAHOMA	1-5	GE	514	21,053	0.95	20,000	13.8	1.175	35.0	600,000	11-10	113.5
WEST POINT	GEORGIA	1	GE	327	3,750	0.90	3,375	4.2	1.10	44.0	275,000	15-5X16-1	53
WEST POINT	GEORGIA	2,3	GE	100	36,842	0.95	35,000	13.8	1.175	44.0	35,200,000	40-8	381.1
WHITNEY	TEXAS	1,2	A-C	128.6	16,667	0.90	15,000	13.8	1.10	45.0	11,000,000	33-0	215.5
WOLF CREEK	KENTUCKY	1-6	GE	105.9	50,000	0.90	45,000	13.8	1.10	43.0	55,000,000	40-0	560

TABLE B-3

PUMP-TURBINE PERFORMANCE DATA

POWER PLANT	STATE	UNIT	TYPE	MFR.	RPM	TURBINE DATA RATING HP	TURBINE DATA RATING HEAD (FT)	TURBINE DATA MAX. HEAD (FT)	TURBINE DATA MIN. HEAD (FT)	PUMP DATA RATING C.F.S.	PUMP DATA RATING HEAD (FT)	PUMP DATA MAX. HEAD (FT)	PUMP DATA MIN. HEAD (FT)
CARTERS	GEORGIA	3,4	FRANCIS	A-C	150	173,000	345	427.5	320	4,435	347	327	327
CLARENCE CANNON	MISSOURI	2	FRANCIS	A-C	75	42,900	75	107	69	5,500	60	60	60
DEGRAY	ARKANSAS	2	FRANCIS	NNS&D	128.6	44,500	171	216	144	1,900	170	170	150
TRUMAN	MISSOURI	1 - 6	ADJ.BL.INCL.AX.	BLH	100	42,400	42.5	79.2	41	4,500	50	50	44.4

TABLE B-4

PUMP-TURBINE WEIGHTS AND DIMENSIONS

POWER PLANT	STATE	UNIT NOS.	TYPE	UNIT SPACING C TO C (FT)	C DIST. TO BOT'M OF DRAFT TUBE (FT)	C DIST. TO MIN. T.W., 1 UNIT GENERATING (FT)	C DIST. TO MIN. T.W., PUMPING MODE (FT)	RUNNER DISCHARGE DIAMETER (FT.-IN.)	RUNNER INLET DIAMETER (FT.-IN.)	TOTAL WEIGHT (TONS)
CARTERS	GEORGIA	3,4	FRANCIS	63	28	-21.8	-21	14 - 2	20 - 8	797
CLARENCE CANYON	MISSOURI	2	FRANCIS	82	50	-9	0	19 - 3	22 - 6.4	1,036
DEGRAY	ARKANSAS	2	FRANCIS	49	32	-1	-1	11 - 3.8	15 - 9.5	32
TRUMAN	MISSOURI	1 - 6	ADJ.JBL.INCL.	52 & 57	35	-30.4	-26	21 - 2	N/A	683.5

TABLE B-5

GENERATOR-MOTOR DATA

POWER PLANT	STATE	UNIT NOS.	MFR.	RPM	RATING KVA	RATING PF	RATING KW	RATING KV	S-C RATIO NLT	TRANSIENT REACTANCE NNT (PERCENT)	HP	PF	KV	WK2 NLT (LB.-FT2)	HOUSING DIAMETER (FT.-IN.)	TOTAL WEIGHT (TONS)
CARTERS	GEORGIA	3 - 4	GE	150	131,579	0.95	125,000	13.8	1.175	34.0	185,000	0.95	13.8	90,000,000		651.0
CLARENCE CANNON	MISSOURI	2	GE	75	28,421	0.95	27,000	13.8	1.175	42.0	40,463	0.95	13.5	53,980,000	45 - 0	432.5
DEGRAY	ARKANSAS	2	A-C	128.6	29,474	0.95	28,000	13.8	1.175	33.0	43,240	0.95	13.5	19,200,000	39 - 0	274.0
TRUMAN	MISSOURI	1 - 6	GE	100	28,386	0.95	26,967	13.8	1.175	45.0	36,000	0.95	13.8	26,000,000	40 - 0	328.0

APPENDIX C

TURBINE SELECTION CHARTS AND DIMENSIONS

DATA AND DIMENSIONAL RATIOS - D_{TH} = 1.0

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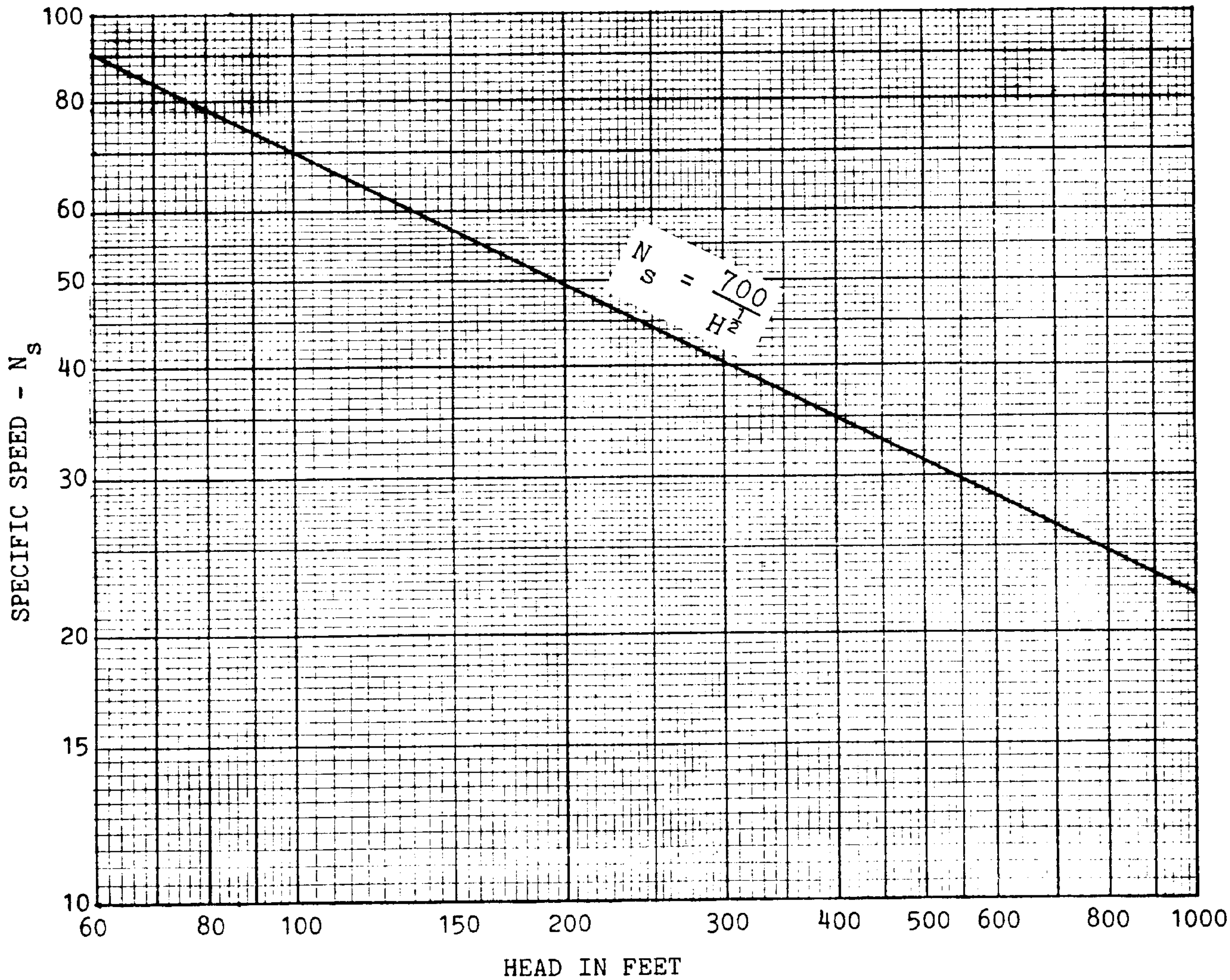


Figure 1. FRANCIS TYPE TURBINES

BEST EFFICIENCY AND PHI

SPECIFIC SPEED VS. HEAD

Note: Plotted points are from existing installations.

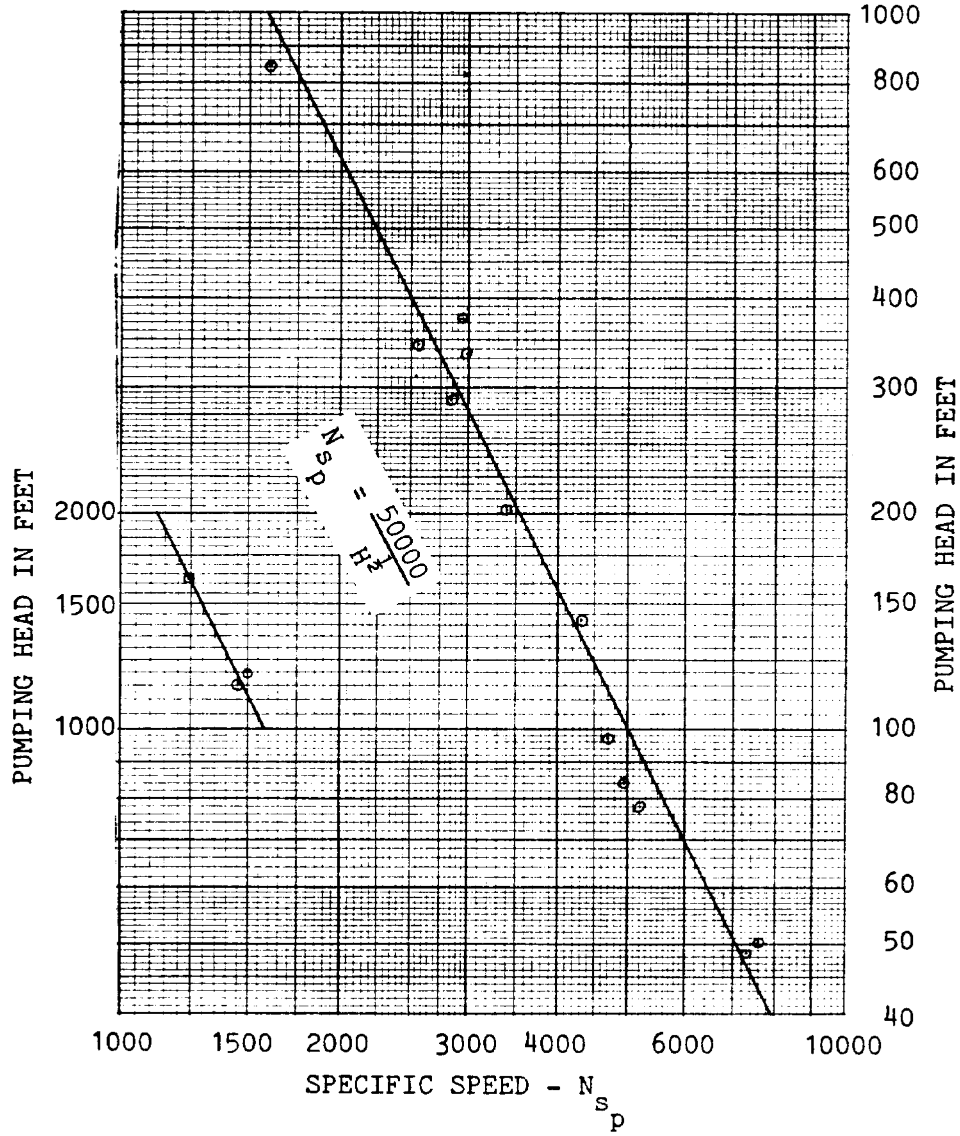
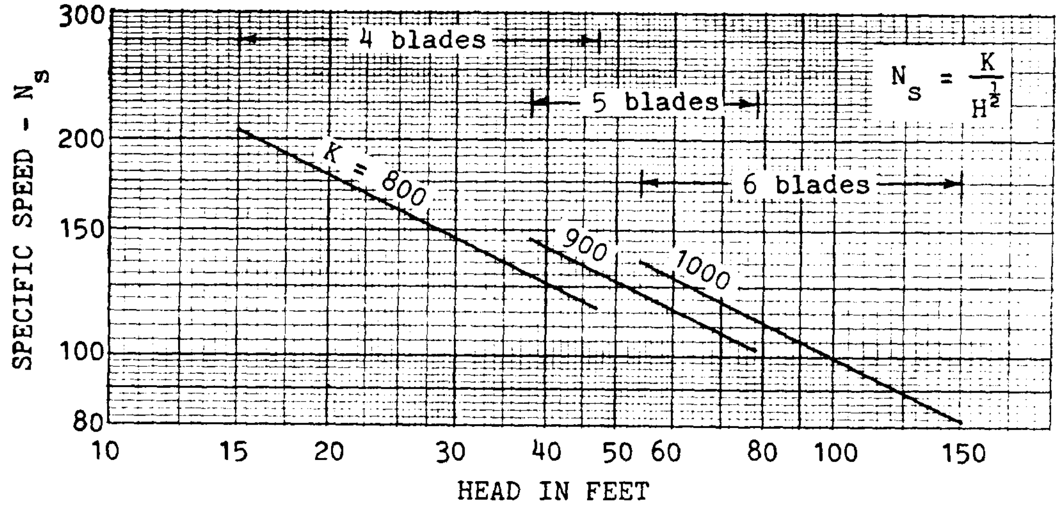


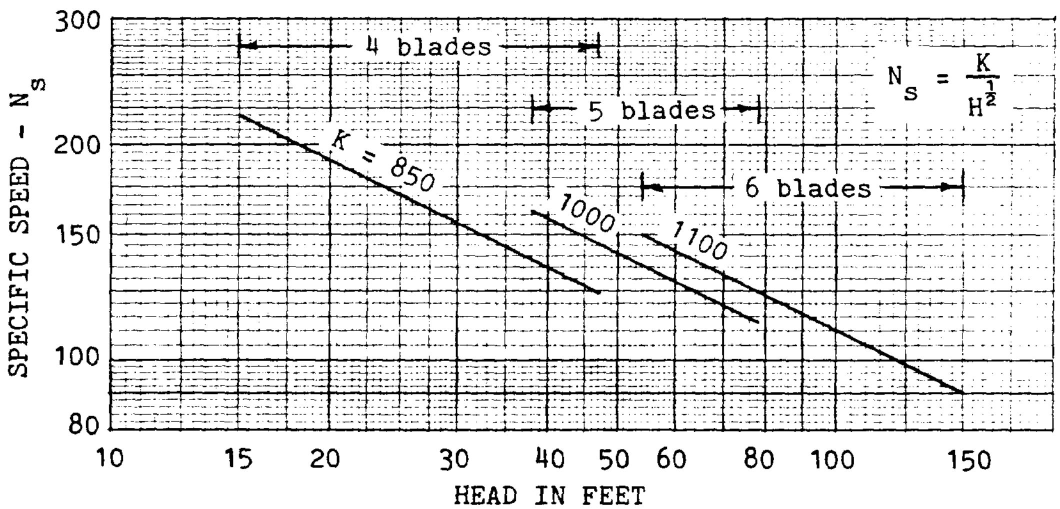
Figure 2. FRANCIS TYPE PUMP-TURBINES

RATED PUMPING CONDITIONS

PUMPING HEAD VS. SPECIFIC SPEED



FIXED BLADE RUNNER



KAPLAN (ADJUSTABLE BLADE) RUNNER

Figure 3. PROPELLER TYPE TURBINES

RATED CONDITIONS

SPECIFIC SPEED VS. HEAD

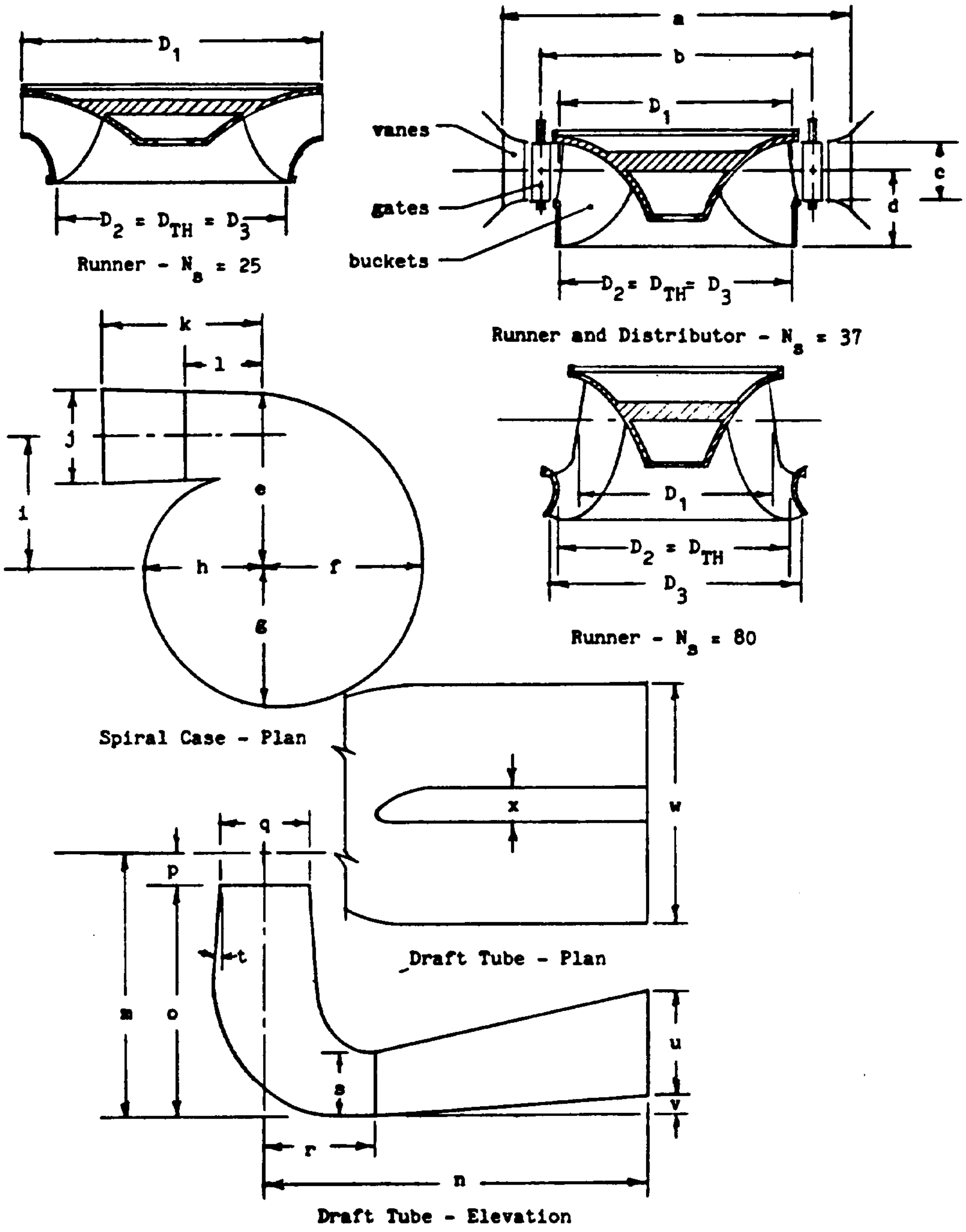


Figure 4. FRANCIS TYPE TURBINES - DIMENSIONS AND PROPORTIONS

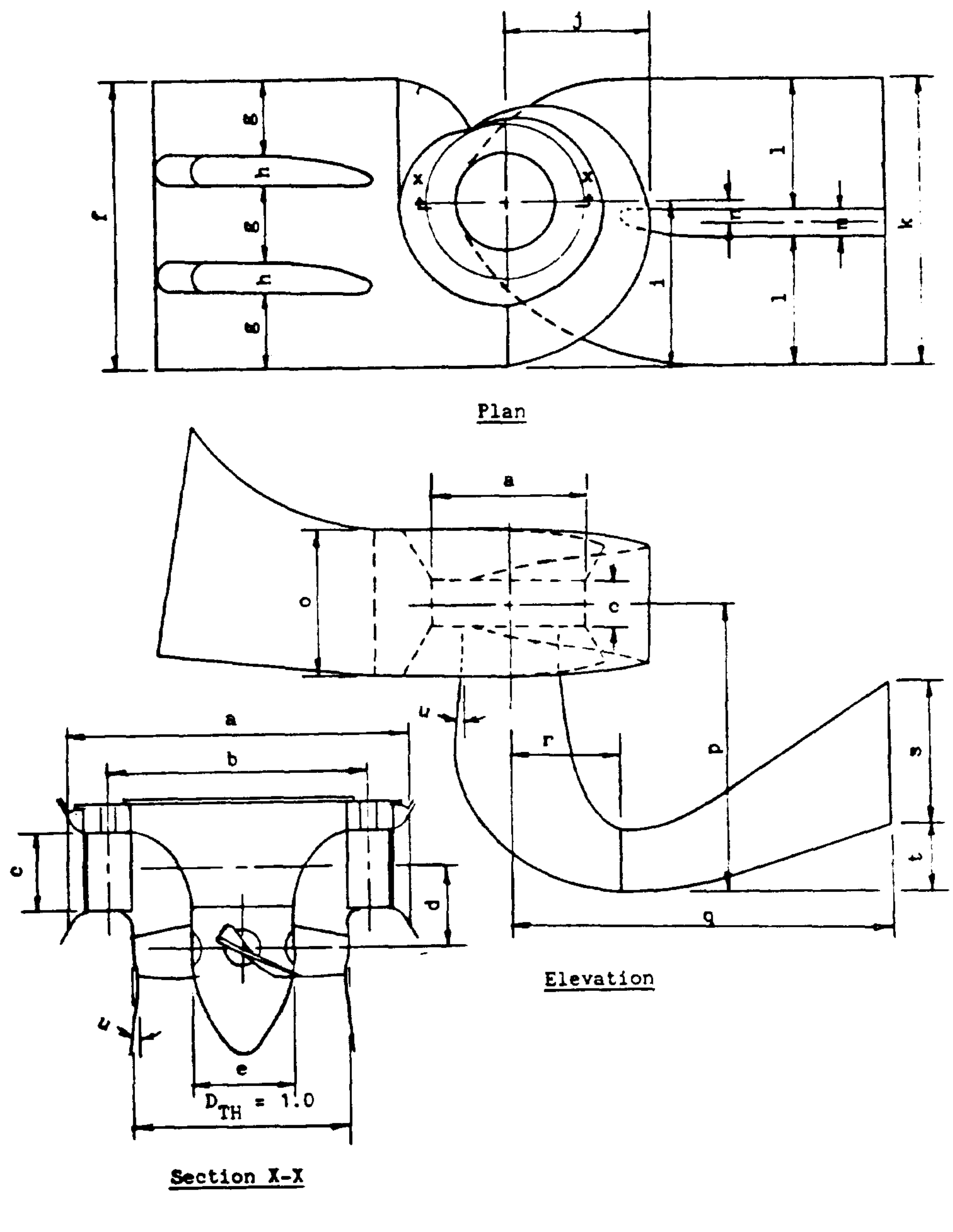


Figure 5. PROPELLER TYPE TURBINES - DIMENSIONS

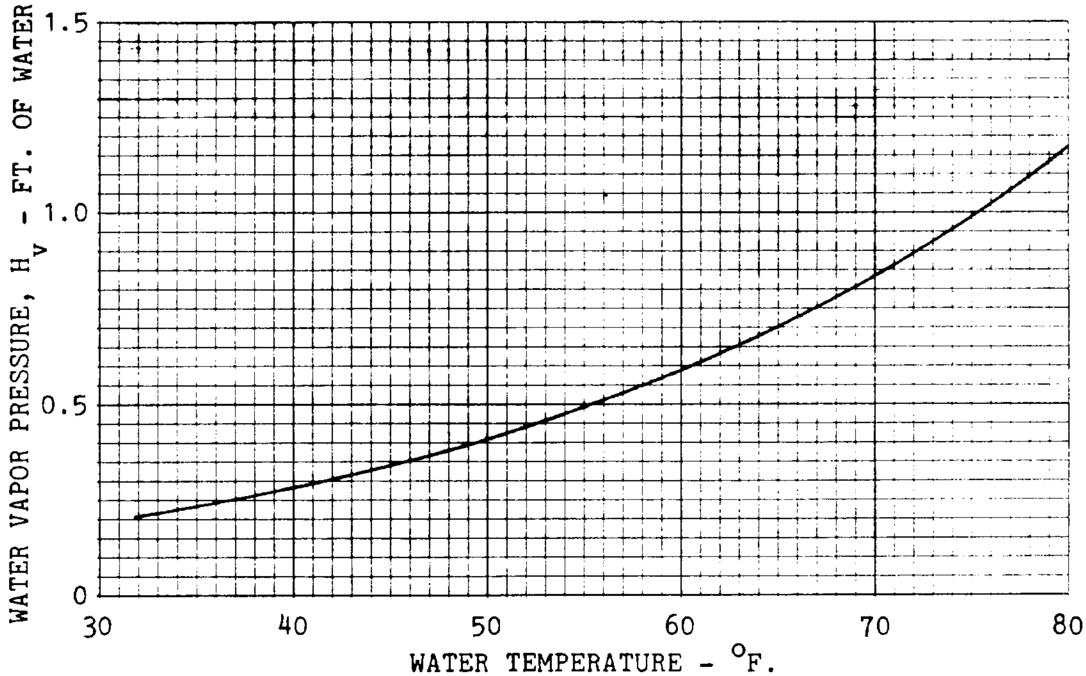


Figure 6(a). WATER VAPOR PRESSURE vs. WATER TEMPERATURE

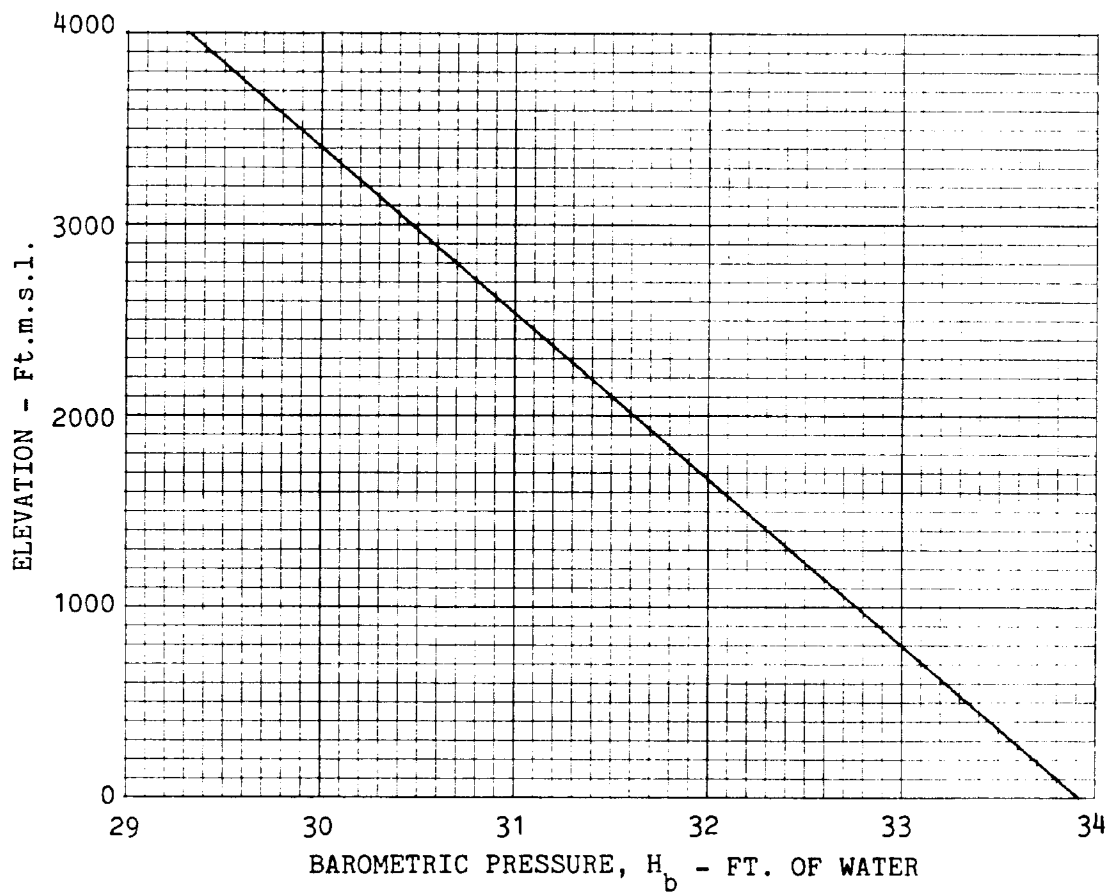


Figure 6(b). BAROMETRIC PRESSURE vs. ELEVATION

TABLE 1

FRANCIS TYPE TURBINES

DATA AND DIMENSIONAL RATIOS - D_{TH} = 1.0

(Refer to Fig. 4)

Fig. No.	F1	F2	F3	F4	F5	F6	F7	F8	(1)
N_s	27.2	34.3	37.2	44.7	50.6	63.1	65.0	80.0	42.8
vanes	22	20	24		20	20	24	10	20
gates	22	20	24	24	20	20	24	20	20
buckets	14	16	15	18	15	15	13	15	17
D_1	1.207	1.064	1.011	0.976	0.920	0.883	1.008	0.859	1.056
$D_2 = D_{TH}$	12"	12.09"	12"	11"	12.52"	12"	13"	13.57"	12"
D_3	1.000	1.000	1.000	1.000	1.018	1.083	1.054	1.136	1.000
a	1.895	1.698	1.498		1.518	1.698	1.590	1.639	1.715
b	1.385	1.256	1.171	1.172	1.150	1.271	1.194	1.217	1.266
c	0.184	0.177	0.231	0.254	0.290	0.333	0.299	0.429	0.160
d	0.320	0.343	0.323	0.352	0.377	0.431	0.429	0.434	0.287
e	1.883	1.803	1.705				1.856		1.793
f	1.738	1.656	1.560				1.773		1.657
g	1.569	1.476	1.384				1.572		1.501
h	1.313	1.198	1.106				1.243		1.274
i	1.440	1.307	1.281				1.254		1.315
j	0.976		1.179				1.155		1.000
k	1.728		2.072				1.933		0.953
l	0.864		0.742						
m	2.876	2.960	2.830		2.717	3.101	3.041	3.229	2.370
n	4.153	4.365	3.982		4.328	4.512	4.999	3.377	3.424
o	2.546	2.609	2.501		2.353	2.647	2.610	2.795	2.079
p	0.330	0.351	0.329		0.384	0.454	0.431	0.434	0.290
q	1.003	1.003	1.006		1.023	1.097	1.060	1.141	1.004
r	1.229	1.284	1.207		1.228	1.626	1.500	1.483	1.260
s					0.747				
t	4°	4.53°	4.68°		4.5°	7°*	6.5°		
u	1.156	1.094	1.281		1.427	1.260	1.244	1.061	0.920
v	0	0.261	0		0.593	0.479	0.305	0	
w	3.375	3.231	3.540		3.039	3.983		4.118	3.500
x	0.313	0.388*	0.337		0.266	0.303*		0.331	0.380

* two piers (1) other designs

TABLE 2

FRANCIS AND PROPELLER TYPE TURBINES
DATA AND DIMENSIONAL RATIOS - D_{TH} = 1.0

(Refer to Fig. 4)

							Propeller Type		
Fig. No.	(1)	(1)	(1)	(1)			FB3	F(1)	K(1)
N_s	48	50.8	57.6	65.2					
vanes	23	20	20	10			(see Table 4, column 3 for other data)	(see Table 5, column 5 for other data)	(see Table 5, column 4 for other data)
gates	24	20	20	20					
buckets	13	17	15	15					
D_1	1.060	0.935	0.968	0.807					
$D_2 = D_{TH}$	13.72"	12.52"	15.75"	14.87"					
D_3	1.000	1.018	1.022	1.081					
a	1.600	1.518	1.542	1.580					
b	1.239	1.150	1.164	1.190					
c	0.267	0.290	0.317	0.403					
d	0.402	0.377	0.399	0.441					
e	1.797						2.199	2.174	1.998
f	1.636						1.989	1.962	1.796
g	1.448						1.715	1.708	1.553
h	1.179							1.305	1.160
i	1.273	(open flume)		(open flume)			1.523	1.458	1.333
j	1.205		1.310				1.587	1.467	1.377
k	1.386		1.577				2.215	1.620	
l									
m	3.117	2.711	2.754	2.646					
n	5.122	4.328	4.054	5.867					
o	2.712	2.333	2.355	2.201					
p	0.404	0.379	0.399	0.445					
q	1.004	1.023	1.024	1.084					
r	1.536	1.228	1.270	1.831					
s									
t	6.9°	4.5°	5.3°	7.6°					
u	1.276	1.238	1.095	1.149					
v	0.313	0.593	0.522	0					
w	3.272	3.039	3.860	3.307					
x	0.321*	0.266	0.328	0.304*					

* Two piers F = fixed blade; K = Kaplan
(1) other designs

TABLE 3

FRANCIS TYPE PUMP TURBINES
DATA AND DIMENSIONAL RATIOS - D_{TH} = 1.0

(Refer to Fig. 4)

Fig. No.	PT1	PT2	PT3	(3)	(3)	(3)	(3)		
N_{s_p} (1)	2150	3200	4700	2600	4500	2650	5100		
vanes	20	14	10	20	14	20	10		
gates	20	28	20	20	28	20	20		
buckets	7	6	6	6	6	6	6		
D_1	1.604	1.317	1.154	1.493	1.135	1.459	1.078		
$D_2 = D_{TH}$	9.995"	11.205"	13.000"	12.328"	12.000"	12.850"	12.030"		
D_3	1.000	1.000	1.000	1.000	1.000	1.000	1.000		
a	2.369	1.832	1.810	2.254	1.712	2.101	1.701		
b	1.876	1.479	1.423	1.785	1.346	1.712	1.342		
c	0.221	0.289	0.315	0.264	0.334	0.253	0.336		
d	0.434	0.399	0.385	0.433	0.385	0.409	0.392		
e	2.159	1.855	2.050	1.967	2.005	1.919	2.004		
f	2.010	1.703	1.887	1.831	1.829	1.784	1.829		
g	1.826	1.521	1.690	1.665	1.615	1.620	1.620		
h	1.570	1.257	1.394	1.435	1.316	1.388	1.325		
i	1.699	1.381	1.455	1.501	1.494	1.470	2.524		
j	1.039	1.071	1.212	0.923	1.351	0.921	1.214		
k	2.647	1.607	1.682	1.622	1.247	1.000	1.666		
l									
m	2.823	2.853	2.690	4.329	2.598	2.353	2.476		
n	10.525	3.893	3.700	5.640	4.610	4.747	3.698		
o	2.382	2.473	2.286	3.863	2.209	1.940	2.035		
p	0.441	0.380	0.404	0.465	0.389	0.413	0.441		
q	1.000	1.000	1.000	1.000	1.000	1.000	1.000		
r	1.299	1.284	1.183		1.173	1.118	1.240		
s			0.638	(2)		0.600			
t	7°	10°	7°	6°	7.4°	7°			
u	2.162	1.238	1.009	1.615	1.507	1.403	1.012		
v	1.941	0.555	0.404	0.470	0.623	0.903	0.389		
w	3.838	3.094	3.630	1.615	3.429	3.600			
x	0.485	0.309*	0.269*	(2)	0.312*	0.424*			

(1) Gpm units at e_{max}.; (2) No pier, as horizontal leg is circular.

*** Two piers ; (3) other designs**

TABLE 4

PROPELLER TYPE TURBINES
DATA AND DIMENSIONAL RATIOS - D_{TH} = 1.0

(Refer to Fig. 5)

Fig. No	FB1	FB2	FB3	K1	K2	K3	K (1)	F-27° (1)	F-30° (1)
vanes	10	24	20	10	20	20	10	24	24
gates	20	24	20	20	20	20	20	24	24
blades	4	5	6	4	5	6	4	5	5
D_{TH}-model	12.44"	12"	12.08"	12.44"	12.51"	12"	12.44"	12"	12"
a	1.571	1.443	1.542	1.578	1.501	1.567	1.571	1.477	1.443
b	1.194	1.148	1.192	1.193	1.151	1.200	1.194	1.178	1.148
c	0.482	0.394	0.404	0.482	0.390	0.406	0.482	0.406	0.394
d	0.427	0.367	0.365	0.427	0.370	0.368	0.422	0.368	0.367
e	0.327	0.350	0.327	0.327	0.400	0.440	0.360	0.390	0.350
f	3.000	2.911	(See Table 2) (See Fig. 4)			3.171	3.062		2.911
g	0.803	0.787				0.857	0.855		0.787
h	0.296	0.275				0.300	0.248		0.275
i	1.727	1.780				1.854	1.800		1.780
j	1.482	1.318				1.515	1.467		1.318
k	3.000	2.912	3.261	3.060	3.127	3.171	3.060	3.307	2.912
l	0.818	1.338	1.415	1.406	0.875	1.457	0.855	0.889	1.338
m	0.273*	0.236	0.431	0.248	0.251*	0.257	0.248*	0.321*	0.236
n	0.227	0.325				0.268	0.269		0.325
o	1.566	1.534				1.671			1.534
p	2.773	2.790	2.791	3.020	2.907	3.030	2.780	2.936	2.790
q	4.500	3.627	3.730	3.900	4.391	3.930	4.060	4.529	3.627
r	1.174	1.130	1.231	1.170	1.323	1.220	1.280	1.303	1.130
s	1.424	1.102	1.139	1.160	1.273	1.200	1.240	1.333	1.102
t	0.758	0.659	0.528	0.690	0.882	0.720	0.500	0.962	0.659
u	7.9°	11°	8.1°	5.5°	7.9°	7°	8.1°	7°	7°

* two piers; F = fixed blade; K = Kaplan
(1) other designs

TABLE 5

PROPELLER TYPE TURBINES

DATA AND DIMENSIONAL RATIOS - D_{TH} = 1.0

(Refer to Fig. 5)

	(1)	(1)	(1)	(1)	(1)				
Ident.	K	F-29.6°	K	K	F				
vanes	20	24	24	24	20				
gates	20	24	24	24	20				
blades	5	6	6	6	8				
D_{TH}-model	12"	12"	12"	12"	12"				
a	1.574	1.443	1.443	1.443	1.580				
b	1.200	1.148	1.148	1.148	1.200				
c	0.406	0.394	0.446	0.446	0.409				
d	0.368	0.367	0.394	0.394	0.551				
e		0.350	0.440	0.440					
f		2.911	2.911	(See Table 2) (See Fig. 4)	(See Table 2) (See Fig. 4)				
g		0.787	0.787						
h		0.275	0.275						
i		1.780	1.780						
j		1.318	1.318						
k		2.912	2.912	2.892	3.394				
l		1.338	1.338	0.787	1.457				
m		0.236	0.236	0.266*	0.480				
n		0.325	0.325						
o		1.534	1.534						
p	3.030	2.780	2.807	2.628	3.028				
q	3.930	3.627	3.627	4.063	4.414				
r	1.220	1.130	1.130		1.221				
s	1.200	1.102	1.102	1.137	1.326				
t	0.720	0.659	0.659	0.545	0.780				
u		7°	11°	4.6°	7°				

* two piers; F = fixed blade; K = Kaplan
(1) other designs

APPENDIX D

MODEL TEST CURVES FOR D_{TH}= 12 INCHES AND H = 1 FOOT AND CRITICAL RUNNER SIGMAS

SECTION I

FRANCIS TURBINE MODEL TEST CURVES FOR D_{TH}= 12 INCHES AND H = 1 FOOT

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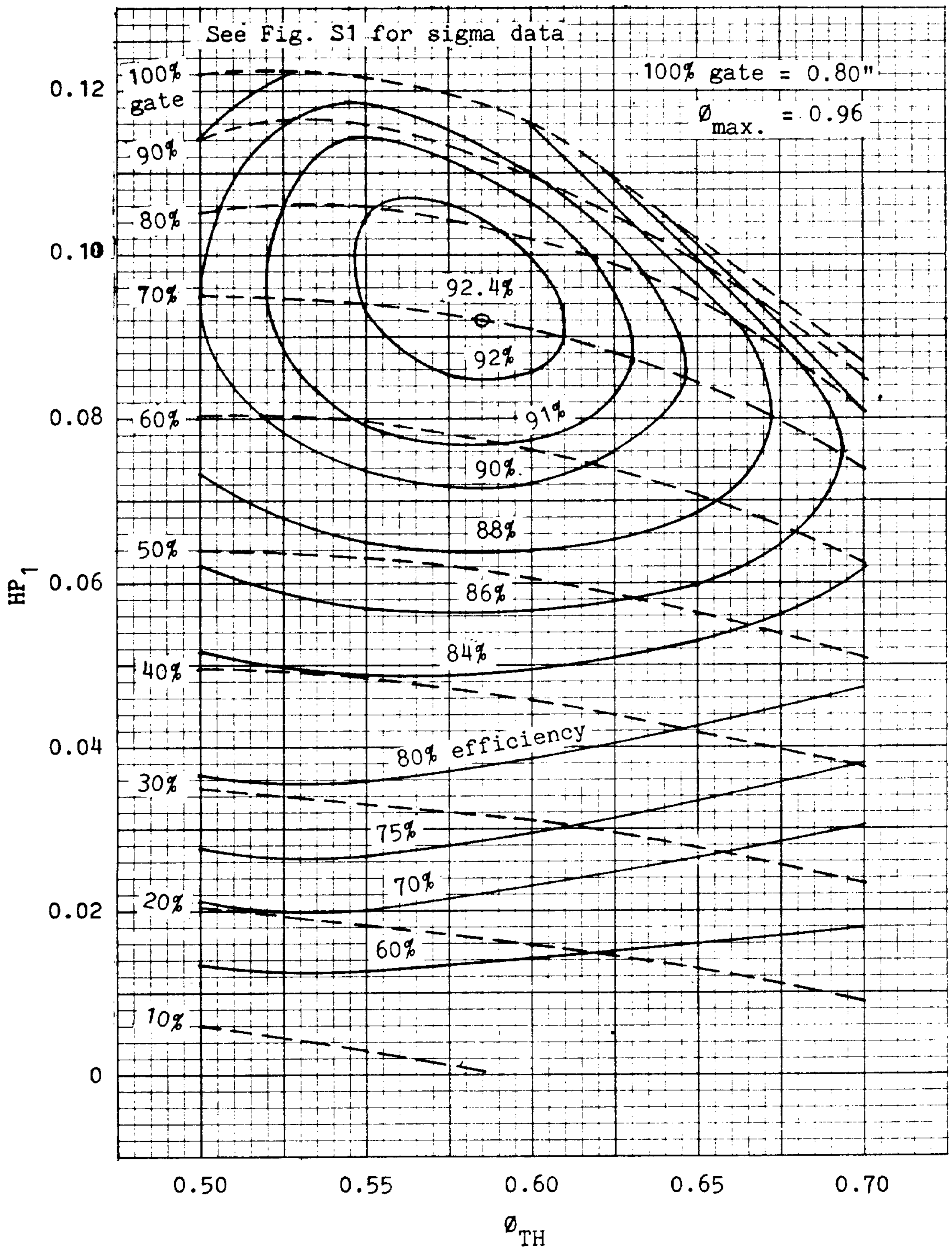

Figure F1. MODEL TEST DATA - FRANCIS TURBINE

D_{TH} = 12"   ONE FOOT HEAD N_s = 27.2

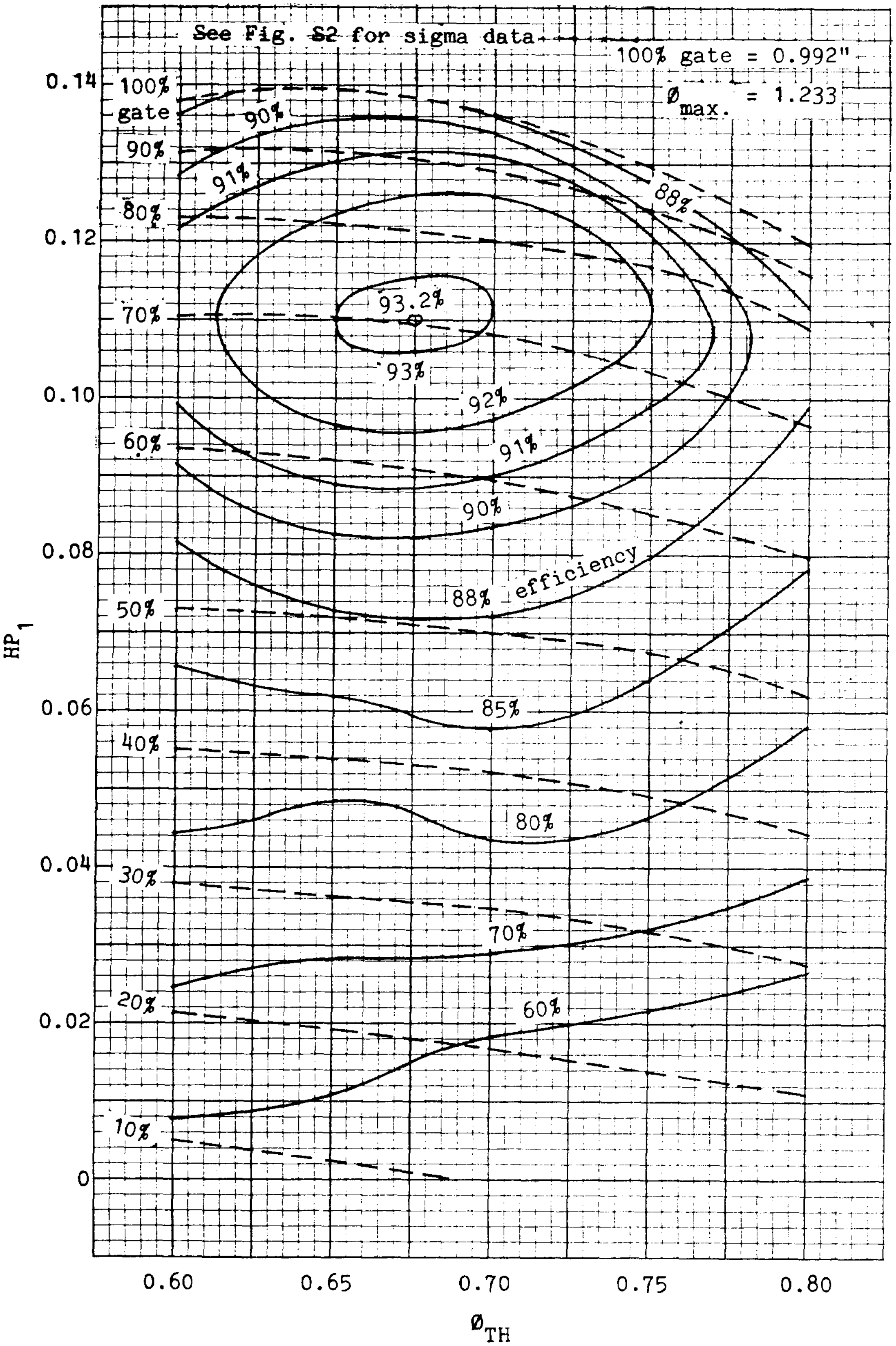

Figure F2. MODEL TEST DATA - FRANCIS TURBINE

D_{TH} = 12" ONE FOOT HEAD N_s = 34.3

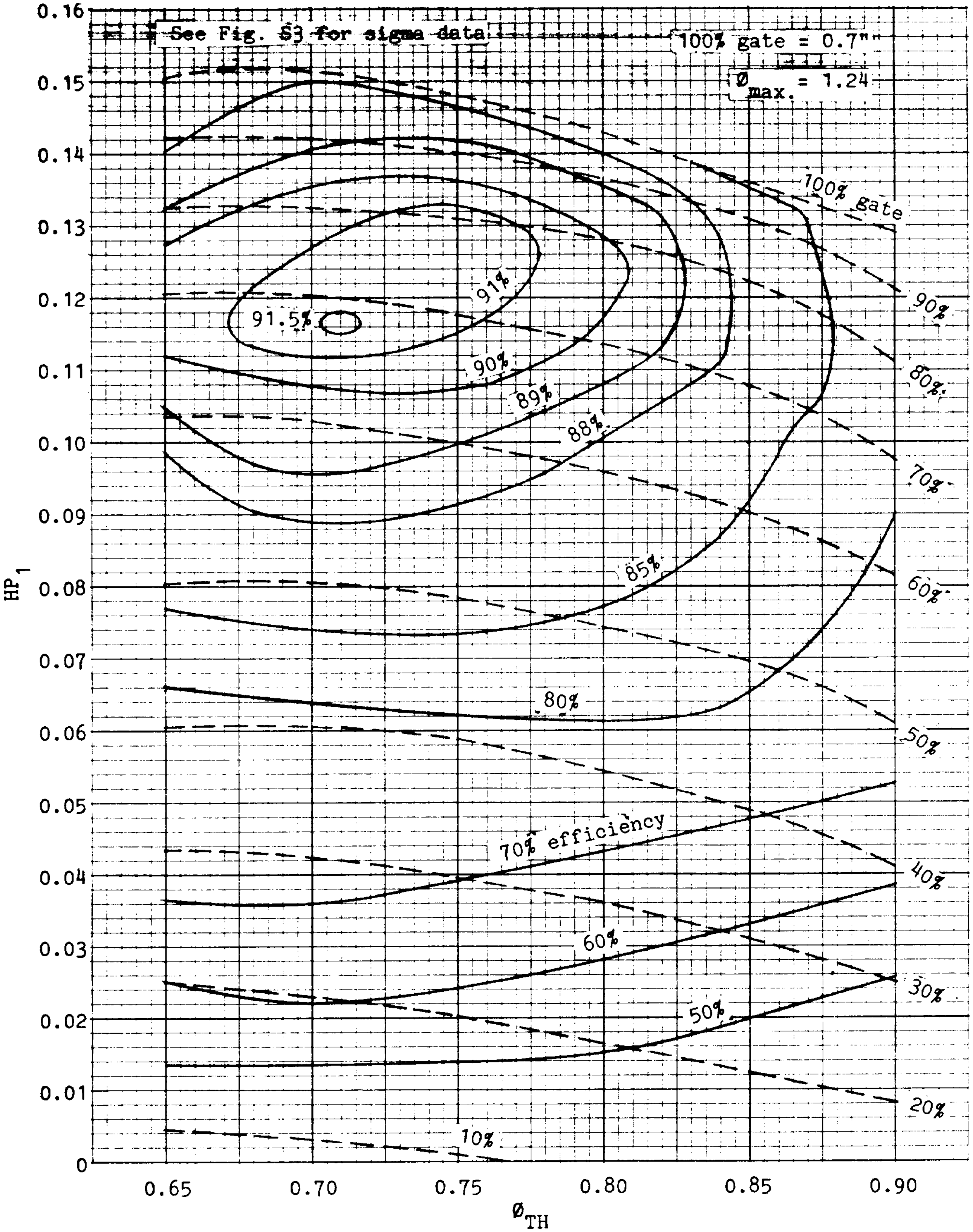

Figure F3. MODEL TEST DATA - FRANCIS TURBINE

D_{TH} = 12" ONE FOOT HEAD N_s = 37.2

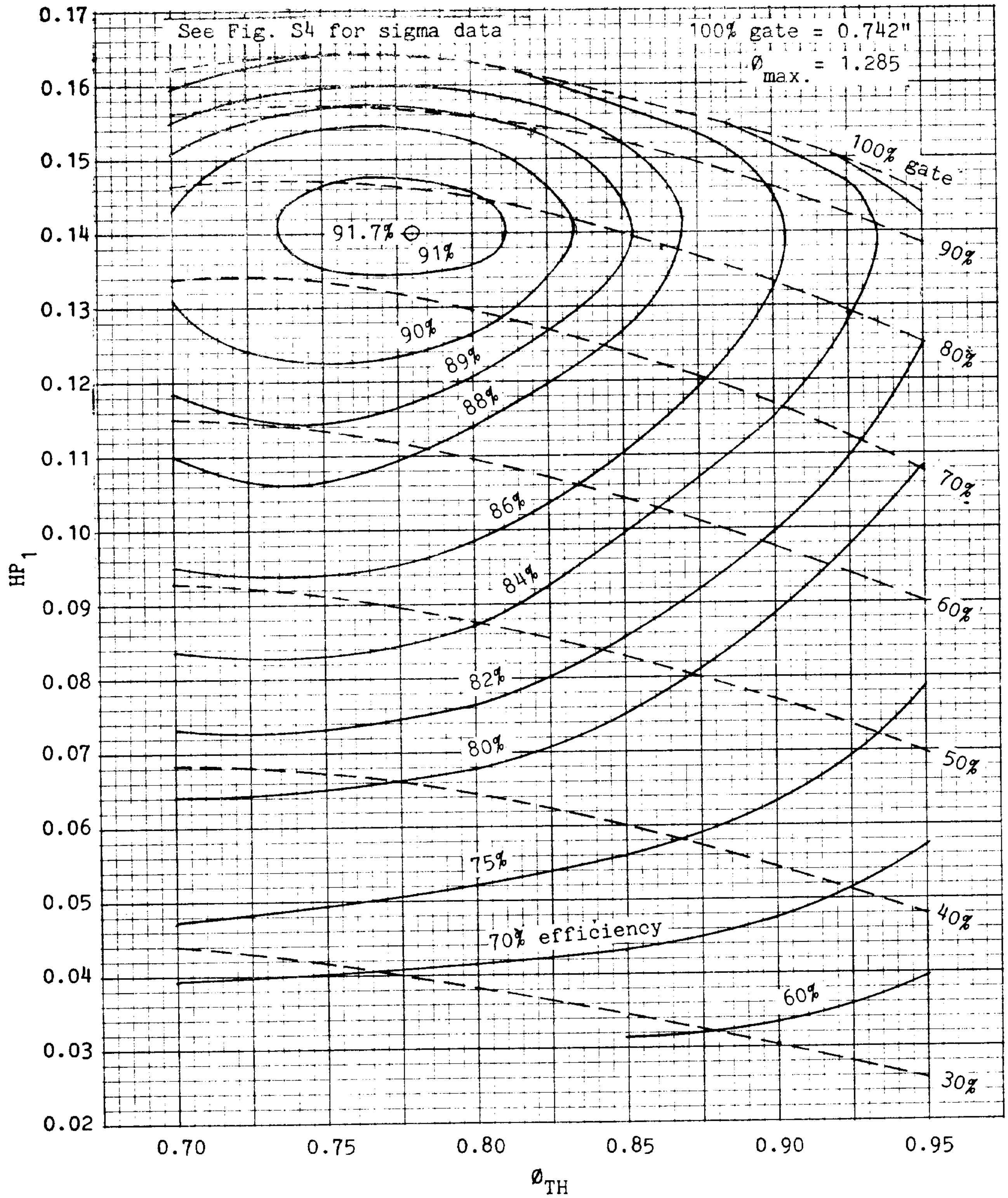

Figure F4. MODEL TEST DATA - FRANCIS TURBINE

D_{TH} = 12"   ONE FOOT HEAD N_s = 44.7

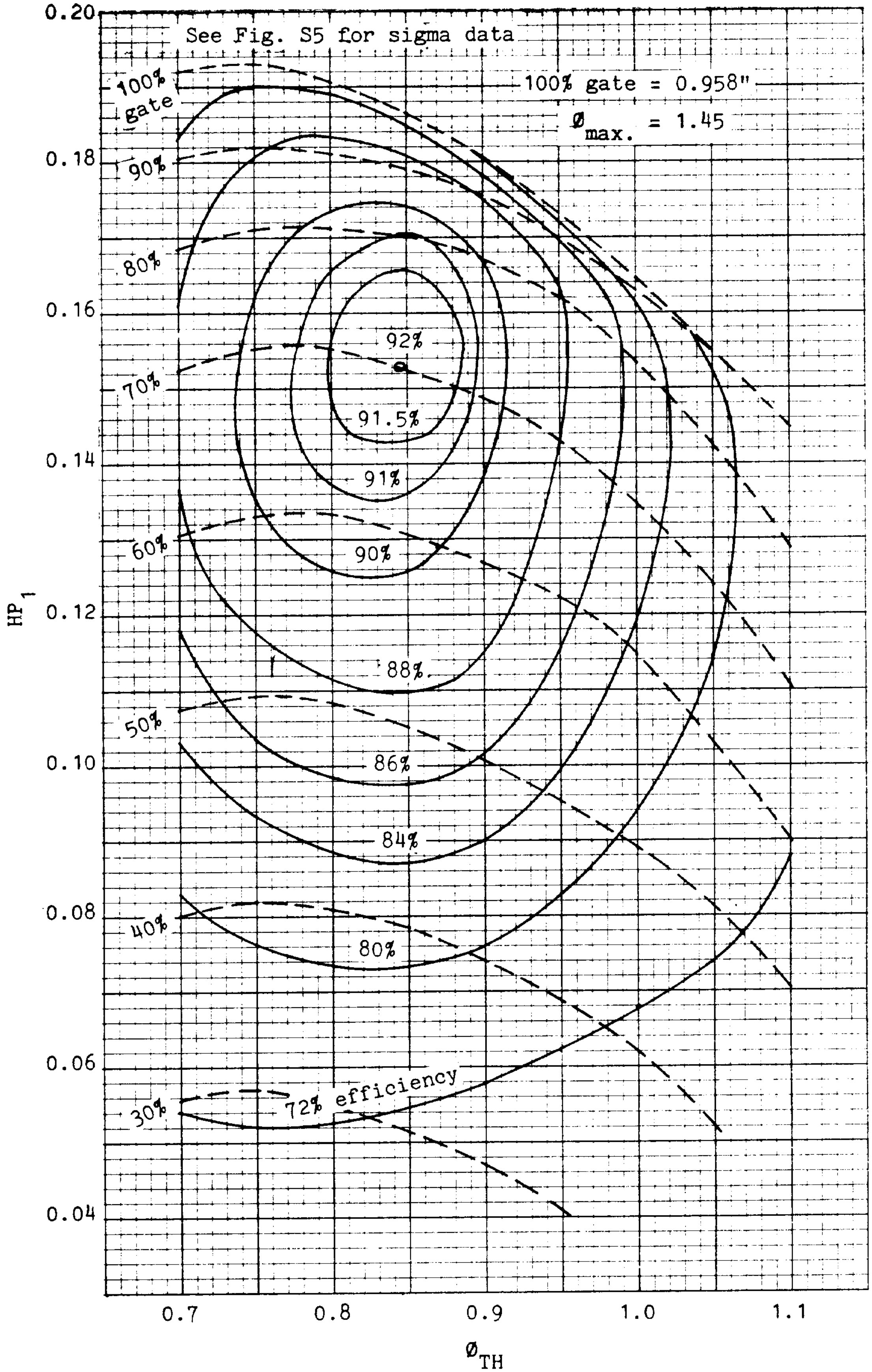

Figure F5. MODEL TEST DATA - FRANCIS TURBINE

D_{TH} = 12" ONE FOOT HEAD N_s = 50.6

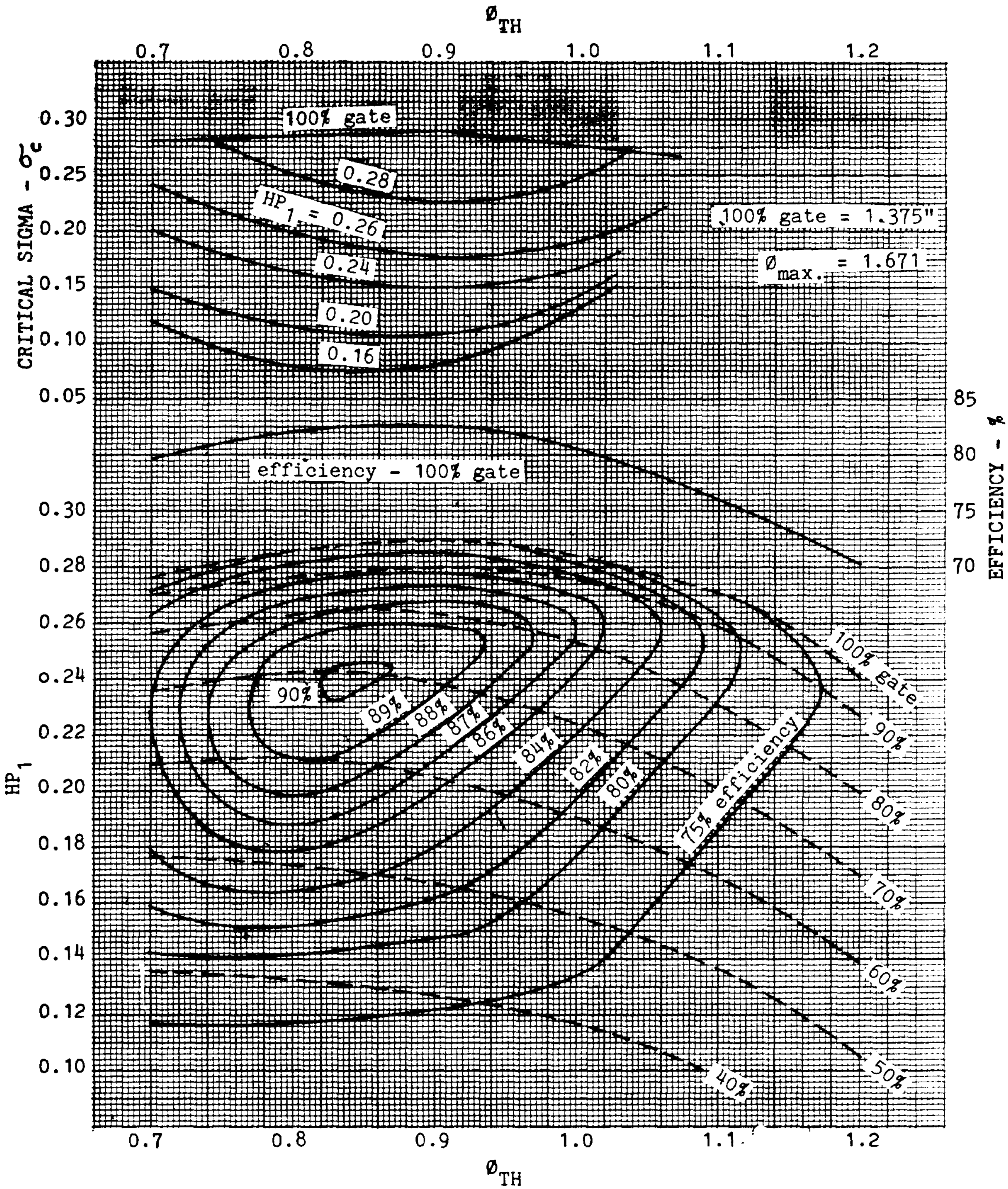

Figure F6. MODEL TEST DATA - FRANCIS TURBINE

D_{TH} = 12" ONE FOOT HEAD N_s = 63.1

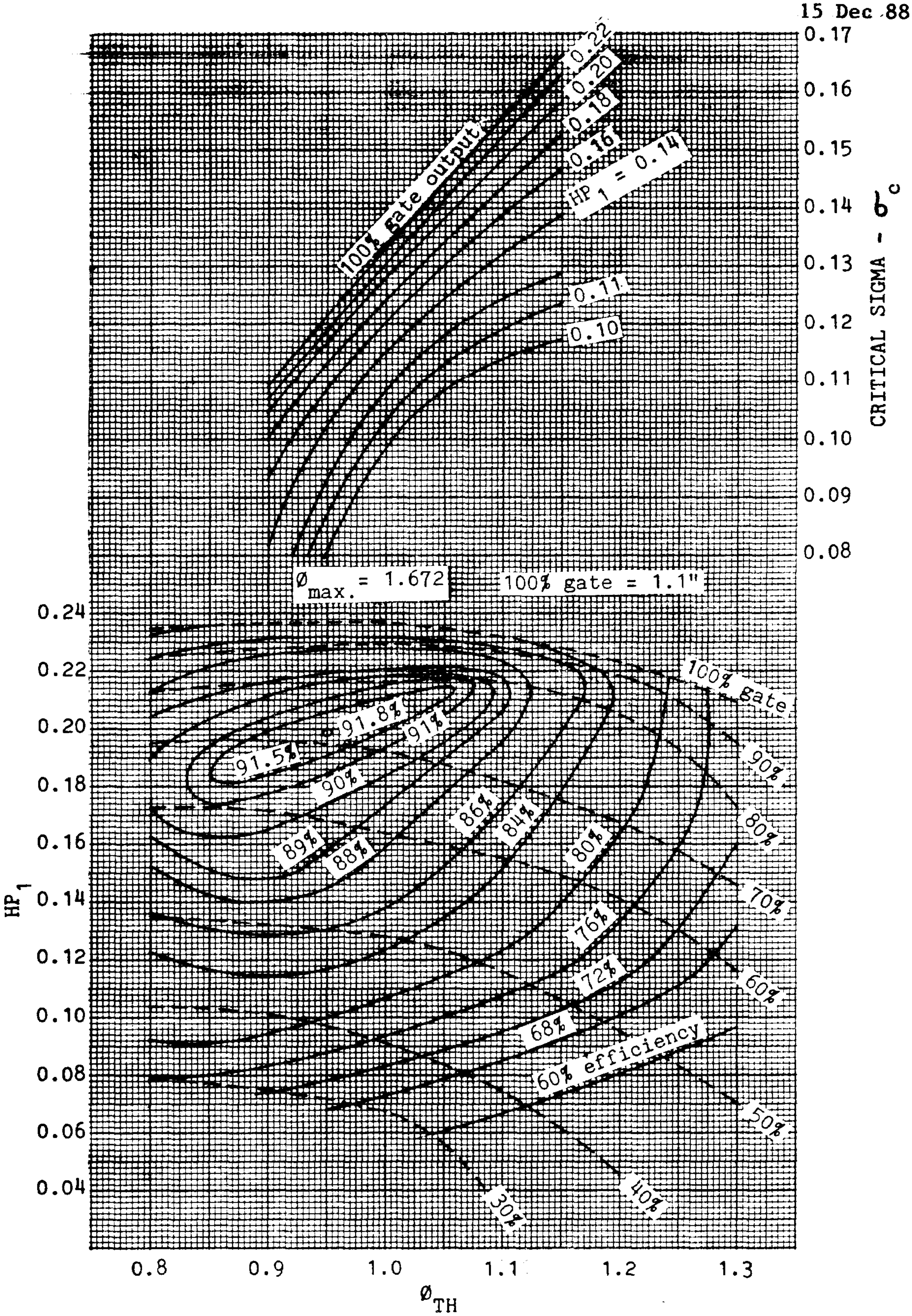

Figure F7. MODEL TEST DATA - FRANCIS TURBINE

D_{TH} = 12" ONE FOOT HEAD N_s = 65

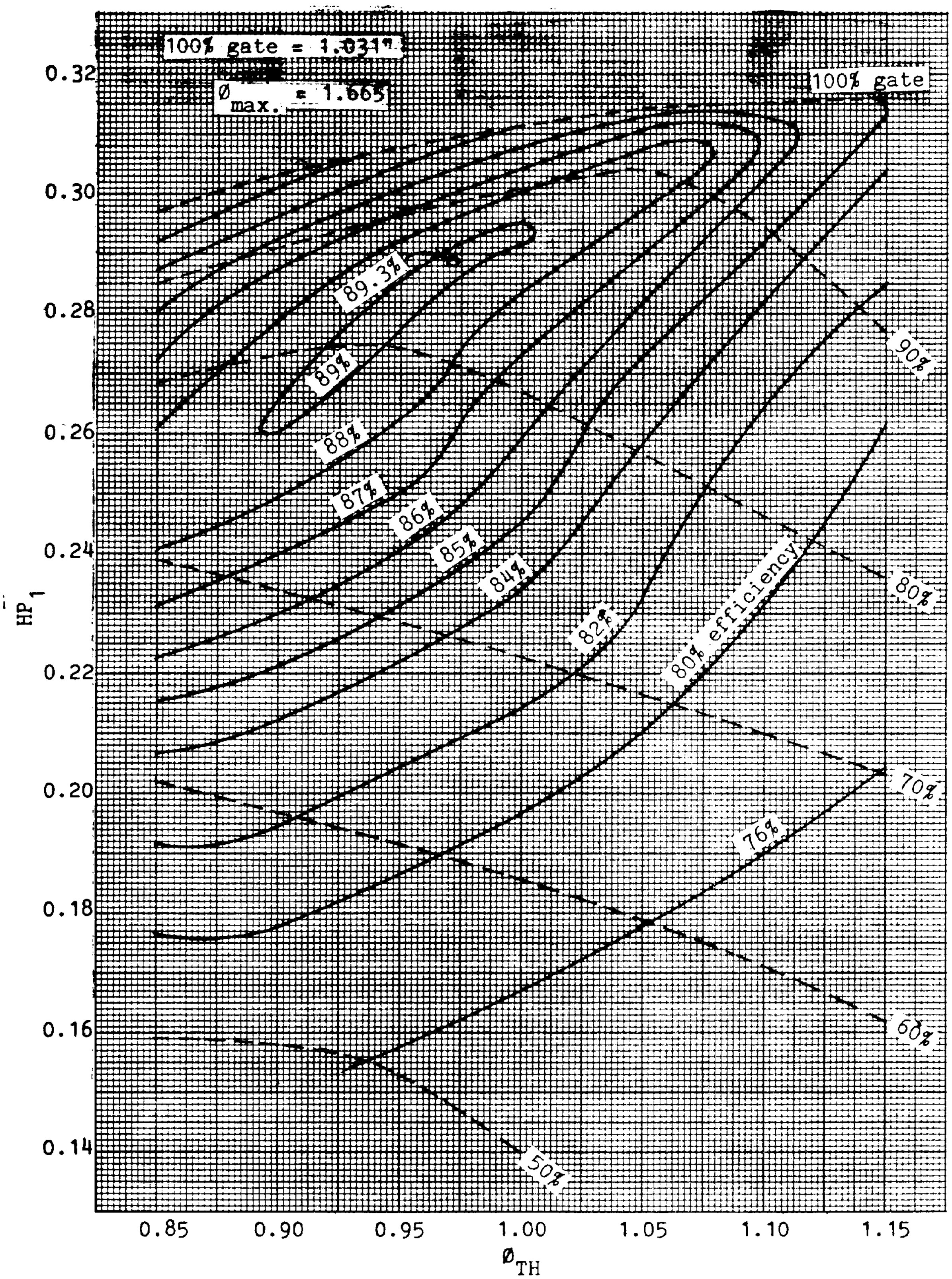

Figure F8. MODEL TEST DATA - FRANCIS TURBINE

D_{TH} = 12" ONE FOOT HEAD N_s = 80

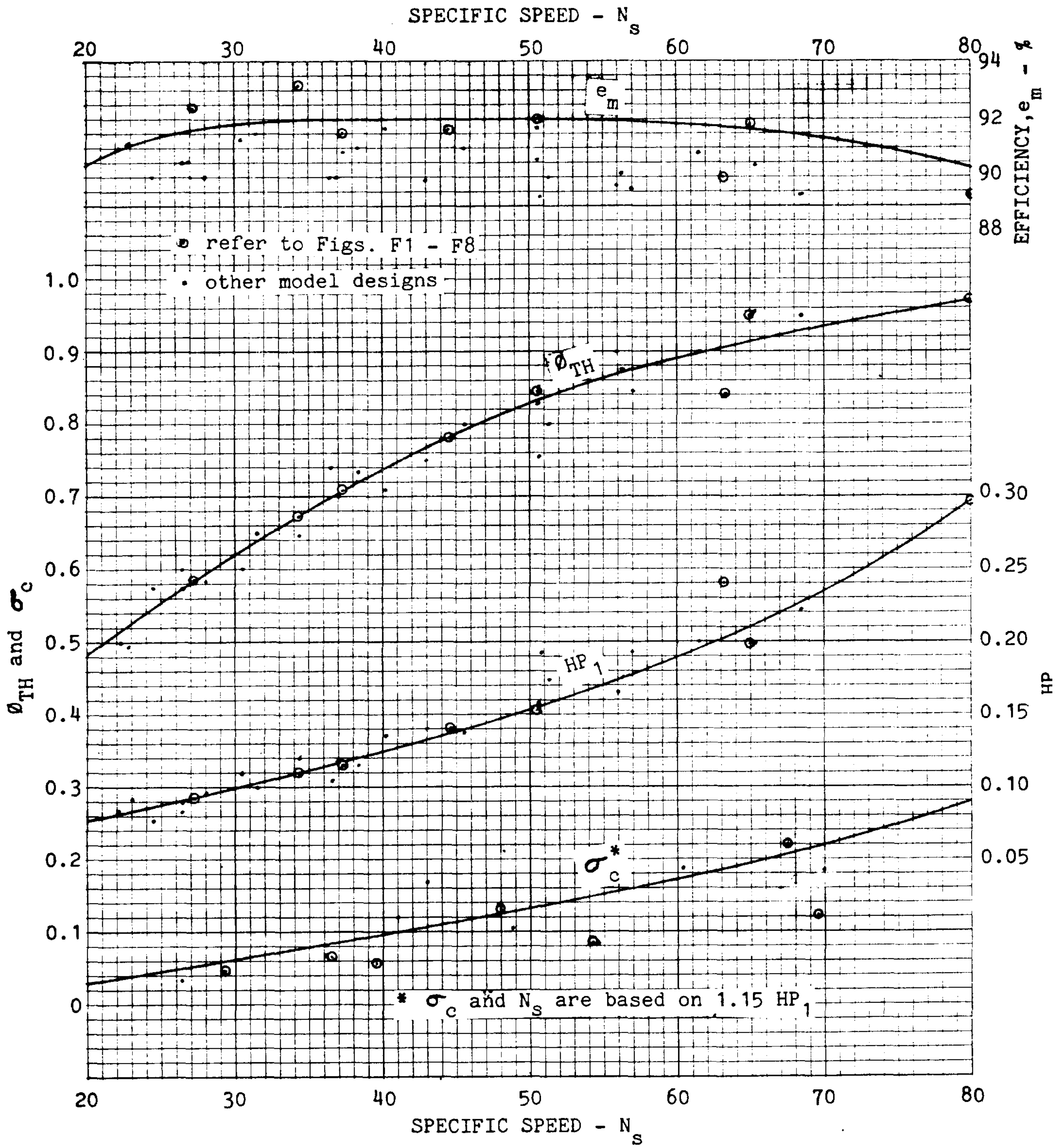

Figure F9. FRANCIS TURBINES; PERFORMANCE HILL DATA - BEST PHI CONDITIONS. D_{TH} = 12" ONE FOOT HEAD

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SECTION II

FRANCIS PUMP-TURBINE MODEL TEST CURVES FOR D_{TH}= 12 INCHES AND H = 1 FOOT

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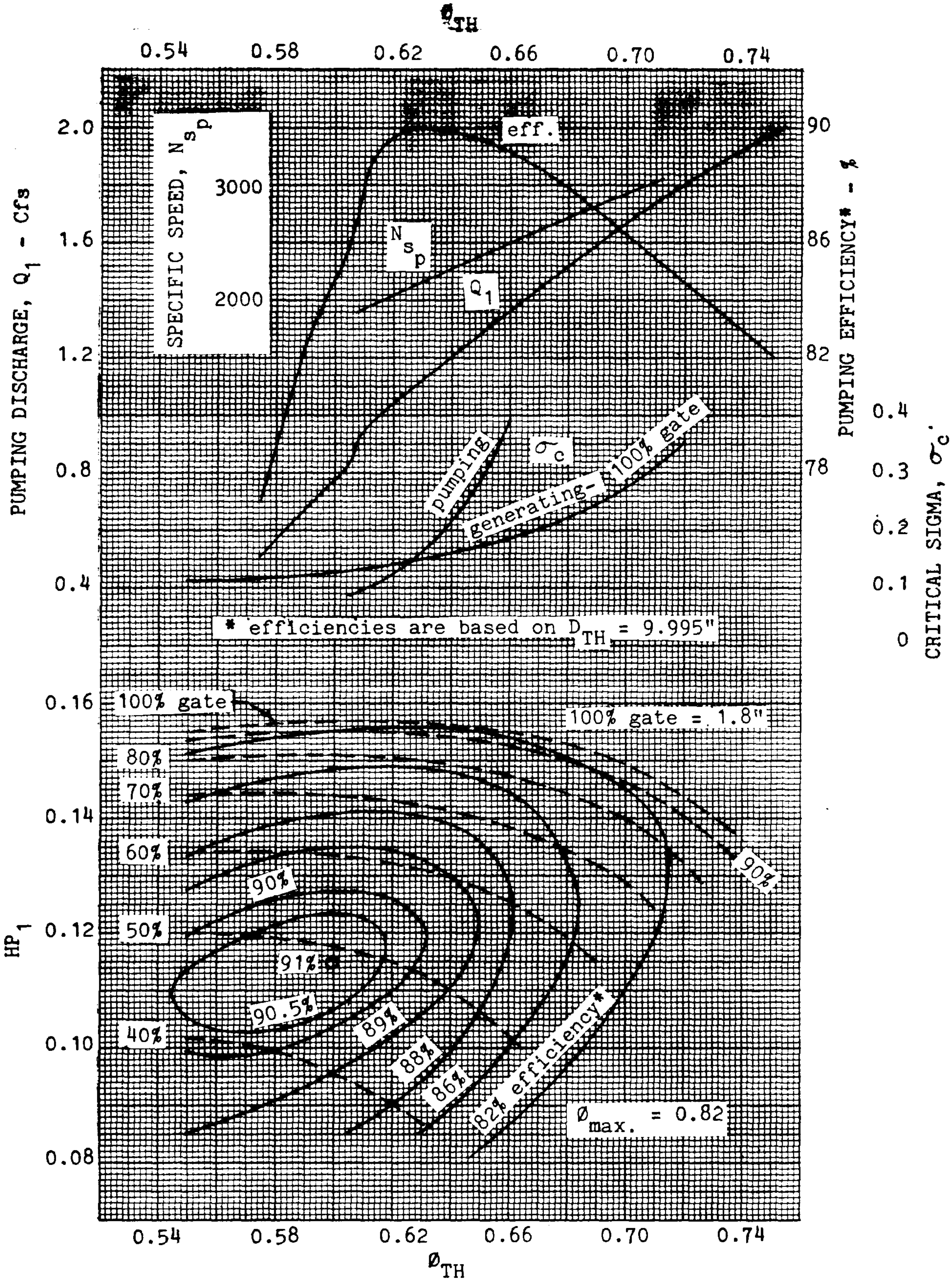

Figure PT1. MODEL TEST DATA - FRANCIS PUMP-TURBINE

D_{TH} = 12" ONE FOOT HEAD N_S = 2,170

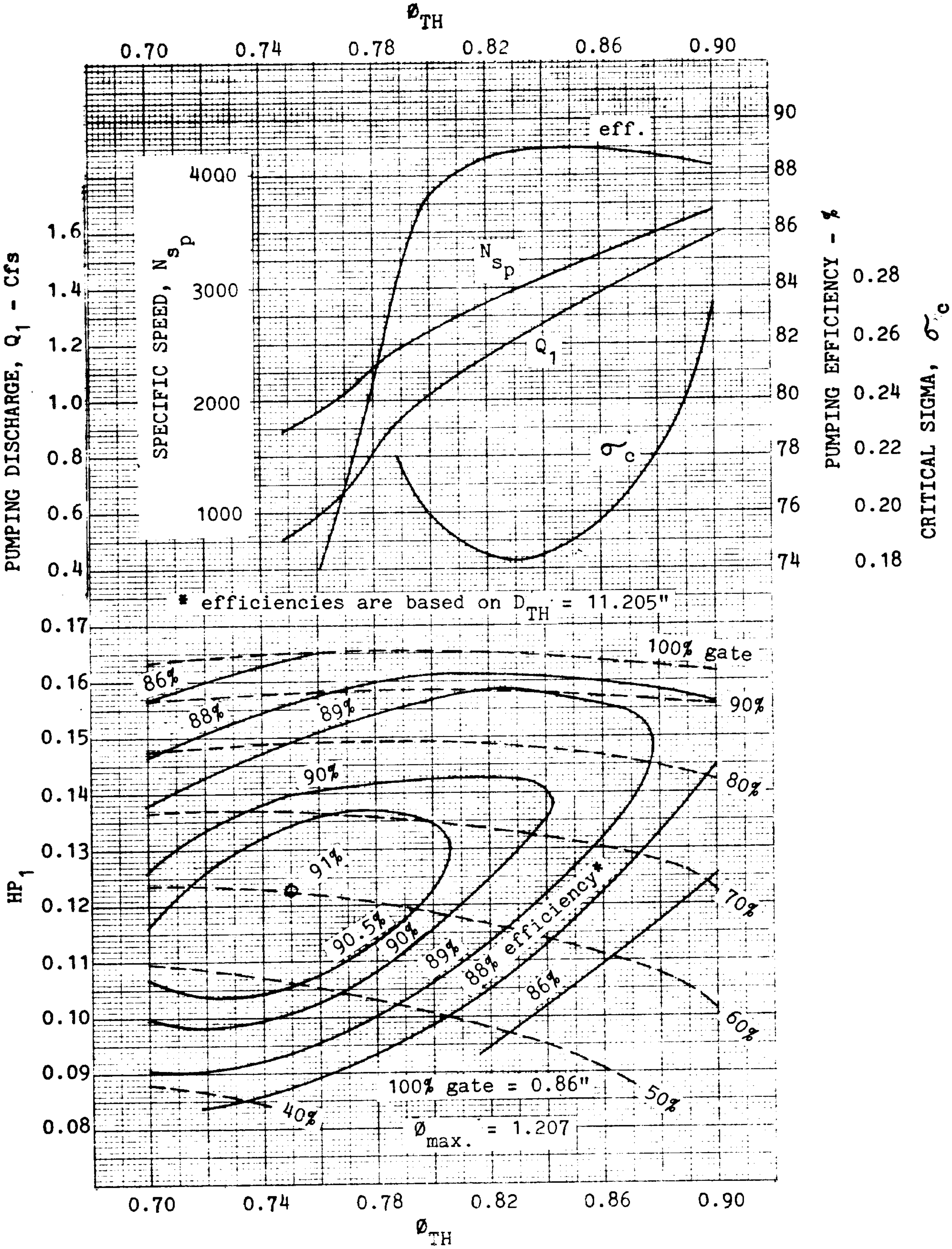

Figure PT2. MODEL TEST DATA - FRANCIS PUMP-TURBINE

D_{TH} = 12" ONE FOOT HEAD N_S = 3,160

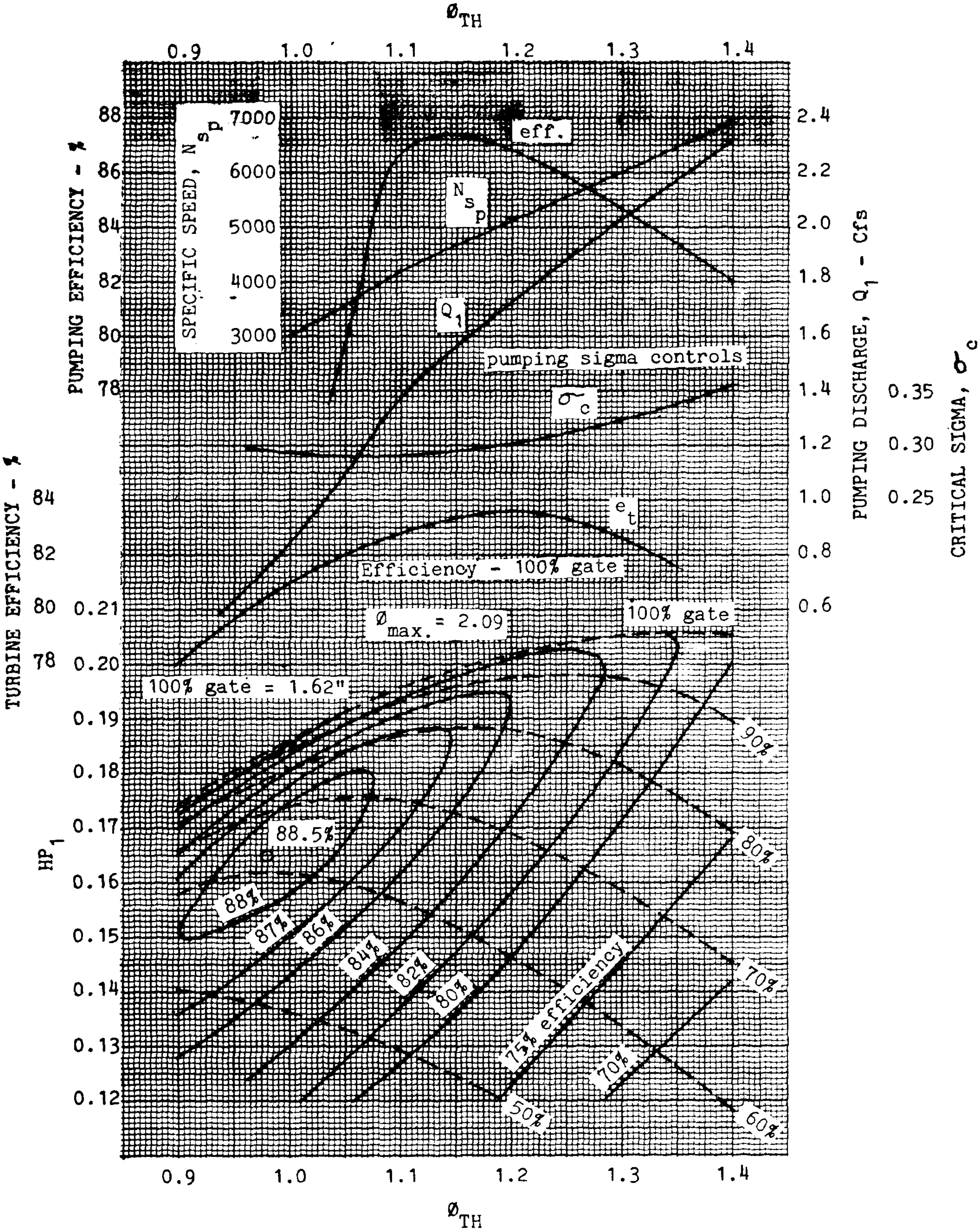

Figure PT3. MODEL TEST DATA - FRANCIS PUMP-TURBINE

D_{TH} = 12" ONE FOOT HEAD N_S = 4,670

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SECTION III

PROPELLER TURBINE MODEL TEST CURVES FOR D_{TH}= 12 INCHES AND H = 1 FOOT

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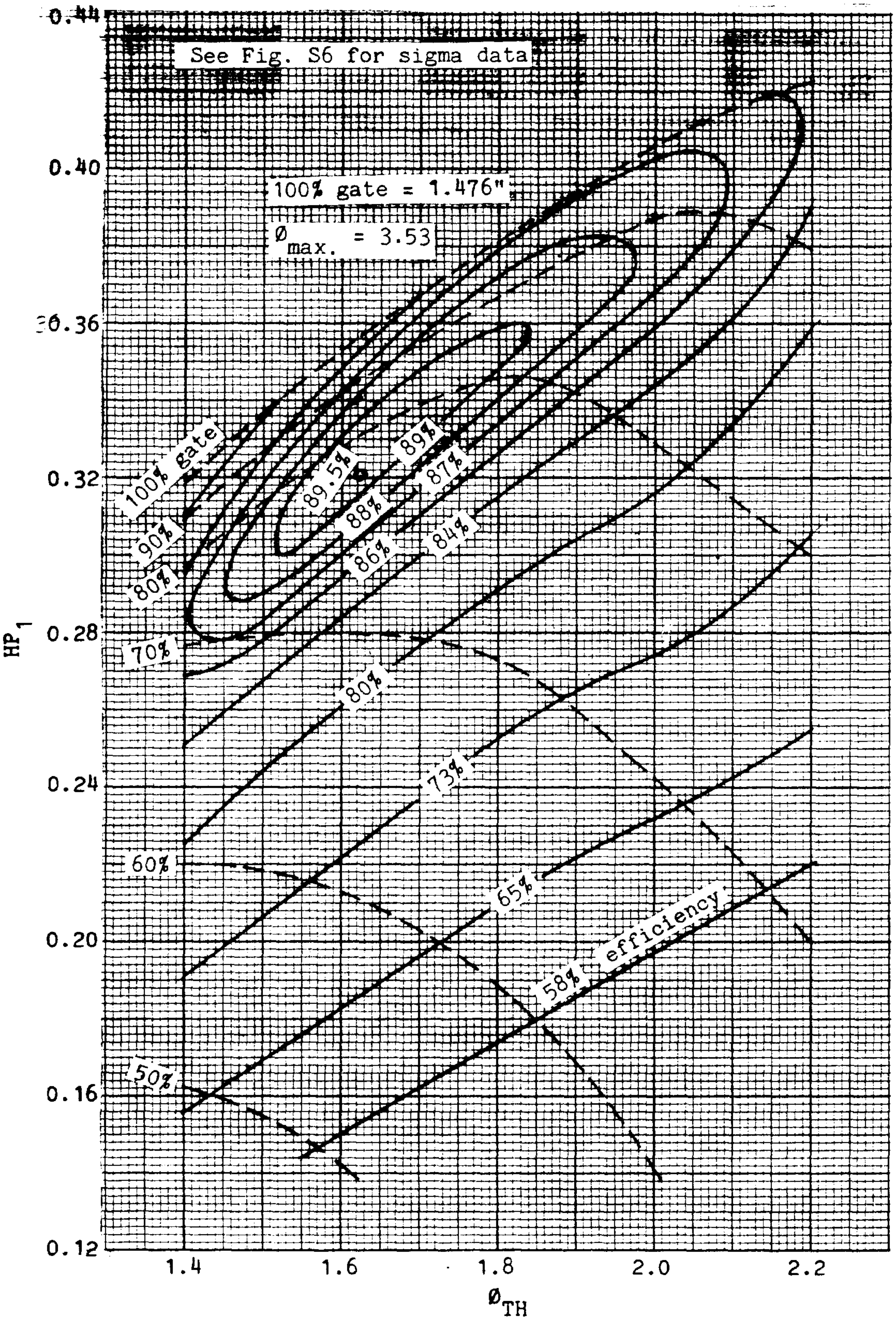

Figure FB1. MODEL TEST DATA - FIXED BLADE PROPELLER TURBINE
FOUR BLADES 24° BLADE ANGLE
D_{TH} = 12" ONE FOOT HEAD

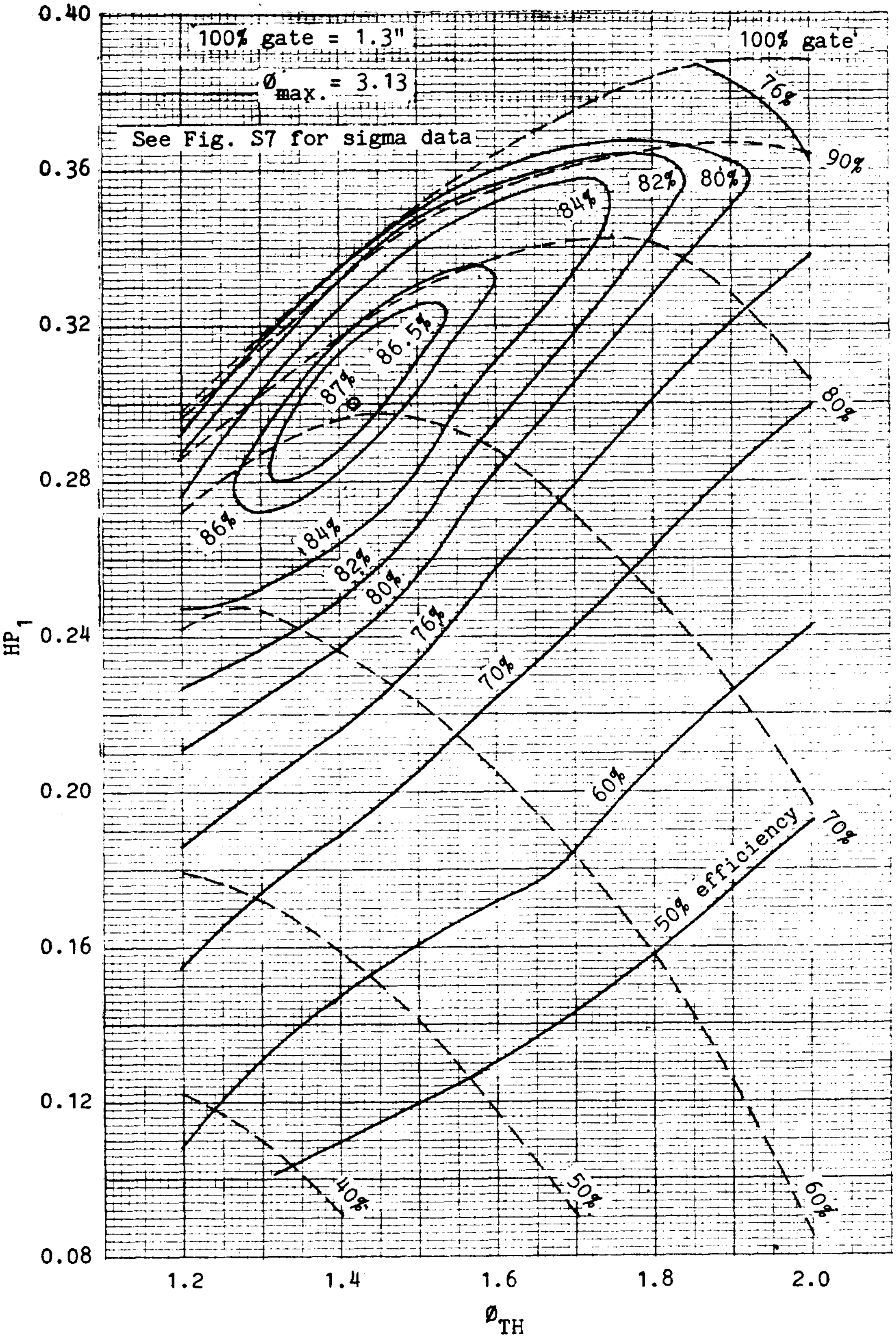

Figure FB2. MODEL TEST DATA - FIXED BLADE PROPELLER TURBINE

FIVE BLADES 31° BLADE ANGLE

D_{TH} = 12 " ONE FOOT HEAD

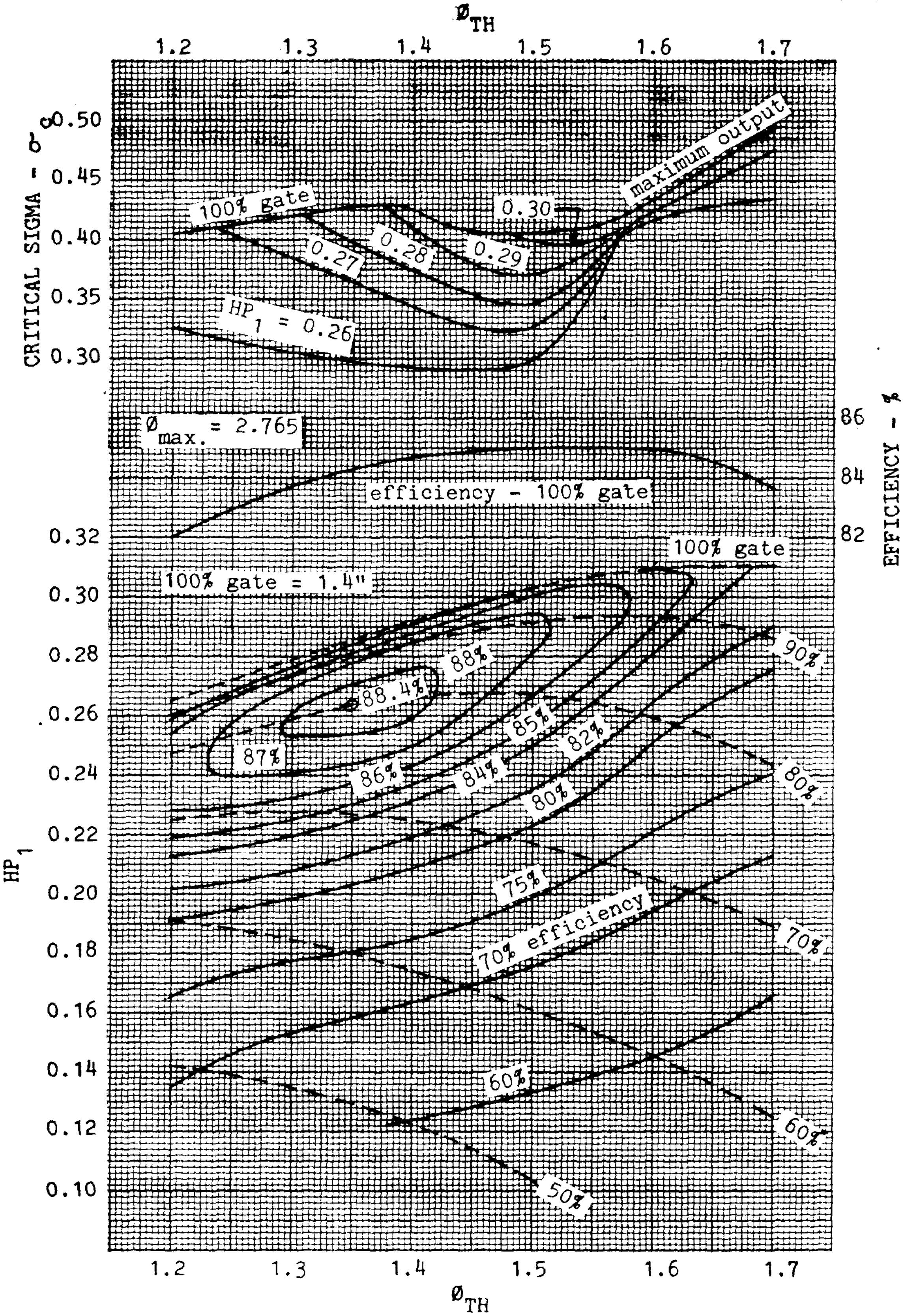

Figure FB3. MODEL TEST DATA - FIXED BLADE PROPELLER TURBINE
SIX BLADES 27° BLADE ANGLE
D_{TH} = 12" ONE FOOT HEAD

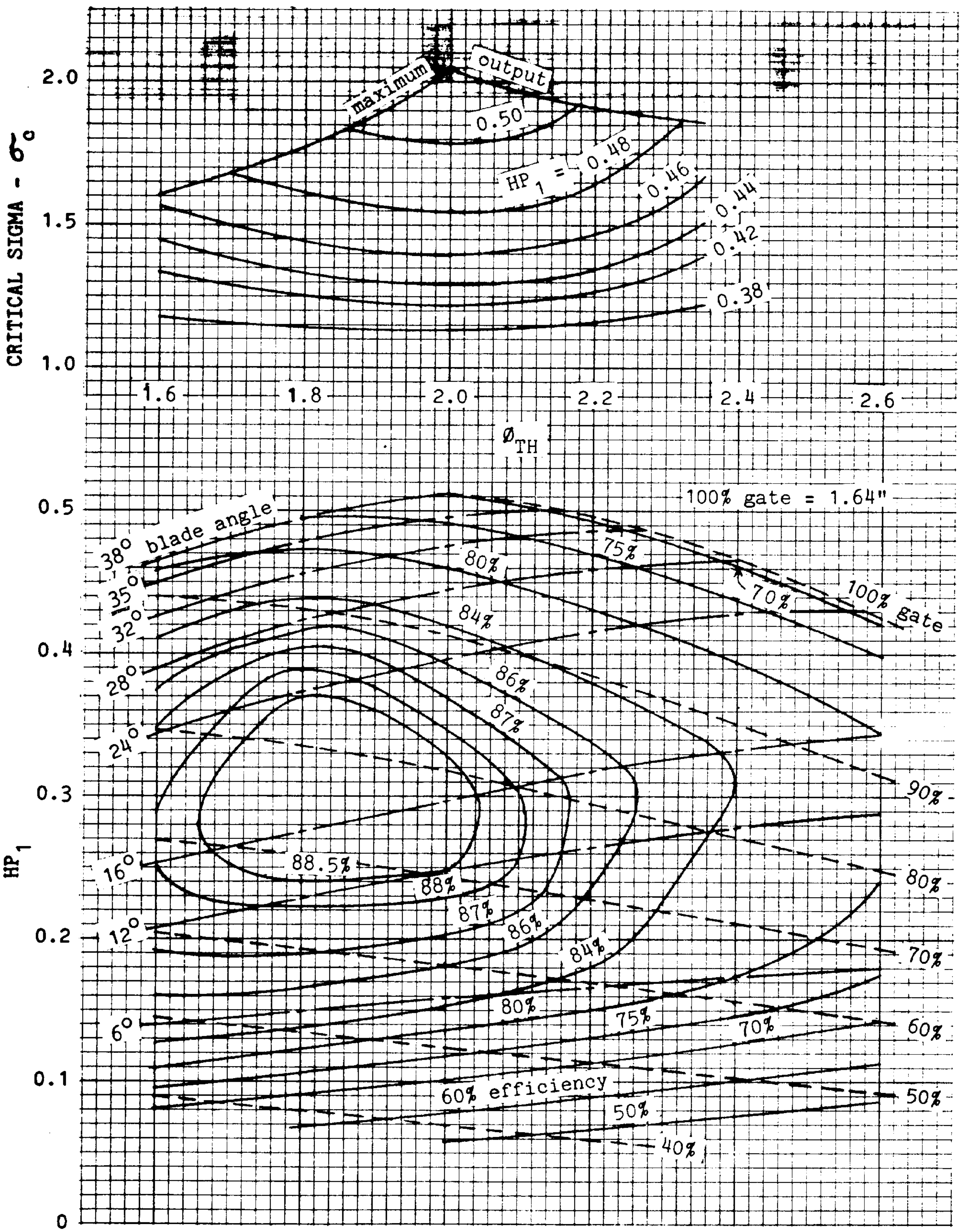

Figure K1. MODEL TEST DATA - FOUR BLADE KAPLAN TURBINE

D_{TH} = 12" ONE FOOT HEAD

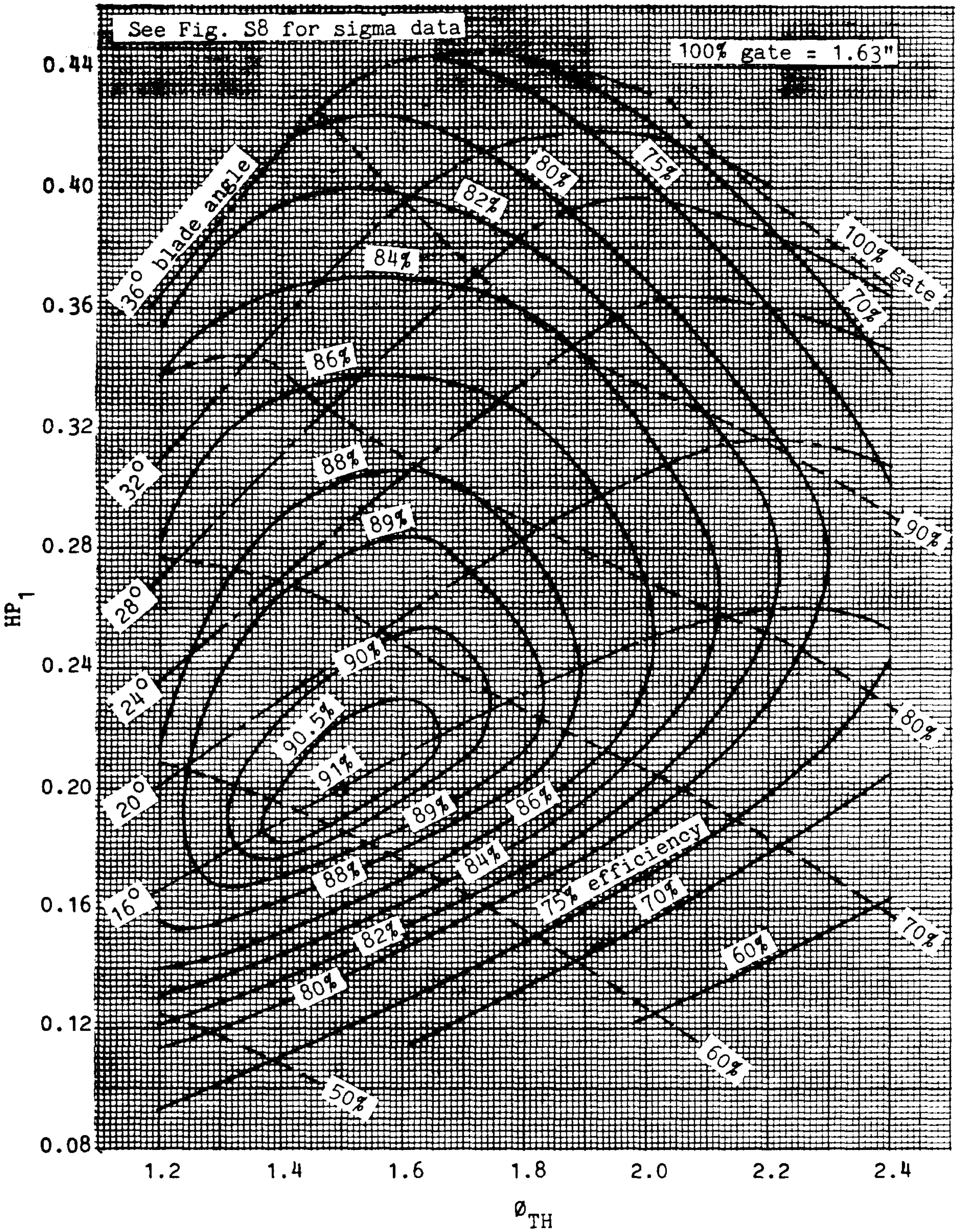

Figure K2. MODEL TEST DATA - FIVE BLADE KAPLAN TURBINE

D_{TH} = 12" ONE FOOT HEAD

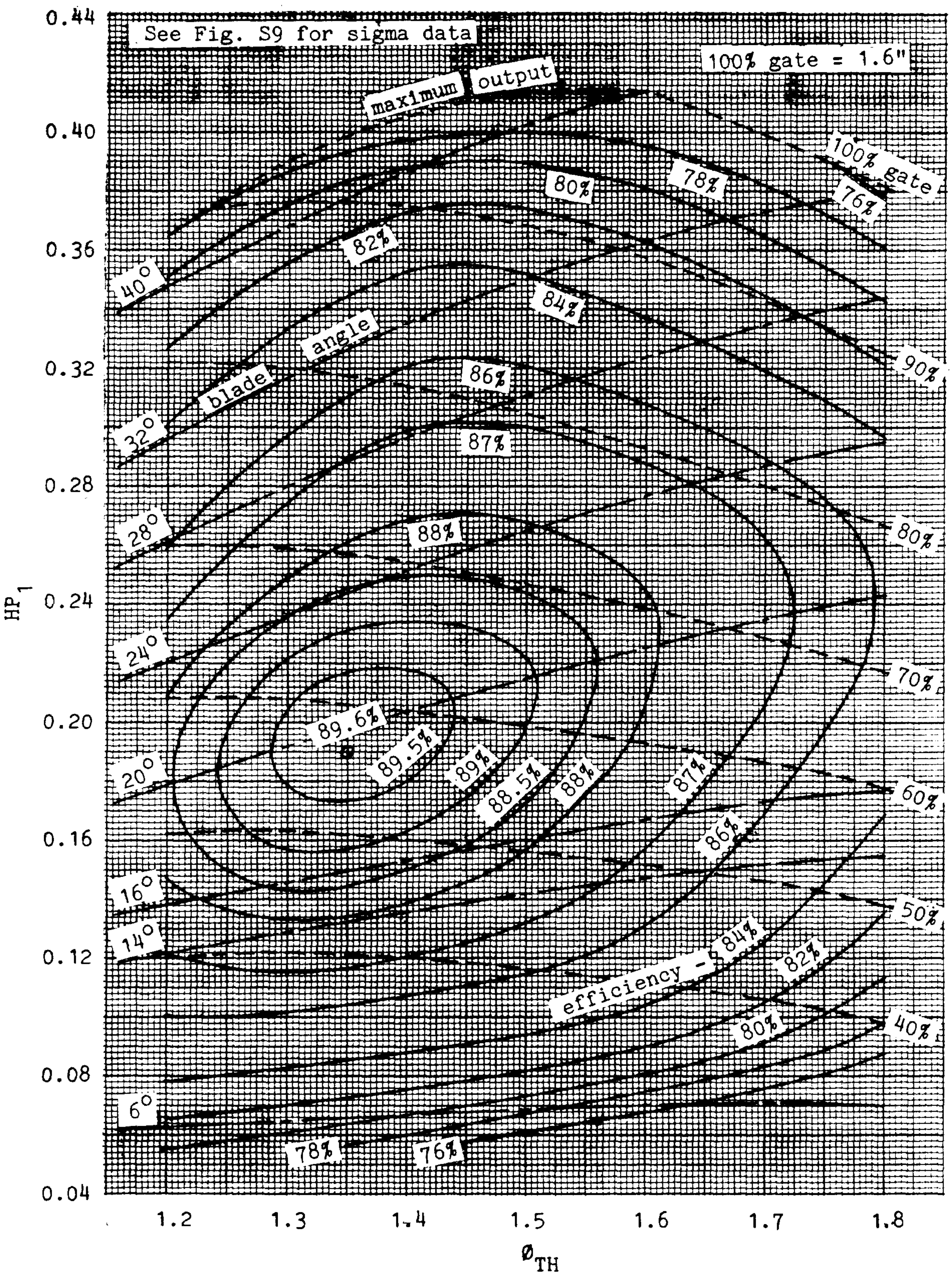

Figure K3. MODEL TEST DATA - SIX BLADE KAPLAN TURBINE

D_{TH} = 12" ONE FOOT HEAD

SECTION IV

CRITICAL RUNNER SIGMAS

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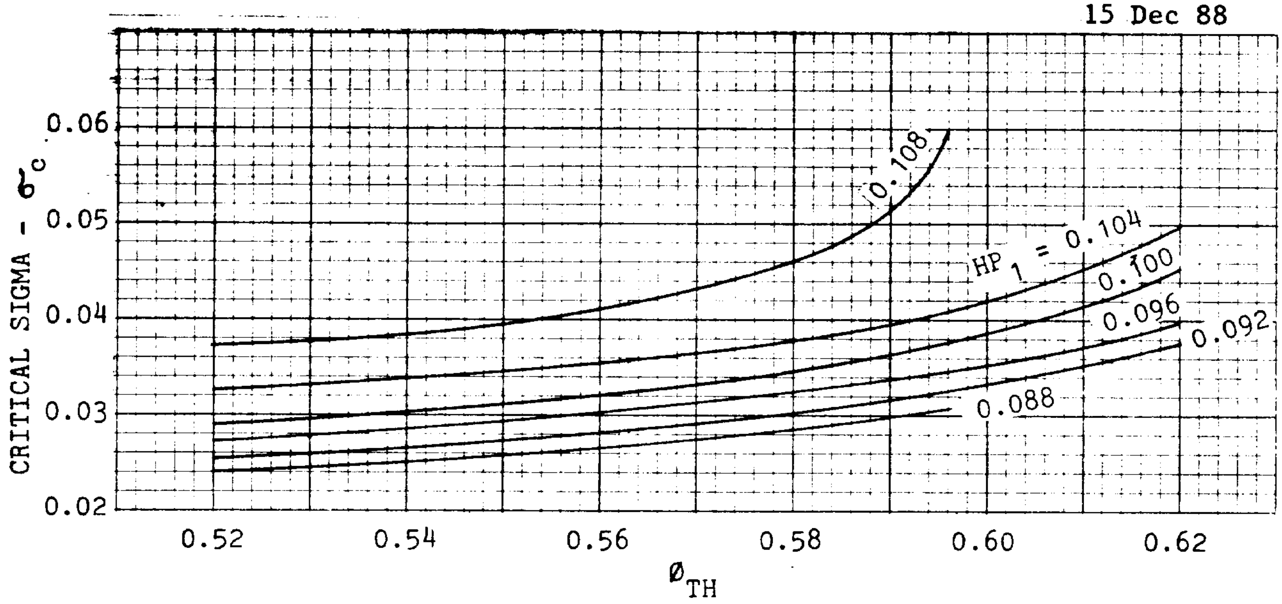


Figure S1

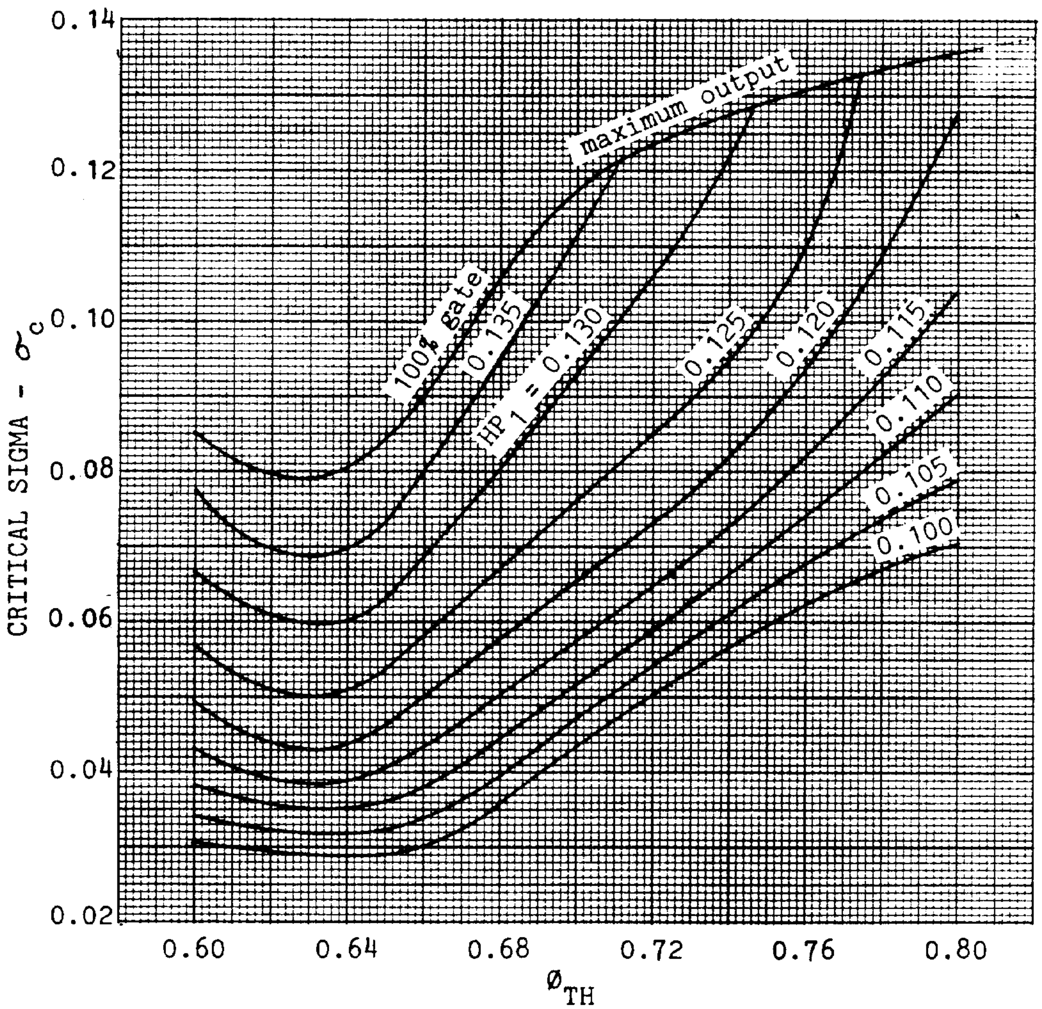


Figure S2

FRANCIS TURBINES

CRITICAL RUNNER SIGMAS

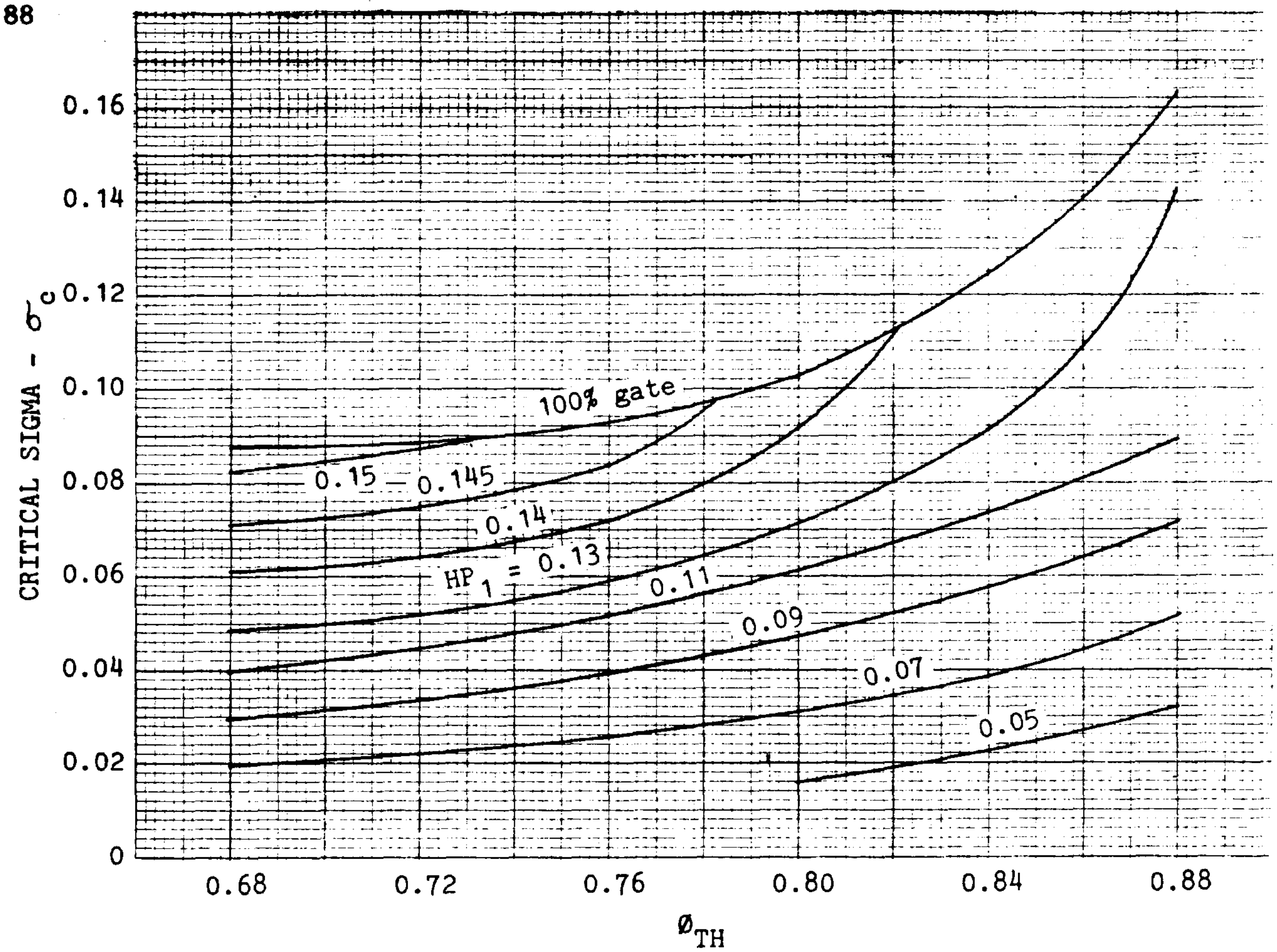


Figure S3

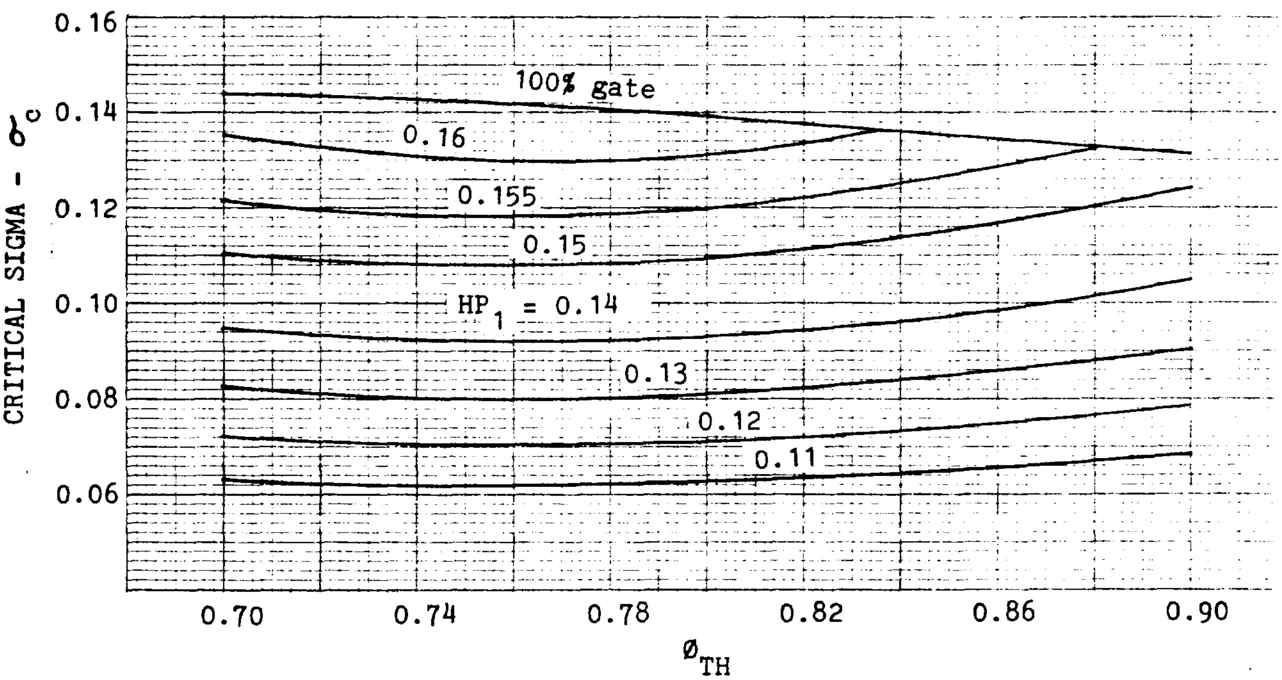
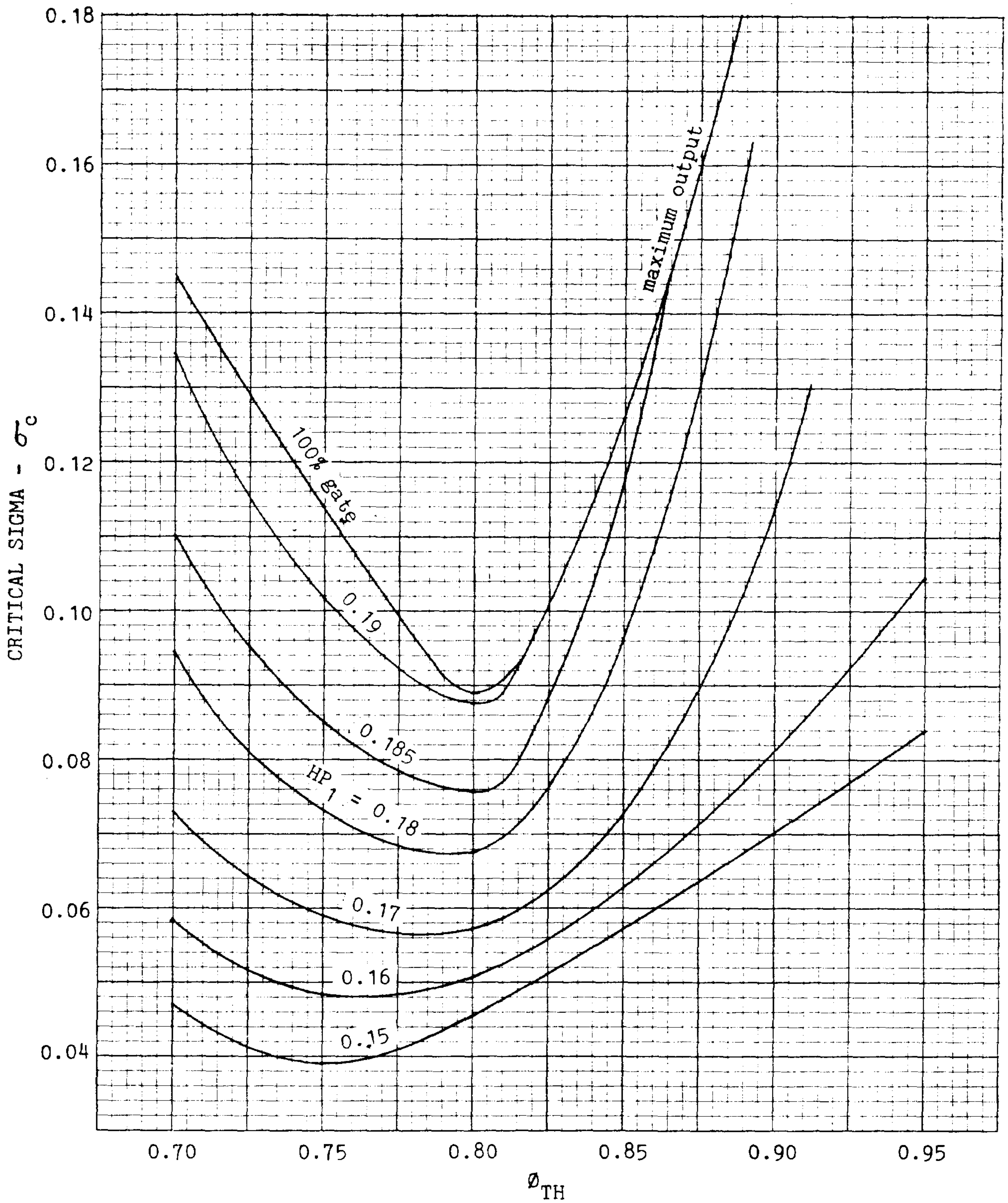


Figure S4

FRANCIS TURBINES

CRITICAL RUNNER SIGMAS



FRANCIS TURBINE

CRITICAL RUNNER SIGMAS

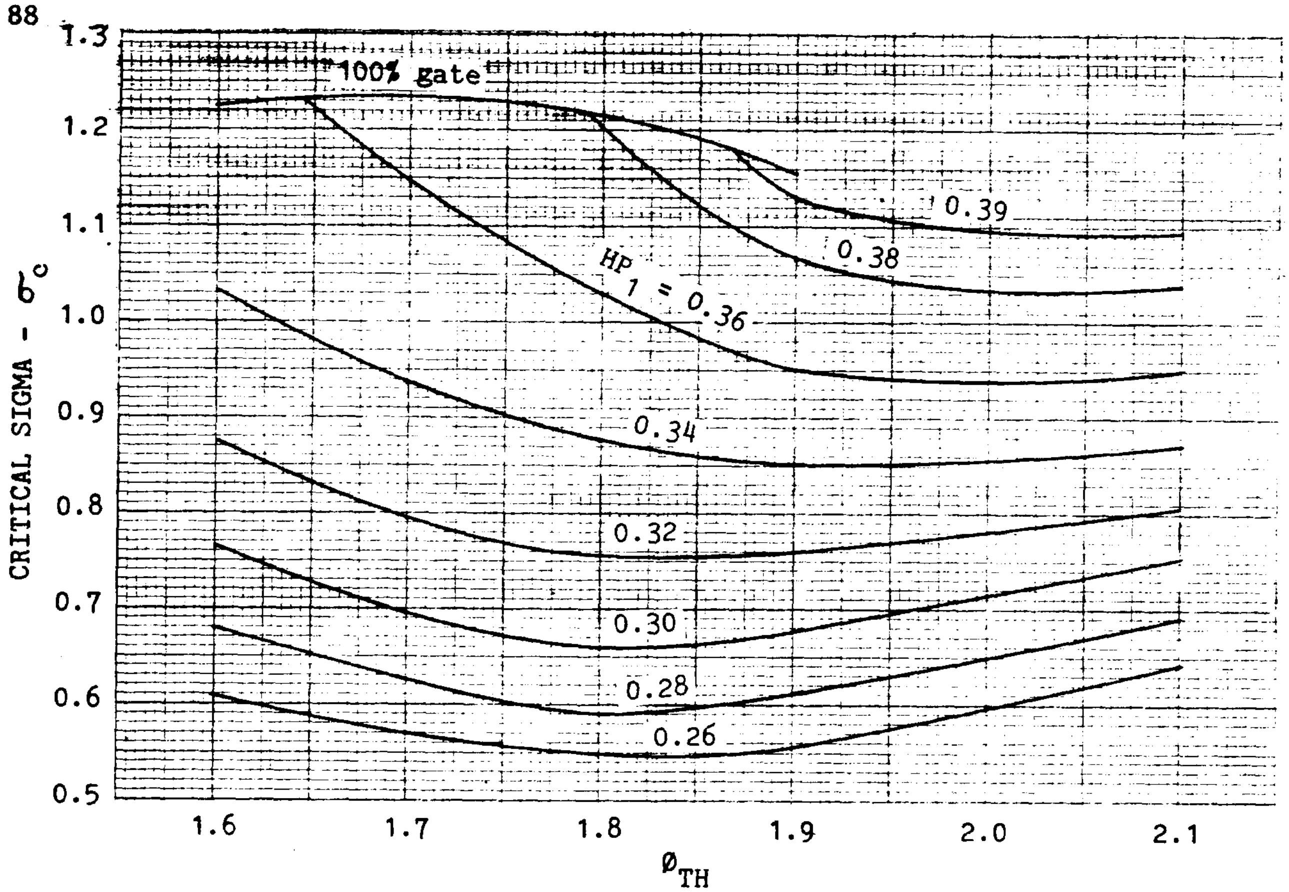


Figure S6

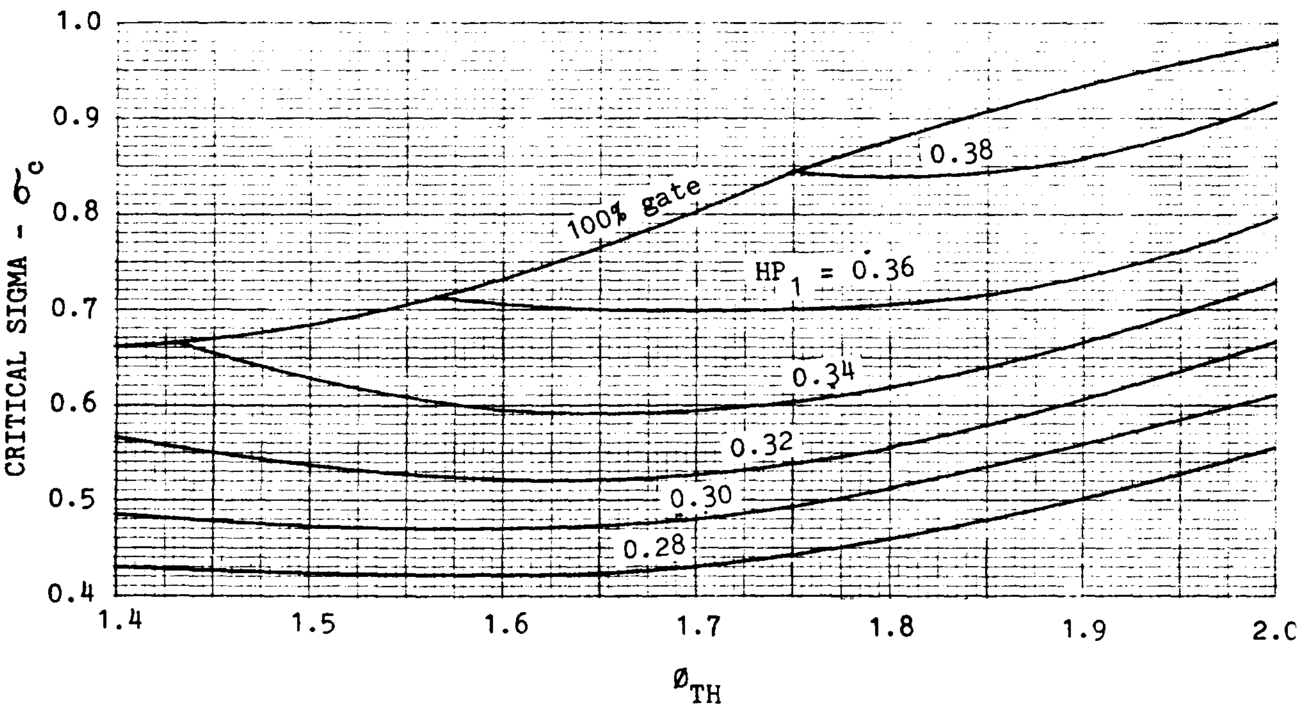
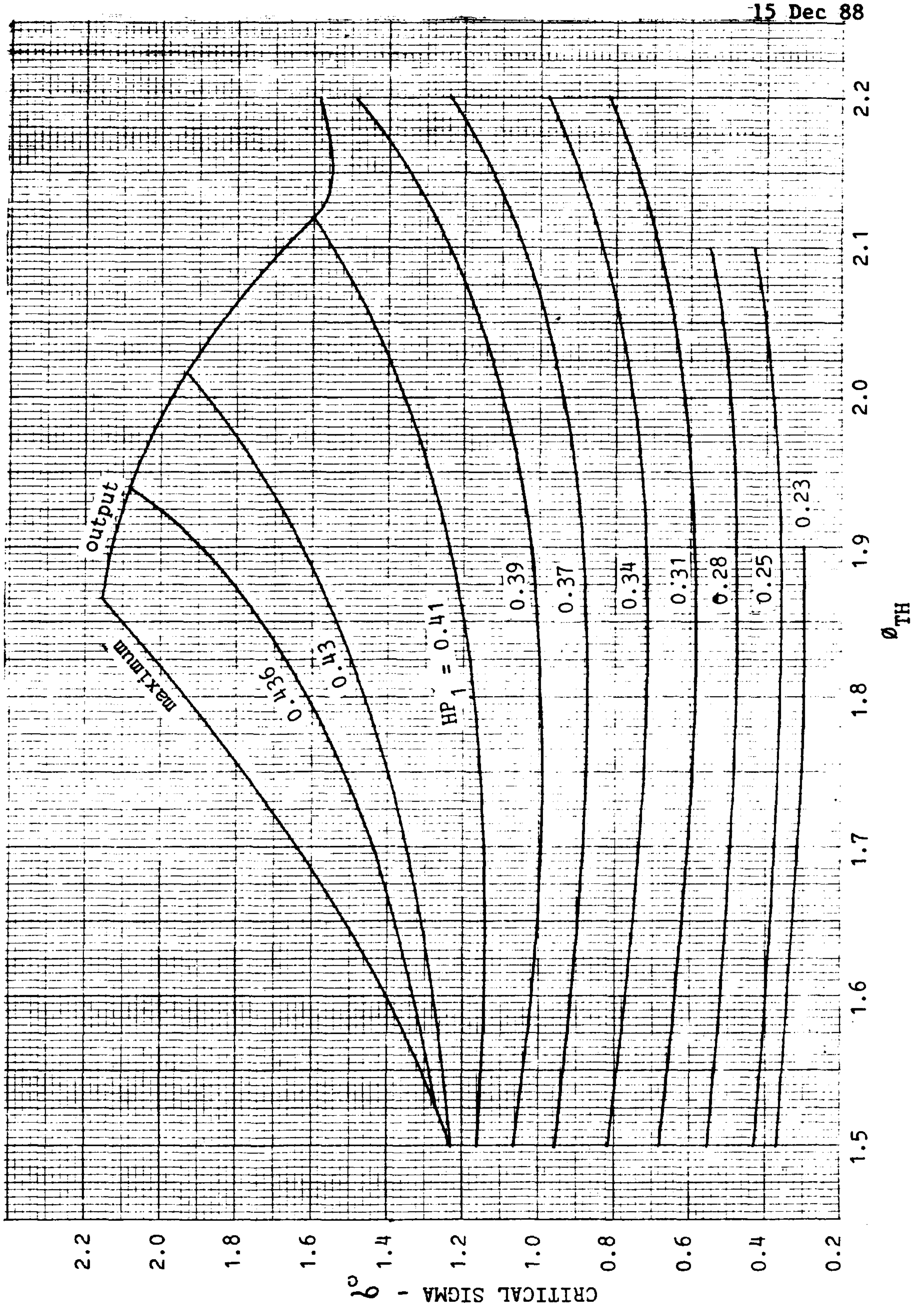


Figure S7

FIXED BLADE PROPELLER TURBINES

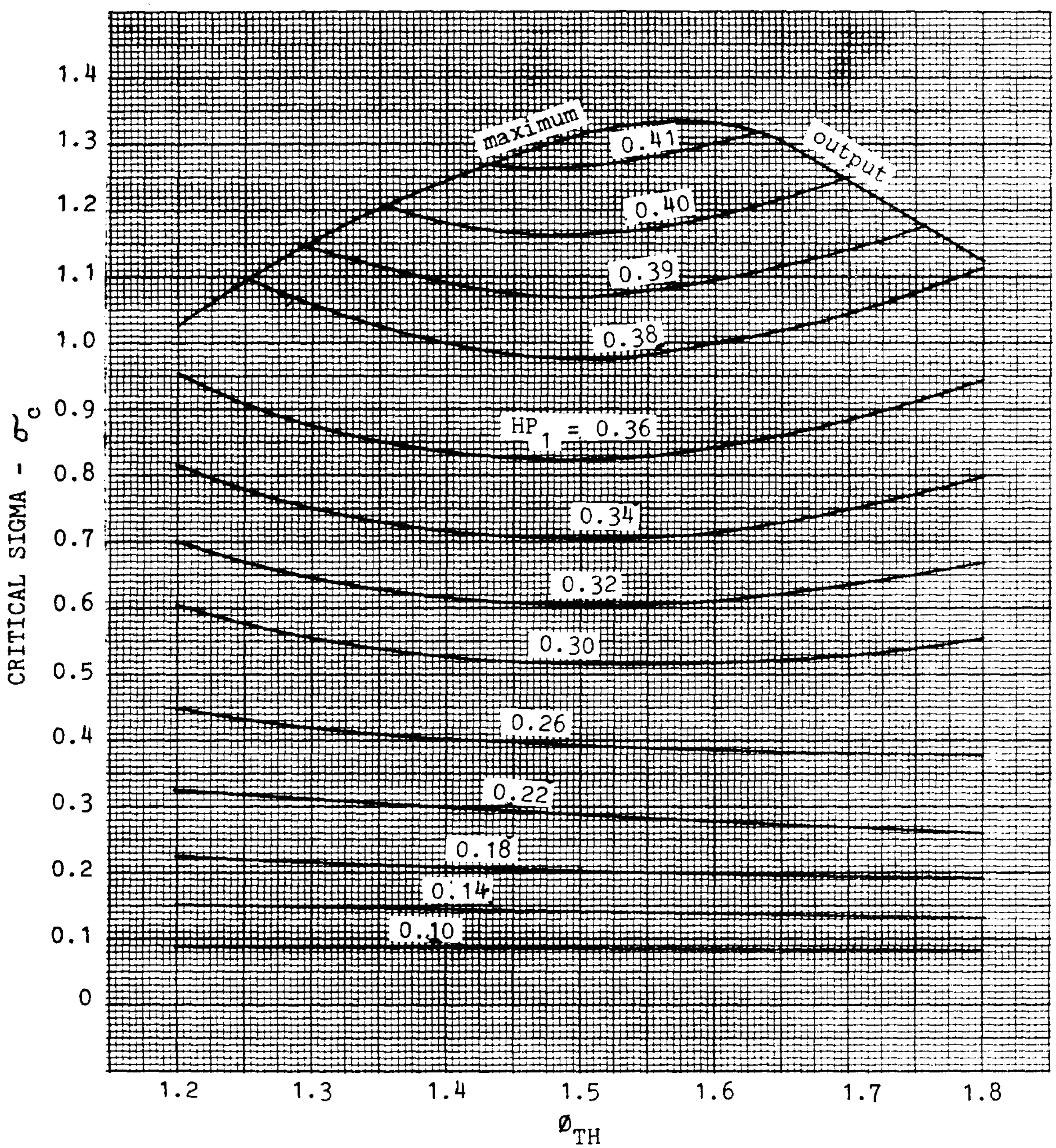
CRITICAL RUNNER SIGMAS



KAPLAN TURBINE

CRITICAL RUNNER SIGMAS

Figure S8



KAPLAN TURBINE

CRITICAL RUNNER SIGMAS

APPENDIX E

SAMPLE CALCULATIONS

SECTION I

FRANCIS TYPE TURBINES AND PUMP-TURBINES

1. DESIGN REQUIREMENTS:

a. Powerplant capacity - 150,000 KW

b. Installation: 2 pump-turbines and one conventional turbine.

c. Pumping requirements: 3,600 cfs each at a dynamic head of 153 feet. The heads vary between 137.5 and 160 feet. The minimum tailwater level for pumping is Elev. 540 ft. m.s.l.

d. Generating requirements: The 3 units must have an aggregate dependable capacity of 150,000 KW. The 3 units must also be capable of producing 170,500 KW at a net head of 137.4 feet. The net heads vary between 132.5 and 151.9 feet. The average (rated) net head is 144.8 feet.

2. SELECTION OF PUMP-TURBINES.

a. Reference Figure 2, Appendix C to note that the recommended specific speed, N_{sp} for the 153 foot rated pumping head has a value of about 4,000.

b. Referring to the model curves in Appendix D, select the design shown on Figure PT3 as the best choice for this specific speed.

c. At maximum efficiency, note the following: E_1 = 87.4 percent, Q_1 = 1.57 cfs and ϕ_{TH} = 1.15.

d. Calculate D_{TH}:

$$3{,}600 = 1.57 \left(\frac{D_{TH}}{12}\right)^2 (153)^{1/2}$$

$$D_{TH} = 163 \text{ inches}$$

e. Calculate speed, N:

$$N = \frac{1838\ (1.15)\ (153)^{1/2}}{163} = 160 \text{ rpm}$$

Round to 163.6 rpm

f. The calculation of the runner throat diameter and associated synchronous speed generally requires an iterative solution. Further iterations are required until the selected value for ϕ_{TH} and associated value of Q_1 produce a value for D_{TH} which substituted in the speed equation, step e, yields a synchronous speed. The following approximation is used to calculate the next trial value of ϕ_{TH}:

$$\phi_{TH} = 1.15 \ \frac{163.6}{160} = 1.176$$

The necessary iterations for this case are as follows:

Step	ϕ_{TH}	Q_I	D_{TH}	N
1	1.150	1.57	163.4	160.0
2	1.176	1.65	159.4	167.8
3	1.160	1.59	162.4	162.4
4	1.165	1.61	161.3	164.2
5	1.162	1.60	161.8	163.2

The accuracy in reading the model test data does not allow a closer determination of D_{TH} or N from step 5. Therefore, the solution indicates D_{TH} = 162 inches for N = 163.6 rpm.

g. At this point the user should make a cursory examination of the pumping efficiencies for other heads with a view to, perhaps, changing the speed to alter the head-efficiency characteristic. In this example, the following relationships are noted:

$$\phi_{TH} = \frac{163.6\ (162)}{1838\ (H)^{1/2}} = \frac{14.42}{H^{1/2}}$$

H	ϕ_{TH}	E_1
160.0	1.140	87.4
153.0	1.166	87.3
137.5	1.230	86.3

This relationship is satisfactory and the balance of the example is completed on the basis of D_{TH} = 162 inches and N = 163.6 rpm.

h. Calculate efficiency step-up.

$$E_2 = 100 - (100 - E_1) \left(\frac{D_m}{D_p}\right)^{0.2}$$

Where, max. E_1 = 88.5 percent (generating); D_m = 12 inches; D_p= 162 inches.

$$E_2 = 100 - (100 - 88.5) \left(\frac{12}{162}\right)^{0.2} = 93.2 \text{ percent}$$

$$\text{step-up} = (2/3)\ (93.2 - 88.5) = 3.1 \text{ percent}$$

i. The expected pumping discharge is calculated to include the effect of the higher prototype expected efficiency as follows:

$$Q_{2_C} = Q_1 \left(\frac{162}{12}\right)^2 (H)^{1/2} \frac{E_2}{E_1}$$

j. The required pumping horsepower is calculated as follows:

$$HP = \frac{Q_{2_C}\ H_w}{550\ \ E_2}$$

k. The required setting of the runner is controlled by the maximum head-minimum tailwater condition. For maximum head, ϕ_{TH} = 1.14 and from Figure PT3, ${}_c$ = 0.295.

$$\sigma_c = \frac{H_b - H_v - H_s - \text{safety}}{H}$$

Refer to Figure 6, Appendix C: For tailwater Elev. 540, H_b = 33.3 feet and a water temperature of 70° F., H_v = 0.8 feet.

$$\text{safety margin} = 0.2\ D_i + 0.4\ H^{1/2}$$

Refer to Table 3, Appendix C, noting that D_1 = 1.154

therefore, $D_i = 1.154 \frac{162}{12} = 15.6$ feet

subs: safety margin = $0.2\ (15.6) + 0.4\ (160)^{1/2} = 8.2$ feet

The required submergence is calculated as follows:

$$0.295 = \frac{33.3 - 0.8 - H_s - 8.2}{160}$$

$H_s = -22.9$ feet

The distance, a, between the bottom of the runner and the distributor centerline is calculated using the ratio, d, from Table 3, Appendix C, as follows:

d = 0.385

a = 0.385 (162/12) = 5.2 feet

The elevation of the distributor centerline is calculated as follows:

Elev. = tailwater Elev. + H_s + a

Elev. = 540 + (-22.9) + 5.2 = 522.3 ft. m.s.l.

1. The expected pumping performance is as follows:

Head	137.5	145.0	153.0	160.0
Q_{TH}	1.230	1.198	1.166	1.140
Q_1	1.82	1.72	1.62	1.53
E_1	86.3	86.8	87.3	87.4
Q_{2_C}	4,029	3,909	3,777	3,652
E_2	89.4	89.9	90.4	90.5
HP	70,200	71,420	72,410	73,140

3. GENERATING CYCLE.

a. The prototype expected performance is calculated as follows:

$$\phi_{TH} = \frac{162\ (163.6)}{1838\ (H)^{1/2}} = \frac{14.42}{(H)^{1/2}}$$

$$HP_2 = HP_1 \left(\frac{162}{12}\right)^2 (H)^{3/2} = 182.25\ (HP_1)\ (H)^{3/2}$$

$$Q_2 = \frac{550\ HP_2}{62.3\ (H)\ E_2} = 8.828\ \frac{HP_2}{H\ E_2}$$

$E_2 = E_1 + 3.1$ percent

Head	ϕ_{TH}	Percent gate	HP_1	E_1	HP_2	Q_2	E_2
132.5	1.253	100	0.204	83.3	56,710	4,375	86.4
		90	0.198	84.6	55,040	4,180	87.7
		80	0.185	83.9	51,420	3,940	87.0
		70	0.164	80.9	45,590	3,615	84.0
		60	0.140	76.0	38,920	3,280	79.1
137.4	1.230	100	0.203	83.5	59,590	4,420	86.6
		90	0.198	85.0	58,120	4,240	88.1
		80	0.187	84.7	54,890	4,015	87.8
		70	0.166	82.1	48,730	3,675	85.2
		60	0.143	78.0	41,970	3,325	81.1
145	1.198	100	0.202	83.6	64,280	4,515	86.7
		90	0.197	85.2	62,690	4,320	88.3

Head	ϕ_{TH}	Percent gate	HP_1	E_1	HP_2	Q_2	E_2
		80	0.188	85.9	59,820	4,090	89.0
		70	0.169	83.8	53,780	3,765	86.9
		60	0.147	80.0	46,780	3,425	83.1
		50	0.119	74.0	37,870	2,990	77.1
151.9	1.170	100	0.200	83.5	68,240	4,580	86.6
		90	0.197	85.2	67,220	4,425	88.3
		80	0.189	86.5	64,490	4,180	89.6
		70	0.171	85.0	58,340	3,850	88.1
		60	0.150	81.8	51,180	3,505	84.9
		50	0.123	76.8	41,970	3,055	79.9

b. The maximum runaway speed is calculated as follows:

Refer Figure PT3 to note that $\phi_{max} = 2.09$

$$N_{max} = \frac{1838\ (2.09)\ (151.9)^{1/2}}{162} = 292 \text{ rpm}$$

c. The guaranteed capacities at the 132.5 foot and 137.4 foot net head conditions are calculated at 98 percent of the 100 percent gate capacities indicated in above tabulation. The guaranteed capacities for the conventional unit at these two heads are as follows:

KW output = $0.98\ (0.746)\ E_g\ HP_2$

KW output = $0.98\ (0.746)\ (0.97)\ HP_2 = 0.709\ HP_2$

Head - feet	132.5	137.4
Plant output - KW	150,000	170,500
Pump-turbines - KW	80,400	84,500

Conventional - KW	69,600	86,000
Conventional - HP	96,180	118,850
Expected - HP	98,140	121,280

4. CONVENTIONAL FRANCIS TURBINE.

a. The relationship for N_s vs. Head shown on Figure 1, Appendix C insures designs with moderate speeds and relatively shallow submergences. In a mixed installation with pump-turbines and conventional turbines, the inherent deeper submergences required of the former generally dictates a variation of this conservative approach. This is necessary to provide a more balanced equipment layout and avoid exaggerated levels for the generator-motors and generators. For this reason the "K" value used in Figure 1 is increased to, say, a value of 800. The corresponding N_s for H = 144.8 feet is 66.5.

b. Referring to the model curves in Appendix D, it may be noted that the designs shown on Figures F6 and F7 are within the range of this specific speed. A comparison of these designs indicates that the former has higher unit power with attendant higher critical sigmas, whereas the latter has higher overall efficiencies with lower critical sigmas and reduced unit power. The former design, Figure F6, is selected for the following reasons. The higher unit power will result in a smaller runner throat diameter with consequent smaller physical dimensions of the turbine to more nearly approach the physical dimensions of the pump-turbines. The higher critical sigmas require deeper submergences, however, this is not inappropriate in view of the deep submergence of the pump-turbines.

c. The method for sizing this unit differs from the conventional approach for Francis turbines. In this instance, the output required at the 137.4 feet critical net head dictates the size. This output is associated with the full gate capacity at a value of ϕ_{TH} slightly higher than the best ϕ_{TH} to be associated with the average head of 144.8 feet. For the latter condition a first value of ϕ_{TH} = 0.86 is chosen. The corresponding value for the 137.4 foot head condition is calculated as follows:

$$\phi_{TH} = 0.86 \left(\frac{144.8}{137.4}\right)^{1/2} = 0.883$$

From Figure F6 for ϕ_{TH} = 0.883, the 100 percent gate HP = 0.29. This is associated with the required expected output of 121,280 HP to calculate D_{TH} as follows:

$$121,280 = 0.29 \left(\frac{D_{TH}}{12}\right)^2 (137.4)^{3/2}$$

$$D_{TH} = 193.4 \text{ inches}$$

d. Calculate the speed, as follows:

$$N = \frac{1838\ (0.833)\ (137.4)^{1/2}}{193.4} = 98.4$$

Round to nearest synchronous speed = 100 rpm

This speed and the D_{TH} calculated above are first values of an iterative solution similar to that described in 2.f. of this example. The necessary iterative steps are as follows:

Step	ϕ_{TH}	HP_1	D_{TH}	N
1	0.883	0.29	193.4	98.4
2	0.897	0.29	193.4	100

Round D_{TH} to 193.5 inches

e. The expected prototype output is calculated as follows:

$$HP_2 = HP_1 \left(\frac{193.5}{12}\right)^2 (H)^{3/2} = 260.02\ (HP_1)\ (H)^{3/2}$$

f. The efficiency step-up is calculated, using the procedure established in 2.h. of this example, as follows:

$$E_2 = 100 - (100 - 90) \left(\frac{12}{193.5}\right)^{0.2}$$

$$E_2 = 94.3 \text{ percent}$$

$$\text{step-up} = (2/3)\ (94.3 - 90) = 2.9 \text{ percent}$$

g. The expected discharge is calculated as follows:

$$Q_2 = \frac{550 \quad HP_2}{62.3 \ (H) \ E_2} = 8.828 \ \frac{HP_2}{(H) \ E_2}$$

h. The guaranteed capacity required at the 137.4 feet critical head is 118,850 HP. The generator output is 86,000 KW. The generator nameplate rating is 86,000 KW at 0.95 p.f. or 90,526 KVA. The turbine is designed to mechanically withstand operation at the generator nameplate rating at 1.0 p.f. or 125,100 HP. The turbine setting is predicated on the availability of 118,850 HP at the critical and higher heads. Although the critical head conditions will generally dictate the setting, it is recommended that other conditions be checked to assure that the critical sigma characteristics of proposed design or unusual tailwater conditions do not alter this normal circumstance.

i. The procedure described in 2.k. of this example is used to establish the turbine setting. The dimensionless ratios for calculating the dimensions D_i and a are obtained from Table 1, Appendix C. The results of pertinent calculations are tabulated as follows:

Net head, feet	137.4	144.8	151.9
T.W. elev.,ft.m.s.l.	550.8	548.5	541.6
HP	118,850	118,850	118,850
ϕ_{TH}	0.898	0.875	0.854
HP_1	0.284	0.262	0.244
σ_c	0.245	0.1830	0.160
H_b, feet	33.3	33.3	33.3
H_v, feet	0.8	0.8	0.8
D_i, feet	14.2	14.2	14.2
Safety margin, feet	7.5	7.7	7.8
H_s, feet	-8.7	-1.7	+0.4
a, feet	6.9	6.9	6.9

Dist. elev, ft. m.s.l.	549.0	553.7	548.9

It is to be noted that the conditions at the critical and maximum heads dictate about the same setting.

j. The maximum runaway speed is calculated as follows:

From Figure F6, $\phi_{max} = 1.671$

$$N_{max} = \frac{1838\ (1.671)\ (151.9)^{1/2}}{193.5} = 195.6 \text{ rpm}$$

k. The expected prototype performance is tabulated below:

Head	ϕ_{TH}	HP_1	E_1	HP_2	Q_2	E_2
132.5	0.915	0.123	75	48,780	77.9	4,170
		0.148	80	58,690	82.9	4,715
		0.185	84	73,370	86.9	5,625
		0.217	87	86,060	89.9	6,375
		0.241	89	95,570	91.9	6,930
		0.258	89	102,320	91.9	7,415
		0.273	87	108,260	89.9	8,025
		0.285	84	113,020	86.9	8,665
		0.290	82.5	115,010	85.4	8,970
137.4	0.898	0.122	75	51,090	77.9	4,215
		0.148	80	61,980	82.9	4,805
		0.211	87	88,360	89.9	6,315

Head	ϕ_{TH}	HP_1	E_1	HP_2	Q_2	E_2
		0.235	89	98,410	91.9	6,880
		0.259	89	108,460	91.9	7,585
		0.274	87	114,740	89.9	8,200
		0.285	84	119,350	86.9	8,825
144.8	0.875	0.120	75	54,370	77.9	4,255
		0.146	80	66,150	82.9	4,865
		0.174	84	78,830	86.9	5,530
		0.203	87	91,970	89.9	6,235
		0.227	89	102,840	91.9	6,825
		0.260	89	117,790	91.9	7,815
		0.273	87	123,680	89.9	8,390
151.9	0.854	0.119	75	57,930	77.9	4,320
		0.144	80	70,100	82.9	4,915
		0.170	84	82,750	86.9	5,535
		0.197	87	95,900	89.9	6,200
		0.219	89	106,610	91.9	6,740
		0.239	90	116,340	92.9	7,280
		0.246	90	119,750	92.9	7,490

5. PROTOTYPE DIMENSIONS. The prototype dimensions of the pump-turbines can be calculated from the dimensionless ratios shown in Table 3, Appendix C. Similar dimensions for the Francis turbine can be calculated from the ratios shown in Table 1, Appendix C.

SECTION II

FIXED BLADE PROPELLER TURBINE

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1. DESIGN REQUIREMENTS.

a. Powerplant capacity: 65,000 KW with 2 units.

b. Generator requirements:

(1) Nameplate rating: 36,111 KVA, 0.9 pf., 32,500 KW, 13.8 KV and 60 Hz.

(2) Must be designed for continuous operation at rated KVA, voltage, p.f. and frequency.

c. Turbine requirements:

(1) Net heads: 78 foot rated, 60 foot minimum and 100 foot maximum.

(2) Require 30,000 HP guaranteed output at 60 foot net head.

(3) Turbine output is limited to 49,900 HP (rated KVA at 1.0 p.f.).

2. TURBINE SELECTION.

a. To utilize the capability of a generator mated with a fixed blade propeller turbine, the turbine at or near best efficiency at rated head should have an output of 95 percent of the horsepower equivalent of the generator rating:

$$HP = \frac{0.95\ (32,500)}{0.746\ (0.97)} = 42,600$$

b. Reference Figure 3, Appendix C to note that the rated head condition and the wide head range for this unit dictates a 6 blade runner. The recommended specific speed at the 78 foot rated head is calculated as follows:

$$N_S = \frac{1,000}{(78)^{1/2}} = 113.2$$

c. The speed is calculated as follows:

$$N = \frac{113.2\ (78)^{5/4}}{(42{,}600)^{1/2}} = 127.1$$

Round to nearest synchronous speed = 128.6 rpm

Corrected N_s = 114.5

d. For preliminary studies requiring only an approximate speed and runner throat diameter, the following empirical formula for ϕ_{TH} may be used to calculate the diameter:

$$\phi_{TH} = 0.089\ (114.5)^{0.58} = 1.391$$

$$D_{TH} = \frac{1838\ (1.391)\ (78)^{1/2}}{128.6} = 175.6 \text{ inches}$$

e. The appropriate model test curves for these conditions are shown on Figure FB3. A curve of best efficiency is constructed from the following data taken from the efficiency contours:

ϕ_{TH}	1.230	1.290	1.350	1.420	1.515
HP_1	0.244	0.256	0.264	0.272	0.290

The location of the design point along this curve is determined by iteration. This is accomplished by substituting associated values of HP_1 and ϕTH in the following formula for specific speed:

$$N_s = 153.17\ (\phi_{TH})\ (HP_1)^{1/2} = 114.5$$

The approximate value ϕ_{TH} = 1.391 from (d) above is used in the first step of the iterative process as follows:

ϕ_{TH}	1.391	1.440	1.430
HP_1	0.2685	0.2745	0.2730
N_s	110.4	115.6	114.5

The design point is located at $\phi TH = 1.430$ and $HP_1 = 0.2730$. The runner throat diameter for this preliminary selection is calculated as follows:

$$42,600 = 0.273 \left(\frac{D_{TH}}{12}\right)^2 (78)^{3/2}$$

$$D_{TH} = 180.6 \text{ inches}$$

f. It may be noted from inspection of Figure FB3 that the design point calculated above is located to the right of bet The model efficiency at this point is $E_1 = 87.9$ percent, which is less than the 88.4 percent peak efficiency. The peak efficiency at the rated conditions can be improved by selecting the next lower synchronous speed, 120 rpm, and repeating the iterative solution for the new design point. The calculations and tabulation of the iterative steps are as follows:

$$N_s = \frac{(42,000)^{1/2}}{(78)^{5/4}} (120) = 106.8$$

Use first trial $\phi_{TH} = 1.43\left(\frac{120}{128.6}\right) = 1.334$

ϕ_{TH}	1.334	1.362	1.355
HP_1	0.2620	0.2650	0.2645
N_s	104.6	107.4	106.7

Calculate the runner throat diameter:

$$42,600 = 0.2645\left(\frac{D_{TH}}{12}\right)^2 (78)^{3/2}$$

$$D_{TH} = 183.5 \text{ inches}$$

g. The second selection matches the peak efficiency of this design to the rated conditions. This is accomplished by selecting a larger, lower speed unit. At this point in the selection process, the user must evaluate the increased capital costs of the larger unit against the benefits of the higher efficiency. The costs should include the effects on the powerhouse structure, excavation taking into account any change in the turbine setting, generator cost, . . . etc. The latter selection is arbitrarily used in the remainder of this example.

h. The model test curves must be checked to assure that the 30,000 HP guaranteed output at 60 foot minimum net head can be developed with the proposed design. The necessary calculations in this determination are as follows:

$$\phi_{TH} = \frac{120\ (183.5)}{1838\ (H)^{1/2}} = \frac{11.98}{(H)^{1/2}} = 1.547$$

$$30{,}000 = HP_1 \left(\frac{183.5}{12}\right)^2 (60)^{3/2}$$

$$HP_1 = 0.2760$$

Referring to Figure FB3 at $\phi_{TH} = 1.547$, note that the full gate (100 percent) output is $HP_1 = 0.3070$.

$$\text{percent margin} = \frac{0.307}{0.276}\ (100) = 111.2 \ \text{percent}$$

The design meets the requirement that the expected full gate output is at least 2 percent greater than the guaranteed output.

i. The prototype expected performance is calculated as follows:

$$\phi_{TH} = \frac{11.98}{(H)^{1/2}}$$

$$HP_2 = HP_1 \left(\frac{183.5}{12}\right)^2 (H_2)^{3/2} = 233.84\ (HP_1)\ (H_2)^{3/2}$$

$$E_2 = 100 - (100 - 88.4)\left(\frac{12}{183.5}\right)^{0.2} = 93.3 \text{ percent}$$

$$\text{step-up} = (2/3)\ (93.3 - 88.4) = 3.3 \ \text{percent}$$

$$E_c = E_1 + 3.3 \text{ percent}$$

$$Q_2 = \frac{550\ HP_2}{62.3\ E_c\ H_2} = 8.828\ \frac{HP_2}{E_c\ H_2}$$

H_2	ϕ_{TH}	HP_1	E_1	HP_2	E_c	Q_2
60	1.547	0.184	70	20,000	73.3	4,015
		0.208	75	22,610	78.3	4,015
		0.234	80	25,430	83.3	4,490
		0.247	82	26,840	85.3	4,630
		0.264	84	28,690	87.3	4,835
		0.272	85	29,560	88.3	4,925
		0.282	86	30,650	89.3	5,050
		0.304	86	33,040	89.3	5,445
		0.307	85	33,360	88.3	5,560
78	1.356	0.159	70	25,610	73.3	3,955
		0.181	75	29,160	78.3	4,215
		0.204	80	32,860	83.3	4,465
		0.214	82	34,470	85.3	4,575
		0.226	84	36,410	87.3	4,720
		0.231	85	37,210	88.3	4,770
		0.238	86	38,340	89.3	4,860
		0.244	87	39,310	90.3	4,925
		0.254	88	40,920	91.3	5,070

H_2	ϕ_{TH}	HP_I	E_1	HP_2	E_C	Q_2
		0.264	88.4	42,530	91.7	5,250
		0.271	88	43,650	91.3	5,410
		0.278	87	44,780	90.3	5,615
		0.282	86	45,430	89.3	5,755
		0.285	85	45,910	88.3	5,885
		0.287	84.4	46,230	87.7	5,965
100	1.198	0.135	70	31,570	73.3	3,800
		0.165	75	38,580	78.3	4,350
		0.191	80	44,660	83.3	4,735
		0.202	82	47,240	85.3	4,890
		0.213	84	49,810	87.3	5.035
		0.219	85	51,210	88.3	5,120

j. The setting of the turbine depends upon the output requirements and the related head-tailwater conditions. The setting is generally predicated on the tailwater level with one unit operating. For this example it is assumed that it is desired to operate the unit at generator rating and 0.9 p.f. under the rated and higher heads. The corresponding turbine output is 44,500 HP. Under normal circumstances the rated condition dictates the setting. However, it is good practice to check the other head conditions to assure that unusual sigma characteristics or head-tailwater relationships do not alter this normal circumstance. The output requirements at the lower heads are assumed to vary directly with the head between the 44,500 HP at 78 foot and the 30,000 HP guaranteed output at 60 foot head. The relationship between tailwater level and discharge is linear between the following sets of conditions:

Tailwater Elev.- ft.m.s.l.	543.0	545.7
Discharge - cfs	5,000	11,000

k. The following steps are required to establish the turbine settings on the basis of the conditions set forth above:

$$\phi_{TH} = \frac{11.98}{(H)^{1/2}}$$

$$HP_1 = \frac{HP_2}{233.84\ H^{3/2}}$$

From Figure FB3, pick off σ_c' and E_1

$E_c = E_1 + 3.3$ percent

$$Q_2 = \frac{8.828\ (HP_2)}{E_c\ (H_2)}$$

$$\text{T.W. Elev.} = \frac{Q_2}{2222} + 540.75$$

$$\sigma_c = \frac{H_b - H_v - H_s - \text{safety}}{H_2}$$

From Figure 6, Appendix C: H_b = 33.3 feet, H_v = 0.8 feet (70° F.)

safety = $0.2\ D_{TH} + 0.7\ (H_2)^{1/2}$

Distr. centerline elev. = T.W. elev. + H_s + a

$a = (d)D_{TH}$

Refer to Table 4 and Figure 5, Appendix C to note d = 0.365

a = 0.365 (183.5/12) = 5.6 feet

ϕ_{TH}	1.547	1.432	1.356	1.263	1.198
HP_2	30,000	38,600	44,500	44,500	44,500
HP_1	0.276	0.282	0.276	0.223	0.190
E_1	85.4	87.4	87.3	85.1	80.0
$_c$	0.375	0.365	0.385		
E_c	88.7	90.7	90.6	88.4	83.3
Q_2	4,975	5,370	5,560	4,935	4,715
T.W. Elev.	543.0	543.2	543.3	543.0	542.9
Safety	8.5	8.9	9.2	9.7	10.1
H_s	1.5	-2.0	-6.8		
Distr. Elev.	550.1	546.8	542.1		

l. As generally expected, the rated conditions dictate the turbine setting. The HP_1 values shown in above tabulation at the higher heads are well below the range of sigma values shown on Figure FB3. This is due to the fact that HP_1 varies with the inverse of $H^{3/2}$. Since the tailwater levels do not vary substantially at the higher heads, the plant sigma with the distributor set at Elevation 542.1 varies only with the inverse of H and sufficient submergence is assured.

m. The cavitation limits for the higher heads can be established for the selected setting by deriving a relationship for $_c$ in terms of head then entering the critical sigma curves on Figure FB3 to estimate the corresponding value of HP_1. This procedure is as follows:

$$\phi_{TH} = \frac{1.98}{(H_2)^{1/2}}$$

$$\sigma_c = \frac{H_b - H_v - (\text{Distr. El.} - \text{T.W. El.} - a) - \text{safety}}{H_2}$$

By substituting known values and allowing a constant tailwater level at Elev. 543, this equation becomes:

$$c = \frac{35.9 - 0.7\ (H_2)^{1/2}}{H_2}$$

A summary of the maximum output limits is as follows:

H_2	$\emptyset_{TH}$	c	HP_1	HP_2
80	1.339	0.370	0.270	45,180
82	1.323	0.361	0.268	46,530
84	1.307	0.351	0.266	47,890
86	1.292	0.342	0.264	49,230
86.5	1.288	0.340	0.264	49,660

The limiting output of 49,500 HP can be developed at 86.5 feet and the higher heads without cavitation.

n. The prototype maximum runaway speed is estimated as follows:

$$N_{max} = \frac{1838\ \emptyset_{max}\ \ (H)^{1/2}}{D_{TH}}$$

Refer to Figure FB3 to note that $\emptyset_{max} = 2.765$

$$N_{max} = \frac{1838\ (2.765)(100)^{1/2}}{183.5} = 277 \text{ rpm}$$

o. As an exercise, the user may elect to analyze the merits of the first selection with N = 128.6 rpm and D_{TH} = 180.6 inches. This will familiarize the user with the formulas and procedures required to develop the necessary data for a given design.

p. The prototype dimensions of the principal parts and water passages of the turbine can be calculated from the dimensionless ratios shown in Table 4 Appendix C.

APPENDIX E

SECTION III

ADJUSTABLE BLADE PROPELLER TURBINE

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1. DESIGN REQUIREMENTS.

a. Powerplant capacity: 69,000 KW with 2 units

b. Generator requirements:

(1) Nameplate rating: 36,320 KVA, 0.95 p.f., 34,500 KW, 13.8 KV and 60 hz.

(2) Must be designed for continuous operation at rated KVA at rated voltage, p.f. and frequency.

c. Turbine requirements:

(1) Net heads: 70 foot rated, 53 foot minimun and 88 foot maximum.

(2) Require 33,200 HP guaranteed output at 53 foot net head.

(3) Turbine output is limited to 50,190 HP (rated KVA at 1.0 p.f.)

(4) The setting requires a concrete semi-spiral case.

2. TURBINE SELECTION.

a. The turbine output required to match the generator rating is calculated as follows:

$$HP = \frac{34,500}{0.746\ (0.97)} = 47,680$$

b. Refer to Figure 3, Appendix C to note that the rated head condition and the wide head range for this unit dictates a 6 blade runner. The recommended specific speed at the 70 foot rated head is calculated as follows:

$$N_S = \frac{1,100}{(70)^{1/2}} = 131.5$$

c. The speed is calculated as follows:

$$N = \frac{131.5\ (70)^{5/4}}{(47,680)^{1/2}} = 121.9$$

Round to nearest synchronous speed = 120 rpm

Corrected N_s = 129.5

d. For preliminary studies requiring only an approximate speed and runner throat diameter, the following empirical formula for $O/\backslash_{TH}$ may be used to calculate the diameter:

$$\phi_{TH} = 0.049\ (129.5)^{0.695} = 1.440$$

$$D_{TH} = \frac{1838\ (1.440)\ (70)^{1/2}}{120} = 184.5 \text{ inches}$$

e. The appropriate model test curves for these conditions are shown on Figure K3, Appendix D. The design point for rated conditions is to be located along the on-cam 32° blade angle curve. The location of the design point is determined by iteration. This is accomplished by substituting associated values of HP_1 and ϕ_{TH} in the following formula for specific speed:

$$N_s = 153.17\ \phi_{TH}\ (HP_1)^{1/2}$$

The approximate value ϕ_{TH} = 1.440 from d above is used in the first step of the iterative process as follows:

ϕ_{TH}	1.440	1.450	1.445
HP_1	0.342	0.343	0.342
N_s	129.0	130.1	129.4

The design point is located at ϕ_{TH} = 1.445 and HP_1 = 0.342. The runner throat diameter for this preliminary selection is calculated as follows:

$$47{,}680 = 0.342 \left(\frac{D_{TH}}{12}\right)^2 (70)^{3/2}$$

$$D_{TH} = 185 \text{ inches}$$

f. The location of this design point with reference to the extremes in head conditions should be checked as follows:

$$\phi = \frac{120\ (185)}{1838\ (H_2)^{1/2}} = \frac{12.08}{(H_2)^{1/2}}$$

$$HP_1 = \frac{(HP_2)}{(185/12)\ (H)_2^{3/2}} = \frac{(HP_2)}{(237.67)^{3/2}}$$

(1) At the 53 foot minimum head a guaranteed output of 33,200 HP is required:

$$\phi_{TH} = \frac{12.08}{(53)^{1/2}} = 1.659$$

$$HP_1 = \frac{33{,}200}{237.67\ (53)^{3/2}} = 0.362$$

Refer to Figure K3 at ϕ_{TH} = 1.659 to note that the full gate (100 percent) HP_1 = 0.405

$$\text{percent margin} = \frac{0.405}{0.362}\ (100) = 111.9 \text{ percent}$$

The design meets the requirement that the expected full gate output is at least 2 percent greater than the guaranteed output.

(2) For the 88 foot maximum head, check the efficiencies for generator rated load:

$$\phi_{TH} = \frac{12.08}{(88)^{1/2}} = 1.288$$

$$\text{Rated } HP_1 = \frac{47{,}680}{237.67\ (88)^{3/2}} = 0.243$$

From Figure K3, E_1 = 88.1 percent
This efficiency is considered satisfactory.

(3) From Figure K3 it is noted that the best ϕ_{TH} = 1.35, which corresponds to a net head of 80.1 feet. At maximum efficiency, E_1 = 89.6 percent, the corresponding prototype expected output is 32,370 HP. This corresponds to about 68 percent of generator rated load.

g. It is recommended that alternate designs be investigated before making a final selection. In this instance the adjacent synchronous speeds, 112.5 and 128.6 rpm. should be investigated. The balance of this example, however, will proceed on the basis of N = 120 rpm and D_{TH} = 185 inches.

h. The prototype expected performance is calculated as follows:

$$\phi_{TH} = \frac{12.08}{(H)^{1/2}}$$

$$HP_2 = HP_1\ (185/12)^2 = 237.67\ HP_1\ (H_2)^{3/2}$$

$$E_2 = 100 - (100 - 89.6)\ (12/185)^{0.2} = 94 \text{ percent}$$

$$\text{set-up} = (2/3)\ (94.0 - 89.6) = 2.9 \text{ percent}$$

$$E_C = E_1 + 2.9 \text{ percent}$$

$$Q_2 = \frac{550\ HP_2}{62.3\ (E_C)\ H_2} = 8.828\ \frac{HP_2}{E_2\ H_2}$$

H_2	ϕ_{TH}	HP_1	E_1	HP_2	E_C	Q_2
53	1.659	0.072	76	6,600	78.9	1,390
		0.080	78	7,340	80.9	1,505
		0.086	80	7,890	82.9	1,580
		0.098	82	8,990	84.9	1,760
		0.115	84	10,550	86.9	2,015
		0.151	86	13,850	88.9	2,590
		0.184	87	16,870	89.9	3,120

H_2	ϕ_{TH}	HP_1	E_1	HP_2	E_C	Q_2
		0.276	87	25,310	89.9	4,680
		0.298	86	27,330	88.9	5,110
		0.327	84	29,990	86.9	5,735
		0.352	82	32,280	84.9	6,320
		0.372	80	34,110	82.9	6,840
70	1.444	0.058	76	8,070	78.9	1,285
		0.063	78	8,770	80.9	1,385
		0.070	80	9,740	82.9	1,480
		0.078	82	10,860	84.9	1,610
		0.091	84	12,670	86.9	1,835
		0.110	86	15,310	88.9	2,165
		0.124	87	17,260	89.9	2,415
		0.145	88	20,180	90.9	2,795
		0.158	88.5	21,990	91.4	3,030
		0.173	89	24,080	91.9	3,300
		0.233	89	32,430	91.9	4,440
		0.249	88.5	34,660	91.4	4,770
		0.271	88	37,720	90.0	5,220
		0.302	87	42,040	89.9	5,885
		0.324	86	45,100	88.9	6,385
		0.355	84	49,410	86.9	7,155

H_2	ϕ_{TH}	HP_1	E_1	HP_2	E_c	Q_2
88	1.288	0.060	80	11,770	82.9	1,420
		0.069	82	13,540	84.9	1,595
		0.082	84	16,090	86.9	1,855
		0.101	86	19,820	88.9	2,230
		0.115	87	22,560	89.9	2,510
		0.134	88	26,290	90.9	2,895
		0.144	88.5	28,250	91.4	3,095
		0.158	89	31,000	91.9	3,375
		0.190	89.5	37,280	92.4	4,040
		0.218	89	42,770	91.9	4,660
		0.232	88.5	45,520	91.4	4,985
		0.245	88	48,070	90.9	5,290
		0.270	87	52,970	89.9	5,900

h. The setting of the turbine depends upon the output requirements and the related head-tailwater conditions. The setting is generally predicated on the tailwater level with one unit operating. For this example it is assumed that it is desired to operate the unit at the generator rating of 0.95 p.f. under the rated and higher heads. The corresponding turbine output is 47,680 HP. Under normal circumstances the rated condition dictates the setting. However, it is good practice to check the other head conditions to assure that unusual sigma characteristics or head-tailwater relationships do not alter this normal circumstance. The output requirements at the lower heads are assumed to vary directly with the head between the 47,680 HP at 70 foot and the 33,200 HP guaranteed output at 53 foot head. The relationship between tailwater level and discharge is linear between the following sets of conditions:

Tailwater Elev.- ft. m.s.l.	500.0	543.5
Discharge- cfs	5,000	11,000

i. The following steps are required to establish the turbine settings on the basis of the-conditions set forth above:

$$\phi_{TH} = \frac{12.08}{(H_2)^{1/2}}$$

$$HP_1 = \frac{HP_2}{237.67 \quad (H_2)^{3/2}}$$

From Figure K3, pick off σ_c and E_1 for above ϕ_{TH} and HP_1 values.

$$E_c = E_1 + 2.9 \text{ percent}$$

$$Q_2 = 8.828 \frac{HP_2}{E_c H_2}$$

$$\text{T.W. Elev.} = \frac{Q_2}{1714} + 537.1$$

$$\sigma_c = \frac{H_b - H_v - H_s - \text{safety}}{H_2}$$

From Figure 6, Appendix C: H_b = 33.3 feet, H_v = 0.8 feet (70o F.)

safety = $0.2\ D_{TH} + 0.7\ (H_2)^{1/2}$

Distributor centerline Elevation = T.W. Elevation + H_s + a

$a = (d)\ D_{TH}$

Refer Table 4 and Figure 5, Appendix C to note that d = 0.368

$a = 0.368\ (185/12) = 5.7$ feet

Refer to Figure S9 for values of critical runner sigma.

H_2	53	62	70	80	88
ϕ_{TH}	1.659	1.534	1.444	1.351	1.288
HP_2	33,200	40,870	47,680	47,680	47,680
HP_1	0.362	0.352	0.342	0.280	0.243
E_1	81.0	83.6	84.7	87.3	88.2
c	0.880	0.780	0.725	0.475	0.375
E_c	83.9	86.5	87.6	90.2	91.1
Q_2	6,590	6,730	6,860	5,830	5,250
T.W. Elev.	540.9	541.0	541.1	540.5	540.2
Safety	8.2	8.6	9.0	9.4	9.7
H_s	-22.3	-24.5	-27.2	-14.9	-10.2
Distr. Elev.	524.3	522.3	519.6	531.3	535.7

j. As generally expected, the rated conditions dictate the turbine setting. This is due to the fact that HP_1 varies with the inverse of $(H)^{3/2}$. Since the tailwater levels do not vary substantially at the higher heads, the plant sigma with the distributor set at Elev. 519.6 varies only with the inverse of H and sufficient submergence is assured.

k. The cavitation limits for the higher heads can be established for the selected setting by deriving a relationship or σ_c in terms of head, then entering the critical sigma curves on Figure S9 to estimate the corresponding value of HP_1 This procedure is as follows:

$$\phi_{TH} = \frac{12.08}{(H_2)^{1/2}}$$

$$\sigma_C = \frac{H_b - H_v \text{ (Distr. El. - T.W. El. -a) - safety}}{H_2}$$

By substituting known values and allowing a constant tailwater level at Elev. 540, this equation becomes:

$$\sigma_C = \frac{56 - 0.7\ (H_2)^{1/2}}{H_2}$$

A summary of the maximum output limits is as follows:

H_2	ϕ_{TH}	σ_C	HP_1	HP_2
72	1.424	0.695	0.337	48,930
72	1.404	0.675	0.332	50,230

The limiting output of 50,190 HP can be developed at 74 feet and the higher heads without cavitation with distributor centerline Elev. 519.6.

1. The prototype dimensions of the principal parts and water passagaes of the turbines can be calculated from the dimensionless ratios shown in Table 4, Appendix C.

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