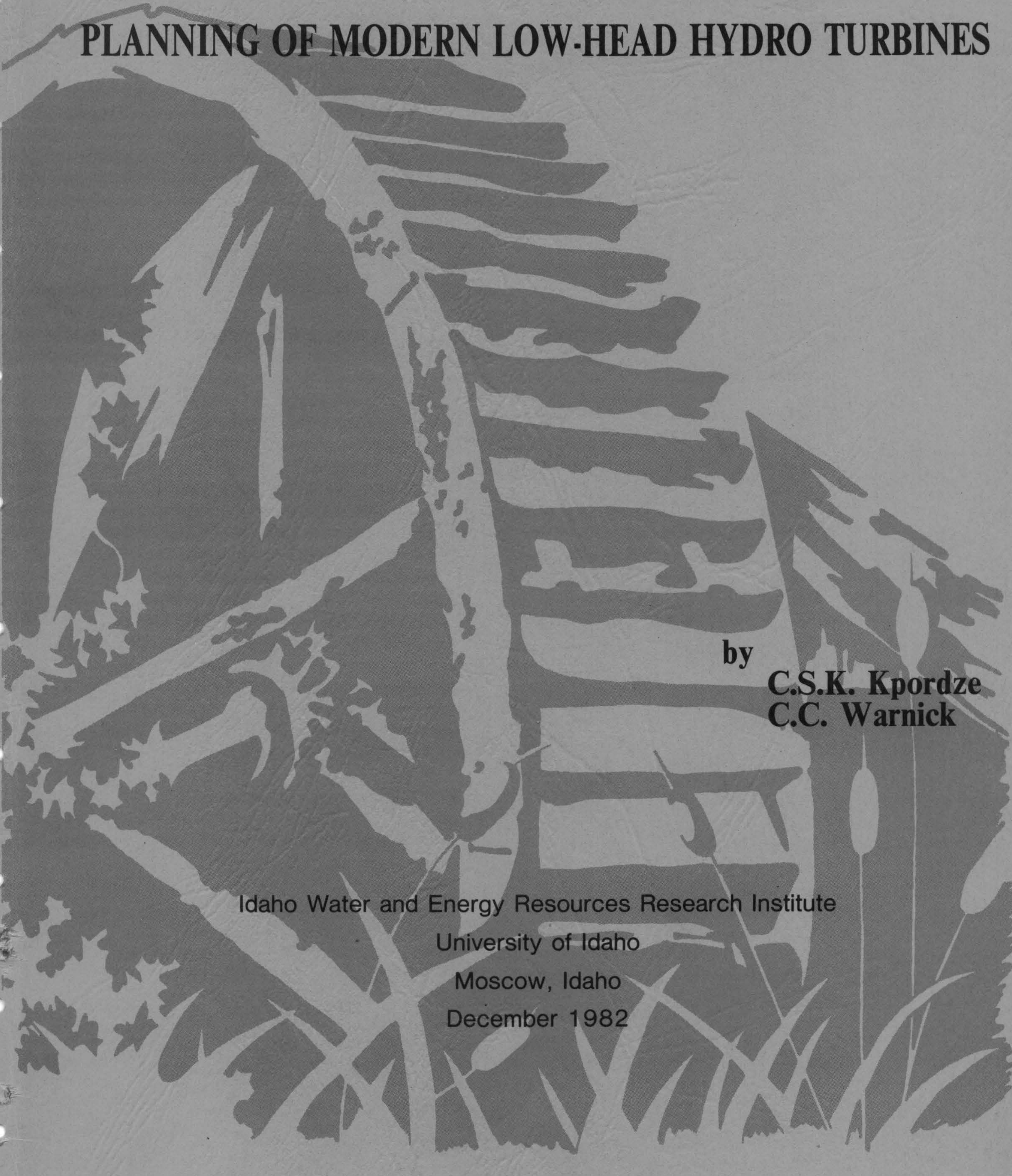
Research Technical Completion Report Project A-077-IDA

EXPERIENCE CURVES FOR FEASIBILITY STUDIES AND PLANNING OF MODERN LOW-HEAD HYDRO TURBINES



by

C.S.K. Kpordze
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Idaho Water and Energy Resources Research Institute
University of Idaho
Moscow, Idaho

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ABSTRACT

This reportdcontains the results of an extensive investigation of the characteristics of over two hundred low-head turbines manufactured all over the world that have been installed or that are due to be installed in hydropower plants between 1953 and 1984. The research focused mainly on bulb turbines, with horizontal shaft arrangement and tubular turbines with their shafts either horizontal or inclined at an angle to the horizontal. The characteristics of the other types of low-head turbines are not presented because adequate data could not be collected on their characteristics during this study period. The characteristics on bulb and tubular turbines are presented in the form of statistical diagrams and regression equations suitable for preliminary design and feasibility studies of low-head hydro projects. Nomographs have been developed for displaying the relationships between the various turbine characteristics and comparing the important dimensions and parameters of turbines which have found common application in the hydro-power technology. New simplified parametric ratio for selection of turbines have been developed that should expedite preliminary selection study for hydropower projects.

ACKNOWLEDGEMENTS

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1. INTRODUCTION

During the 1980's there will be an urgent search for alternate sources of energy and the exploitation of small scale energy resources all over the world. The present escalating fuel prices coupled with spiralling energy demands have brought a great desire for energy independence and interest in exploiting reliable and renewable energy sources. One of the proven and most efficient sources of energy which could readily be exploited is the hydroelectric power. In the developed countries, most of the ideal sites for high head installations have already been developed; while in the developing countries, lack of capital might make large scale hydro developments out of their reach.

Low-head hydroelectric developments could prove to be the key to maintaining and improving the present standards of living all over the world. Low-head hydro sites are natural sites which are still available in sufficient quantity and adequatley distributed to ensure the full use of the hydraulic resources in accordance with development programs designed to meet consumer needs. A very efficient class of the turbo machines for supplying power at low-head installations is small-scale propeller turbines with their axis horizontal or slightly inclined from the horizontal. This class of turbines is not new but the recent advances in the technology and the state-of-the-art of their development have made it an attractive alternative to other types of turbines. In the low-head range between 3 and 46 meters (10 and 150 feet), small scale propeller turbines are gaining increasing interest throughout the world and are often selected in place of vertical shaft-Kaplan machines to improve the feasibility of low-head schemes or tidal plants (IECO, no date).

In preliminary design and feasibility studies, it is necessary for the engineer to get information on power output of a given plant, the synchronous speed, an estimate of runner diameter, size of civil works components, preliminary costs and to decide on particular arrangements for the hydro-power plant units before final selection is made (Warnick, in press). The manufacture of turbines is usually based on similitude which is the theory and art of predicting prototype condition from model observations. In order to put such information in a form suitable for engineers and planners to use, statistical methods of analysis have been used to correlate the salient parameters and constants of a specific class of turbines manufactured for hydro-power plant installations currently in operation. The curves resulting from such statistical analysis are known as experience curves. Experience curves have been developed for the Francis, Pelton, and the vertical-shaft Kaplan turbine (de Siervo et al., 1976, 1977, 1978). The specific speed has been universally used as the means of relating other important turbine parameters to a common base characteristic.

Experience curves have not yet been developed for the low-head hydro turbines manufactured in recent years. There is therefore a need to develop experience curves for the modern low-head hydro turbines. Very little work has been done in trying to relate entrance works of hydro-power installation dimensions to the characteristics of the turbine runner and this could be very useful in helping to standardize that portion of hydropower design and construction. Experience curves as conceived in this research fill a present gap in the needed tools for the preliminary design of hydro power plants and provide engineers with the latest information for the preliminary design and feasibility studies of low-head power plants. The investment costs of low-head hydroplants are relatively high; of these the electro-mechanical equipment and the civil works are generally the major portion. One of the ways of reducing these costs is to use the experience and the information accumulated in the manufacture of the equipment and the installation of the existing low-head hydroplants all over the world, in the planning, feasibility studies and design of new lowhead hydropower installations. In this way the best choice of turbine and configuration can be determined rapidly for any site.

Objectives

The general purpose of this study was to carry out an extensive investigation on the new low-head turbines manufactured all over the world and develop statistical diagrams or experience curves for feasibility studies and planning of future hydroelectric developments.

Specific objectives were to:

(i) Study bulb turbines with horizontal shaft arrangements and develop experience curves relating such parameters as effective head, unit speed, unit power, unit discharge, cavitation coefficient and draft static head to specific speed of the turbine.

(ii) Study the same as above for tubular type hydropower development.

(iii) Develop graphical nomographs for displaying the various relationships developed in items (i) and (ii).

(iv) Develop empirical equations for simple use in design and planning offices of agencies and companies involved in hydro power development.

(v) Compare results with experiènce curves for impulse turbines, Francis turbines and conventional Kaplan turbines that are now available in the engineering literature.

2. LITERATURE REVIEW AND THEORY

2.1 Hydraulic Turbine Terminology

A hydraulic turbine is a class of machines characterized by the transference of energy between a continuous fluid stream and a rotor. The technological objective in the case of turbines is to convert the pressure, potential and kinetic energy of water into mechanical energy, through a rotor known as turbine runner, by utilizing the difference in the elevation between two water levels or reservoir levels. The rotary action of the turbine in turn drives an electrical generator that produces electrical energy. The difference in elevation between the water above the turbine, headwater, and below the turbine, tailwater, is called gross head. The gross head as shown in Figure (2.1), represents the energy expended by a unit weight of water as it moves from the surface of the headwater to the surface of the tailwater.

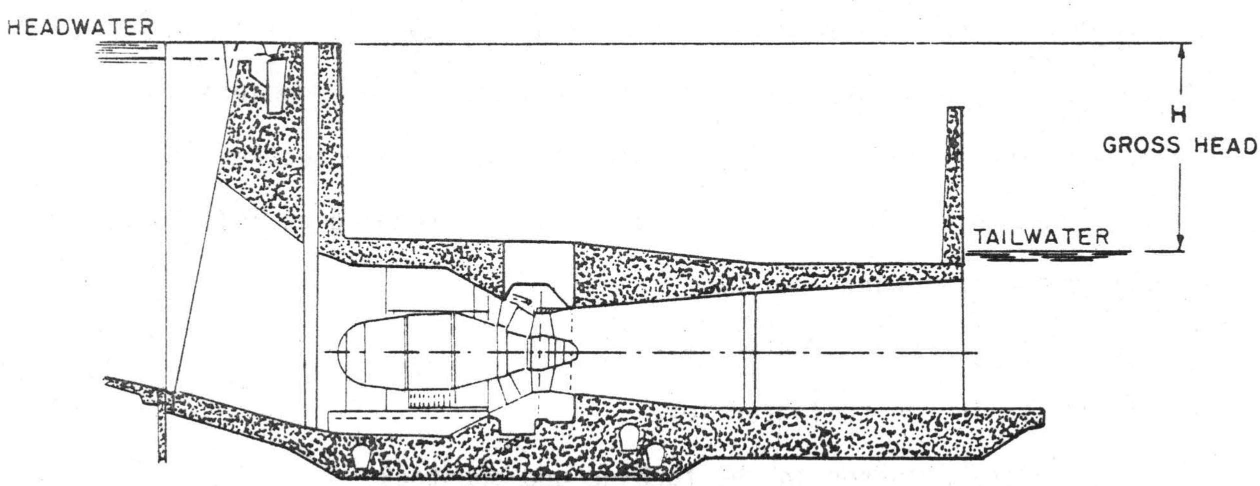


Figure 2-1. Schematic diagram of a power plant showing gross head, headwater and tailwater.

The estimation of the hydroelectric potential at a site is usually explained in terms of work, power and energy. These words are defined as follows:

Work is the transferred energy and is the product of force and distance moved in the direction of the force.

Power is the rate of transferring energy or work per unit time.

Energy is the capacity to do work or the time integral of power. The power developed by the water depends both on discharge and effective head.

Discharge is the volume rate of flow with respect to time through the plant. All the power available in the water is not completely utilized in the turbine due to inevitable losses in the system. Some of the losses which occur when a turbine unit is running at full load, are due to the following:

(a) Fall in the headwater level.

(b) Rise in the tailwater level.

(c) Friction losses in the turbine and penstock.

(d) Other losses through screens, sluice gates, etc.

The difference between the gross head and the sum of all the losses in head is known as the effective head. A related word, design head, is the effective head for which the turbine is designed for best speed and best efficiency.

The theoretical power output at the generator terminals is given by:

$$\text{Power, } P(KW) = \rho g Q H = 9.81 Q H \quad (KW) \tag{2.1}$$

Where ρ = mass density of water, (kg/m^3)

g = acceleration due to gravity, (m/sec^2)

Q = the discharge in (m^3/sec.)

H = the effective head in meters

In hydro power language, power is usually measured in both kilowatts (KW) and horsepower (HP) units.

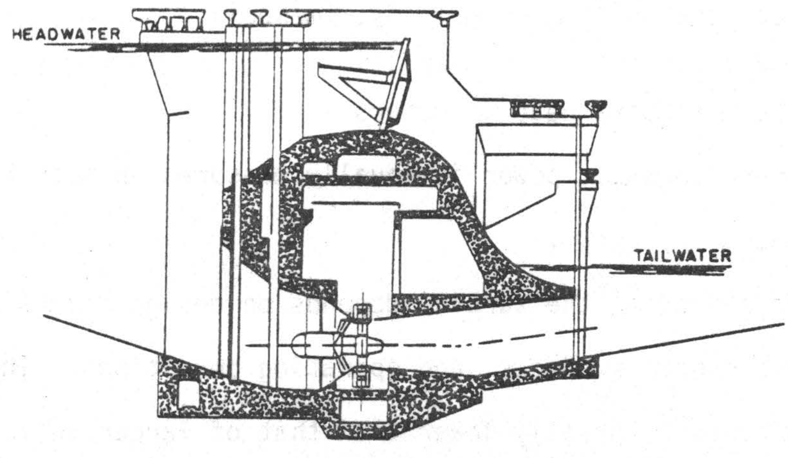
The efficiency of the turbine depends on design dimensions, condition of the runner surfaces, and operating conditions. The efficiency of small turbines is usually lower than that of larger units since the roughness of the water-filled passageways is relatively higher, the Reynolds number is smaller and the mechanical losses are relatively larger.

2.2 Types of Propeller Turbines Used in Low-Head Hydro Installations

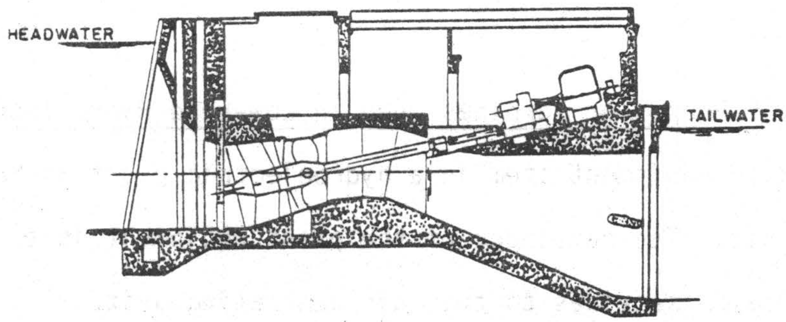
The major equipment item in a hydroelectric plant is the turbine/-generator unit. The remainder of the plant equipment is to control protect and provide services to the main generating unit.

There are two fundamental types of hydraulic turbines, the impulse and reaction. Impulse turbine is one which utilizes the kinetic energy of a high velocity jet of water to transform the water energy into mechanical energy. The reaction turbine develops power from the combined action of pressure energy and kinetic energy of water. Reaction turbines are subclassified as propeller and Francis types (C.C. Warnick, in press). For low-head hydro power plants, the small-scale propeller type turbine is the most often used in new installations. The three kinds of low-head small propeller turbines considered in this study (shown in Figure 2-2) are:

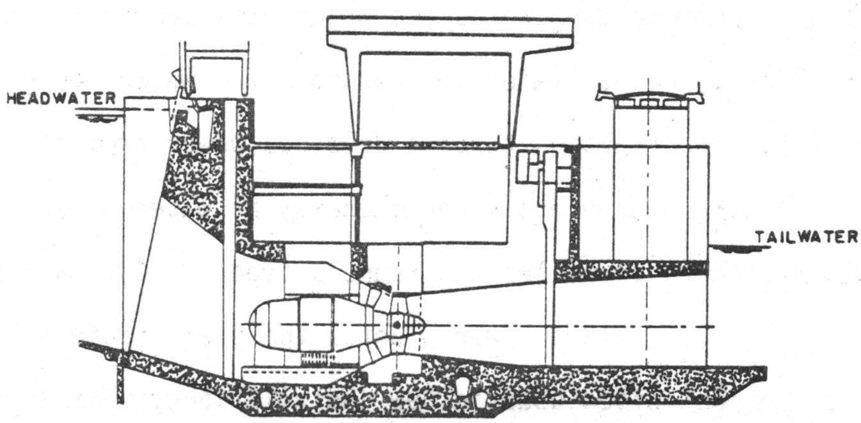
1. The Rim-generator type.
2. The Tubular type.
3. The Bulb type.



RIM-GENERATOR TURBINE



TUBULAR TURBINE



BULB TURBINE

Figure 2-2. Types of low-head turbine installations.

Rim-generator turbine

The rim-generator turbine, which has the generator rotor mounted on the periphery of the turbine runner blades, is one of the early horizontal shaft types; patented by the late Mr. L.F. Harza in 1919 and 1924 (Neyrpic, 1964 and Mosonyi, 1963). It is well suited for operation on a small power system because there is adequate space on the periphery of the runner for a large generator with a sufficiently large rotational inertia. The rim-generator turbine offers a potential saving in powerhouse costs because it is a compact unit and has no driving shaft and the generator stator is also simpler, since it need not conform with the water passages. The main disadvantage of the Rim-generator type is that completely reliable methods of supporting the generator from the water passage have not been satisfactorily proven in service. In general it has been found that a rim-generator type is limited by economy and potential leakage at the circumferential seals to smaller installations with a maximum runner diameter of 3 to 10 meters (10 to 33 feet) (Neyrpic, 1964 and Mosonyi, 1963).

Tubular turbine

Tubular turbines are horizontal or slant mounted turbines with fixed or adjustable blade propeller runners. The generators are either directly coupled to the turbine shaft or connected through a speed increaser. The generators of tubular turbines are located outside the water passageways which result in a longer shaft and larger floor space requirements than the other low-head types. It has the advantage of a relatively simple seal arrangement, as compared to the rim-generator type, and also no generator cooling problem as in the case of bulb turbine . It combines the straight-through flow advantage of rim-generator and the standard design of an external generator. This turbine type has a high rotational inertia thus the generator can be reduced in size by the addition of a gear box to increase its speed. The largest tubular turbines in physical size and amongst the largest in unit power in the world up to date were the units installed in the Ozark plant in the U.S.A. These units have runners of 8 meters diameter.

Bulb turbine

Bulb turbines are horizontal units which have wicket gates and fixed or adjustable-blade propeller runners directly connected to the generator. The generator is enclosed in a water-tight structure (Bulb) located within the water passage usually on the upstream side of the turbine runner. The water passages must be large enough to accommodate the generator and the required excavation is generally somewhat deeper than that of the tubular turbine. The straight-flow water passageway also minimizes the head loss.

The maximum economic size is generally in the range of 7 to 7.5 meter (23 to 25 feet) runner diameter and with a maximum head of 15 to 18 meters (50 to 60 feet), (IECO, no date). The minimum size is determined by the requirements for access into the Bulb; the practical limit being reached when the runner approaches a diameter of 3 to 4 meters (10 to 13 feet). The largest bulb turbines and generators in physical size ever built in the world up until now and the first installed in the United States is the Rock Island Hydroelectric Plant which was commissioned in 1978. These units have runner diameters of 7.4 meters and are rated at 53 MW per unit. The bulb turbine is a compact, self-contained, operationally flexible installation. The main advantage over the other types of small propeller turbines is its good operating record, since it lacks the seal problems that have occurred with the rim-generator type or alignment problems evident in tubular type turbine installations. There are some disadvantages, such as poor access to the generator and difficulty in generator cooling.

2.3 History of Development of Low-head Turbines

The history of hydraulic machinery could be traced as far back as the invention of the water wheel used in the ancient times for lifting water from a lower to a higher elevation in irrigation systems. Later the water wheel provided power by direct connection or with pulley and gear systems to drive various machines such as grist mills and textile mills. The earliest record of the horizontal shaft Roman water wheel is found in the writings of Marcus Virtruvius dated probably about A.D. 27 (Wilson, 1975). One of the first major advances came in 1827 when Fourneyron invented a hydraulic machine designed for driving processing machinery and which could use heads of 20 to 30 meters (66 to 98 feet) or more to produce several hundred horsepower. Later, heads as high as 200 meters (656 feet) and 500 meters (1640 feet) were equipped with turbines by Berges (Neyrpic, 1964). He selected natural sites where the length of the supply and tailrace channel could be reduced to a minimum, by using a penstock. This idea led the pioneers in hydropower development to seek "ideal" sites where maximum drop occurred for a given horizontal distance. Since the higher the head, the greater the amount of energy that could be obtained from a unit volume of water, hydro power development and research were concentrated mainly on the high head and large capacity installations. In 1919 L.F. Harza produced an idea for a horizontal shaft annular alternator, the spider, which was transformed into a series of blades acted upon by water flow.

This idea developed into the rim-generator patented to him about 1920. This type of units was highly promoted in Europe during World War II and has become a perfected unit in the case of the STRATFLO units of Escher Wyss (Neyrpic, no date).

Recent development and manufacture of low-head turbine units has been reported in almost all of the developed countries. However, as early as 1943, much of the sustained effort in low-head turbine development was done in France where engineers and manufacturers have sought to produce a rational and economic solution to the problems raised by tidal power installations, in connection with the Rance tidal project (Neyrpic, 1964) and (Barberis, 1965). The present advances made in the state-of-the-art of the technology of small propeller turbines such as the Bulb turbine and the tubular turbine will probably be sustained since small-scale hydropower development is becoming an attractive energy production alternative.

2.4 Hydraulics of Hydropower

Hydraulics of hydropower deals with the transfer of energy between a continuous water stream and the turbine runner blades. The energy transfer occurs when water flows from the intake zone through the tortuous water passages past the runner blades to the tailwater zone. The three fundamental approaches usually used to develop the hydraulic theory of hydropower engineering are:

(1). Energy-Work approach

(2). Bernoulli-Energy Equation approach

(3). Kinetic Theory approach

Energy-Work approach.

This approach considers the work done in each time interval, dt, by an elemental volume of water, dv, in moving from the surface of the head water, position 1, to the surface of the tailwater level, position 2 (Fig. 2-3).

Work = Force x Distance

$$dW = \rho g dv\ H \tag{2.2}$$

Where dW = work done by elemental mass of water, (Joules)

ρ = density of water, (Kg/m^3)

g = acceleration of gravity, (m/sec^2)

H = vertical distance moved by the elemental volume, (m)

dv = elemental volume, (m^3)

The power extracted by the turbine runner is the rate of doing work and can be represented mathematically as follows:

$$\text{Power} = \text{Work/ Time} \tag{2.3}$$

$$dP = dW/\ dt$$

$dP = \rho g\ dQ\ H$ Where dQ = elemental discharge

Summing the elemental power components of the total discarge, Q, passing through the turbine unit gives theoretical power as:

$$P = \rho g Q H \tag{2.4}$$

If the system losses are considered, the power output at the shaft is given by:

$$P\ (KW) = \rho g Q H \eta = 9.81\ QH\eta \tag{2.5}$$

where η = total efficiency of the plant.

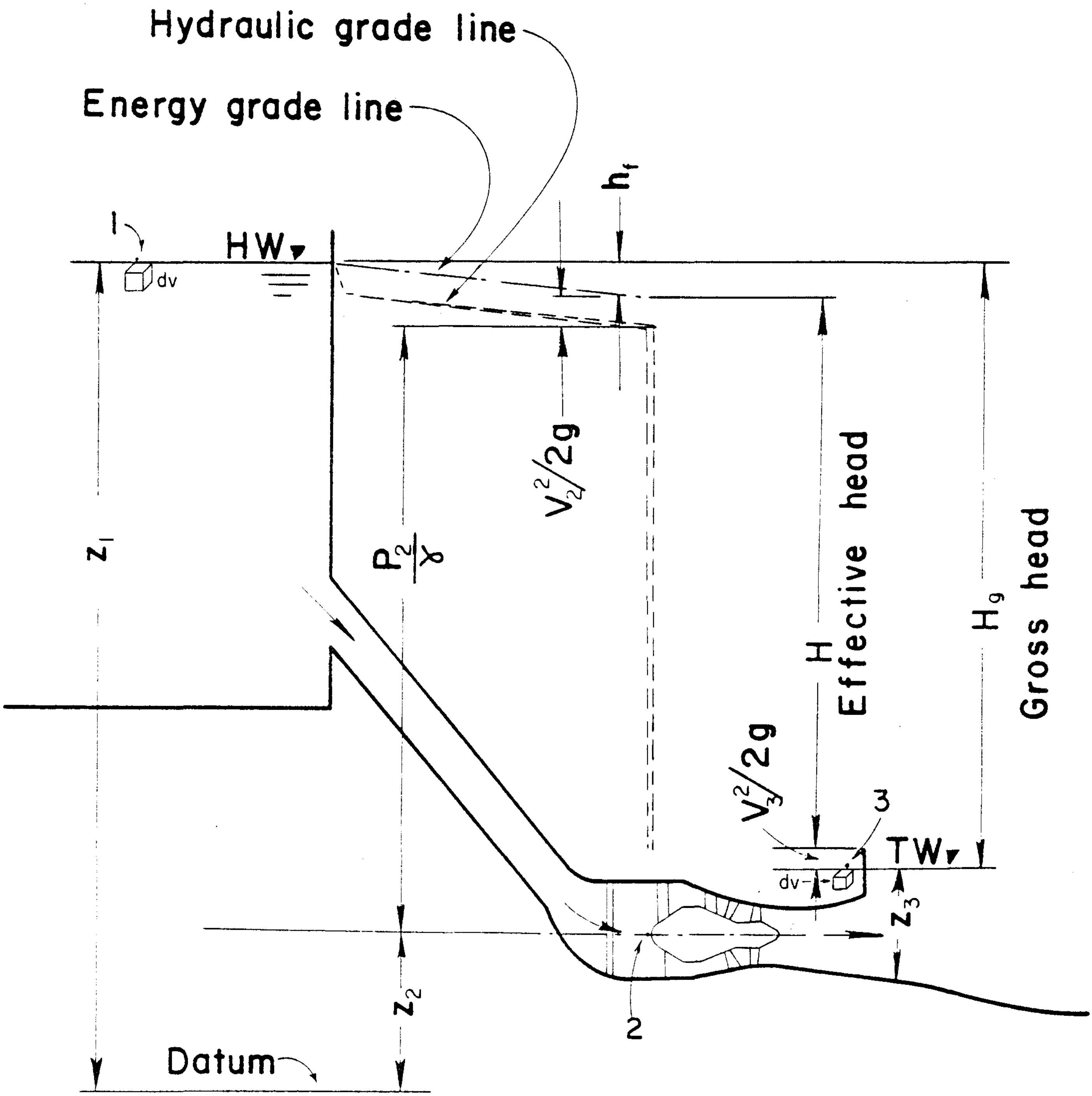


Figure 2-3. Diagram for developing energy-work approach for hydro-power theory.

Bernoulli-Energy equation approach.

According to the Bernoulli theorem, which expresses the law of energy conservation in hydromechanics, the total energy of a unit weight of water along a streamline is constant. Mathematically, the Bernoulli equation states that the sum of the component energies of pressure, position and kinetic energy is constant.

$$\frac{P}{\gamma} + Z + \frac{V^2}{2g} = E = \text{constant.} \tag{2.5}$$

where

P = Average pressure, (N/m^2)
Z = Height above mean sea level (m)
V = Average water velocity, (m/sec.)
γ = Specific weight (ρg), (N/m^3).
E = Total energy (m of water)

Considering the movement of a unit weight of water from position 1 to position 2, (Figure 2-4) then

$$\frac{P_1}{\gamma} + Z_1 + \frac{V_1^2}{2g} = \frac{P_2}{\gamma} + Z_2 + \frac{V_2^2}{2g} + H = E \tag{2.6}$$

Where subscripts 1 and 2 denote component variables at positions 1 and 2 (in Figure 2-4), respectively.

Representing the losses in the system by h_f then

$$\frac{P_1}{\gamma} + Z_1 + \frac{V_1^2}{2g} = \frac{P_2}{\gamma} + Z_2 + \frac{V_2^2}{2g} + H + h_f = E \tag{2.7}$$

(see Figure 2.3)

The effective head is given by

$$H = Z_1 - Z_2 - \frac{V_2^2}{2g} - h_f \tag{2.8}$$

The energy per unit time, power, is therefore represented by:

$$P(KW) = gQ\rho H\eta = 9.81\ QH\eta \tag{2.9}$$

Which is the usual expression for the power output from the shaft

Where η = the total efficiency.

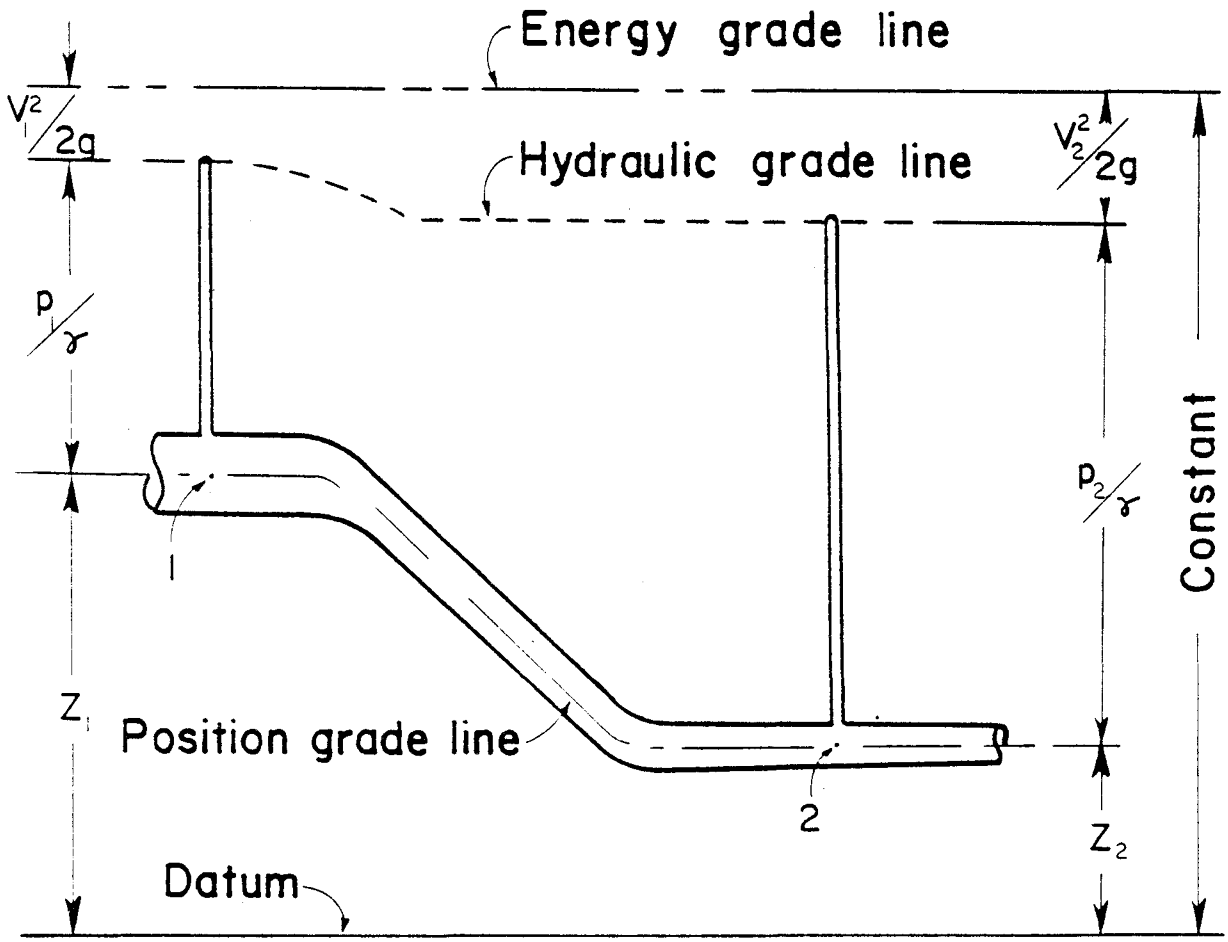


Figure 2-4. Bernoulli diagram relating energy gradeline and hydraulic gradeline.

Kinetic theory approach.

The Kinetic theory approach of explaining the energy converting action between the water stream flowing through a power plant and turbine runner blades is based on the following:

(1) One-dimensional calculation of a mean flow-through velocity at the runner inlet and outlet.

(2) Euler approximation, namely, that when leaving a row of blades, the relative fluid velocity follows the blade contour.

When the water passes through the runner blades, the stream deviates from its initial direction and its pressure on the blades causes their rotation, thereby creating a torque upon the turbine shaft (Kovalev, 1961).

The torque (T), imparted by the water to the runner is given by the equation:

$$T = \frac{W}{g}\left(r_1 V_1 \cos \alpha_1 - r_2 V_2 \cos \alpha_2\right) \tag{2.10}$$

(see Figure 2.5)

Where $W = \gamma Q$ = quantity of water flowing, (Kg/sec)

r_1 = radius of runner in meters at the periphery where the water first strikes the runner vane,(m).

V_1 = absolute velocity of the water at the entrance to the runner (m/s).

α_1 = angle that the absolute velocity vector V_1 makes with tangent to runner at water particle entrance (degrees)

r_2 = radius of the runner in meters at the point where water leaves the runner, (m).

V_2 = absolute velocity of the water at the exit to the runner, (m/s)

α_2 = angle that the absolute velocity vector V_2 makes with tangent to runner at water particle exit, (degrees).

The power developed at the turbine shaft is given by :

$$P = T\omega = \gamma \frac{Q}{g}(r_1 V_1 \cos\alpha_1 - r_2 V_2 \cos\alpha_2)\omega \qquad (2.11)$$

where ω = the angular velocity, (rad/sec).

Since the peripheral velocities at the entrance and exit of the runner are $r_1\omega = u_1$ and $r_2\omega = u_2$ respectively and $\cos\alpha_1 = \cos\alpha_2 = 1$, then the power developed is expressed by

$$P = T\omega = \gamma\frac{Q}{g}(v_1 u_1 - v_2 u_2) = \gamma\, QH\eta \qquad (2.12)$$

Therefore $(v_1 u_1 - v_2 u_2)/g = H\eta$ in the usual expression for power.

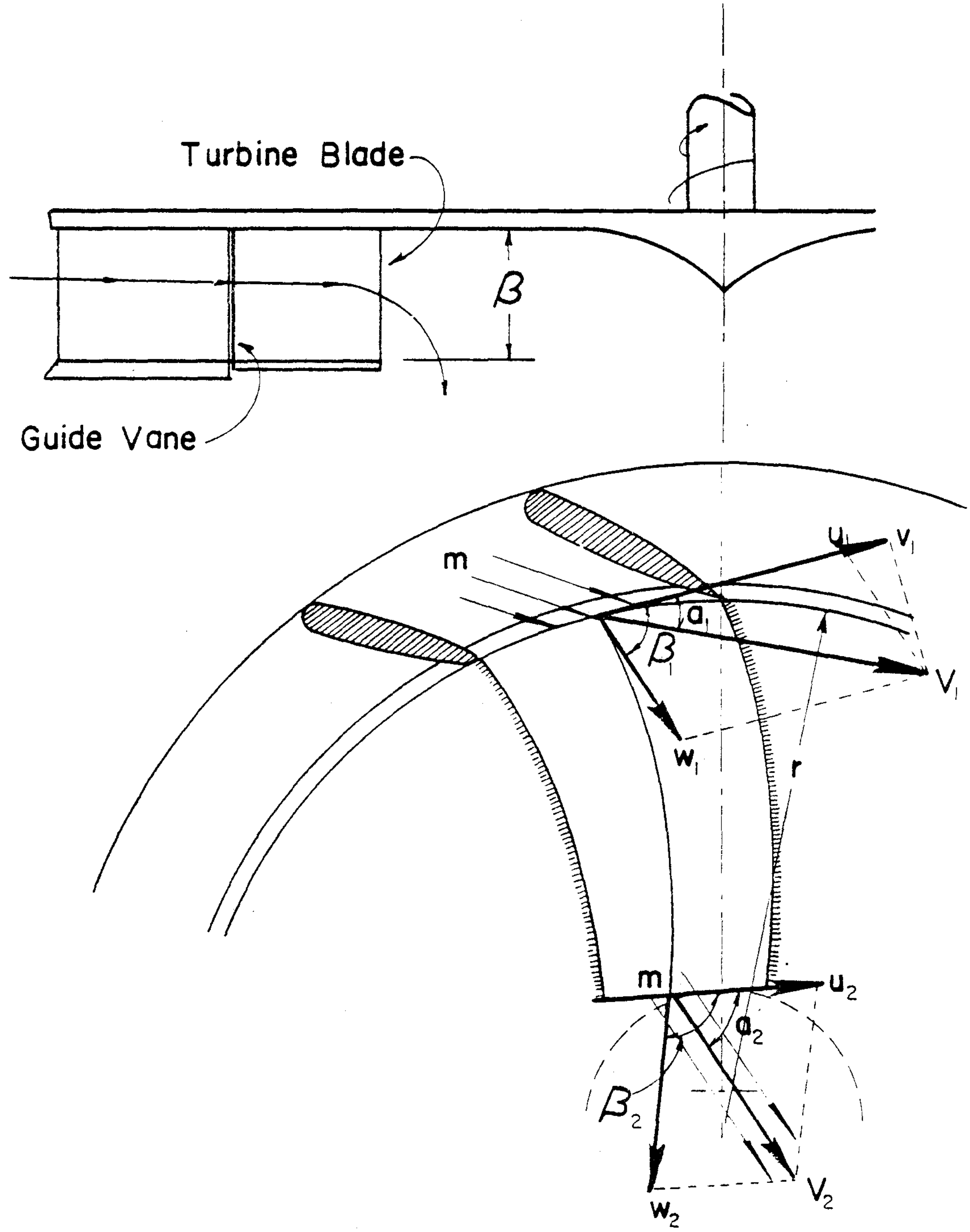


Figure 2-5. Vector diagram of water action on reaction turbine.

2.5 Similarity Laws

Every water turbine is designed to operate under a preferred or design head. Under this head, it will pass a certain discharge of water and it should run at its correct speed. If the head is higher, it will discharge more water, it should run faster, and it will develop more power and vice versa. The externally observable variables which characterize the performance of turbines are: head, discharge, speed of rotation and torque delivered. A certain number of these parameters can be varied independently but the turbine is found to behave deterministically, so that a complete set of independent variables may be found in the knowledge of which all the others could be predicted. Laws of prediction, similarity laws, based on theory and observation of model performance, have been developed for characterizing and predicting turbine performance of units of different size and type or units of similar design to those that have already been built.

In model studies of flows involving complex boundary conditions, the basic requirement is that the model be an exact geometric replica of the prototype (geometric similarity) and that the ratios of corresponding velocities and accelerations be the same throughout the flow field (Kinematic similarity). In order to maintain geometric and kinematic similarities between flow situations, the forces that act on corresponding masses in the model and prototype must be in the same ratio throughout the entire flow field (Dynamic similarity). Consequently, the flow patterns will be the same in the model as in the prototype if both geometric similarity and dynamic similarity are satisfied. When turbines of different sizes are designed to have corresponding linear dimensions with common geometric ratio, the turbines are said to be homologous. The power output, speeds and flow characteristics are proportional and the turbines tend to have equal efficiencies.

In order for performance curves resulting from similarity laws to be of practical use, they must be represented by a single curve relating a dimensionless net force to a dimensionless parameter expressing the characteristics of the flow and the properties of the fluid, i.e., in the form y = f(x) where y is a nondimensional combination containing one dependent variable (Csanady, 1964). By using the concept of dynamic similarity, performance laws could be formulated in the one-parameter form, valid for incompessible, nonviscous and non-cavitative fluids, possible choices for y and x are:

$$\frac{P}{\rho D^5 N^3} = f\left(\frac{N\ P}{\rho^{1/2}(gH)^{5/4}}\right) \tag{2.13}$$

Which is Power coefficient = f(Power speed ratio).

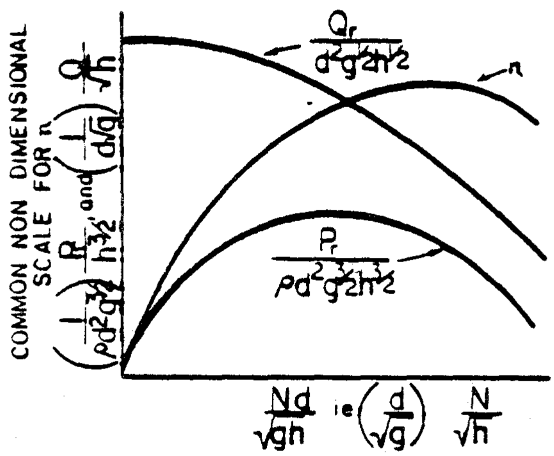
Barr (1966) in a discussion of a paper by Jones (Jones, 1964) proposed the adoption of a fundamental approach to the derivation of similarity criteria which could be applied to any fluid flow situation. This approach uses a dynamic velocity or water spouting velocity, as a basis of measure for scaling of systems where dynamic similarity is intended. A convenient first step in such a system is to form dynamic velocities. A dynamic velocity is proportional to the velocity that would be attained by a representative element were it to move from rest through a representative distance under the influence of an active force. By assuming geometrically similar family of turbomachines and that the ratio of the imposed to dynamic velocities be constant,

$$V_i/V_h = N\ D/\sqrt{gh} = \text{constant} \tag{2.14}$$

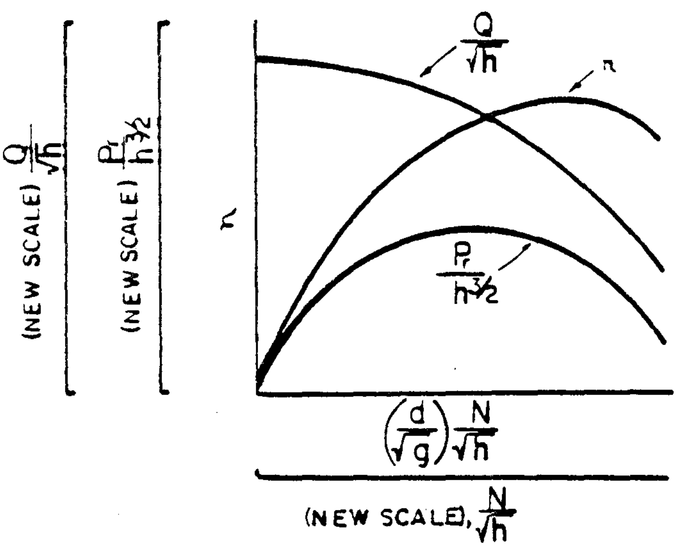
where V_i = imposed velocity, m/sec

V_h = dynamic velocity.

Barr formulated expressions for relevant turbine constants by correlating data relating to power output or design discharge in dimensionless groups against the nondimensional reduced speed, $ND/\sqrt{gh}$, the best value of which determines the best operational condition. He also showed that for turbomachines operating on the Earth's surface with water as the fluid flowing through the system, separate ordinate scales are necessary for his proposed system to be reduced to the presently used definition of unit conditions. (see Fig 2-6)



A plot of resultant power versus non dimensional speed.



New scale plot.

Figure 2-6. A plot of Jones' proposed new approach to specific speed.

2.6 Turbine Constants and Empirical Equations

The similarity laws discussed above are developed and presented in a series of formulas known as turbine constants. Traditionally the turbine constants in both the English system and the Metric system of units have dimensions. However recently the various turbine manufacturers have put forth an international system for expressing turbine constants (Allis Chalmers, no date). The International system used dimensionless ratios and metric, SI, units for various parameters. This new system is more convenient to use because similitude reasoning proves that these dimensionless numbers remain constant for a particular machine shape if the machine is run at optimum unit speed value.

Some dimensionless turbine constants in the International system of units are given below.

Unit speed, the ratio of the peripheral speed of runner to the theoretical spouting velocity of water, is given by:

$$\omega_{ed} = \omega\ D \ / \ \sqrt{gH} \tag{2.15}$$

where ω_{ed} = unit speed
ω = angular velocity of runner, (rad/sec)
D = reference diameter of runner, (m)
g = acceleration of gravity, (m/sec^2)
H = design head, (m)

It can be shown that ω_{ed} is dimensionless as follows:

$$\omega_{ed} = \omega\ D/\sqrt{gH}\ \ \frac{(\text{rad/sec})\ (\text{m})}{[(\text{m/sec}^2)(\text{m})]^{1/2}} = \frac{(1/T)\ (L)}{[(L/T^2)(L)]^{1/2}} = \frac{(1/T)\ (L)}{(1/T)\ (L)} = \text{dimensionless}$$

where T = Time units
L = Length units

Historically the unit speed has been defined in the metric form as:

$$N_{11} = \frac{N\ \ D}{H^{0.50}} \tag{2.16}$$

where the terms are as previously defined, note the N_{11} is not dimensionless (Warnick, in press).

Unit discharge, is expressed as

$$Q_{ed} = Q/D^2\ (gH)^{0.5} \tag{2.17}$$

where Q_{ed} = unit discharge.

Q = design discharge flowing through the turbine, (m^3/sec).

Likewise, historically unit discharge has been defined in the metric form as follows:

$$Q_{11} = \frac{Q}{D^2\ H^{0.50}} \tag{2.18}$$

where the terms are as previously defined. Note, the Q_{11} is not dimensionless (Warnick, in press)

Unit power is expressed as

$$P_{ed} = P/\rho D^2 (gH)^{3/2} \tag{2.19}$$

where P_{ed} = unit power

P = turbine power output, (watts)
ρ = mass density of water, (Kg/m^3)
g = acceleration of gravity, (m/sec^2.)

Likewise, historically unit power has been defined in the metric form as follows:

$$P_{11} = \frac{P}{D^2\ H^{1.50}} \tag{2.20}$$

where the terms are as previously defined. Note P_{11} is not dimensionless (Warnick, in press).

Specific speed. The term "specific speed" was introduced into turbine construction to characterize the hydraulic properties of a turbine in terms of speed and discharge capacity, as well as to compare various turbines and runners. The specific speed is numerically equal to the rotational speed of a turbine of a given series which develops a unit of power under a unit head. The specific speed varies with changes in the operating conditions of the turbine, i.e., power or speed at given head. Hence different types of turbines may be compared in terms of specific speed only if they are designed for predetermined operating conditions. The development of the expression for specific speed is as follows:

Using equations 2.9 and 2.19

$$P = \rho g Q H \eta \qquad (2.5)$$

$$P_{ed} = \frac{\rho g Q H \eta}{\rho D^2 (gH)^{3/2}} \qquad (2.19)$$

and substituting $\omega_{ed} \sqrt{gH}/\omega$ for D.

$$P_{ed} = \frac{\rho g Q H \eta}{\rho \{\frac{\omega_{ed}}{\omega} \sqrt{gH}\}^2 (gH)^{3/2}}$$

grouping the constants, P_{ed}, ω_{ed}, and on one side,

$$\frac{P_{ed} \quad \omega_{ed}^{2}}{\eta} = \frac{\omega Q^{1/2}}{(gH)^{3/2}}$$

taking the square root of both sides of the equation gives the specific speed equation as:

$$\omega_s = \frac{\sqrt{(P_{ed} \, \omega_{ed}^{2})}}{\eta} = \frac{\omega Q^{1/2}}{(gH)^{3/4}} \qquad (2.21)$$

Recognizing $P_{11} = \frac{P}{D^2 \, H^{3/2}}$ from Equation 2.20 and substituting D^2 from Equation 2.16 in Equation 2.19 the following equation is obtained.

$$P_{11} = \frac{P}{(\frac{N_{11}^{2}}{N^2} \, H) \, H^{1.50}} \qquad (2.22)$$

or

$$P_{11}\,N_{11}^{\;2} = \frac{N^2\,P}{H^{\,2.50}}$$

By taking the square root of both sides of the equation we have the traditional form of the specific speed

$$N_s = (P_{11}\,N_{11}^{\;2})^{0.50} = \frac{N\,P^{0.50}}{H^{1.25}} \tag{2.23}$$

where the terms are as previously defined. Note again that N_s is not dimensionless (Warnick, in press)

This form of specific speed is most widely used and has come to be recognized throughout the turbine industry. Because the turbine manufacturers have developed all their test data using this form of specific speed it is likely to be continued in use.

An American form of specific speed uses the power term expressed in horsepower and the head term expressed in feet. It should be noted that in converting from the dimensionless form of specific speed W_s to the N_s that one is expressed in terms of turbine discharges and the other in terms of turbine output. To be able to convert from one system to the other the actual turbine efficiency must be known.

In addition to the International system of expressing the specific speed, Csanady (1964) proposed a similar expression for specific speed using the English system of units for the parameters as given below:

$$\Omega = \omega_q^{\;1/2} \;/\; (gh_n)^{3/4} \tag{2.24}$$

where Ω = specific speed
ω = angular velocity of the runner, (radians/sec)
q = water discharge, (ft^3/sec)
h_n = net head, (ft.)
g = acceleration of gravity, (ft/sec^2)

Other variations of turbine constants have been developed for the convenience of analyzing certain characteristics of turbines. Table 2-1 page 39 is a summary of the various forms of the turbine constants in use. Equations for each and conversions for the specific speed from one system to another is also given.

Specific speed selection is very important in the design of turbines. It permits a general comparison of all classes and series of turbines and their classification according to rotational speed and discharge capacity. In common practice, however, turbines are selected by means of universal characteristic curves which permit a thorough quantitative analysis of turbine parameters under any operating conditions. Such curves are known as Hill Curves. An excellent explanation of the Hill Curves has been presented by (Warnick, in press).

2.7 Cavitation, Turbine Selection and Setting

Cavitation

Cavitation is a very important consideration in the design, selection and setting of turbines; cavitation is normally defined as the formation of the vapor phase in a liquid. The term cavitation (originally coined by R.E. Froude) can imply anything from initial formation of bubbles (inception) to large-scale attached cavities (supercavitation). Cavitation can affect the performance of turbines by decreasing the power output and efficiency of the turbine. Performance breakdown, noise, vibration and erosion in turbo machinery and large valves are all associated with cavitation. From an engineering point of view the basic concern is, will cavitation occur? And if cavitation cannot be avoided due to site conditions, economic or other operational considerations, then can the turbine still function properly (Arndt, 1981)?

Research has shown that when water has a high velocity or when a solid body moves rapidly within it, the continuity of the flow is disturbed and vapor-filled pockets appear in the areas of high velocity. This phenomenon is known as cavitation. The possibility of cavities forming in the stream can be shown by referring to the equation of conservation of energy. From the Bernoulli equation:

$$\frac{P}{\gamma} + Z + \frac{V^2}{2g} = \text{constant} \tag{2.25}$$

where

$\frac{P}{\gamma}$ = pressure head, (m)

Z = elevation head, (m)

$\frac{V^2}{2g}$ = velocity head, (m)

For Z = 0, the Bernoulli equaiton becomes:

$$\frac{P}{\gamma} + \frac{V^2}{2g} = \text{constant.} \qquad (2.26)$$

At a point where V increases, the velocity head, $\frac{V^2}{2g}$, increases while the pressure head, $\frac{P}{\gamma}$, decreases. When the absolute pressure decreases to the value of the vapor pressure of the water for a given temperature, vapor filled pockets (cavities) are rapidly formed within the stream. When these cavities or vapor bubbles are carried away by the stream into areas of lower velocity and increased pressure, the vapor bubbles collapse instantaneously. The pressures generated by the collapse of the bubble may reach extremely high values of the order of 15,000 atmospheres. Although sometimes referred to as "cold boiling", cavitation is distinguished from boiling in the sense that the former is induced by the lowering of hydrodynamic pressure, whereas the latter is induced by the raising of vapor pressure to some value in excess of the hydrodynamic pressure. The two phenomena are related. Cavitation inception and boiling can be compared in terms of the vapor-bubble dynamics of sub-cooled and superheated liquids (Plesset, 1957). Quite often a clear distinction between the two types of phenomena cannot be made. This is especially true for cavitation in liquids other than cold water. Ordinary liquids can sustain tension and more than one type of cavitation process can occur in the same flow field. Bubble growth can be a result of formation of the vapor phase, or be due to the release of dissolved gas, or can be some combination thereof. It is not always possible to clearly distinguish between vaporous and so-called gaseous cavitation (Arndt, 1981). Experiments have also shown that high pressures and temperatures occur when the bubble is compressed. The compression of the cavitational bubbles also sets up electrical phenomena which create bubble luminescence. Observations showed that the cavitation bubbles oscillate continuously and resonance occurs under certain flow conditions (Kovalev, 1961).

In reaction turbines, (Francis or Propeller turbines), the water is discharged from the runner into the tailwater through the draft tube. When a particle of water flows into a reaction turbine, its velocity increases through the constricted water-passages, and the surrounding pressure decreases. If at any point between the turbine inlet and the draft tube outlet, the pressure falls below the vapor pressure, a vapor filled bubble or cavity will form. If this cavity is carried on to a point where the pressure increases, normally on the surface of the turbine runner, guide vanes or fixed boundaries of the water-passages, it will collapse and deliver a severe blow to the metal. As millions of these pin-point blows are struck in the same area, the surface begins to deteriorate forming needle point holes (pitting), that is the damaging action of cavitation. The second stage of cavitation is characterized by a loss of efficiency, severe vibration and a sudden drop in power output, accompanied by the metal turning into a spongy form which, if not repaired or replaced can cause serious problems (Kovalev, 1961) and (Gilkes, no date).

Turbine setting and selection

Reaction turbines may be operated with the center line of the runner blades installed either above or below the normal tailwater surface elevation. The effective head for developing power is unaffected by the setting, provided the turbine is submerged below a specified minimum elevation. USBR monograph No. 20 recommends 0.3m (1 foot) below the elevation at which cavitation damage and loss of performance have approached unacceptable values. The 0.3m (1 foot) margin allows for variation of atmospheric pressure and minor variations in runner characteristics. However, unstable operation and/or excessive pitting of the runner and discharge ring may develop due to cavitation if the setting of the turbine is not properly matched with the speed of the runner and the flow patterns that persist during turbine operation. A cavitation coefficient (number), σ, has been developed from the results of model tests and prototype performance to select a safe setting for reaction turbines (e.g., Thoma's coefficient). This cavitation coefficient is generally referred to as plant sigma and defined as follows:

$$\sigma_p = \frac{H_a + H_v - H_s}{H} \qquad (2.27)$$

where σ_p = plant cavitation coefficient (plant sigma)

H_a = atmospheric pressure head in meters of water

H_v = vapor pressure in meters

H_s = static draft head in meters (it is positive above the tailwater elevation and negative when below tailwater elevation)

H = turbine effective head in meters.

σ_p is usually termed the cavitation coefficient of the power station or plant sigma, because it depends only upon net head (H), site elevation and static draft head (H_s), i.e., the three main parameters of the plant.

For bulb, tubular and rim-generator turbines, the static draft head is considered as the difference in elevation between the highest point of the runner blades and the minimum tailwater elevation. This is due to the fact that in these turbines, the most unfavorable cavitation conditions occur on the periphery of the blades at their highest point. The static draft head is considered positive if the minimum tailwater elevation is below the reference elevation mentioned above; conversely, the static draft head is negative if the minimum tailwater elevation is above the reference elevation of the runner blades.

Reaction turbines have critical cavitation coefficients, σ_{cr}, (critical sigma) which are functions of design of the runner, the static draft head, (H_s) and speed of the runner. This critical value is usually established by laboratory tests on turbine models. The critical sigma is considered as a performance boundary such that for $\sigma_p > \sigma_{cr}$ no cavitation occurs, while for $\sigma_p < \sigma_{cr}$ cavitation effects occur which could lead to performance degradation, noise and vibration (Arndt, 1981).

In order to eliminate cavitation, selection of the turbine speed (N), throat diameter (D) and setting could be made from the critical sigma (σ_{cr}) and the plant sigma (σ_p) values. The best selection criteria is to choose a turbine with high plant sigma. To increase the plant sigma, it is necessary to increase the absolute value of the static draft head (H_s), as can be seen in equation 2.27. This however involves a greater amount of civil works on the erection of the substructure. Therefore, an important consideration in the selection of a turbine for a proposed hydroelectric project is to choose or specify turbine with the best cavitational properties as it permits the construction cost of the plant to be reduced.

The selection of a turbine setting is often a compromise between site conditions, economic and other operational factos. The lower the setting, the greater the unit discharge through the turbine and therefore the greater the turbine speed, resulting in a smaller turbine/generator unit and lower cost. However, the lower the setting, the higher the cost of civil works. As an example, if a power station is to be located on a soft gravel river bed, it will be of advantage to select a low-setting as the cost of the additional civil works will usually be less than the reduction of cost due to selection of a smaller unit. While in the case of a rocky subsoil the cost of the excavation will be very high and it will be a better choice from an overall project cost view point to select a high setting.

Other factors that affect selection of sigma value and setting are as follows:

1. Large changes in tailwater level most often call for a low setting.
2. Lowest tailwater level may be associated or not with the highest head and this will result in an important change in the setting of two identical units.
3. Part load operation may also be a factor to consider. The sigma value for very small discharges has to be kept higher to prevent the occurrence of unusual cavitation patterns (Neyrpic, correspondence).

2.8 Selection of Period of analysis using Cluster Analysis

Design and manufacture of hydroelectric turbines is usually based on theory and observation of model data. It is therefore a combination of science and art of modeling. Furthermore, the values of the turbine characteristics depend on a number of factors including: the level of workmanship of the workers in the manufacturing company, the advancement in the turbine design and manufacture technology, management policies, economic factors and project site characteristics. Therefore, the turbine data collected from different turbine manufacturers over a period of years need to be classified into periods of similar turbine design characteristics, before useful performance curves can be developed for the turbines.

Cluster analysis in this research was used to group the turbine data into periods of similar turbine design characteristics. Cluster analysis is a means of classifying observations (in this case turbine characteristics) on the basis of similarity (Anderberg, 1973). This method was considered a valid statistical technique for classifying the turbine data into periods of similar turbine design characteristics. In this study, the type of cluster analysis technique used is similar to the weighted pair-group method used by Davis (Davis, 1973). The data base of four turbine characteristics on 190 bulb turbines manufactured all over the world, was treated as a 4 x 190 matrix. The four turbine characteristics used were: specific speed, head, unit discharge and unit power. Using a computer, the 4 x 190 matrix was partisioned into a 4 x n_1 and 4 x n_2 submatrices based on the date of commissioning of the turbines. Where n_1 denotes number of bulb turbines put into service during the periods of time under consideration and n_2 denotes 190 - n_1. The only restriction placed on the value of n_1 was that n_1 be greater than 15 ($n_1 > 15$). The analysis procedure was started from the earliest date among the turbine commissioning dates, 1953 to the next date, say, 1960 such that n_1 was greater than 15. Then linear regression analysis was performed on the resulting 4 x n_1 and 4 x n_2 matrices and the corresponding correlation coefficients noted for each of the four groups of characteristics. The value of n_1 was then increased by increasing the period of analysis and the correlation coefficients recomputed and compared with the previously computed values. This process was repeated until the resulting correlation coefficients were less than the next previously computed values. Then the period of analysis was taken as constituting the first sample period. The procedure was repeated to determine the next period of turbine design characteristics. The second trial period was selected to include one year after the first period up to the year such that n_1 for the second time interval exceeded 15 turbine characteristics. Two such periods identified for the 190 bulb turbines were: 1953 to 1965, constituting the first sample period, and 1966 to 1984, the second sample period. The two above mentioned periods were then used to group all the turbine characteristics throughout the rest of the analysis to determine experience curves for low-head hydroelectric turbines. The only modifications made were in the cases where the characteristics curves resulting from the regression analysis for the two periods were so close as to justify representation by a single regression curve or the number of turbine characteristics in each time period were too few to justify the group classification. In all such cases the period of analysis was taken to include 1953 to 1984.

2.9 Methods of Data Analysis

In order to discuss the types of interpolation functions used to develop the experience curves some definitions are necessary.

Curve fitting, is the general method of finding equations for approximating curves which fit given sets of data points plotted on a rectangular coordinate.

Regression, is one of the main purposes of curve fitting. It is used to estimate one of the variables (the dependent variable) from the other (independent variable). The process of estimation is often referred to as regression. If Y is to be estimated from X by means of some equation, the equation is called regression curve of Y on X.

Correlation, is the degree of relationship between variables. When only two variables are involved, the relationship is called simple regression and simple correlation. When more than two variables are involved, the relationship is known as multiple regression and multiple correlation (Spiegel, 1961). Sometimes it helps to plot the scatter diagrams in terms of transformed variables. For example if log Y vs X leads to a straight line, $\log Y = a + bX$ will be used as an equation for the approximating curve. The types of equations used in this study are:

Linear regression; $Y = a + bX$ (2.28)

Exponential curve fit; $Y = ae^{bX}$ (2.29)

Power curve fit; $Y = aX^{b}$ (2.30)

Logarithmic curve fit; $Y = a + b\ln X$ (2.31)

Where a and b are constants.

The data shown in Tables A-1 and A-2 in appendix represents some of the outstanding low-head turbines manufactured all over the world between the years 1953 to 1982. The data used in the analysis were screened to include only turbines having complete information; those having incomplete information or unusual operating characteristics were eliminated. The resulting sets of data were analyzed using a computer system known as "Statistical Analysis System" (SAS), developed by SAS Institute, Inc. of North Carolina, USA. The above named group of programs was run on IBM Virtual Machine Facility/370 (CMS). The SAS computer system is set up to perform linear regression analysis, to plot data values and to print out any desired input or computed values. In order to use the transformed variable models, the data must be transformed and arranged in the appropriate linear model form. The selection of turbine constants used in the linear regression models was based on the turbine constants currently used in practice and the type of information needed for preliminary investigation or feasibility studies of hydroelectric projects.

Table 2-1 Comparison of turbine constants in different systems of units and forms of equations

Parameter	American system hp, inch, CFS, ft, rpm		European system Kw, m, m^3/sec, rpm		Dimensionless system	
	Designation	Formula	Designation	Formula	Designation	Formula
Speed ratio	ϕ	$\phi = \frac{dn}{43.368(h)^{0.5}}$	k_u	$k = \frac{D_3 N}{60(2gH)^{0.50}}$	ω_{ed}	$\omega_{ed} = \frac{\omega D}{(gH)^{0.5}}$
Unit speed	N_1	$N_1 = \frac{dN}{h^{0.5}}$	--	--	ω_{ed}	$\omega_{ed} = \frac{\omega D}{(gH)^{0.5}}$
Unit discharge	q_1	$q_1 = \frac{q}{d^2\ h^{0.5}}$			Q_{ed}	$Q_{ed} = \frac{Q}{D^2(gh)^{0.5}}$
Discharge coefficient	--	--	--	--	$Q_{\omega d}$	$Q_{\omega d} = \frac{Q}{\omega D^3}$
Unit torque	--	--	--	--	T_{ed}	$T_{ed} = \frac{T}{\rho D^3\ gH}$
Torque coefficient	--	--	--	--	$T_{\omega d}$	$T_{\omega d} = \frac{T}{\rho\omega^2\ D^5}$
Energy coeficient	--	--	--	--	$E_{\omega d}$	$E_{\omega d} = \frac{gH}{(\omega D)^2}$
Unit power	P_1	$P_1 = \frac{P}{d^2 h^{1.5}}$	--	--	P_{ed}	$P_{ed} = \frac{P}{\rho D^2 H^{1.5}}$
Power coefficient	--	--	--	--	$P_{\omega d}$	$P_{\omega d} = \frac{P}{\rho\omega^3\ D^5}$
Specific speed	n_s	$n_s = \frac{n\ P^{0.5}}{h^{1.25}}$	N_s	$N_s = \frac{n\ P^{0.5}}{H^{1.25}}$	ω_s	$\omega_s = \frac{\omega\ Q^{0.5}}{(gH)^{0.75}}$
Conversion term	$n_s = 0.262\ N_s$		$N_s = 157.453\,\omega_s^{1.002}$ - Bulb turbine $N_s = 169.687\,\omega_s^{0.937}$ - Tubular turbine			

H = net head, m of water; h = net head, ft of water; d = runner diameter in inches, D = runner diameter in m; q = discharge in cfs, ft^3/sec; Q = discharge in m^3/sec; ω = angular velocity, rad/sec; T = torque kgm; g = acceleration due to gravity, m/sec^2; ρ = mass of density of water, kg/m^3.

3. DATA ACQUISITION

The goal in collecting data for this research was to acquire an accurate and representative set of information on all low-head turbines manufactured in the world. To fulfill this objective different methods were used to acquire data.

Initially data were obtained through personal visits paid by Professor C.C. Warnick to the major American and European turbine manufacturing companies in connection with a manuscript he prepared for a hydro-power Engineering text book (in press). Additional data were obtained from technical publications and through correspondence with the turbine manufacturing companies and agencies which own or operate hydropower plants. The greater portion of the data obtained from technical publications was verified for accuracy through correspondence with the manufacturers. In cases where there were disparities in a set of data obtained from different sources, the set of data supplied by the manufacturers was adopted and used so that the resulting experience curves will represent the existing characteristics of the turbines as they were designed and manufactured.

4. RESULTS AND DISCUSSIONS

4.1 Introduction

The results in this chapter represent an extensive investigation of more than two hundred outstanding low-head turbines manufactured all over the world that have been installed or that are due to be installed in hydropower plants between 1953 and 1984. Due to insufficient data collected on the rim-generator units, the characteristics of only bulb and tubular turbines are reported. The characteristics of these turbines are presented in the form of statistical curves drawn by simple regression procedures, using digital computer programs known as "Statistical Analysis System." Since the main purpose of this research is to produce experience curves for planning and feasibility studies, the statistical diagrams presented resulted from correlation analysis among turbine characteristics used more often by hydraulic turbine manufacturers, designers and planners. However, the dimensionless forms of the characteristics developed in chapter two of this text can be deduced from the results presented.

During the collection and screening of turbine data, classification of turbines was based mainly on date of commissioning of turbine units and periods of similar turbine characteristics and not on detailed turbine features such as number of blades, type of generator or mode of connection between turbine and generator. It was assumed that the turbine parameters supplied by the manufacturers were the appropriate values and the rated values of the parameters represent the values of turbine characteristics at best efficiency under full load. Some of the salient hydraulic turbine characteristics presented in chapter two are expressed below:

Unit speed, $N_{11} = NDH^{-0.50}$ (2.16)

Unit discharge, $Q_{11} = Q\ D^{-2.0}\ H^{-0.50}$ (2.18)

Unit Power, $P_{11} = P\ D^{-2.0}\ H^{-1.50}$ (2.20)

Specific Speed, $N_s = N\ P^{0.50}\ H^{-1.25}$ (2.23)

Cavitation Coefficient $\sigma = \dfrac{H_a - H_v - H_s}{H}$ (2.27)

The terms in the above relations are as previously defined.

The available data on 190 Bulb and 38 tubular turbine installations shown in the appendix have been classified into groups using cluster analysis. The dates of commissioning of the turbine units were used as the classification parameter. Regression analysis was made separately for the groups of Bulb turbines and also for the tubular turbines.

The criteria for determining the degree of relation between two turbine characteristics or group of characteristics is the value of correlation coefficient. Correlation coefficient values of +1 or -1 denotes perfect linear correlation while values of correlation coefficient between +1 and -1, a measure of linear dependence that exist between two parameters or groups of parameters.

The statistical diagrams resulting from regression analysis performed on the groups of the characteristics are grouped together on pages 58 through 82 on Figures 4-1 through 4-23. The corresponding correlation coefficients, standard deviations, sample period, number of turbine installations per sample period and turbine types are shown on pages 53 through 57 in Tables 4-1 through 4-5. The range of operation of Bulb and Tubular turbines produced by some major turbine manufacturers and efficiency versus year graphs are shown in Figures 4-24 through 4-32; these curves can be found at the end of this chapter.

4.2 Specific Speed

The specific speed has been universally used as the turbine parameter to which all the other turbine characteristics have been related. In most hydro projects the three design parameters usually required at the beginning of the project are normally; the design head, power and discharge. It is therefore desirable to seek relations between specific speed and these known parameters. In order to obtain regression of specific speed on design head, the specific speed can be expressed as some function of design head as shown below:

$$N_s = N\, P^{0.50}\, H^{-1.25} = f(H) \qquad (4.1)$$

The specific speed as a function of design head is plotted as Figures 4-1 and 4-15 which show that the specific speed tends to increase with decreasing design head. It should be noted however that the specific speed does not depend on design head alone but also on turbine speed of rotation and rated power. The results showed that the specific speed of Bulb and Tubular turbines examined in this research had values between 398 to 1814 and 407 to 1007, respectively.

The regression equations for specific speed versus design head shown in Table 4-1 and Figures 4-1 and 4-15, the corresponding correlation coefficients, r, and standard deviations, s, are presented below:

Regression Equations for Bulb Turbines

		r	s
$N_s = 1420.954\ H^{-0.346}$	for 1953-1960	0.36	242.82
$N_s = 967.678\ H^{-1.590}$	for 1961-1970	0.22	113.77
$N_s = 1757.752\ H^{-0.341}$	for 1971-1984	0.47	113.40

Regression Equations for Tubular Turbines

Regression equation for bulb turbines, the corresponding correlation coefficient and standard deviation are:

	r	s
$N_s = 1003.912\ H^{-0.243}$ for 1957-1984	0.52	94.56

Figure 4-2 is a plot of specific speed versus head for eight major low-head turbine manufacturers. The corresponding regression equations, correlation coefficients, standard deviations, and number of units are shown in Table 4-2. The results showed a spread of no correlation to good correlation between specific speed and head, depending on the manufacturer. It appears that the specific speed versus design head relation is not used by all turbine manufacturers in the current state-of-the-art of low-head turbine design and production. It is therefore not surprising that specific speed showed a weak correlation with design head in Figure 4-1 which was plotted using data supplied by about thirty turbine manufacturers. The period 1961 to 1970 plot of specific speed versus design head showed a departure from the form of the plot for the other periods. The occurrence is not readily explained but it might be due to lack of standardization in the low-head turbine industry and research as compared with the degree of standardization achieved in the production of medium and high head turbines, namely: Francis, Pelton and large scale propeller turbines. The results showed that the cluster analysis method was effective in identifying turbines with similar characteristics. Specific speed showed a weak correlation with cavitation coefficient and apparently no correlation with turbine efficiency, diameter and static draft head. All regression equations involving the specific speed and other turbine parameters are grouped together on pages 53 through 57 in Tables 4-1 through 4-5.

Specific Speed and Dimensionless Specific Speed Relations

The specific speed correlated well with the dimensionless specific speed. Using the expression for dimensionless specific speed, a relation between specific speed and discharge, rotational speed and head has been developed. Regression equations relating specific speed to dimensionless specific speed and turbine speed, discharge and head for bulb and tubular turbines are given below.

Regression Equation for Bulb Turbines

Regression equation relating specific speed to the dimensionless specific speed for bulb turbines, the correlation coefficient and standard deviation are:

		r	s
$N_s = 157.453\ (\omega_s)^{1.002}$	for 1953-1984	0.97	35.50

The regression relation that results for specific speed is given by

$$N_s = \frac{2.945\ N\ Q^{0.50}}{H^{0.75}}$$

Regression Equations for Tubular Turbines

Regression equation relating specific speed to the dimensionless specific speed for tubular turbines, the corresponding correlation coefficient and standard deviation are:

		r	s
$N_s = 169.687\ \omega_s^{0.937}$	for 1957-1984	0.98	15.81

The derived relation for specific speed is given by

$$N_s = 4.123\ \frac{N^{0.937}\ Q^{0.468}}{H^{0.702}}$$

4.3 Turbine Unit Constant Terms

Unit power, unit discharge and unit speed are referred to in this thesis as unit constant terms. Hydraulic turbines are normally manufactured using the results of model tests which are often expressed in terms of unit power, unit discharge and unit speed. These unit constant terms are therefore closely associated with prototype performance. Relations between specific speed and the unit constant terms are normally used in hydraulic turbine design instead of relations between specific speed and rated discharge, power and speed. Regression equations relating specific speed to these unit constant terms and between specific speed and other turbine characteristics shown in Tables 4-1 and 4-4 and in the various figures in this chapter are therefore desired in planning and feasibility studies of hydropower projects. These relations have been derived from functional equations of the form:

$$N_s = f(P_{11}) \tag{4.2}$$

$$N_s = f(Q_{11}) \tag{4.3}$$

$$N_s = f(N_{11}) \tag{4.4}$$

Multi-parameter relations among the unit constant terms and between the unit constant terms and other turbine characteristics, sometimes called Hill curves, have been used for selection of turbines for hydroelectric projects. However, Hill curves were not developed in this study because they are usually proprietary information of the turbine manufacturers who should do the final turbine selection for a hydropower project. Therefore, only two-parameter relations suitable for planning and feasibility studies have been developed. Correlation analyses between cavitation coefficient, turbine diameter, efficiency, static draft head and unit constant terms have also been made. The results ranged from good to weak correlation between one of the unit turbine constant terms and specific speed. There exists a very weak relation between the cavitation coefficient and the unit constant terms. No apparent correlations exist between efficiency, diameter and static draft head and the unit constant terms.

Generally, based on correlation coefficient criteria, the results showed that the current low-head hydraulic turbine design and manufacture appears to be based on curves relating unit constant terms to one another and between the unit constant terms and other turbine parameters. Some of the regression equations involving the unit constant terms are shown below:

<u>Regression Equations for Bulb Turbines</u>

The regression equations relating specific speed to unit power, unit discharge and unit speed and relating the unit constant terms to one another for bulb turbines and the corresponding correlation coefficients and standard deviations are given below:

		r	s
$N_s = 59.065\ P_{11}^{0.850}$	for 1953-1984	0.88	63.60
$N_s = 386.656\ Q_{11}^{0.780}$	for 1953-1965	0.75	83.21
$N_s = 391.251\ Q_{11}^{0.823}$	for 1966-1984	0.81	68.57
$N_s = 0.233\ N_{11}^{1.568}$	for 1953-1965	0.86	75.76
$N_s = 4.568 \times 10^{-2}\ N_{11}^{1.897}$	for 1966-1984	0.87	61.03
$P_{11} = 9.051\ Q_{11}^{0.937}$	for 1953-1965	0.95	1.02
$P_{11} = 9.454\ Q_{11}^{0.947}$	for 1966-1984	0.84	2.17
$N_{11} = 59.065\ P_{11}^{0.350}$	for 1953-1984	0.56	13.56
$N_{11} = 126.765\ Q_{11}^{0.349}$	for 1953-1984	0.54	12.91

Regression Equations for Tubular Turbines

Regression equations relating specific speed to unit power, unit discharge and unit speed and relating unit power to unit discharge for Tubular turbines and the corresponding correlation coefficients and standard deviations are:

		r	s
$N_s = 63.876\ P_{11}^{0.824}$	for 1957-1984	0.62	56.66
$N_s = 418.593\ Q_{11}^{0.663}$	for 1957-1984	0.46	65.19
$N_s = 0.810\ N_{11}^{1.309}$	for 1957-1984	0.82	43.01
$P_{11} = 9.601\ Q_{11}^{0.806}$	for 1957-1984	0.90	0.68

4.4 Diameter of Turbine Runner

The diameter of a turbine runner is one of the essential turbine parameters used for planning, feasibility studies and design of low-head hydro projects. It is normal for the overall dimensions of a turbine unit and some standard dimensions of the civil works to be expressed in terms of the runner diameter. The cost of turbine unit and accessories, transportations costs and cost of related civil works are normally related to the turbine and generator size. In the case of bulb turbine units, the bulb diameter and length required to accommodate the generator, the net area of the annular space accommodating the bulb, overall diameter of the water passageway, permissible water velocity in the annular space and the overall draft tube length are examples of the standard dimensions which can be determined from the knowledge of turbine runner diameter, (Sutherland, 1968). It is therefore desirable to correlate the diameter of turbine runner with other turbine parameters. In order to relate the diameter to the unit constant terms, Equations 2.16, 2.18 and 2.20 have been used. Equation 2.20 can be rearranged in the form:

$$D^2 = P/P_{11}\ H^{1.50}$$

$$D = f\ (P/H) \tag{4.5}$$

Similarly using equations 2.16 and 2.18 the turbine diameter can be expressed as some function of discharge and rotational speed as shown below.

$$D^2 = Q/Q_{11}\ H^{0.50} \tag{4.6}$$

$$D = N_{11}\ H^{0.50}/N \tag{4.7}$$

Multiplying equation 4.14 by equation 4.15 gives:

$$D^3 = Q\ N_{11}\ H^{0.50}\ /\ N\ Q_{11}\ H^{0.50}$$

$$D = (Q\ N_{11}/N\ Q_{11})^{1/3}$$

$$D = f\ (Q/N) \tag{4.8}$$

Regression equation for diameter versus power-head ratio and diameter versus discharged - speed ratio (equations 4.5 and 4.8) yielded very good correlation coefficient values. The resulting regression equations are given below:

<u>Regression Equations for Bulb Turbines</u>

Regression equations relating diameter to power-head ratio and discharge-speed ratio for bulb turbines and the corresponding correlation coefficients and standard deviations are:

		r	s
$D = 0.211\ (P/H)^{0.436}$	for 1953-1965	0.92	0.69
$D = 0.193\ (P/H)^{0.439}$	for 1966-1984	0.97	0.48
$D = 4.186\ (Q/N)^{0.309}$	for 1953-1984	0.99	0.70

<u>Regression Equations for Tubular Turbines</u>

Regression equations relating diameter to power-head ratio and discharge-speed ratio and the corresponding correlation coefficients and standard deviations are:

		r	s
$D = 0.190\ (P/H)^{0.456}$	for 1957-1984	0.96	0.51
$D = 4.488\ (Q/N)^{0.331}$	for 1957-1984	0.96	0.17

The author has not come across the above derivations, (equations 4-5 through 4.8) yielding parametric expressions for the diameter in terms of either the rated power and head or discharge and turbine speed of rotation anywhere in the literature. The resulting regression equations yielded very good results. It is believed that the above relation will yield better estimates of the diameter than the procedures which select the diameter from relations involving the specific speed and the rotational speed determined from experience curves which already contain some inherent error.

4.5 Efficiency of Low-Head Hydraulic Turbines

Figures 4-30 through 4-32, grouped together on pages 89 through 91 are graphs of efficiencies of turbines versus year of commissioning of the turbines produced by major turbine manufacturers. Due to the proprietary nature of such turbine data information the manufacturers are not identified with the curves derived from their data. The graphs show that low-head turbines generally have very high efficiencies which are likely due to the "straight flow-through" advantage in the design of the water passageways for these turbines. In a few cases the computed efficiencies were greater than unity. This might be explained by lack of uniformity in the definition of rated values reported by the different turbine manufacturers. The efficiency was correlated with specific speed, the unit constant terms, head, cavitation, coefficient and diameter. There appeared to be no correlation between the turbine efficiency and the above mentioned turbine constants.

4.6 Cavitation Coefficient

Cavitation coefficient is used to select a safe setting for reaction turbines. It is one of the salient performance parameters and it depends mostly on the design head, site elevation and static draft head (see equation 2.27). The proper selection of the cavitation coefficient should help in obtaining cavitation free performance of the turbine. Relations have been sought between cavitation coefficient and other turbine parameters. The results showed that for bulb turbines, only the specific speed, unit discharge and head are related to the cavitation coefficient. Correlation coefficients between 0.51 and 0.74 have been obtained between cavitation coefficient and head and unit discharge in the case of tubular turbines. The resulting regression equations are as follows:

Regression Equations for Bulb Turbines

Regression equations relating cavitation coefficient to specific speed, head and unit discharge for bulb turbines and the corresponding correlation coefficients and standard deviation are:

		r	s
$\sigma = 76.23 \times 10^{-6} N_s^{1.485}$	for 1953-1984	0.53	0.64
$\sigma = 8.256 H^{-0.797}$	for 1953-1984	0.73	0.56
$\sigma = 0.421 Q_{11}^{2.050}$	for 1953-1965	0.57	1.00
$\sigma = 0.486 Q_{11}^{1.331}$	for 1966-1984	0.51	0.53

Regression Equations for Tubular Turbines

Regression equations relating cavitation coefficient to unit discharge and head for tubular turbines and the corresponding correlation coefficients and standard deviation are:

		r	s
$= 0.288\ Q_{11}^{2.148}$	for 1957 - 1984	0.72	0.22
$= 2.711\ H^{-0.506}$	for 1957 - 1984	0.67	0.27

4.7 Regression Equations of Weakly Related Hydraulic Turbine Parameters

In the preceeding paragraphs the regression equations of correlation among some turbine parameters have been presented. In addition to those relations, correlations among some other parameters were examined. Regression analysis yielding correlation coefficients less than 0.40 were considered as relations that should not be used or at least used with great care. The results of such weakly related parameters are recorded in Tables 4-3 and 4-5, on pages 55 and 57.

4.8 Worked Example of Turbine Selection

A worked example demonstrating how the experience curves produced in this thesis can be used in feasibility studies and planning of hydropower porjects is given in the appendix.

Table 4-1. Regression Equations for Bulb Turbines

Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units Per Sample Period	Type of Turbine
$N_s = 1420.954H^{-0.346}$	0.36	242.82	1953-1960	27	Bulb
$N_s = 967.678H^{-0.159}$	0.22	113.77	1961-1970	61	Bulb
$N_s = 1757.752H^{-0.341}$	0.47	113.40	1971-1984	99	Bulb
$N_s = 59.065P_{11}^{-0.850}$	0.88	63.60	1953-1984	188	Bulb
$N_s = 386.656Q_{11}^{0.780}$	0.75	83.21	1953-1965	48	Bulb
$N_s = 391.251\ Q_{11}^{0.823}$	0.81	68.57	1966-1984	123	Bulb
$N_s = 0.233\ N_{11}^{1.568}$	0.86	75.76	1953-1965	56	Bulb
$N_s = 4.568\times10^{-2}N_{11}^{1.897}$	0.87	61.03	1966-1984	131	Bulb
$N_s = 157.453\ \omega_s^{1.002}$	0.97	35.50	1953-1984	173	Bulb
$P_{11} = 9.051\ Q_{11}^{0.937}$	0.95	1.02	1953-1965	48	Bulb
$P_{11} = 9.454\ Q_{11}^{0.947}$	0.84	2.17	1966-1984	123	Bulb
$\sigma = 7.623\times10^{-5}N_s^{1.485}$	0.53	0.64	1953-1984	61	Bulb
$\sigma = 8.256\ H^{-0.797}$	0.73	0.56	1953-1984	61	Bulb
$\sigma = 0.421\ Q_{11}^{2.050}$	0.51	1.00	1953-1965	12	Bulb
$\sigma = 0.486\ Q_{11}^{1.331}$	0.51	0.53	1966-1984	48	Bulb
$\sigma = 4.761\times10^{-3}P_{11}^{2.056}$	0.55	0.97	1953-1965	12	Bulb
$\sigma = 39.389\times10^{-3}P_{11}^{1.184}$	0.52	0.49	1966-1984	48	Bulb
$D = 0.211(P/H)^{0.436}$	0.92	0.69	1953-1965	56	Bulb
$D = 0.193\ (P/H)^{0.439}$	0.97	0.48	1966-1984	131	Bulb
$D = 4.186\ (Q/N)^{0.309}$	0.99	0.70	1953-1984	172	Bulb
$N_{11} = 59.065\ P_{11}^{0.350}$	0.56	13.56	1953-1984	188	Bulb
$N_{11} = 126.765\ Q_{11}^{0.349}$	0.54	12.91	1953-1984	172	Bulb

Table 4-2. Regression Equation for Bulb and Tubular Turbines by Source

Regression Equation	Correlation Coefficient	Standard Deviation	Data Source Number	Number of Units Per Source	Type of Turbine
$N_s = 1492.139H^{-0.275}$	0.36	152.07	1	12	Bulb
$N_s = 1625.000H^{-0.310}$	0.74	70.77	2	10	Bulb
$N_s = 1752.508H^{-0.335}$	0.96	17.00	3	4	Bulb
$N_s = 1268.540H^{-0.253}$	0.31	120.66	4	16	Bulb
$N_s = 1118.139H^{-0.219}$	0.27	125.74	5	11	Bulb
$N_s = 1653.176H^{-0.323}$	0.98	17.92	6	5	Bulb
$N_s = 1551.682H^{-0.344}$	0.65	95.26	7	9	Bulb
$N_s = 795.523H^{-0.054}$	0.05	103.49	8	22	Bulb
$N_s = 1307.36H^{-0.195}$	0.10	277.50	9	14	Bulb
$N_s = 1142.632H^{-0.187}$	0.26	165.29	10	14	Bulb
$N_s = 1194.049H^{-0.338}$	0.61	68.13	11	11	Tubular
$N_s = 1053.040H^{-0.268}$	0.53	103.57	3	22	Tubular
$\eta = 0.744\ D^{0.233}$	0.57	0.09	1	12	Bulb
$\eta = 0.967\ D^{-0.037}$	0.44	0.02	2	10	Bulb
$\eta = 0.886\ D^{0.0085}$	0.004	0.03	3	13	Bulb
$\eta = 0.853D^{0.056}$	0.25	0.04	4	16	Bulb
$\eta = 0.892D^{0.017}$	0.96	0.002	5	2	Bulb
$\eta = 0.820D^{0.070}$	0.66	0.02	6	3	Bulb
$\eta = 0.859D^{0.068}$	0.21	0.07	7	9	Bulb
$\eta = 0.979D^{-0.053}$	0.02	0.12	8	22	Bulb
$\eta = 0.896D^{-0.017}$	0.01	0.04	9	14	Bulb
$\eta = 0.886D^{0.008}$	0.004	0.03	3	13	Tubular
$\sigma = 4.549\times10^{-6}N_s^{1.908}$	0.58	0.84	1	12	Bulb
$\sigma = 313.332\times10^{-6}N_s^{1.274}$	0.92	0.11	2	10	Bulb
$\sigma = 0.097\times10^{-6}N_s^{2.479}$	0.92	0.15	3	4	Bulb
$\sigma = 111.435\times10^{-6}N_s^{1.423}$	0.47	0.47	4	15	Bulb
$\sigma = 80.774\times10^{-6}N_s^{1.491}$	0.44	1.02	5	11	Bulb
$\sigma = 1541.62\times10^{-6}N_s^{1.015}$	0.84	0.20	6	3	Bulb
$H_s = 1.80 - 2.88$	0.33	4.80	1	12	Bulb
$H_s = -19.58 + 8.92$	0.59	2.77	2	10	Bulb
$H_s = -7.77 + 1.12$	0.39	0.61	3	4	Bulb
$H_s = -3.07 + 1.089$	0.17	1.33	4	15	Bulb
$H_s = -2.21 + 0.404$	0.04	2.06	5	11	Bulb
$H_s = -5.121 + 1.741$	0.45	0.87	6	3	Bulb

Table 4-3. Regression Equations for Weakly Related Bulb Turbine Characteristics

Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units Per Sample Period	Type of Turbine
$\eta = 1.033\ N_s^{-0.024}$	0.01	0.05	1953-1965	48	Bulb
$\eta = 0.507\ N_s^{0.089}$	0.04	0.097	1966-1984	124	Bulb
$\eta = 0.926\ P_{11}^{-0.016}$	0.01	0.05	1953-1965	48	Bulb
$\eta = 0.652\ P_{11}^{0.112}$	0.08	0.08	1966-1984	123	Bulb
$\eta = 0.923\ Q_{11}^{-0.068}$	0.08	0.05	1953-1965	48	Bulb
$\eta = 0.964\ Q_{11}^{-0.053}$	0.02	0.09	1966-1984	123	Bulb
$\eta = 0.857\ H^{-0.067}$	0.02	0.05	1953-1965	48	Bulb
$\eta = 0.948\ H^{-0.014}$	0.003	0.09	1966-1984	124	Bulb
$\eta = 0.911\ D^{-0.031}$	0.06	0.05	1953-1965	48	Bulb
$\eta = 0.935\ D^{-0.010}$	0.002	0.09	1966-1984	123	Bulb
$H_s = -0.296-0.286H$	0.123	3.33	1953-1984	61	Bulb
$D = 0.149P_{11}^{1.034}$	0.29	1.20	1953-1965	56	Bulb
$D = 0.332P_{11}^{0.860}$	0.25	1.35	1966-1984	131	Bulb
$D = 4.337\times10^{-3}N_{11}^{1.278}$	0.15	1.31	1953-1965	56	Bulb
$D = 0.814\times10^{-3}N_{11}^{1.679}$	0.18	1.38	1966-1984	131	Bulb
$D = 1.445Q_{11}^{0.919}$	0.27	1.05	1953-1965	48	Bulb
$D = 1.686Q_{11}^{1.144}$	0.37	1.24	1966-1984	123	Bulb

Table 4-4. Regression equation for Tubular Turbines

Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units Per Sample Period	Type of Turbine
$N_s = 1.003.912H^{-0.243}$	0.52	94.56	1957-1984	37	Tubular
$N_s = 418.593Q_{11}^{0.663}$	0.46	65.19	1957-1984	17	Tubular
$N_s = 169.682W_s^{0.937}$	0.98	15.81	1957-1984	17	Tubular
$N_s = 63.876P_{11}^{0.824}$	0.62	56.66	1957-1984	28	Tubular
$N_s = 0.810N_{11}^{1.309}$	0.82	43.01	1957-1984	28	Tubular
$\sigma = 0.288Q_{11}^{2.148}$	0.72	0.22	1957-1984	16	Tubular
$\sigma = 2.711H^{-0.506}$	0.67	0.27	1957-1984	16	Tubular
$\sigma = 1.571\times10^{-3}P_{11}^{2.353}$	0.72	0.22	1957-1984	16	Tubular
$P_{11} = 9.601\ Q_{11}^{0.806}$	0.90	0.68	1957-1984	17	Tubular
$D = 0.190(P/H)^{0.456}$	0.96	0.51	1957-1984	28	Tubular
$D = 4.488(Q/N)^{0.331}$	0.96	0.17	1957-1984	17	Tubular

Table 4-5. Regression Equations for Weakly Related Tubular Turbine Characteristics

Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units Per Sample Period	Type of Turbine
$\eta = 1.495\ N_s^{-0.0822}$	0.06	0.04	1957-1984	17	Tubular
$\eta = 1.201\ P_{11}^{-0.114}$	0.08	0.04	1957-1984	17	Tubular
$\eta = 0.979\ Q_{11}^{-0.194}$	0.34	0.04	1957-1984	17	Tubular
$\eta = 0.832\ H^{0.027}$	0.10	0.03	1957-1984	17	Tubular
$\eta = 0.832\ D^{0.086}$	0.19	0.04	1957-1984	17	Tubular
$D = 1.413\times10^{-2} N_{11}^{1.008}$	0.05	1.67	1957-1984	28	Tubular
$D = 1.756\ Q_{11}^{0.275}$	0.03	0.53	1957-1984	17	Tubular
$D = 0.190\ P_{11}^{0.916}$	0.08	1.64	1957-1984	28	Tubular
$N_{11} = 63.876\ P_{11}^{0.324}$	0.20	15.05	1957-1984	28	Tubular
$N_{11} = 135.094\ Q_{11}^{0.260}$	0.12	16.37	1957-1984	17	Tubular
$\sigma = 31.100\times10^{-5} N_s^{1.242}$	0.26	0.33	1957-1984	16	Tubular

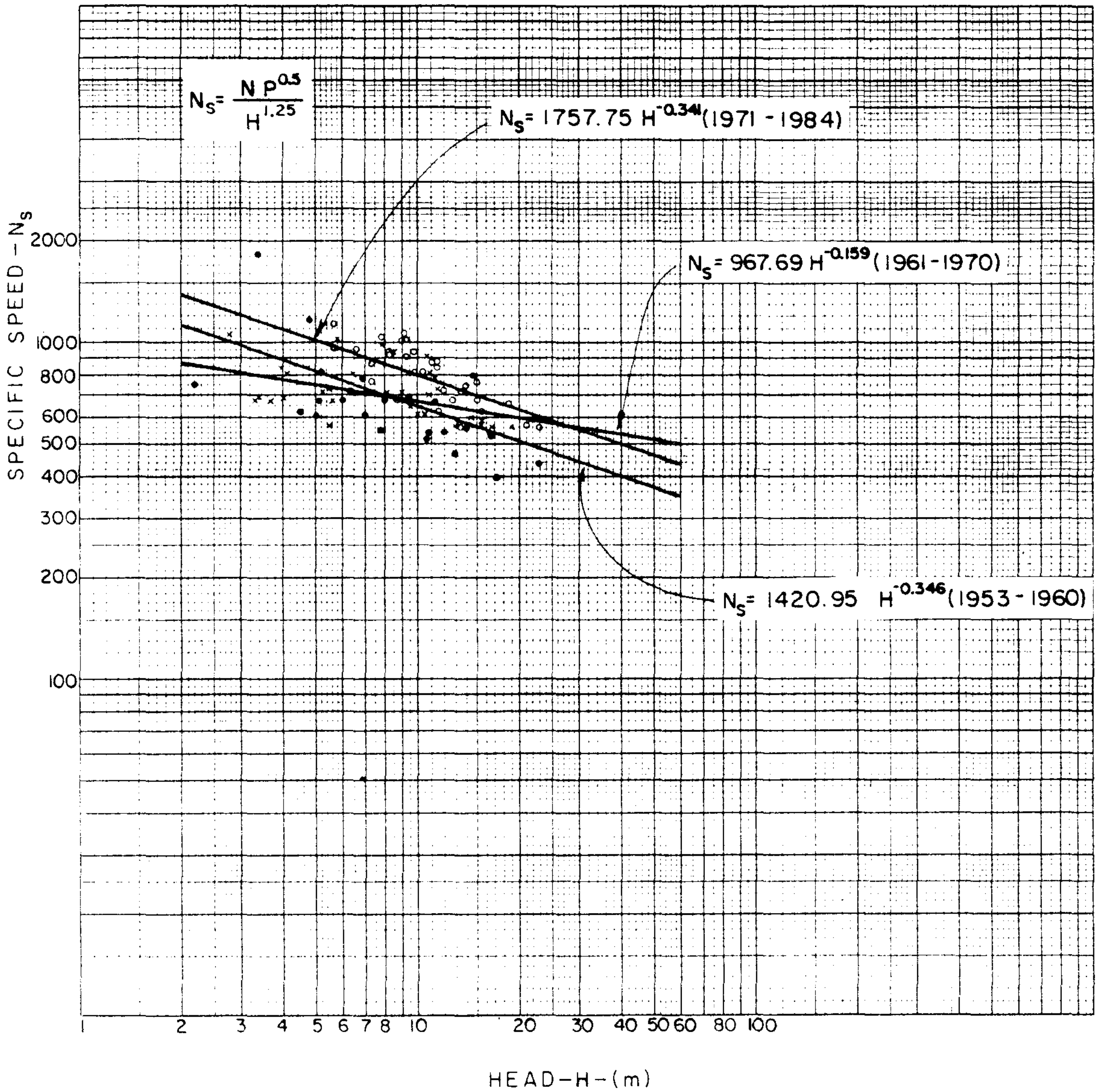


Figure 4-1. Specific speed versus design head for bulb turbines.

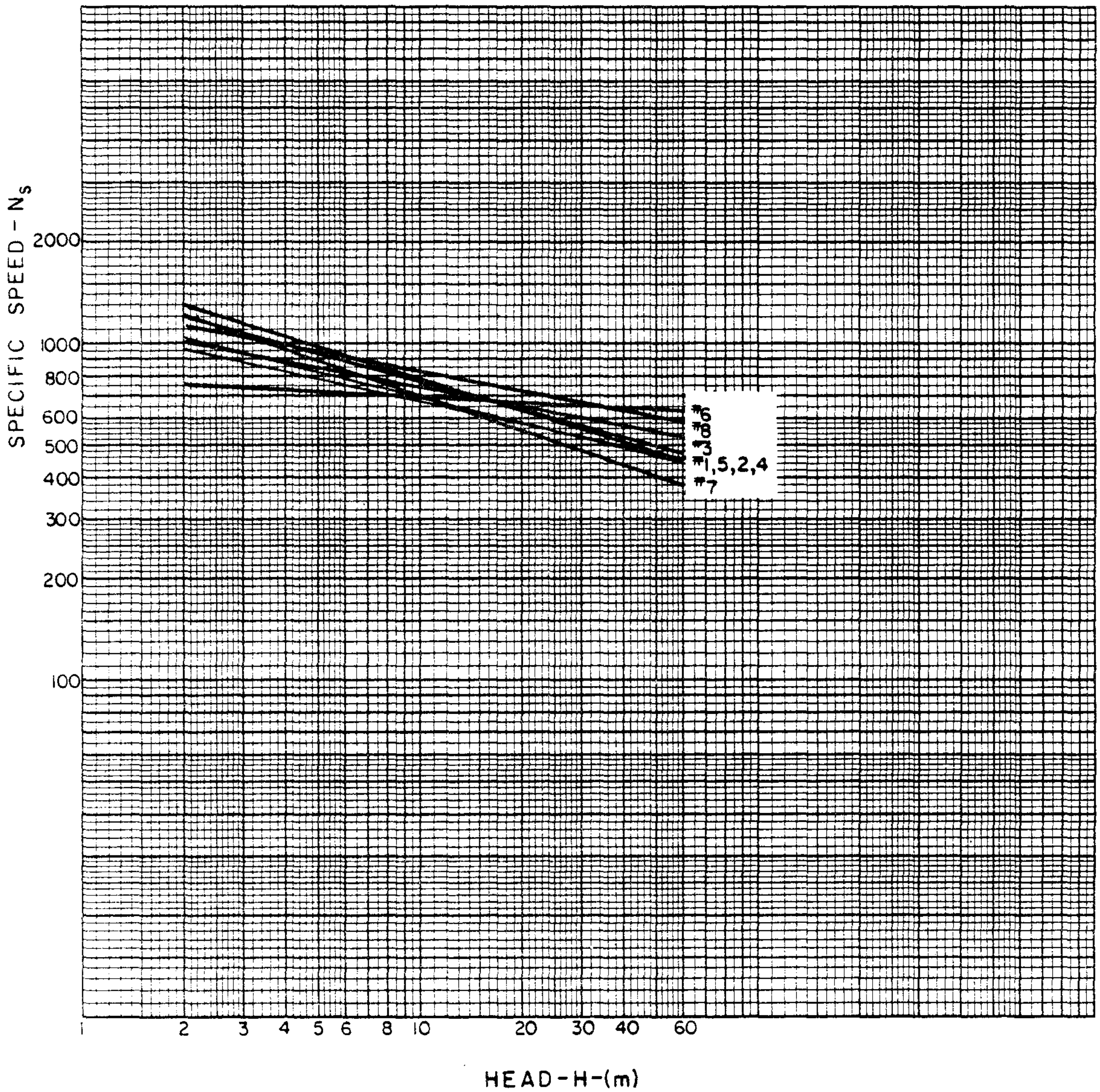


Figure 4-2. Specific speed versus design head for bulb turbines from eight turbine manufacturers.

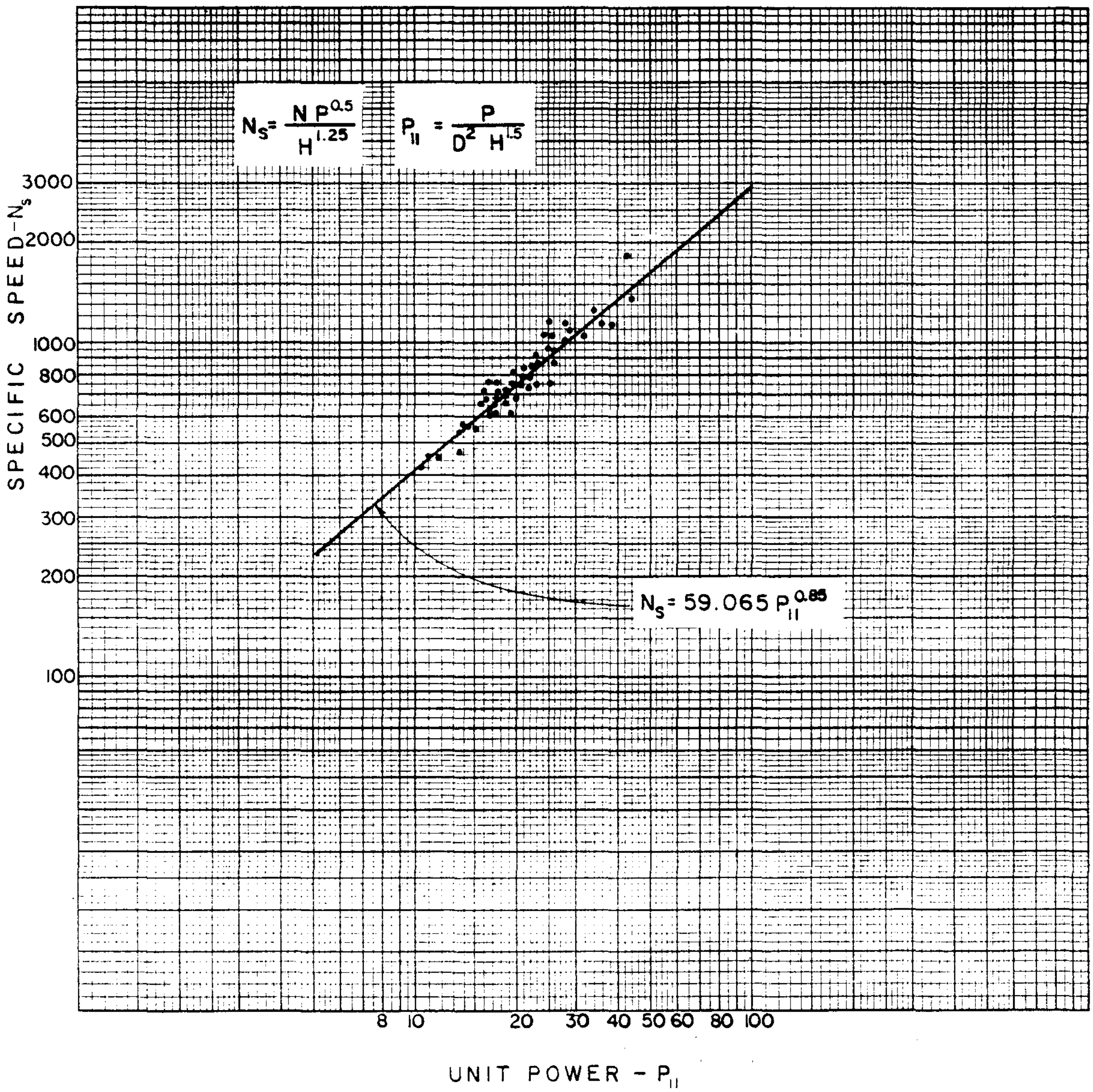


Figure 4-3. Specific speed versus unit power for bulb turbines.

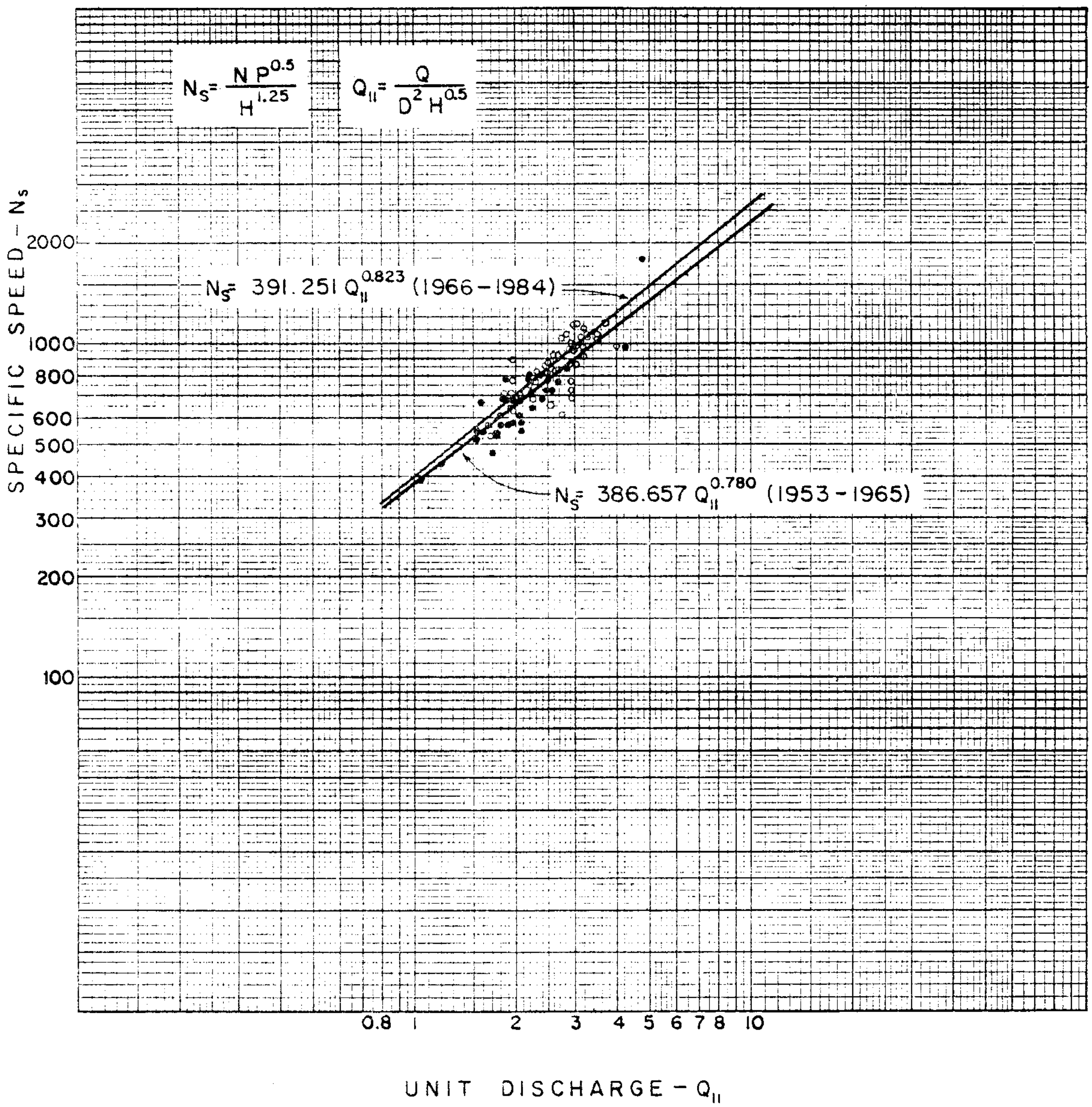


Figure 4-4. Specific speed versus unit discharge for bulb turbines.

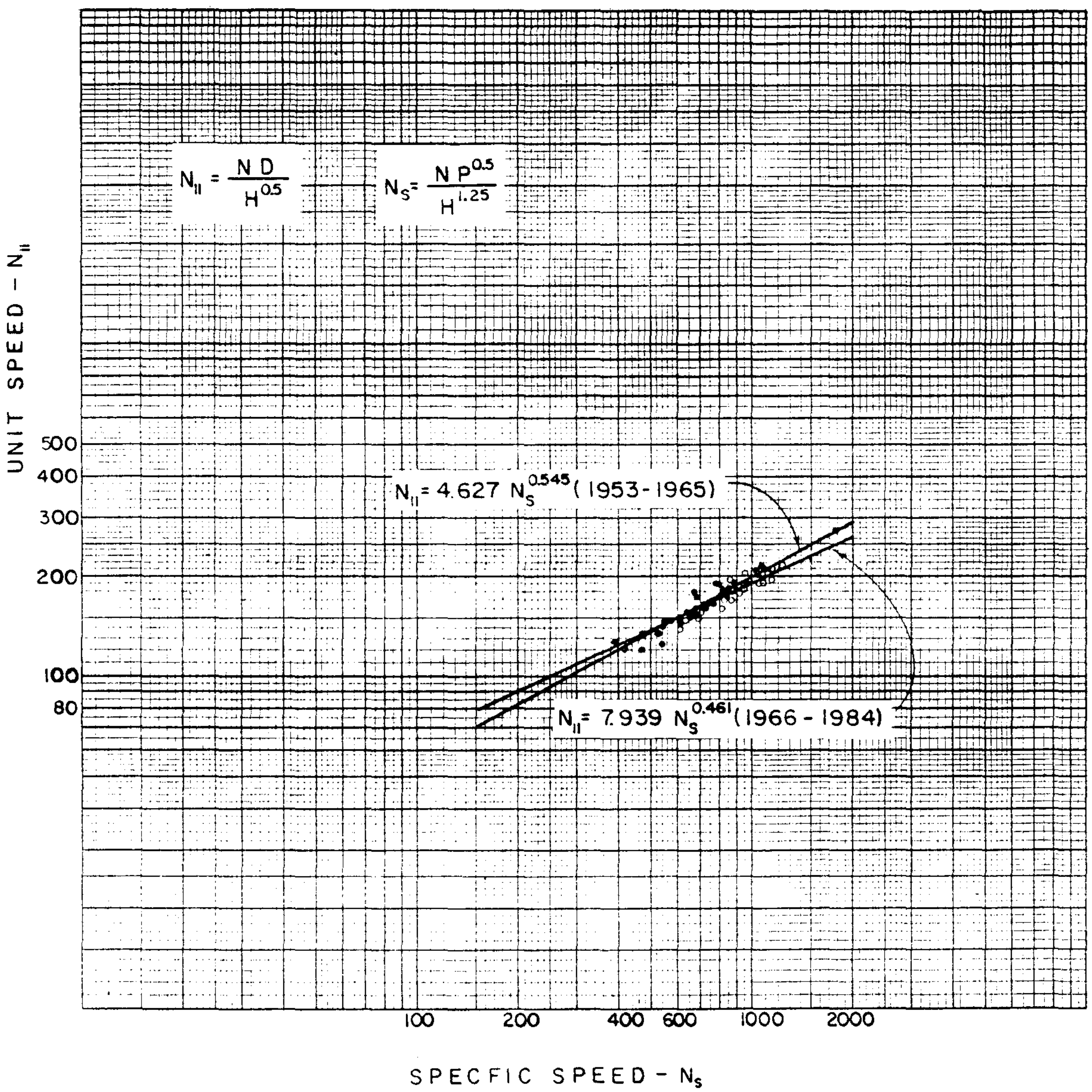


Figure 4-5. Unit speed versus specific speed for bulb turbines.

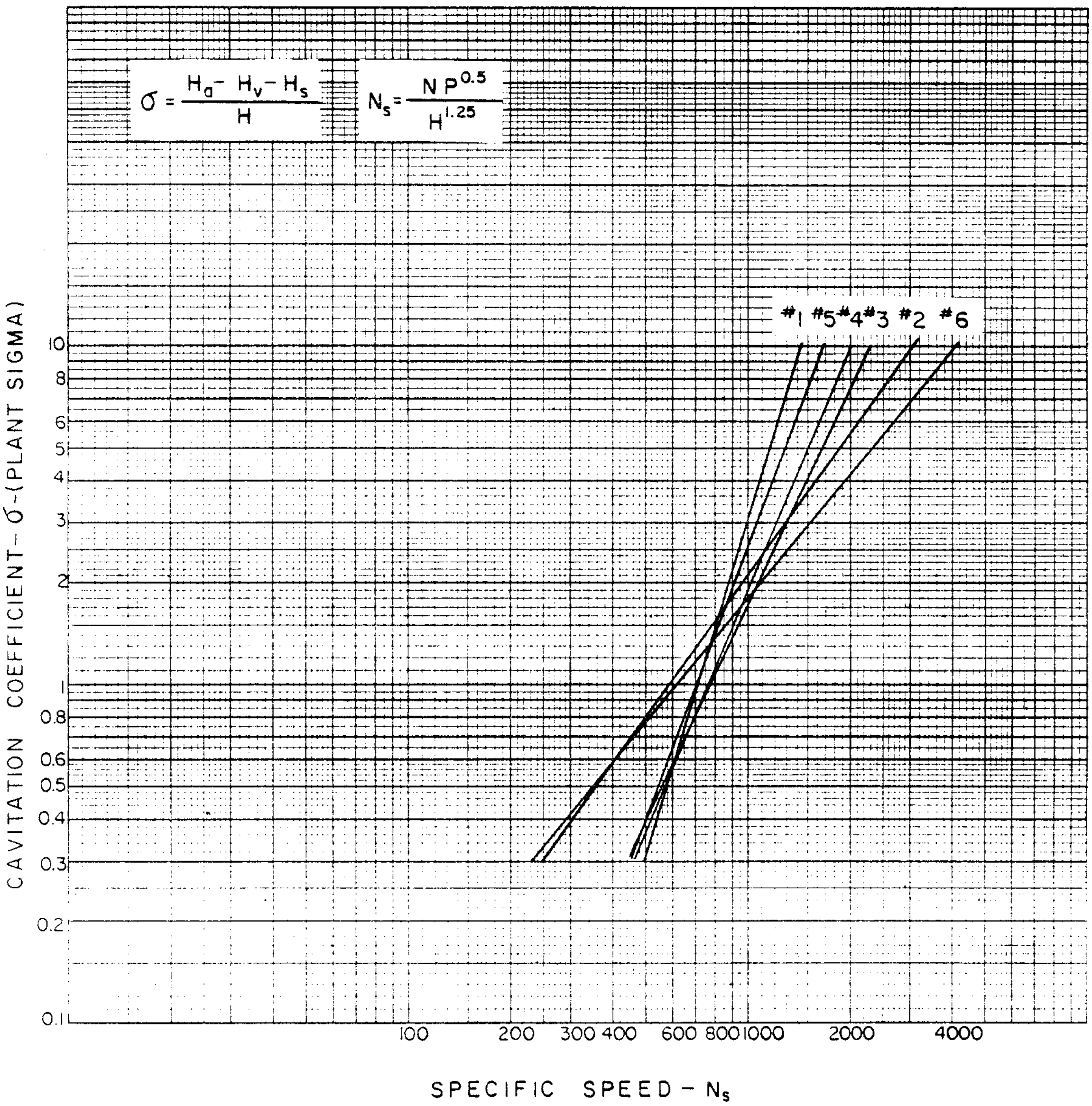


Figure 4-6. Cavitation coefficient versus specific speed for bulb turbines from six different turbine manufacturers.

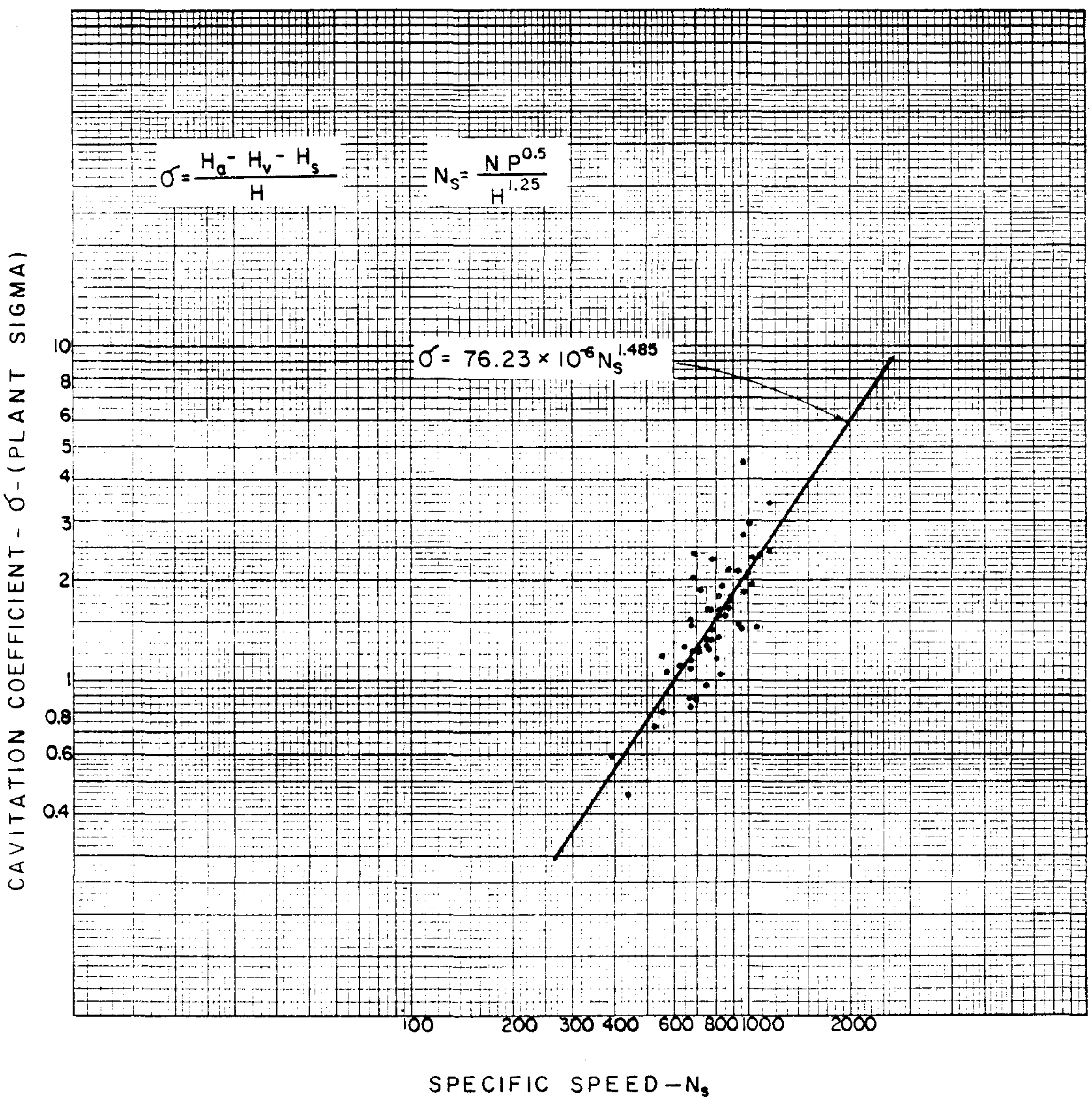


Figure 4-7a. Cavitation coefficient versus specific speed for bulb turbines derived from Figure 4-6.

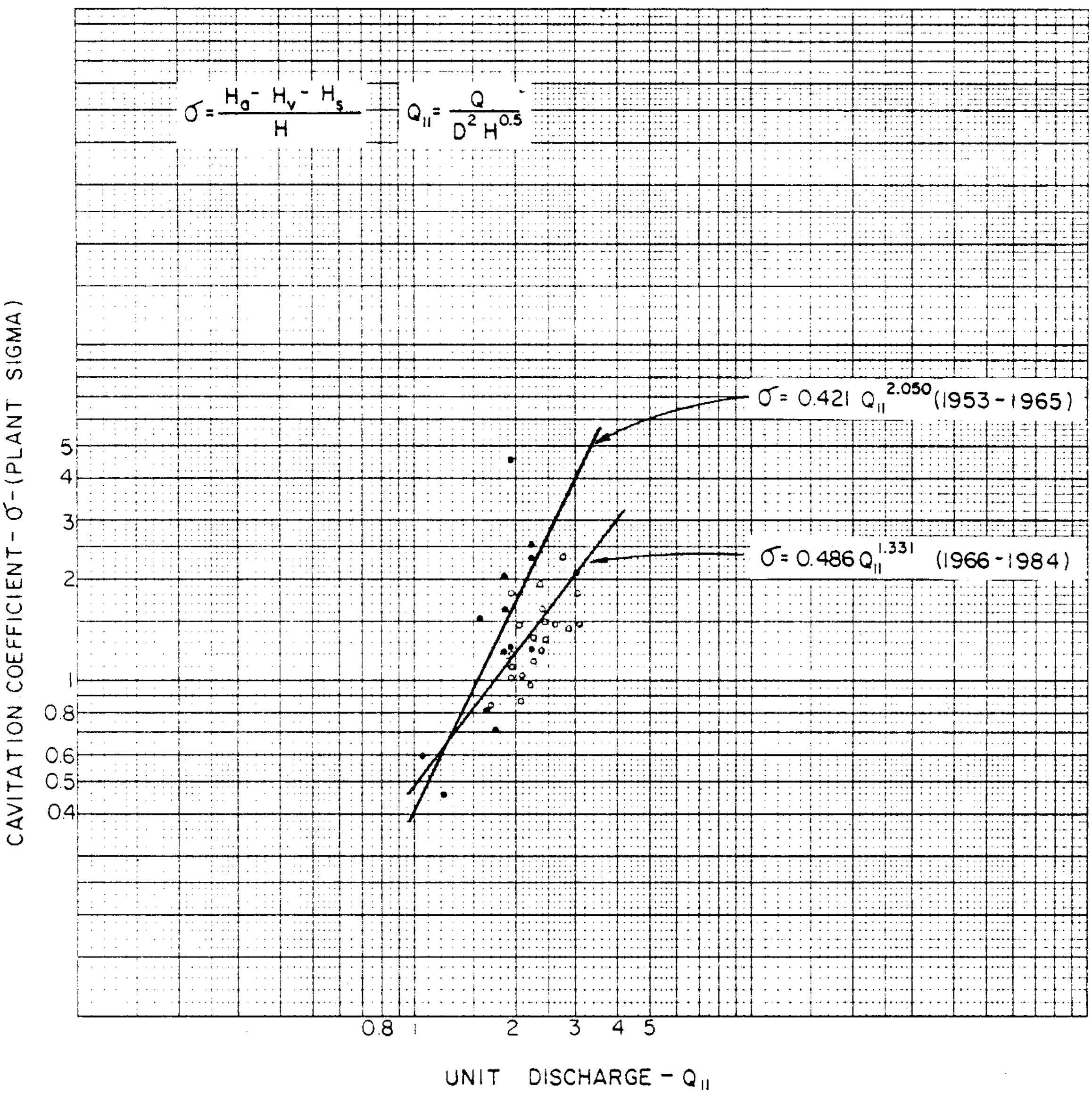


Figure 4-7b. Cavitation coefficient versus unit discharge for bulb turbines.

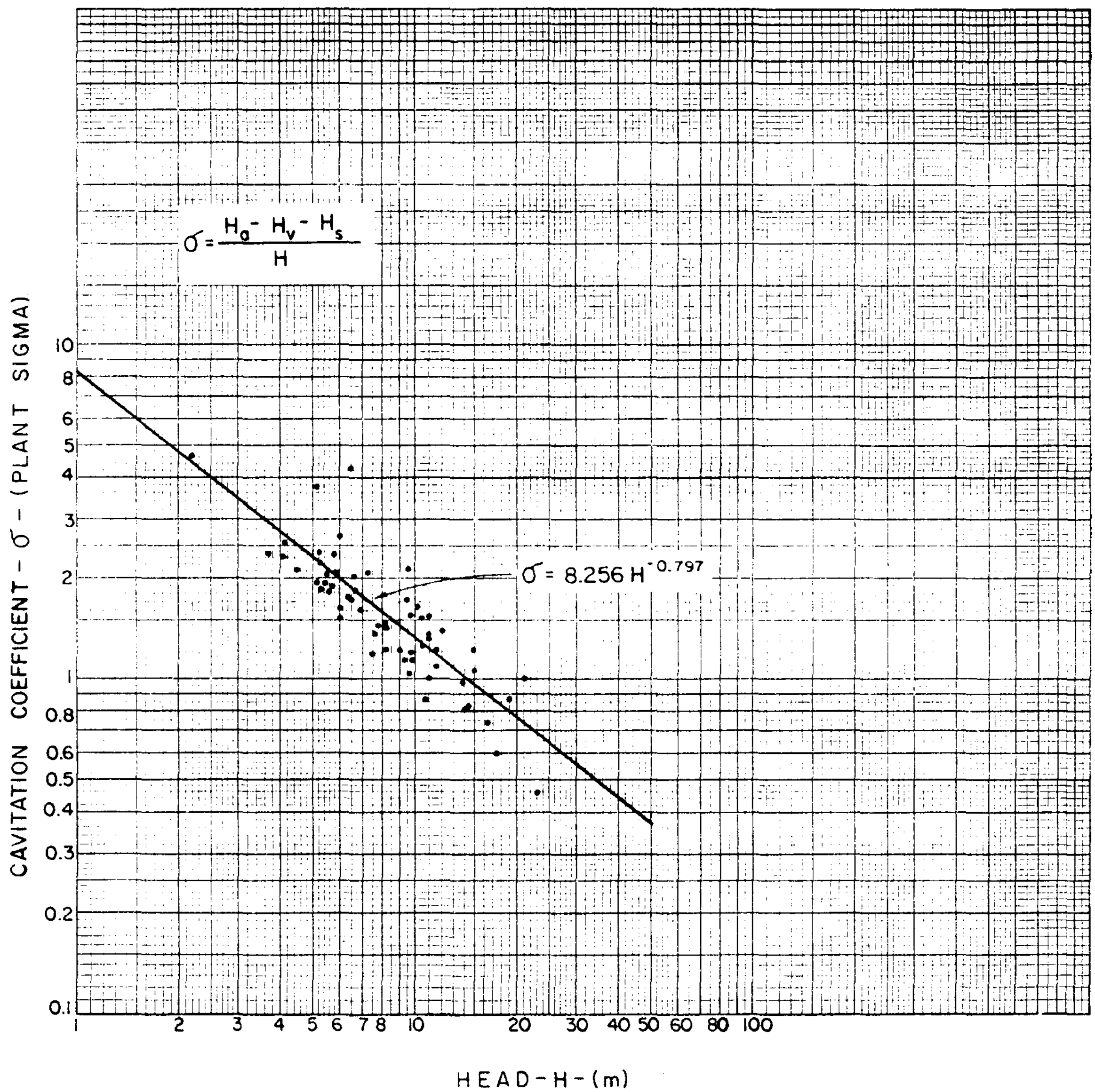


Figure 4-8. Cavitation coefficient versus design head for bulb turbines.

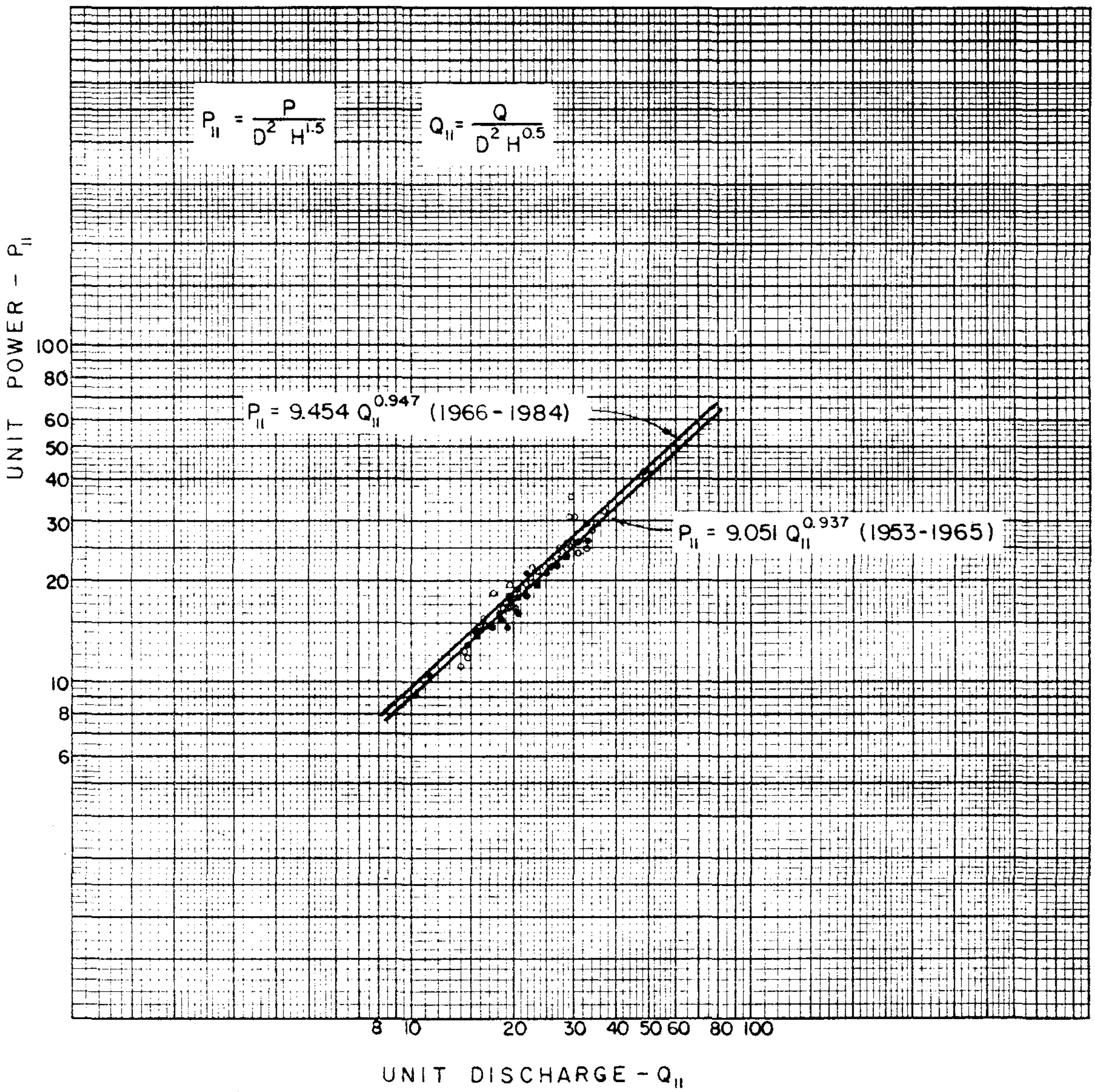


Figure 4-9. Unit power versus unit discharge for bulb turbines.

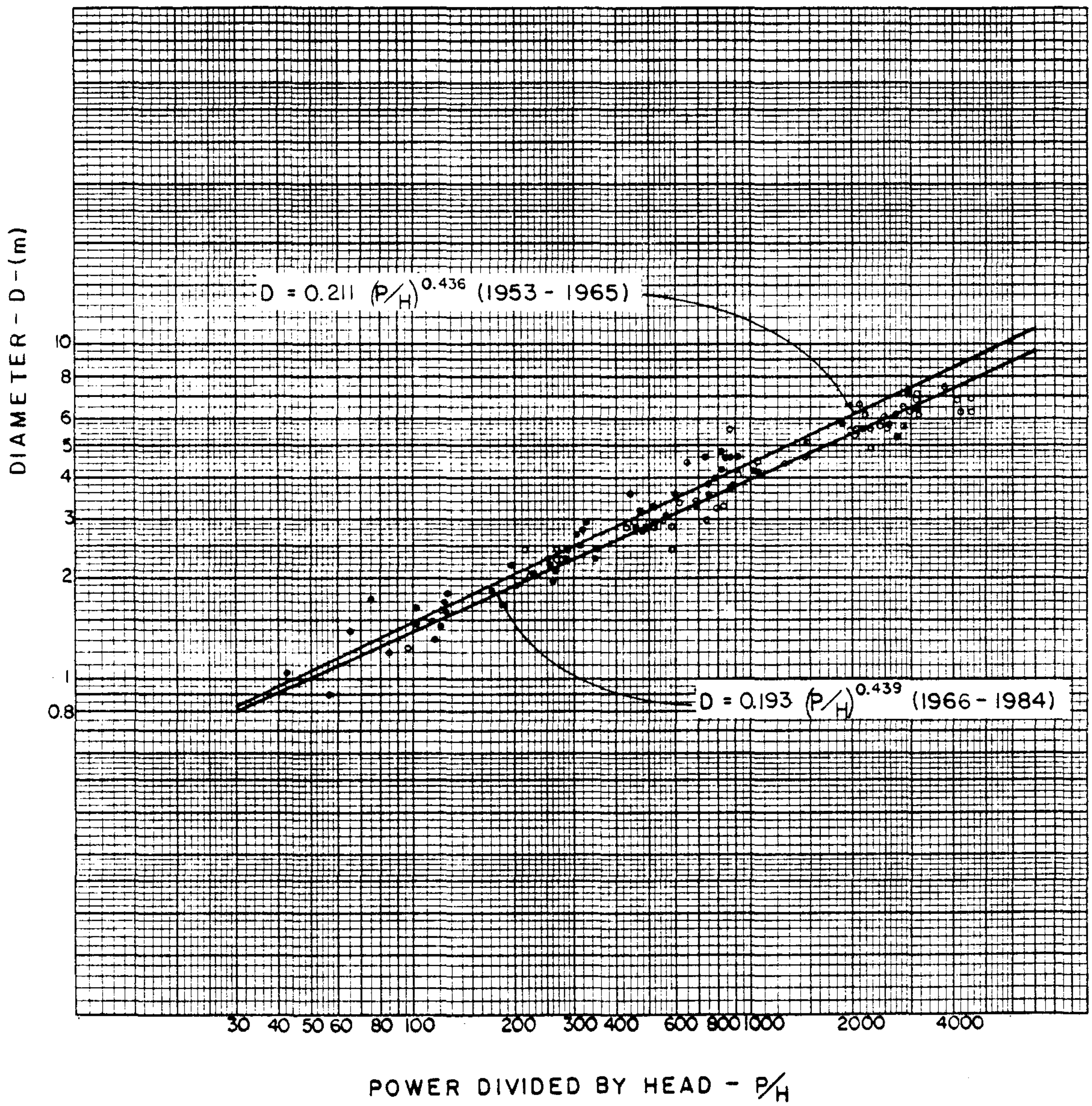


Figure 4-10. Diameter versus ratio of discharge and speed for bulb turbines.

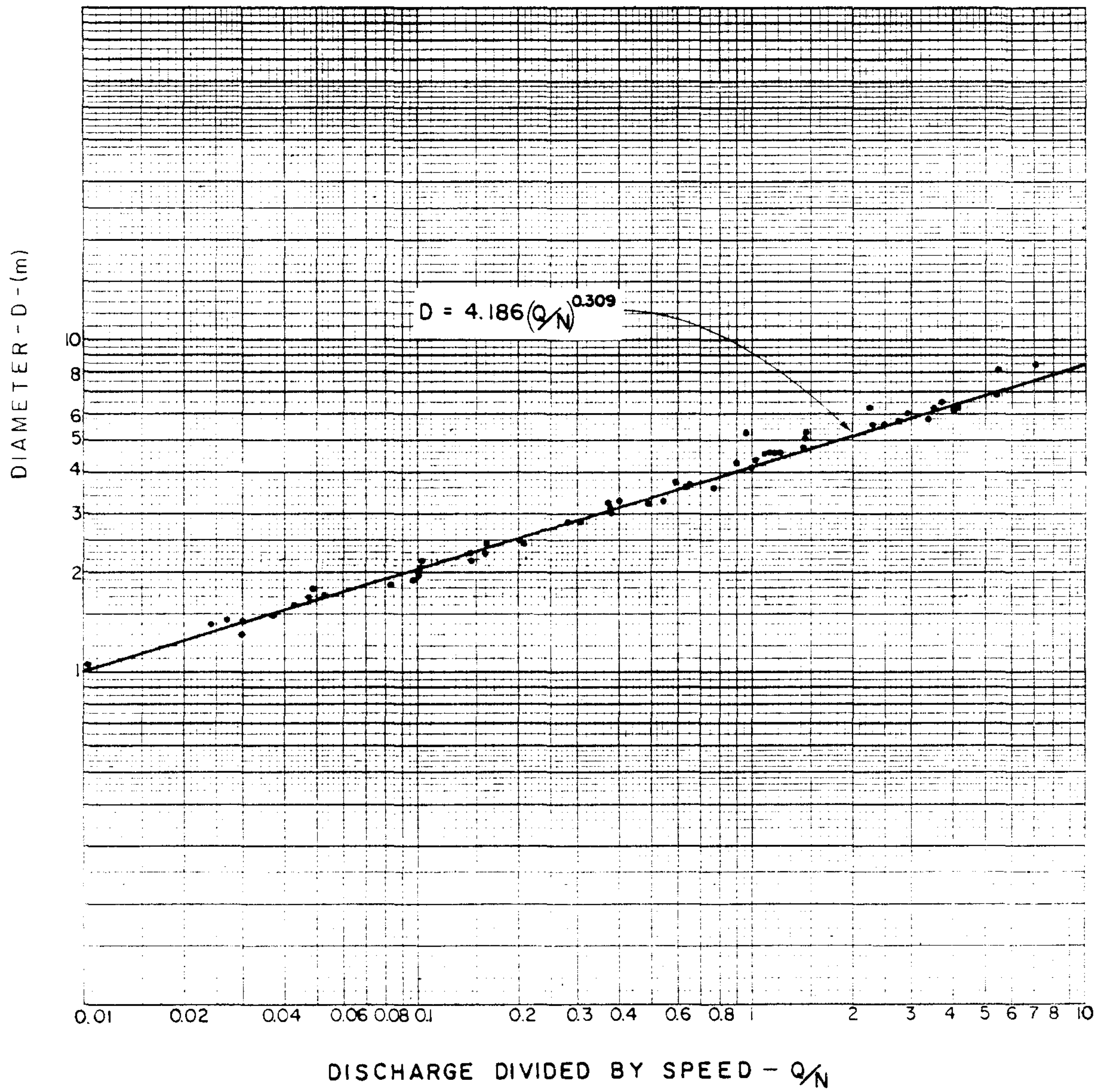


Figure 4-11. Diameter versus ratio of discharge and speed for bulb turbines.

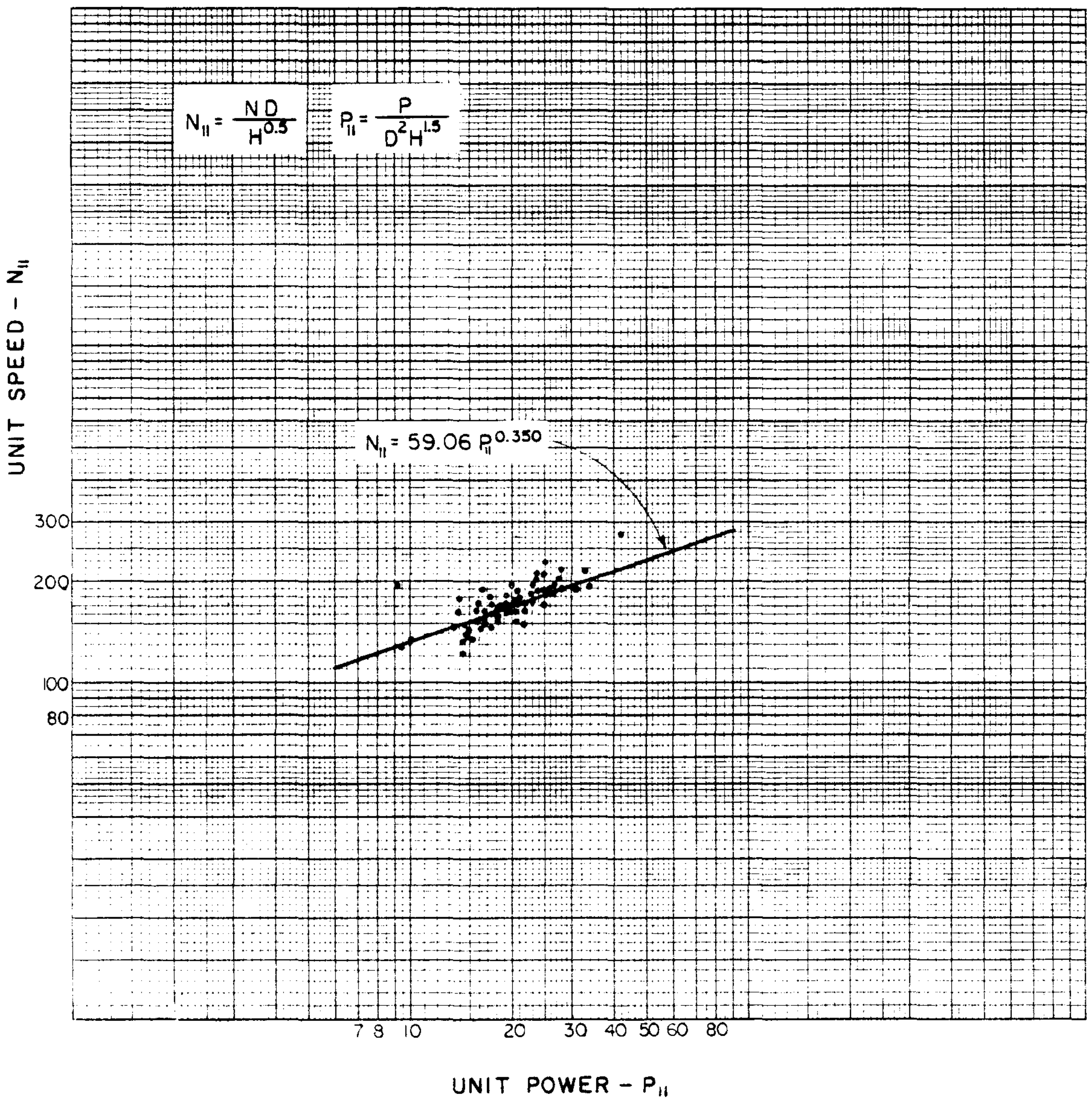


Figure 4-12. Unit speed versus unit power for bulb turbines.

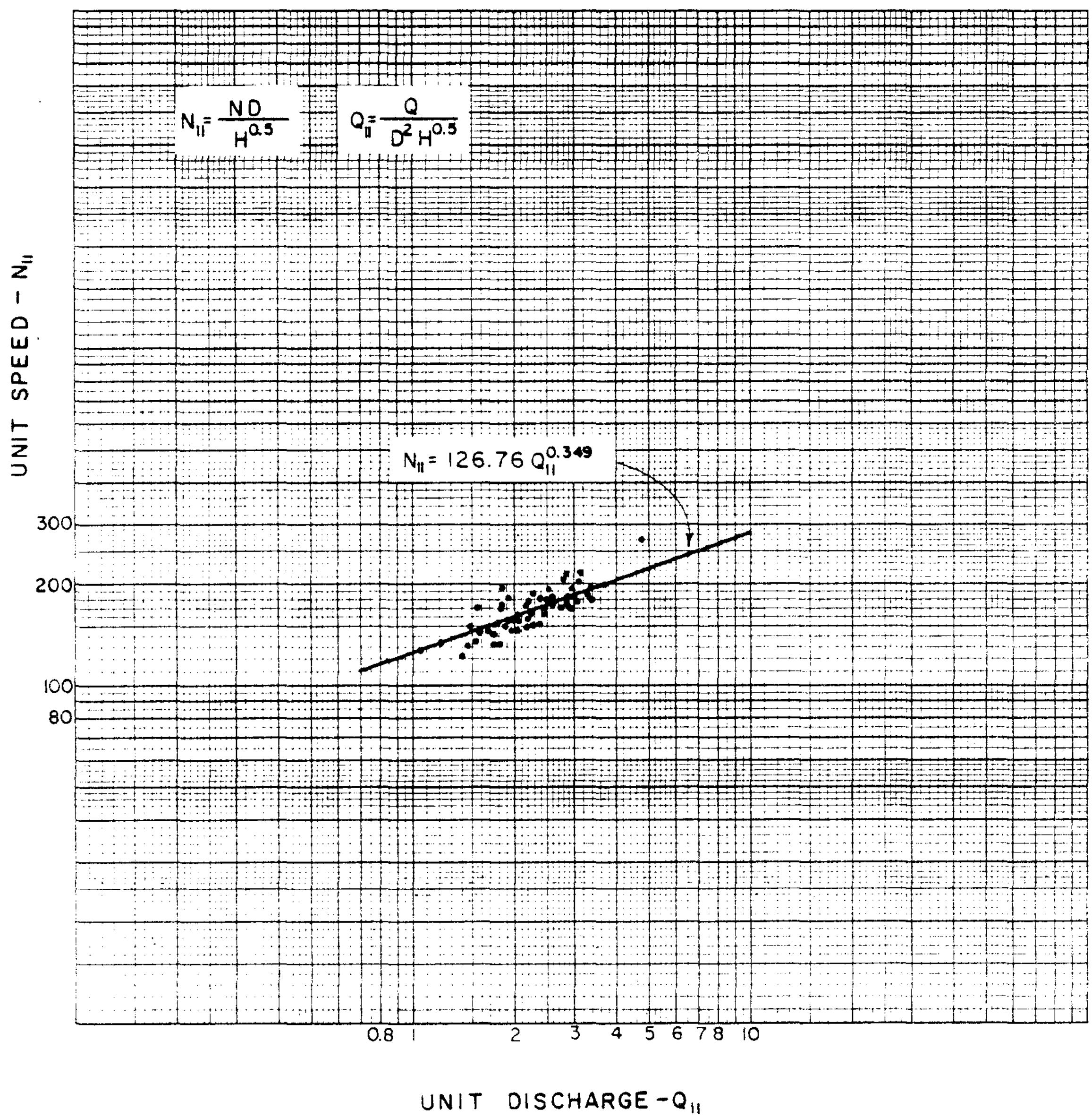


Figure 4-13. Unit speed versus unit discharge for bulb turbines.

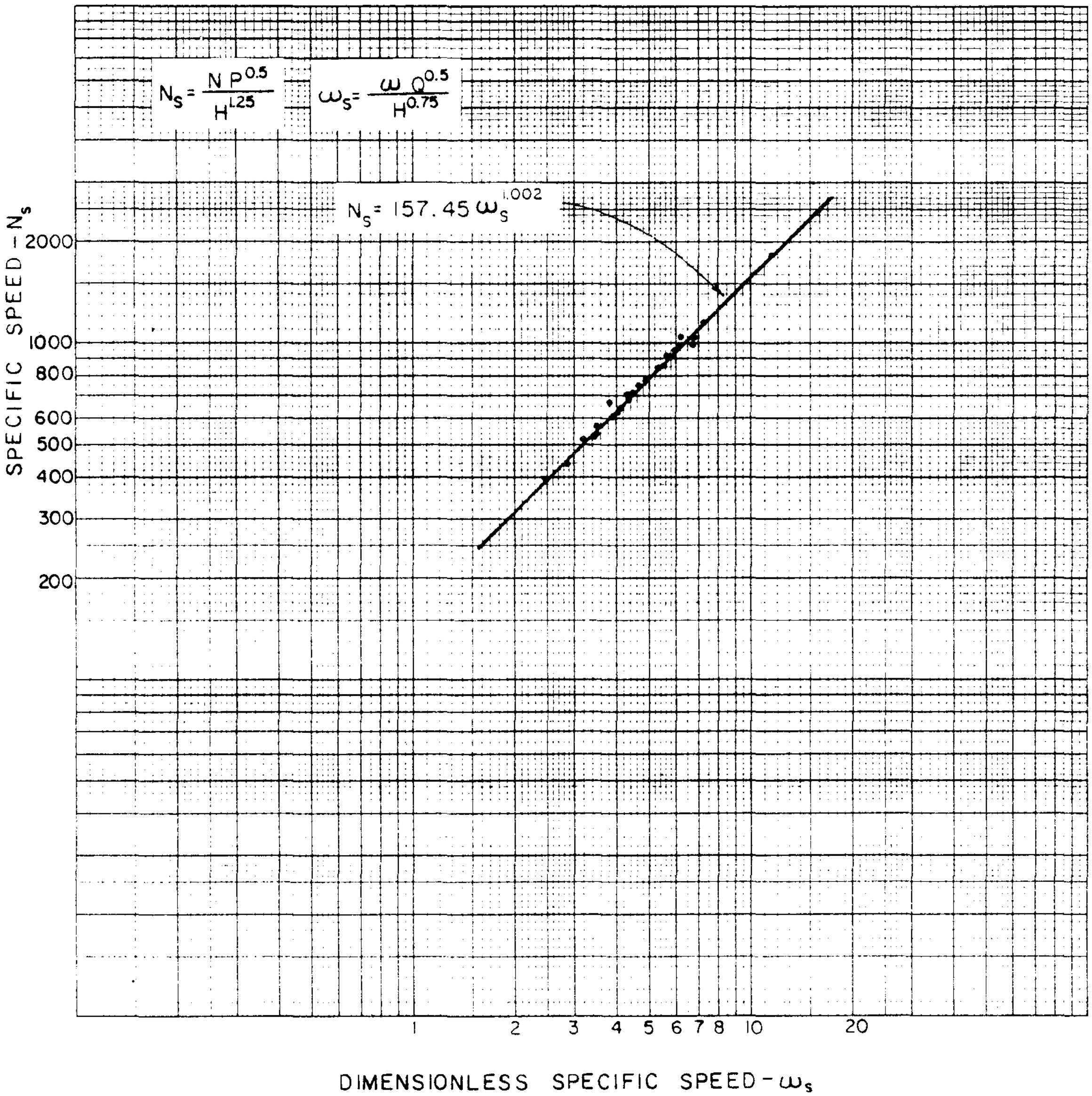


Figure 4-14. Specific speed versus dimensionless specific speed for bulb turbines.

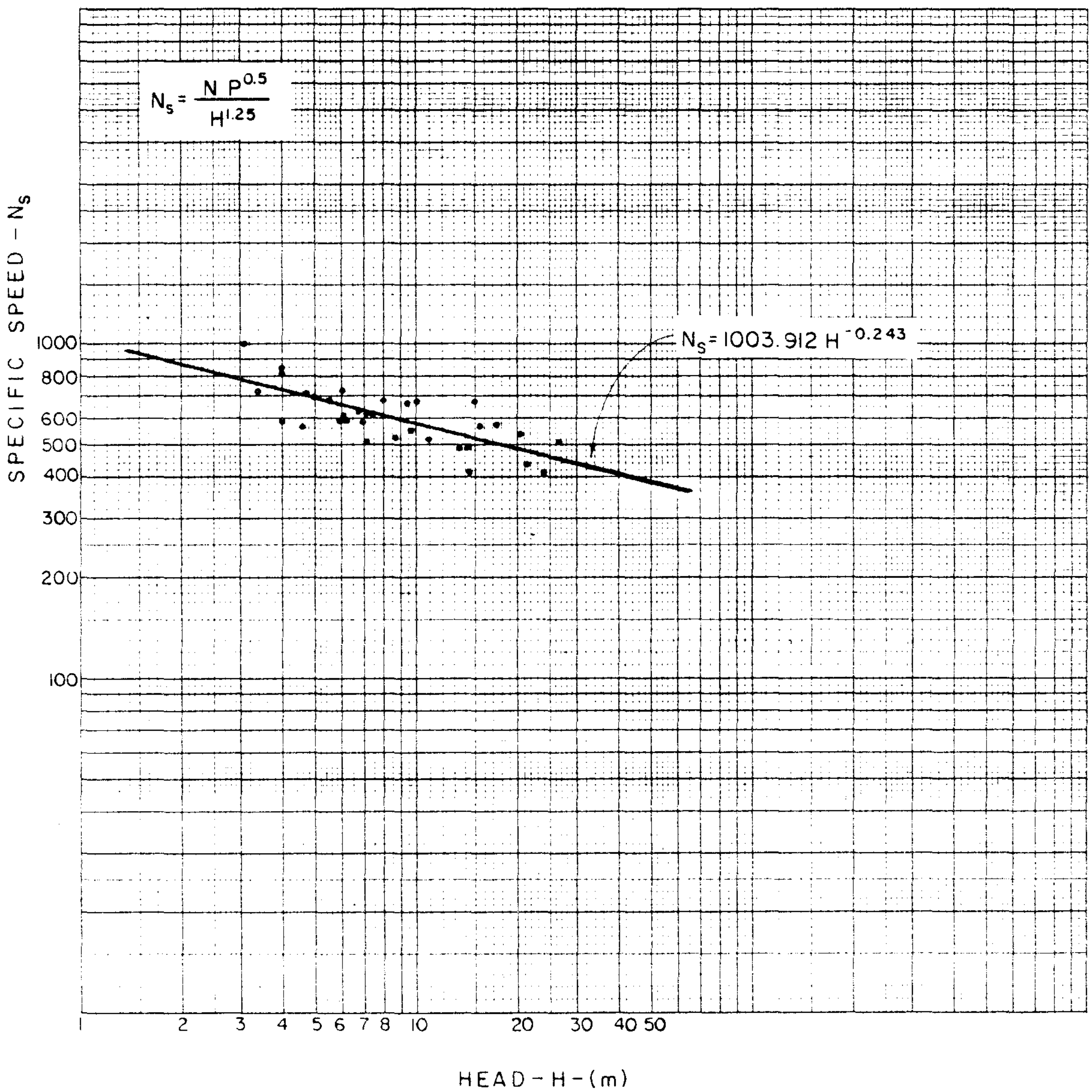


Figure 4-15. Specific speed versus design head for tubular turbines.

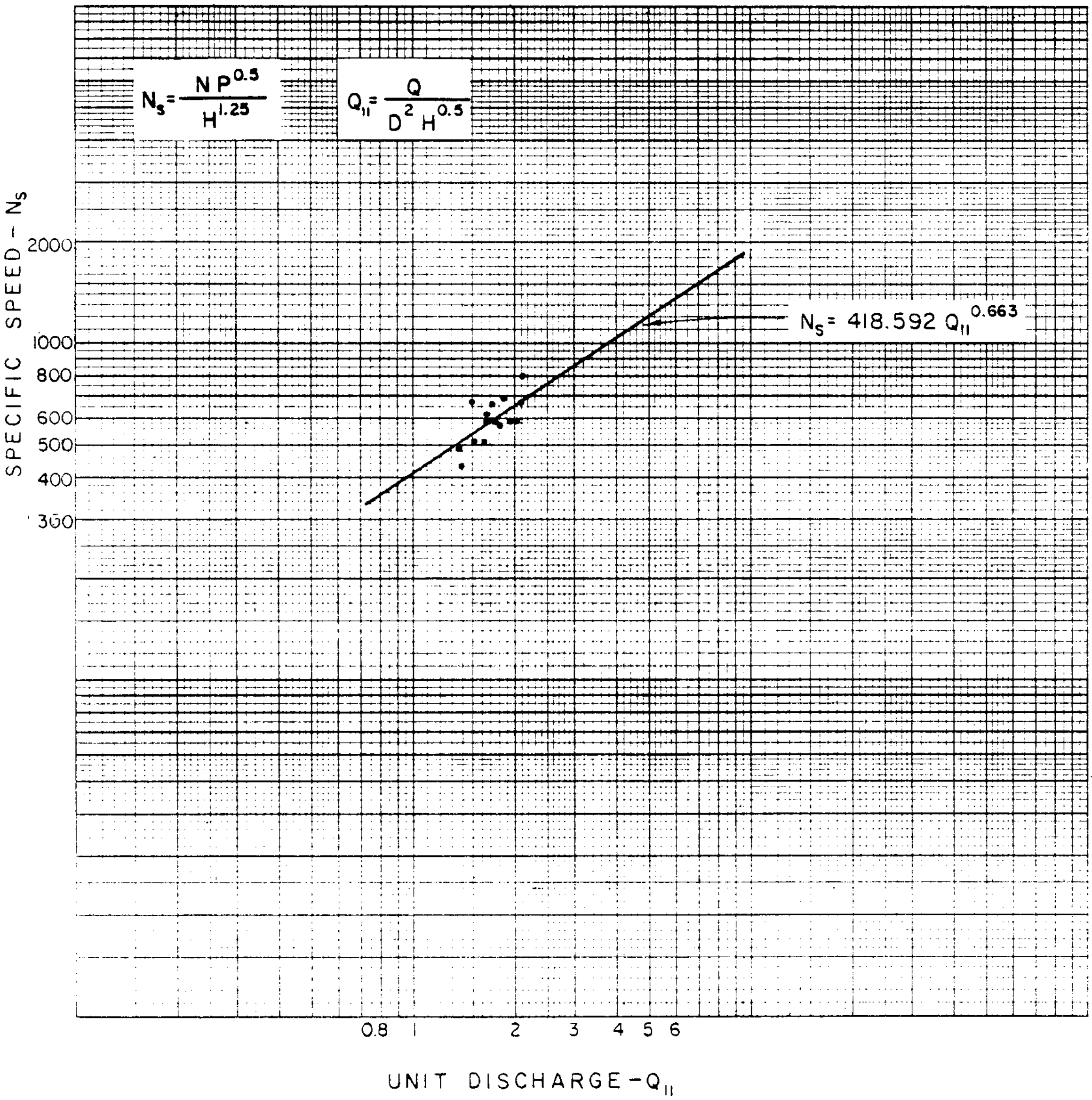


Figure 4-16. Specific speed versus unit discharge for tubular turbines.

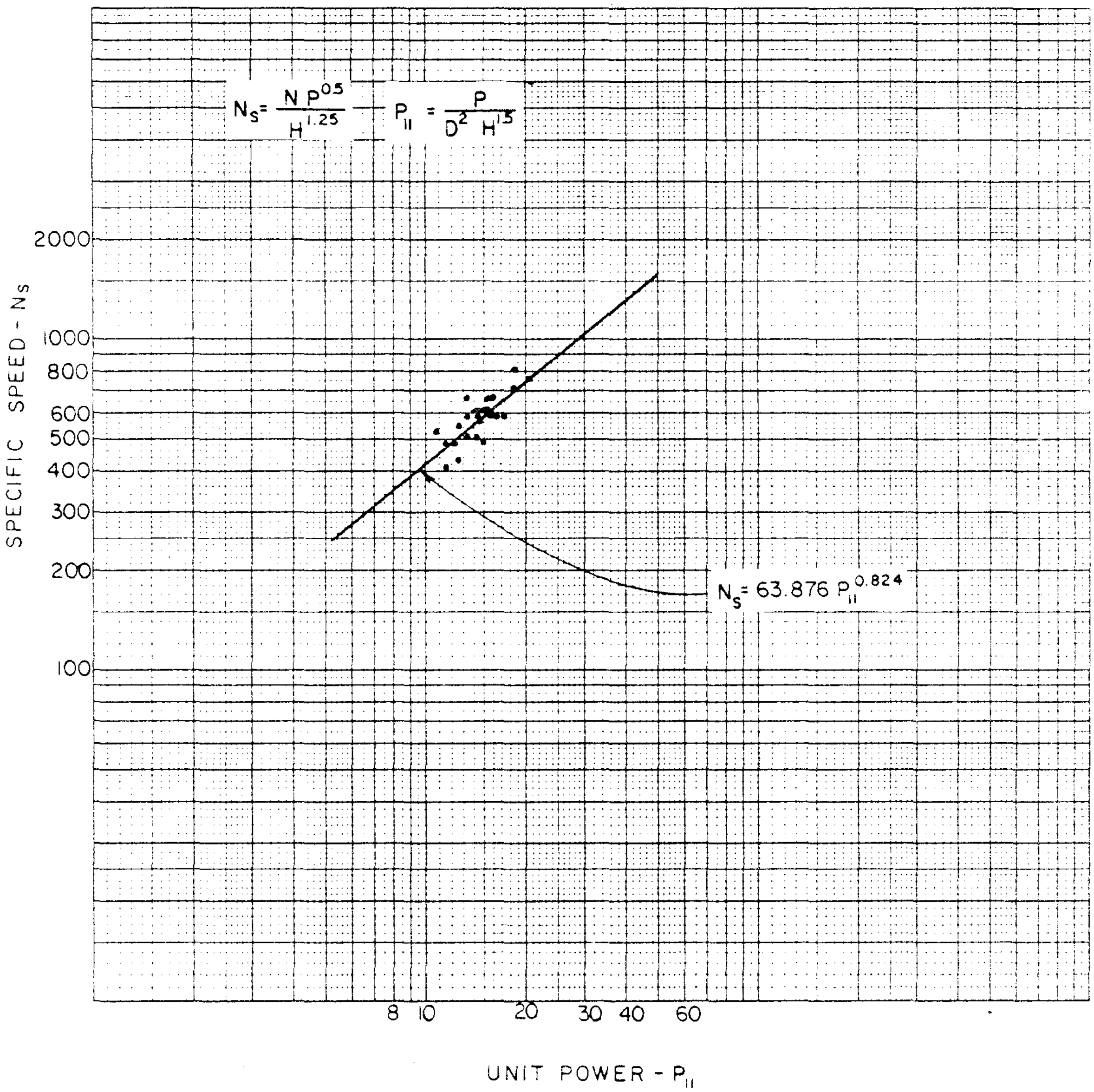


Figure 4-17a. Specific speed versus unit power for tubular turbines.

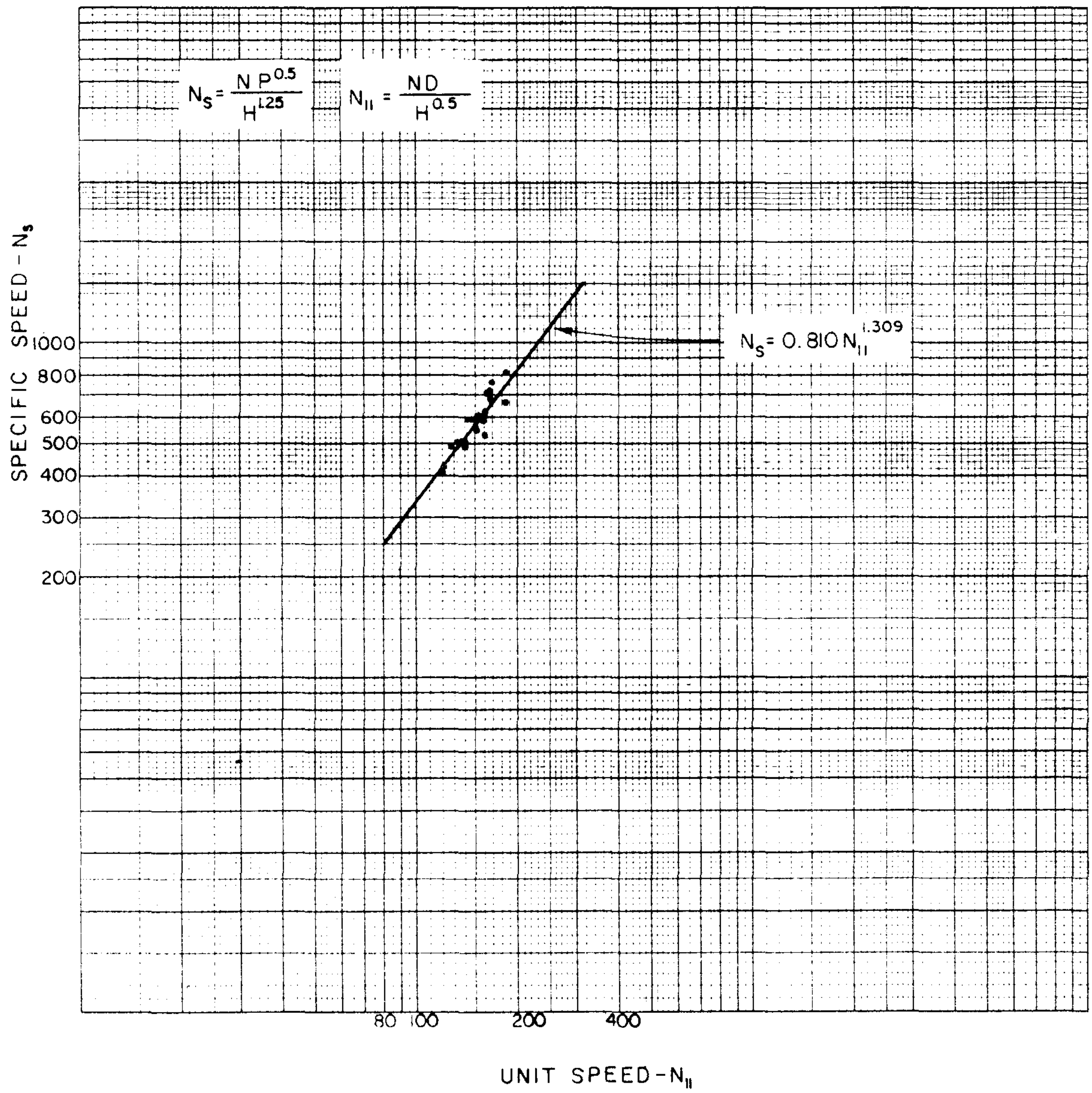


Figure 4-17b. Specific speed versus unit speed for tubular turbines

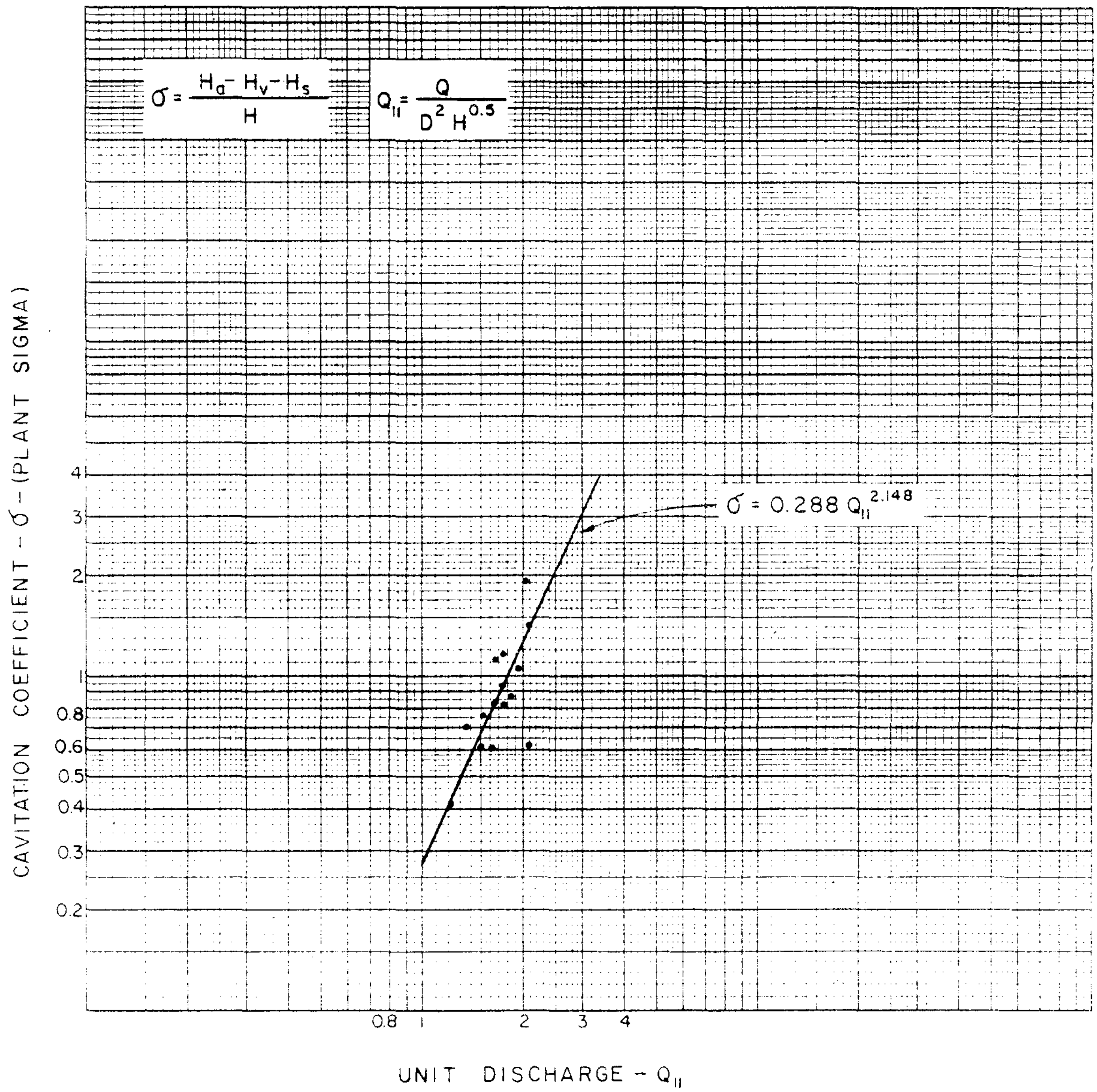


Figure 4-18. Cavitation coefficient versus unit discharge for tubular turbines.

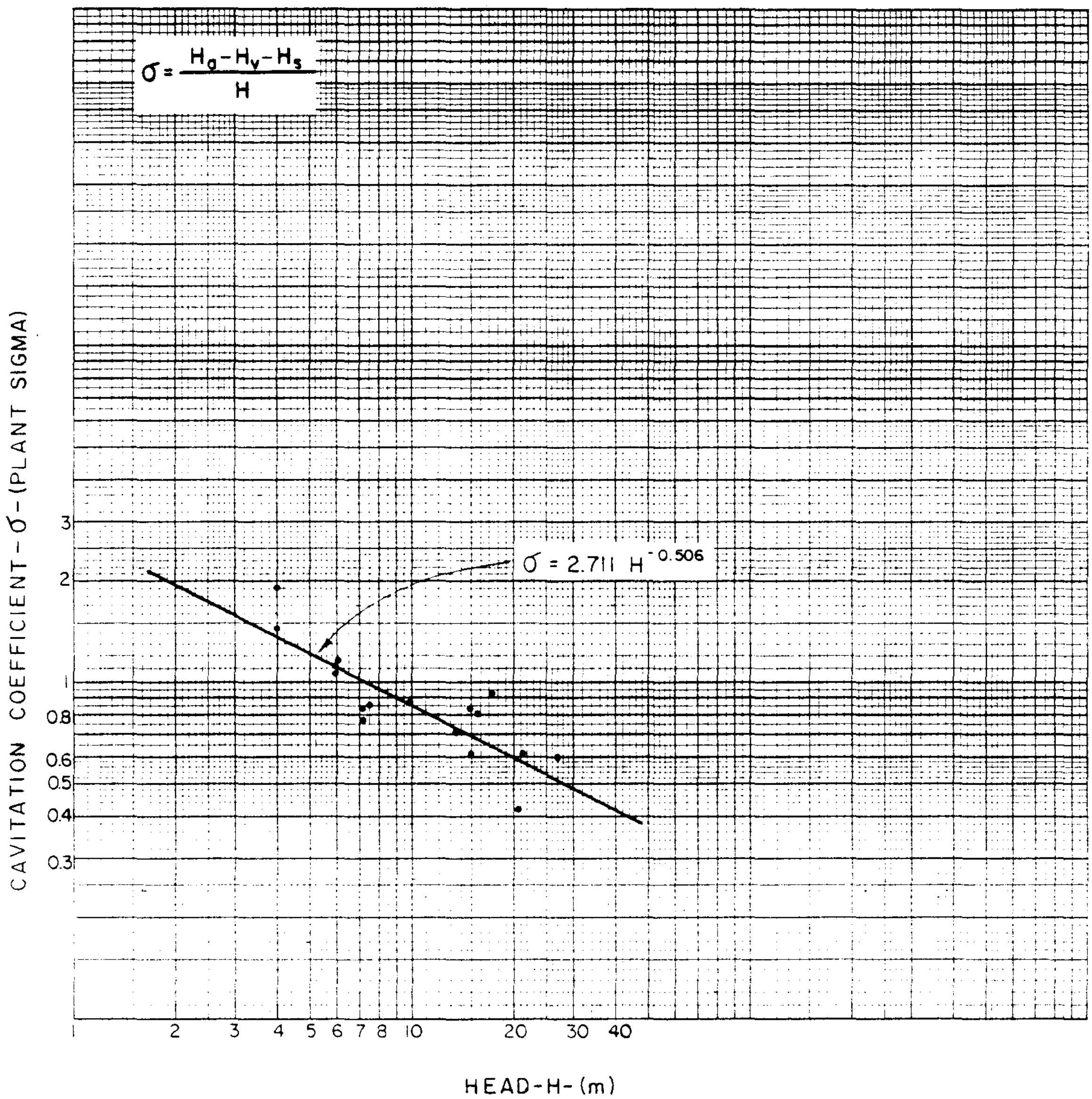


Figure 4-19. Cavitation coefficient versus design head for tubular turbines.

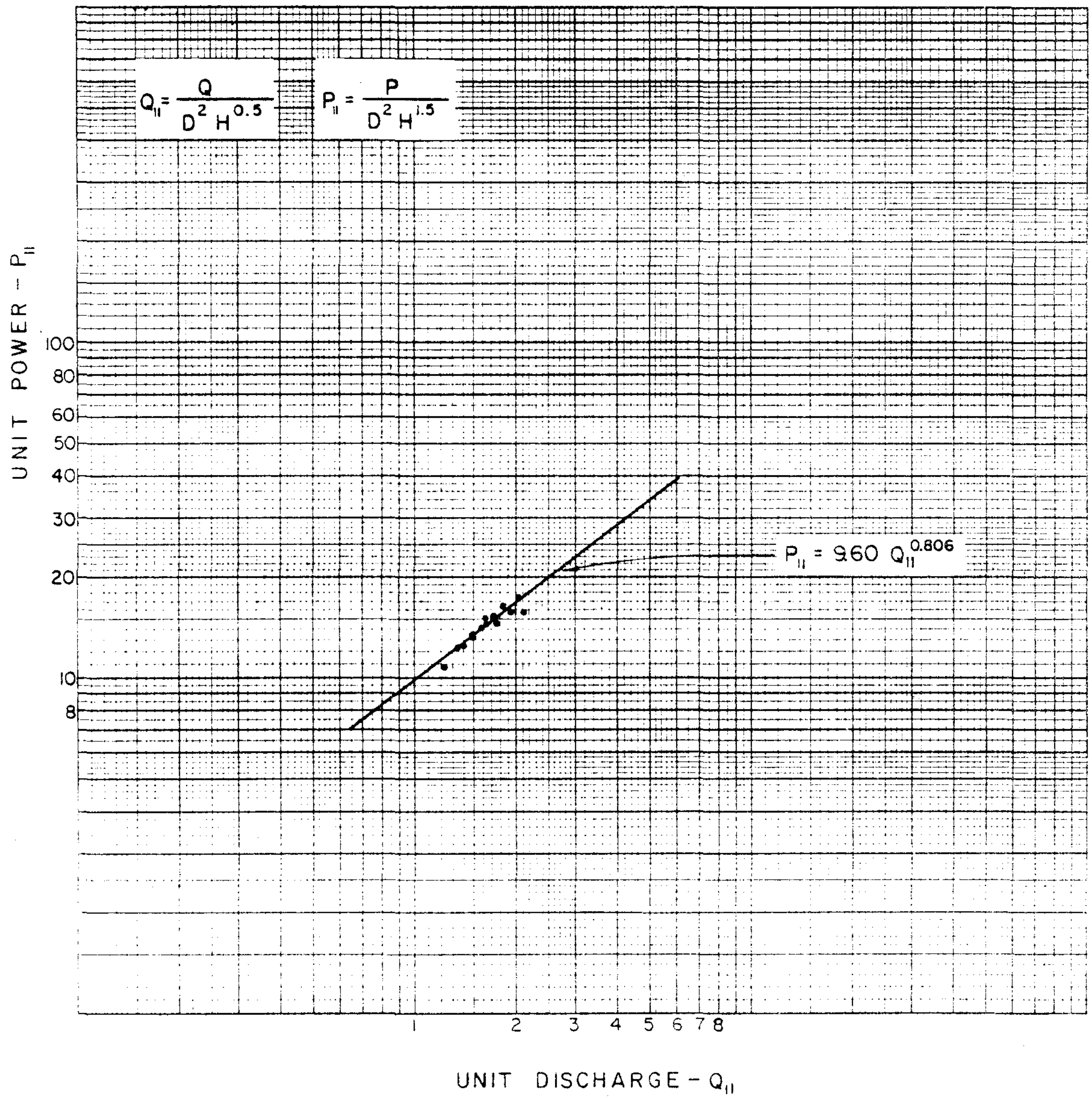


Figure 4-20. Unit power versus unit discharge for tubular turbines.

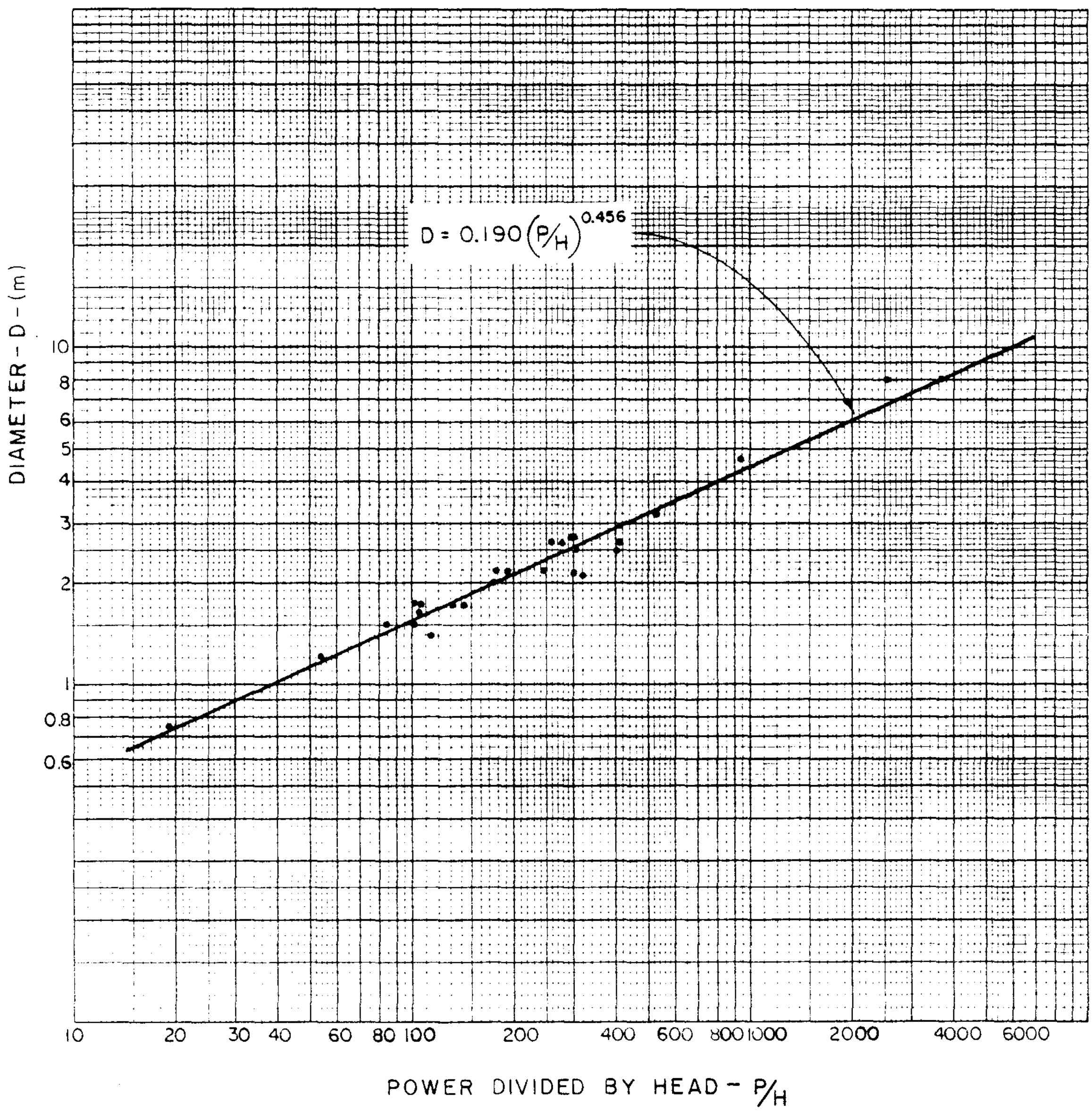


Figure 4-21. Diameter versus ratio of rated power and design head for tubular turbines.

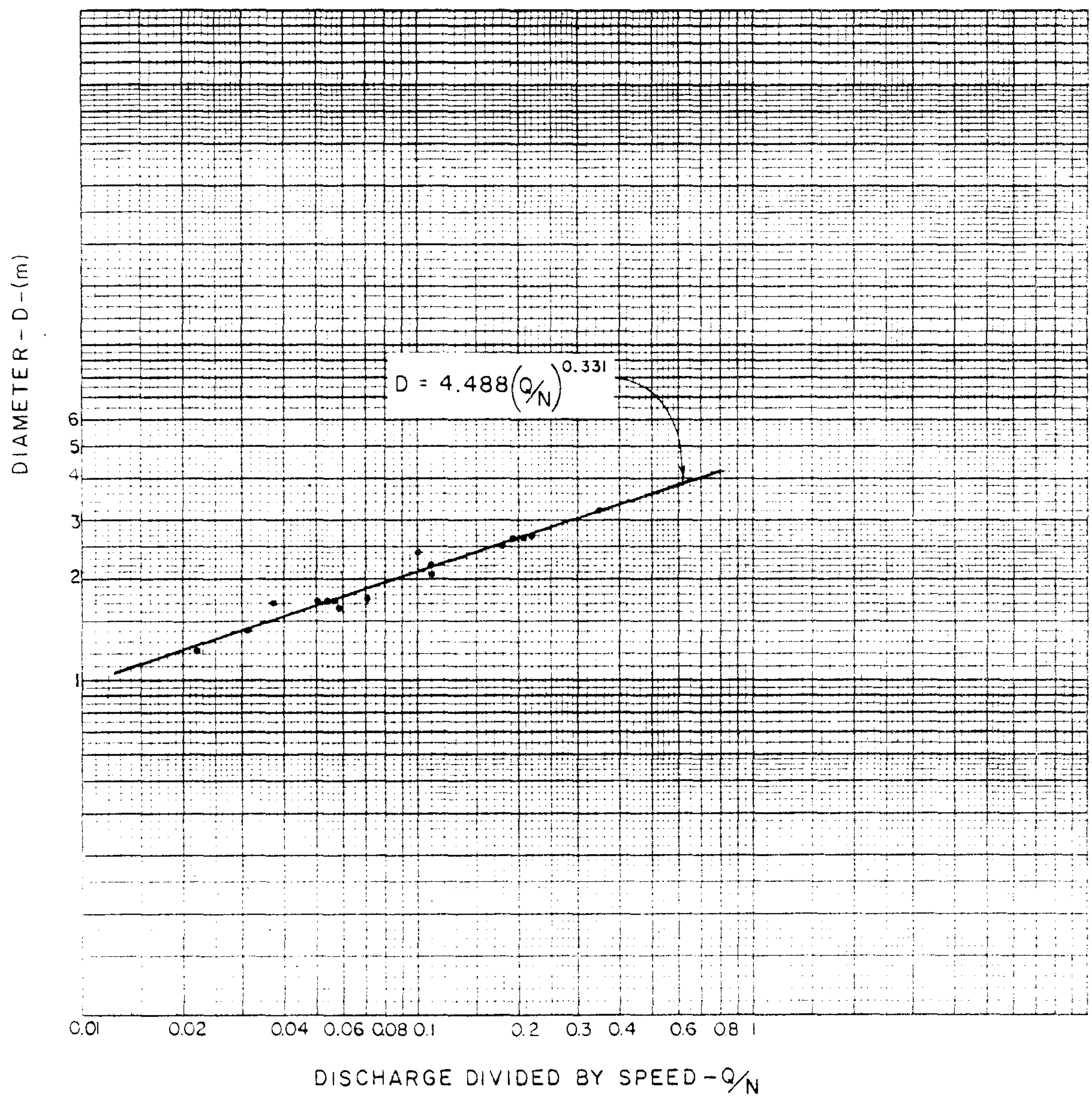


Figure 4-22. Diameter versus ratio of discharge and speed for tubular turbines.

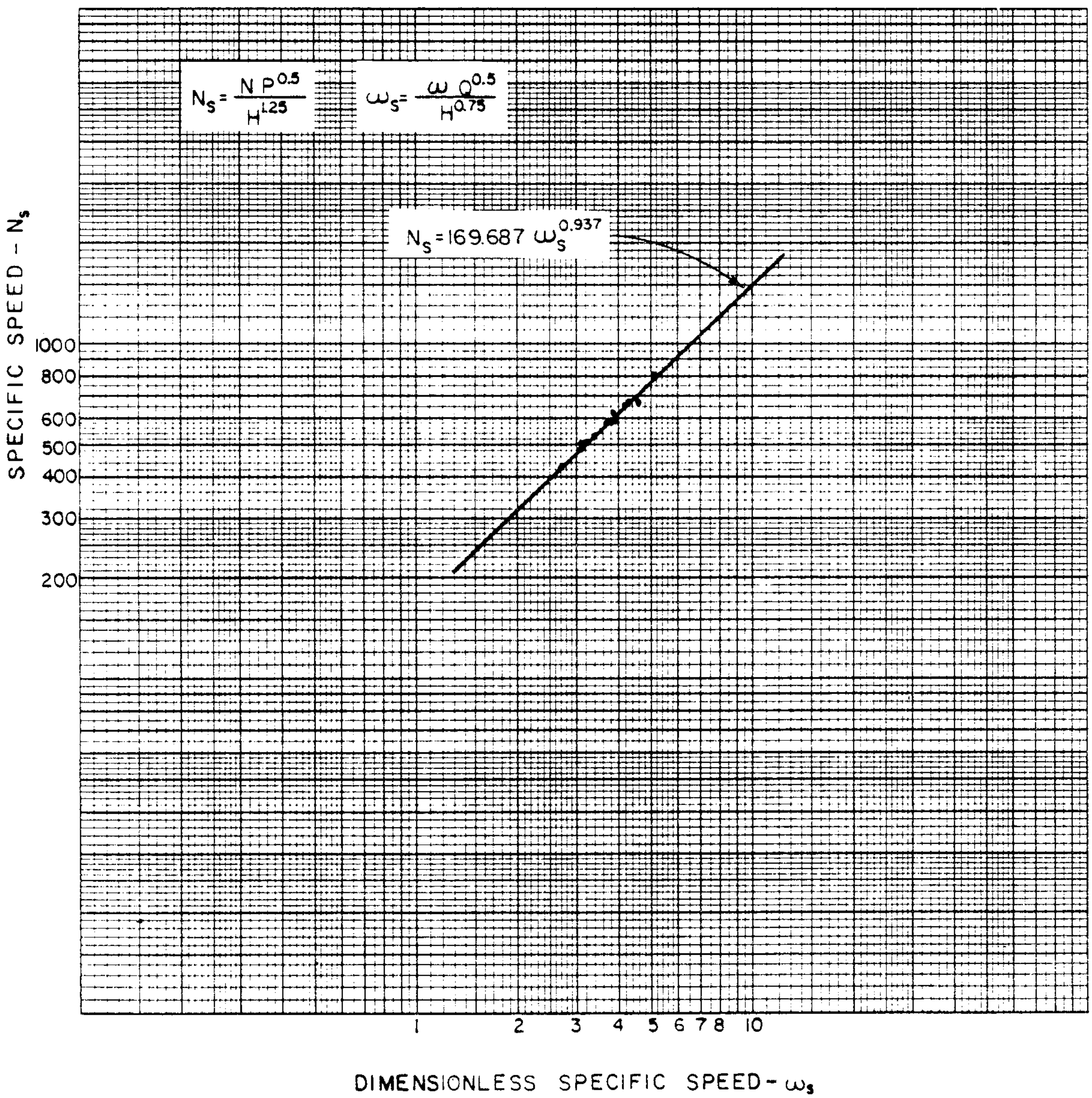


Figure 4-23. Specific speed versus dimensionless specific speed for tubular turbines.

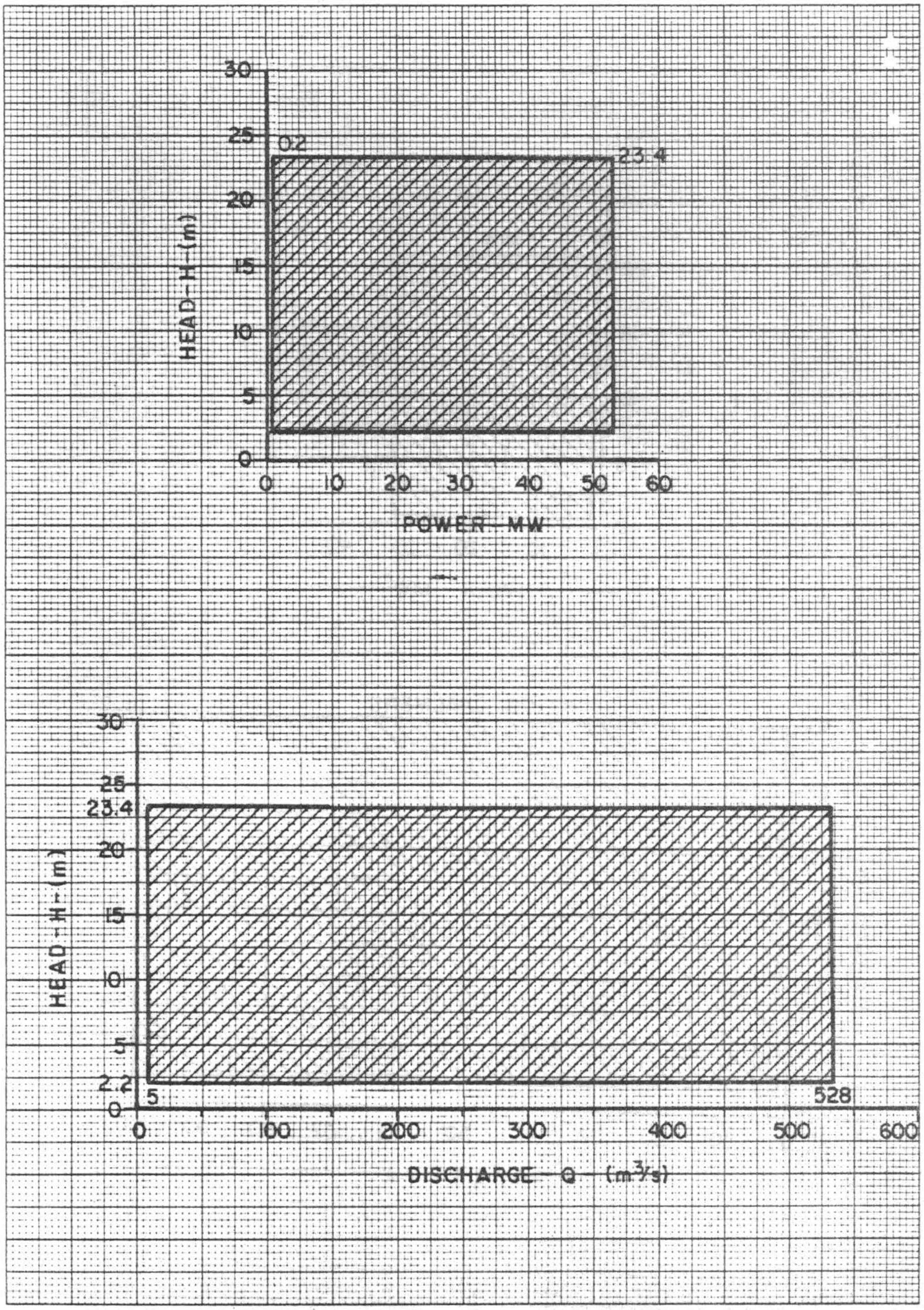


Figure 4 - 24. Bulb turbine range of use selection chart produced from Table A-1.

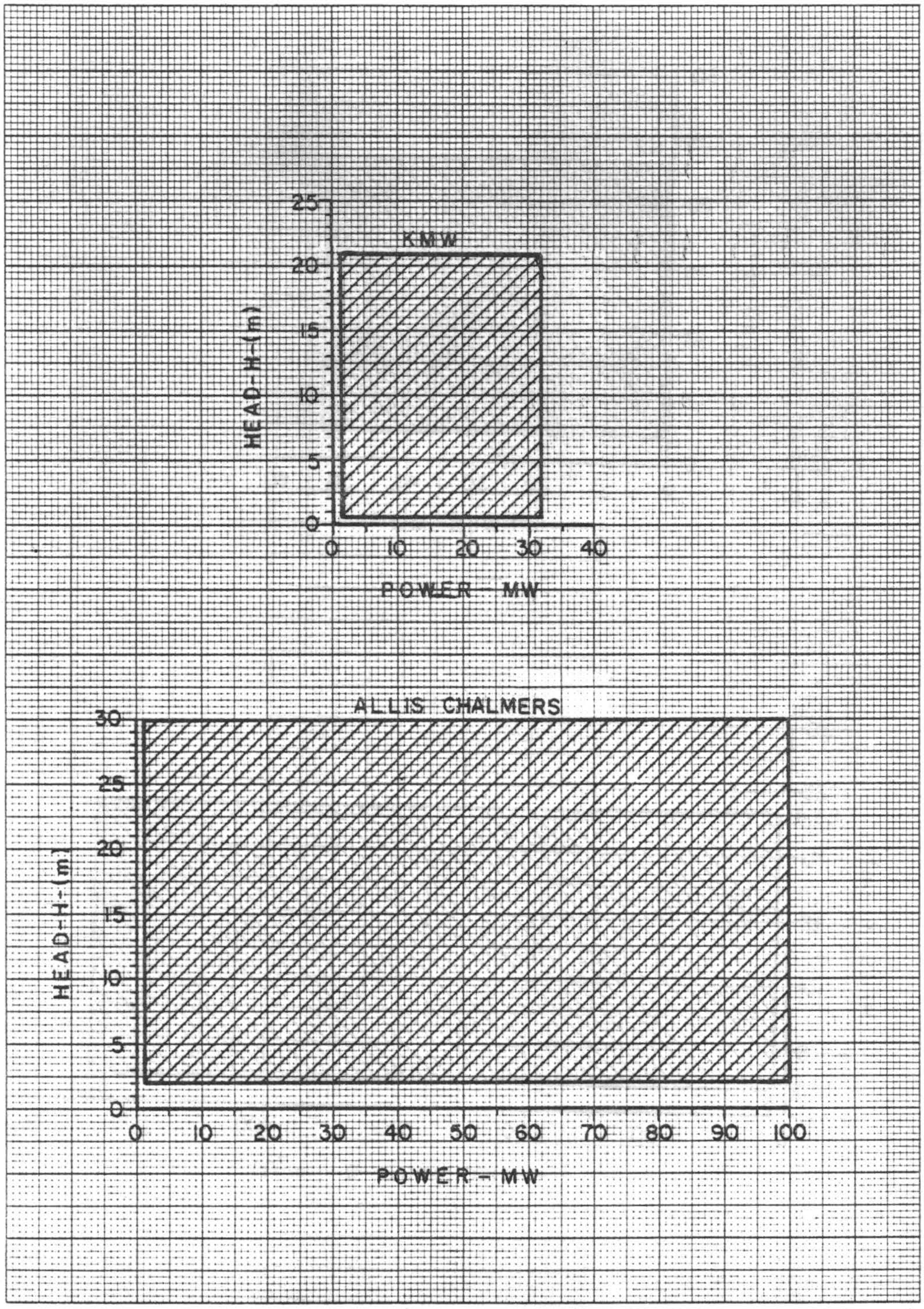


Figure 4- 25. Bulb turbine range of use selection chart for KMW and Allis Chalmers.

Figure 4- 26. Standard bulb turbine range of use selection chart for Neyrpic.

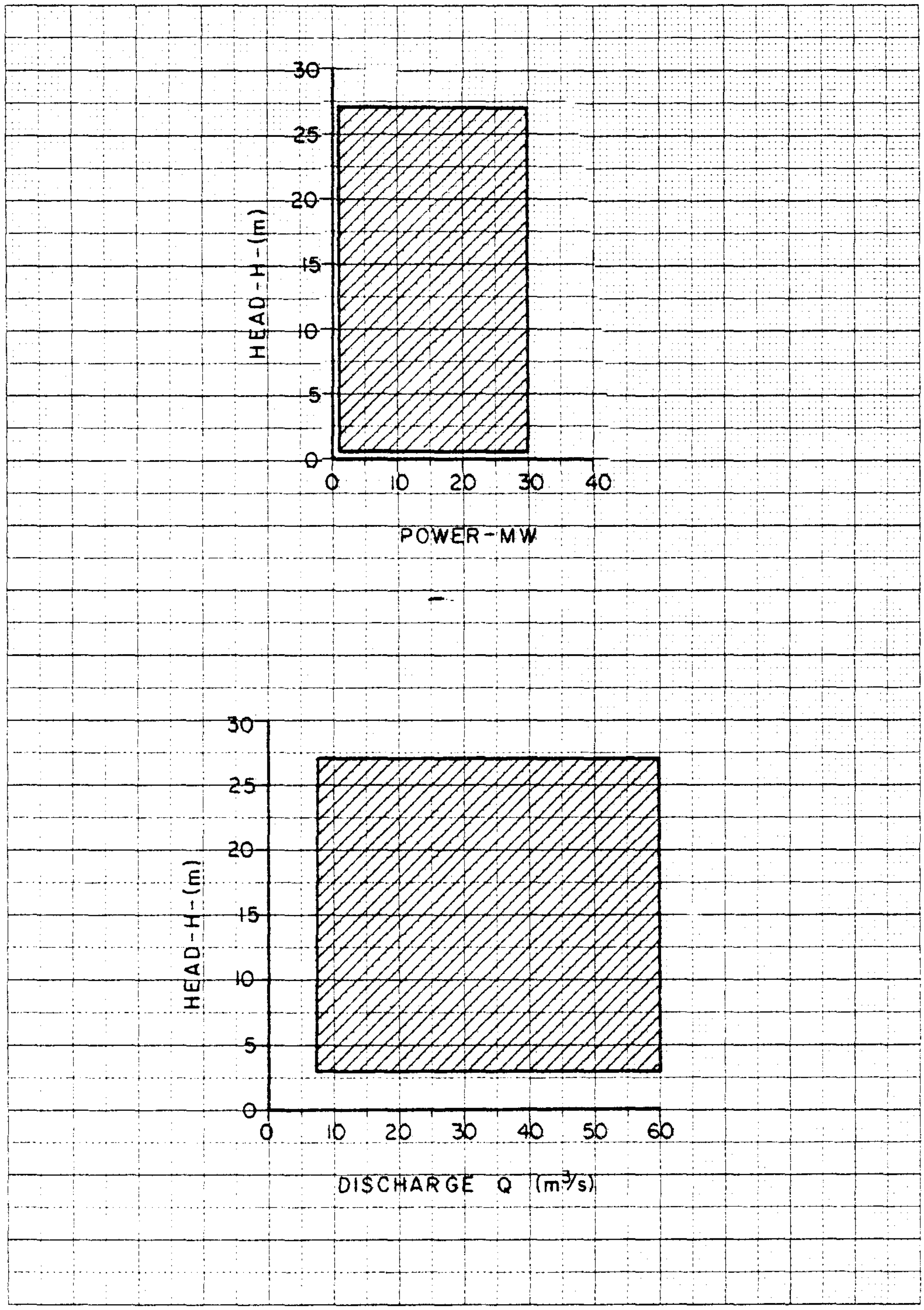


Figure 4-27. Tubular turbine range of use selection chart produced from Table A-2.

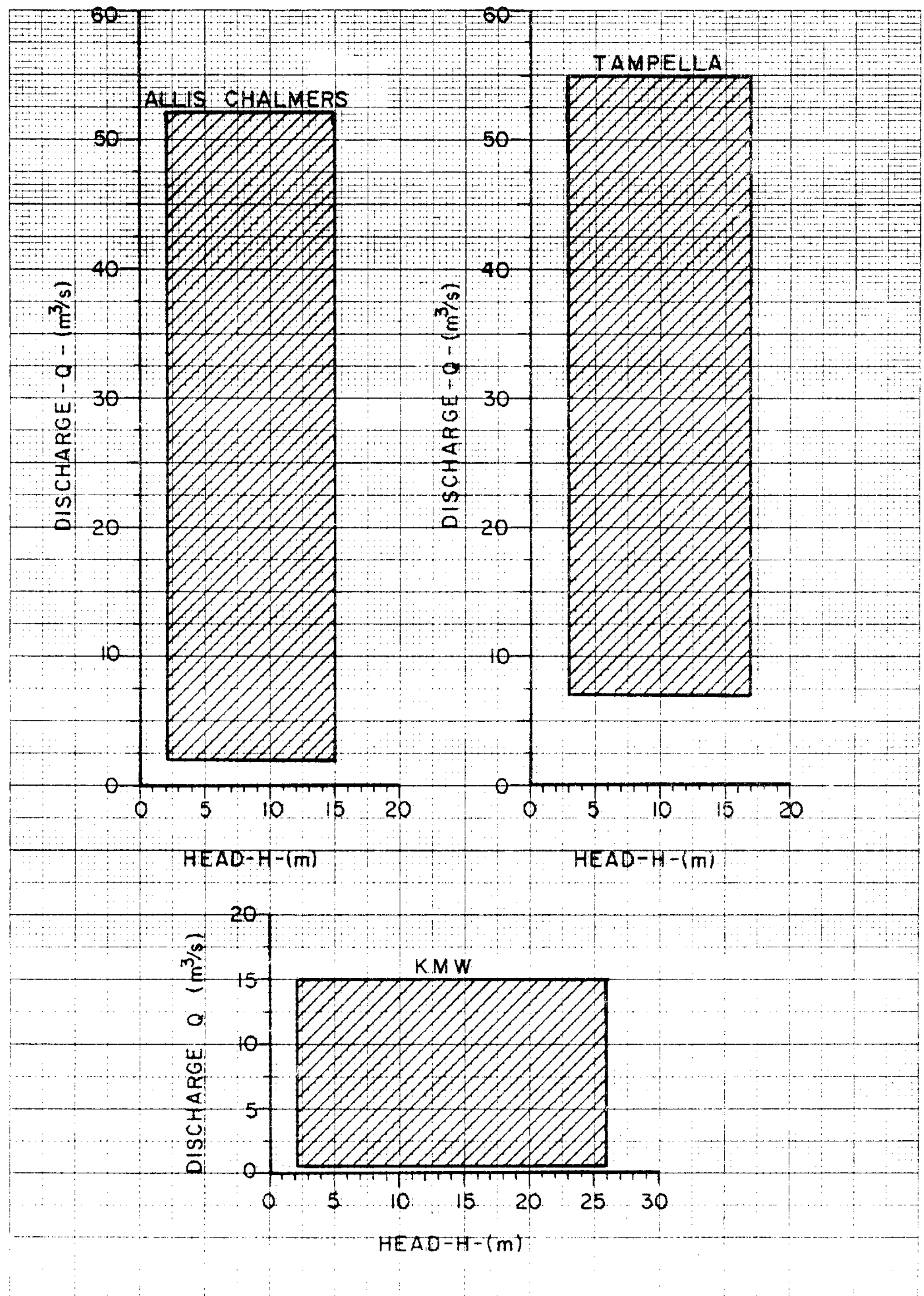


Figure 4 - 28. Standard tubular turbine range of use selection chart for Allis Chalmers, Tampella and KMW.

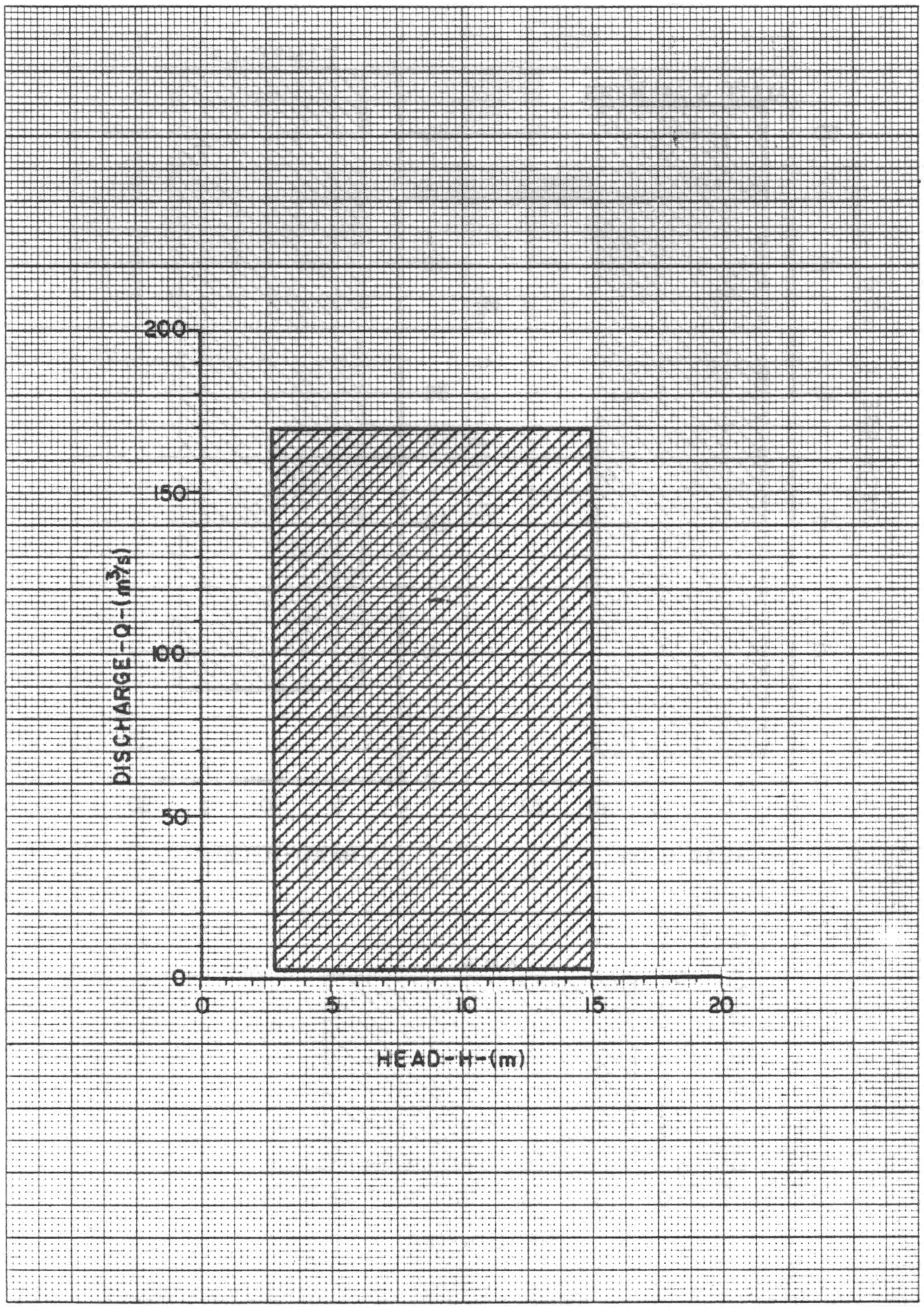


Figure 4-29. Standard Straflo turbine range of use selection chart for Escher Wyss.

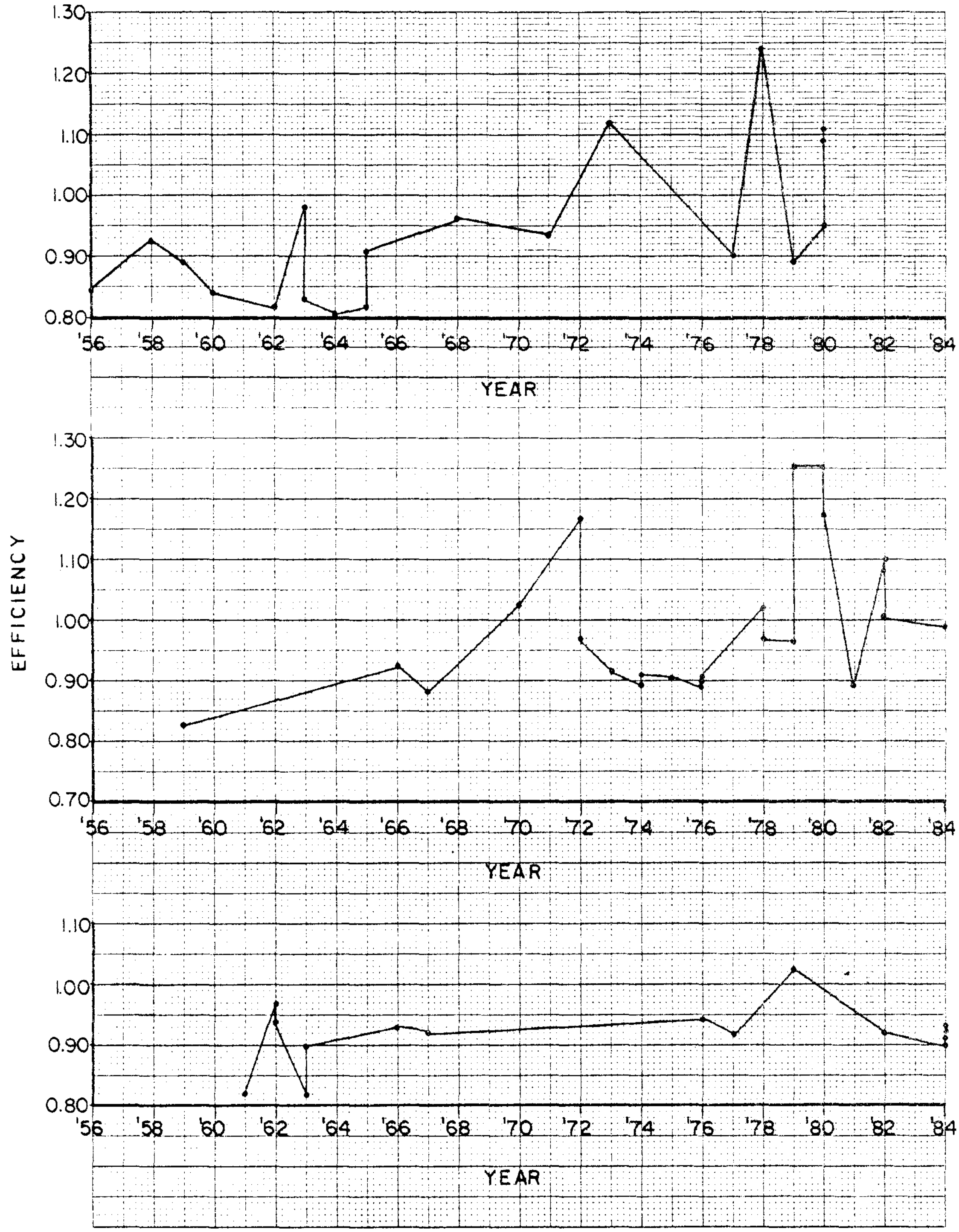


Figure 4-30. Bulb turbine efficiency versus year for some major turbine manufacturers.

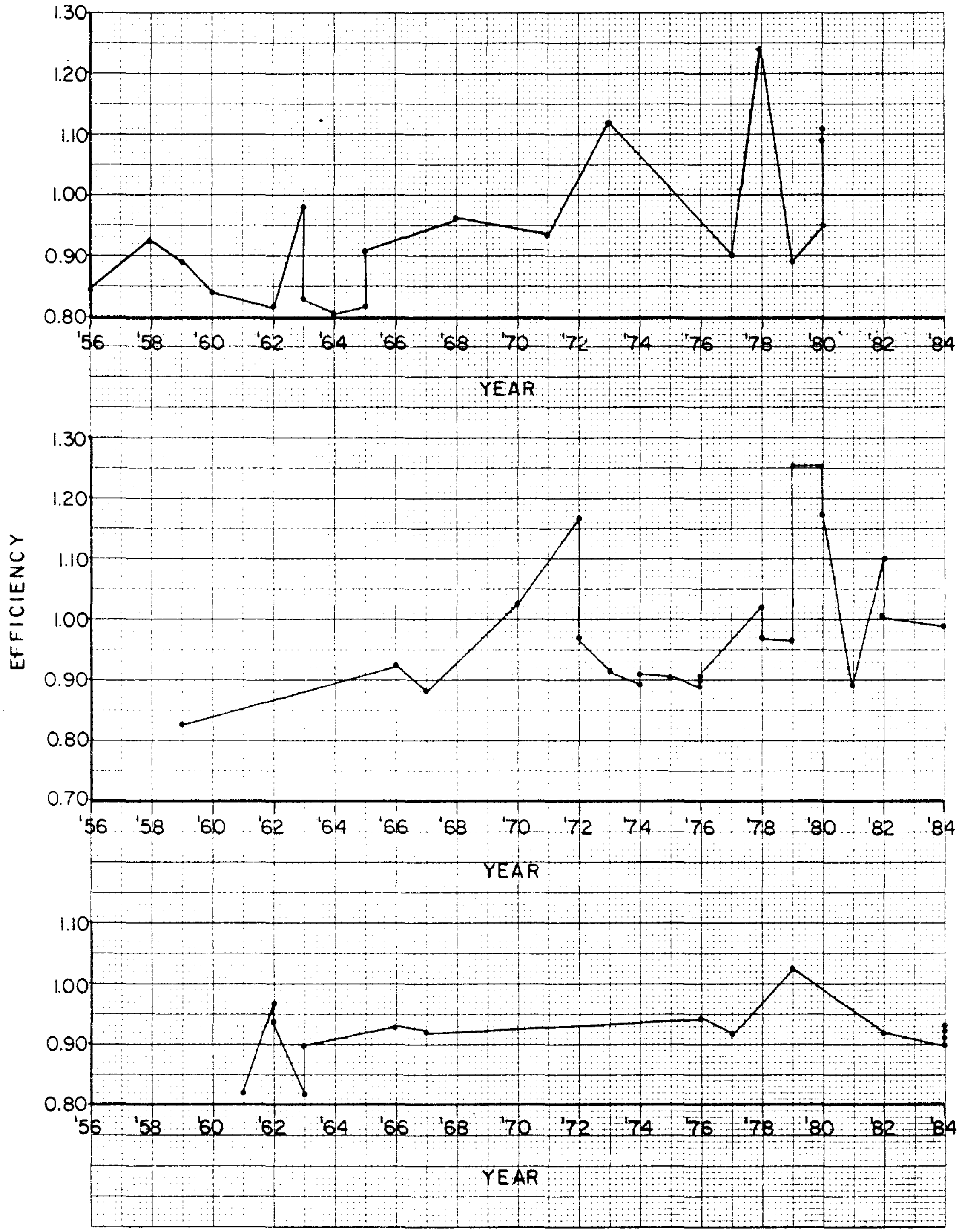


Figure 4-31. Bulb turbine efficiency versus year for some major turbine manufacturers.

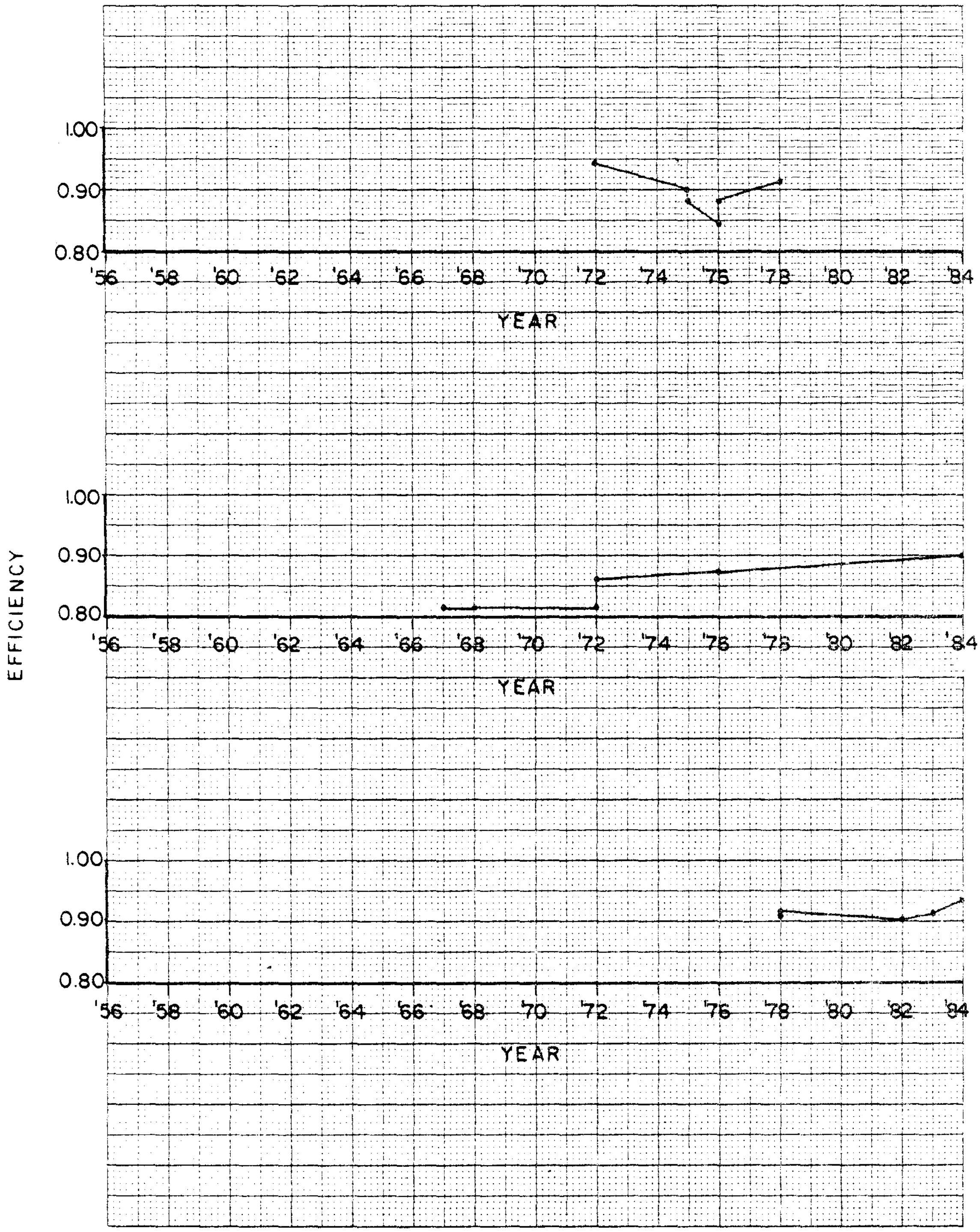


Figure 4-32. Bulb turbine efficiency versus year for some major turbine manufacturers.

5. COMPARISONS OF EXPERIENCE CURVES FOR HYDRAULIC TURBINES

5.1 Introduction

As indicated earlier, the main purpose of developing experience curves for low-head turbines is to fill the present gap in the needed tools for the preliminary design and feasibility studies of low-head power plants and provide engineers with the latest information on the characteristics of low-head turbines. It is therefore of importance to compare the regression equations developed in chapter four of this thesis with some of the regression equations developed by other authors for Francis, Pelton and Propeller turbines. The above mentioned relations were developed by de Siervo et al., (1976, 1977, 1978) and by Sutherland (1968). Sutherland developed some relations for low-head turbines but did not distinguish between the characteristics of the three types of low-head turbines discussed in this study, namely: Bulb, Tubular and rim-generator turbines.

5.2 Comparison of Regression Equations and Nomographs

Figure 5-1 compares experience curves of cavitation coefficient versus specific speed, for the Francis and Propeller turbines, developed by de Siervo and de Leva with those developed for bulb and tubular turbines in this study. Relations with tubular turbines did not yield good correlation coefficients and the regression equation did not show the same trend as those of Francis, Propeller and Bulb turbines. The resulting regression equations are shown below.

Francis turbine $\sigma = 7.54 \times 10^{-5} N_s^{1.410}$ (de Siervo and de Leva, 1977)

Propeller turbine $\sigma = 6.40 \times 10^{-5} N_s^{1.460}$ (de Siervo and de Leva, 1977)

Bulb turbine $\sigma = 7.62 \times 10^{-5} N_s^{1.485}$ (by the author, 1982)

Tubular turbine $\sigma = 31.10 \times 10^{-5} N_s^{1.242}$ (by the author, 1982)

Relations involving specific speed and design head are shown in Figure 5.2. The feasible design range for Propeller, Bulb and Francis turbines are best represented by ranges or solution spaces of possible values of specific speed for a given value of design head or vice versa. The solution spaces are bounded at the top and bottom by regression curves for the latest and earliest group of turbines manufactured, respectively. Regression equations describing the average values of the turbine characteristics are presented below.

Francis turbine $N_S = 3470H^{-0.625}$ 1970-1975
(de Siervo and de Leva, 1976)

Pelton turbine (single jet) $N_S = 85.490H^{-0.243}$ 1965-1977
(de Siervo and de Leva, 1978)

Propeller turbines $N_S = 2419H^{-0.489}$ 1970-1976
(de Siervo and de Leva, 1977)

Bulb turbine $N_S = 1757.75H^{-0.341}$ 1971-1984
(by Author, 1982)

Tubular turbine $N_S = 1003.91H^{-0.243}$ 1957-1984
(by Author, 1982)

Sutherland (1968) published a range of possible specific speeds for a given design head for low-head turbines as presented below.

Low-head turbines N_s range $1000H^{0.333}$ to $2000\ H^{0.333}$

Figure 5-1 shows that Bulb turbines as designed, manufactured and installed have higher cavitation coefficients than either Propeller or Francis turbines. This implies that for the same size machines and design conditions, the Bulb turbines should normally be set at a lower elevation than tubular, vertical propeller, and Francis turbines to obtain adequate protection from cavitation. However, due to the horizontal orientation of the runner and draft tube it is still possible to achieve reasonably cavitation-free operation with less excavation than with the other turbines. The above observations apply to the tubular turbines for specific speeds lower than 1500, when the tubular turbine is compared with either Propeller or Francis turbines. Figure 5-2 shows that for design heads lower than 30m, the specific speed for propeller turbines are higher than those of Bulb and Tubular turbines for the same design head. Since the current lowhead turbine range of operation is below 30 meters, (see figures 4-24 through 4-29) the curves developed in this research reveal that selections of low-head turbines using the characteristic curves for the Propeller turbine will be in error.

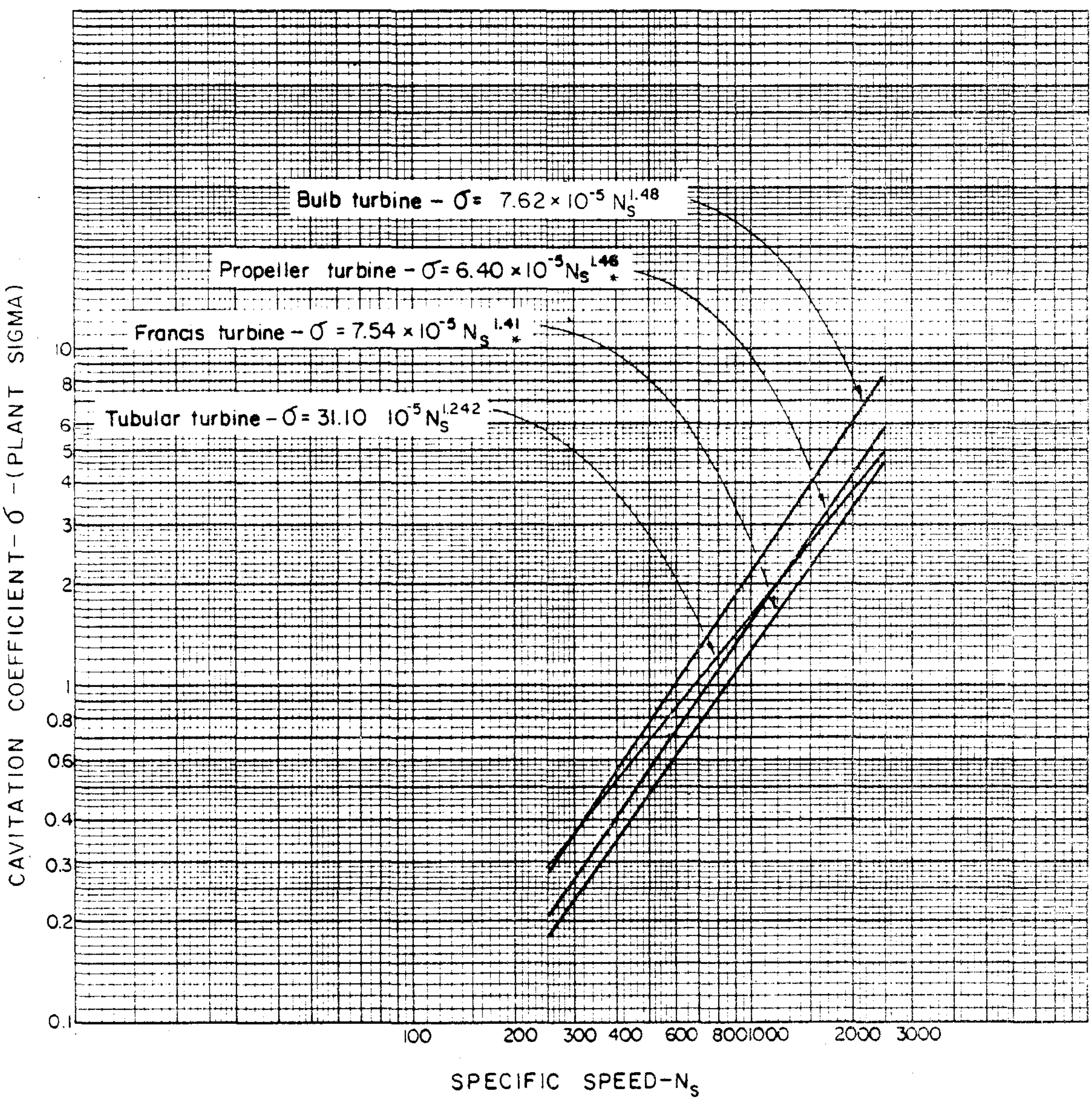


Figure 5-1. Cavitation coefficient versus specific speed for propeller, bulb and Francis turbines.

Sources: * de Siervo, F., Water Power and Dam Construction.

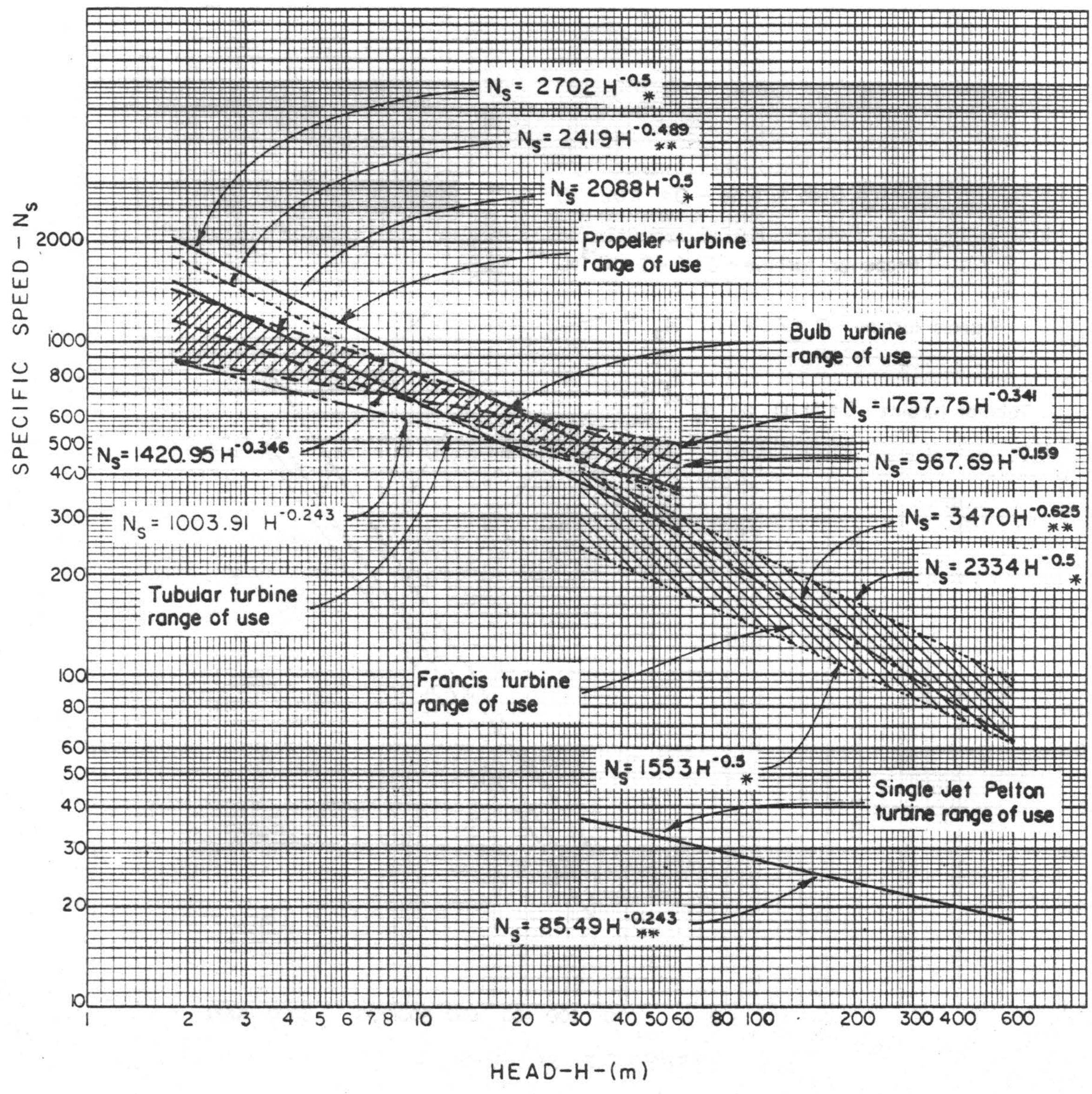


Figure 5-2. Specific speed versus design head for propeller, tubular, bulb, Francis and Pelton turbine ranges.

Sources: ** de Siervo, F., Water Power and Dam Construction;
* USBR, Monogram No. 20.

6. CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

The results of the research work presented in this thesis involved systematic collection of information and study of a sample of bulb and tubular turbines manufactured all over the world that have been installed or that are due to be installed in hydropower plants between 1953 and 1984. The study involved the development of nomographs and linear regression equations correlating the most commonly used hydraulic trubine constants in the hydropower industry. The study has produced a body of observational knowledge against which theories of low-head hydropower turbines can be checked and from which turbine parameters can be estimated for planning and feasibility studies of low-head hydropower plants.

The data used in the study were collected on one hundred and ninety (190) bulb turbines and thirty-eight (38) tubular turbines manufactured by about thirty (30) hydraulic turbine manufacturers. Since low-head hydraulic turbine manufacture has not to any great extent been standardized, design and production standards of such turbines are governed by management policies, design practices and the available level of hydraulic turbine technology and workmanship existing in the various manufacturing companies. The above mentioned factors influence the characteristics of turbines produced. Therefore, it is of importance that a representative sample of data be collected from the industry. On the basis of correspondence with some major European, Japanese and American hydraulic turbine manufacturers, it is believed that the data included in the appendix of this thesis contained the characteristics of over eighty percent (80%) of the most outstanding bulb and tubular turbines manufactured all over the world between 1953 and 1982 and therefore the data constituted a representative sample of the characteristics of bulb and tubular turbines manufactured all over the world between the above mentioned dates.

An acceptable methodology commonly used to analyze data to fit a two-variable linear models is the statistical methods of simple regression analysis. The value of the correlation coefficient and a significant statistic on the regression slope parameter are used as a measure of how well a straight line fits the available data points. Simple regression analysis and electronic computer programs were used in the study in order to obtain predictive relationship that exist between various turbine parameters. The nomographs and regression equations developed in this study were compared with those developed for Francis, Pelton and Propeller turbines by other researchers. The results of such comparisons revealed that generally the characteristics curves developed in this study for bulb and tubular turbines follow the same kind of trends as those developed for the Francis, Pelton and Propeller turbines. However, there are also some very significant differences (figures 5-1 and 5-2, on pages 95 and 96) suggesting that scaled down values of the high and medium head turbines characteristics can not be used to represent the characteristics of low-head turbines. Figure 4-1 on page 58 shows that the average specific speed of bulb turbines placed in service or are due to be placed in service between 1971 and 1984 has increased compared with the average specific speeds of the turbines placed in service between 1953 and 1960. Similarly Figures 4-9 and 4-10 on pages 67 and 68 respectively showed that turbines placed in service or are due to be placed in service between 1966 and 1984 exhibited an increase in unit power and rated power respectively compared with turbines placed in service between 1953 and 1965 all reduced to the same design condition and diameter. The observed increases in specific speed, unit power and rated power can all be related to increases in rated power which in turn can be related to the improvement in the technology of design and manufacture of low-head turbines. The advancement in technology might have resulted partly from the introduction of computer applications in design, improved model testing and more careful production practices of turbines. The curve correlating the specific speed and rated head of turbines placed in service between 1961 and 1970 (figure 4-1 on page 58) showed a different trend in slope from the curves for 1953 to 1960 and 1971 to 1984. The possible causes of this difference could not be readily explained.

Figure 5-1 on page 95 shows that the value of the cavitation coefficient of bulb turbine is higher than those for Francis, Propeller and tubular turbines all considered at the same design conditions. This implies that the bulb turbine can be set at a lower elevation than the tubular, propeller or Francis turbine for the same design conditions. For specific speeds ranging from 300 to 1000 the tubular turbine has shown a higher cavitation coefficient than either the Propeller or Francis turbine.

Figures 4-10 and 4-11 on pages 68 and 69, respectively are graphs of turbine diameter versus power-head ratio and discharge-speed ratio for bulb and tubular turbines. The resulting regression equations yielded very good results with the respective maximum correlation coefficients of 0.97 and 0.99 for bulb turbines and 0.96 and 0.96 for tubular turbines. It is believed that the above relations derived in this thesis will yield better estimates of the turbine diameter than the procedures which select the turbine diameter from relations involving the specific speed and the rotational speed determined from experience curves which already contain some inherent error.

The relations involving cavitation coefficient and other turbine constants yielded correlation coefficients ranging from 0.51 to 0.73 which are lower in value than was expected. Consultations with a few European and American turbine manufacturers revealed that the definition of the reported maximum and minimum tailwater conditions differ among the manufacturers. Therefore, the values of the cavitation coefficient computed using the data supplied by the various turbines manufacturers do not always represent similar operating conditons.

Figures 4-30 through 4-32 grouped together on pages 84 through 91 are graphs of turbine efficiency versus year of installation. These graphs revealed that there is very little, if any, value in relating efficiency of bulb turbines with year of installation because of lack of uniformity among the turbine manufacturers in reporting rated conditions instead of the guaranteed operating conditions. During the study period of the characteristics of bulb and tubular turbines reported in this thesis, the author and Professor Calvin C. Warnick corresponded with a number of major European and American manufacturers and also with some of the authors of publications on hydraulic turbine characteristics. Those enquiries revealed no knowledge of experience curves presently existing for bulb and tubular turbines as reported in this study. Therefore, the nomographs and regression equations presented in this thesis should be useful to engineers, planners and designers in selecting bulb and tubular turbines for low-head hydro projects.

6.2 Recommendations

In the earlier chapters regression equations have been derived with turbine constants normally used in feasibility studies and planning of hydro power projects. New relations between turbine runner diameter and power-head ratio and discharge-speed ratio have also been presented. The experience curves presented are recommended for use in the selection of design speeds, preliminary design turbine diameter and expected rated turbine output when design net head and discharge have been determined at proposed sites.

The results of this research would indicate that a more consistent practice should be sought to report rated output and the operating inputs of net head and discharge to obtain more knowledge as to the efficiencies that are actually attained. The results are not sufficiently conclusive with turbine setting and cavitation characteristics to recommend use of the experience curves.

In order to decrease cost of feasibility studies and planning of low-head hydropower projects, it is useful to have a guide to the required dimensions of the turbine, generator and corresponding housing and concrete structures. Some of the desired dimensions are those of intake structure, turbine position from intake structure, water passage surrounding the turbine, the draft tube dimensions, in case of bulb turbines, bulb dimensions, bulb position from intake structure, and water passage surrounding the bulb. Further research work on the characteristics of low-head turbines needs to be done to relate the above mentioned dimensions of the hydropower installation to the hydraulic turbine diameter. It is believed that such correlations will yield approximate guides to the required dimensions of hydropower units under initial study. Experience curves relating turbine diameter and turbine cost, efficiency and draft tube dimensions should be developed.

A number of turbine manufacturers have produced standard turbine units. The characteristics of these standard units should be compared with the experience curves derived in this research.

More data should be sought on very small minihydro and microhydro low-head units that are becomming available to see whether these very small units follow similar turbine similarity laws and can be characterized by these newly developed experience curves.

The above mentioned type of experience curves lend themselves to being used in programs written for modern high-speed digital computers. Therefore, a further stage of the research should be to write programs for economic analysis of potential low-head hydropower sites. Such programs should comprise subroutines for hydrological studies using flow duration curve method, turbine capacity selection by power duration curve technique, hydraulic turbine selection and cost estimation. The type of the economic analysis programs envisioned, will be simple empirical schemes written for use on both digital computers and programmable hand calculators. The programs, when completed, will provide a valuable tool for the practicing engineer in the field or working in developing countries where capital and technological resources are scarce.

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APPENDIX

WORKED EXAMPLE OF TURBINE SELECTION

During the feasibility studies of a hydropower project, the initial site data available are:

1. Design head
2. Design flow

The main responsibilities of the Engineer or planner with regard to turbine selection are to decide on:

1. Type of turbine required
2. Plant capacity
3. Turbine running speed
4. Turbine specific speed
5. Turbine diameter
6. Plant sigma (σ_p) and turbine setting

The procedure for making the above deicisions based on the nomographs developed in this thesis are given below.

Assume that the given hydropower project is the Melk hydropower plant in Austria with the following site data as given in Table A-1.

Design head = 8.20 m

Design flow = 300.0 m^3/sec.

Procedure

1. Using figures 4-24 through 4-29, select a suitable turbine type for the project. That is either bulb or tubular turbine.
2. Assume a suitable turbine efficiency η, example $\eta = 0.92$.
3. Compute plant capacity per unit (KW) $P = \rho gQH\eta$ (KW)
4. Compute the ratio (P/H) and using figure 4-10, determine diameter, D of the turbine.

5. Compute turbine running speed H from (Q/H) regression equation in Table 4-1 or on Figure 4-11.

6. Compute the synchronous speed, N, for the turbine and generator.

7. Recompute the diameter, D, using the value of the synchronous speed.

8. Recompute rated power.

9. Compute the specific speed, N_S.

10. Using the computed value of the specific speed, compute cavitation coefficient σ_p using Figure 4-7a or regression equation in Table 4-1.

Solution

1. Assume bulb turbine(s) is selected for the project.

2. Assume efficiency $\eta = 0.92$

3. Plant capacity P, is given by:

$$P = \rho g Q H \eta \text{ (KW)}$$

$$= 9.81 \times 300 \times 8.2 \times 0.92 = 2202 \text{ KW}$$

4. $(P/H) = (22{,}202/8.2) = 2707.6$

$$D = 0.193\ (P/H)^{0.439} \quad \text{(see Figure 4-10)}$$

$$D = 0.193\ (2707.6)^{0.439} = 6.20\text{m}$$

5. Running speed, N

$$N = Q/(D/4.186)^{3.236}$$

$$N = 300/(6.2/4.186)^{3.236} = 84.15 \text{ rpm}$$

6. Number of Poles, $N_p = \frac{120\ (f)}{N}$

Where f = frequency = 60 Hertz (for USA)

$$N_p = \frac{120 \times 60}{84.15} = 85.56$$

$$\text{For } N_p = 86, \quad N = \frac{7200}{86} = 83.72 \text{ rpm}$$

$$N_p = 84, \quad N = \frac{7200}{84} = 85.71 \text{ rpm}$$

Rule of Thumb

If net head at site will vary less than 10%, choose the higher speed. If net head will vary more than 10% choose the lower speed.

Assume net head at the Melk site varies more than 10%. Therefore, N = 83.72 rpm is chosen.

7. New Diameter

$$D = 4.186\ (Q/H)^{0.309} \qquad \text{(see Table 4-1)}$$

$$D = 4.186\ (300/83.72)^{0.309} = 6.21\text{m}$$

8. Recompute rated power

$$P = H\left(\frac{D}{0.193}\right)^{2.278}$$

$$P = 8.2\ \left(\frac{6.21}{0.193}\right)^{2.278} = 22{,}283 \text{ KW}$$

9. Specific speed $N_s = \dfrac{N\ P^{0.5}}{H^{1.25}}$

$$N_s = \frac{83.72\ (22{,}283)^{0.5}}{(8.2)^{1.25}} = 901\ \frac{\text{rpm}\ (\text{KW})^{0.5}}{(\text{Meter})^{1.25}}$$

10. Cavitation coefficient. $_p$

$$_p = 76.23 \times 10^{-6}\ (N_s)^{1.485} \qquad \text{(see Table 4-1)}$$

$$_p = 76.23 \times 10^{-6}\ (901)^{1.488} = \underline{\underline{1.86}}$$

SUMMARY OF DESIGN AND COMPUTED PARAMETER

	Design	Computed	% Error
Rated Head, H(m)	8.2	8.2	--
Rated flow, $Q(m^3/s)$	300	300	--
Rated power P, (KW)	22,280	22,283	0.01
Runner diameter D(m)	6.30	6.21	1.43
Running speed N(rpm)	85.7	83.72	2.31
Specific speed N_s	922	901.0	2.28

Table A1.

B U L B   T U R B I N E S

POWER STATION	DATE OF COMMISSIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIAMETER (M)	RUNNING SPEED (RPM)	MANUFACTURER
AUSTRIA								
REUTTE	1956	LECH	6.07	24.0	1210	2.20	165.0	EW
PARTENSTEIN	1963	GR.MUHL	9.60	26.0	2200	2.09	234.0	V
TRAUNLEITEN 2	1965	TRAUN	9.50	15.0	1200	-	-	-
GMUNDEN	1968	TRAUN	9.00	75.0	6520	3.30	136.4	-
URSTEIN	1969	SALZACH	10.90	125.0	12310	4.28	125.0	V
OTTENSHEIM	1973	DANUBE	9.10	250.0	20400	5.60	100.0	AD
GMUNDEN(SUPPL.)	1974	TRAUN	-	-	6120	3.30	136.4	AD
GABERSDORF	1974	MUR	8.61	115.0	9000	4.15	107.1	EW
FELTEN	1976	MURZ	6.40	30.0	1700	2.30	176.5	EW
ALTENWORTH	1976	DANUBE	14.00	300.0	38900	6.00	103.4	V
UBERVOGAU	1977	MUR	7.39	117.6	7690	4.15	107.1	EW
ABWINDEN-ASTEN	1979	DANUBE	7.96	284.0	22730	5.70	93.7	V
ABWINDEN-ASTEN	1979	DANUBE	8.20	270.0	20000	5.70	93.7	VA
MELK	1982	DANUBE	8.20	300.0	22280	6.30	85.7	V
GREIFENSTEIN	-	DANUBE	11.20	350.0	35000	6.50	93.7	V
KLEINMUENCHEN	1978	TRAVN	11.50	65.0	6500	3.15	166.7	VA
BELGIUM								
NEUVILLE-SUR-RUY	1962	-	4.00	75.0	2400	3.60	97.5	EW
CANADA								
JENPEG	1976	-	7.30	448.0	28000	7.50	62.0	LMZ
CENTRALE DE LA RIVIERE STE-MARIE	-	STE-MARIE	5.70	360.0	18000	7.10	64.3	ALLIS
LACHINE	-	ST-LAWRENCE	11.00	400.0	35000	6.90	93.8	ALLIS

Table A1 (continued) B U L B T U R B I N E S

POWER STATION	DATE OF COMMISSIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIAMETER (M)	RUNNING SPEED (RPM)	MANUFACTURER
PEOPLE'S REPUBLIC OF CHINA								
MA JI TANG	1984	ZI SHUI	6.56	310.0	18000	6.30	75.0	V
FINLAND								
ANKKAPURHA	1983	KYMIJOKI	9.80	225.0	19800	5.40	100.0	TAM
VAJUKOSKI	1984	KITINEN	15.00	160.0	22020	4.60	136.0	TAM
FRANCE								
GOLFECH	1973	GARONNE	15.50	180.0	23000	5.10	125.0	N
ARGENTAT	1957	DORDOGNE	16.60	98.5	14350	3.70	150.0	V-C
ARGENTAT	1958	DORDOGNE	17.40	14.45	2220	1.80	300.0	V-C
ARGENATAT	1958	DORDOGNE	16.50	-	14400	3.80	150.0	N
VILLENEUVE-SUR-LOT	1970	LOT	11.30	128.0	14400	4.40	136.6	J
CAMBEYRAC	1957	TRUYERE	10.80	55.0	5000	3.10	150.0	N
CAMBEYRAC	1957	TRUYERE	10.80	55.0	5000	3.30	136.4	J
AMBIALET	1961	TARN	6.50	38.0	2000	2.50	187.0	SW
LA CROUX	1981	TARN	13.60	75.0	9280	3.25	200.0	N
SAINT-MALO	1959	-	3.40	300.0	9000	5.80	88.3	N
LA RANCE	1966	LA RANCE	5.80	191.0	10000	5.35	93.8	V-C
GERSTHEIM	1967	RHINE	11.45	234.0	23800	5.60	100.0	S
STRASBOURG	1970	RHINE	11.70	234.0	24500	5.60	100.0	N
GAMBSHEIM	1974	RHINE	10.35	270.0	24050	5.60	100.0	N
BEAUMONT-MONTEUX	1959	ISERE	11.30	89.0	8500	3.80	150.0	N
PIERRE-BENITE	1966	RHONE	7.80	333.0	20000	6.10	83.8	A
BEAUCAIRE	1970	RHONE	10.70	400.0	35000	6.25	93.8	N
GERVANS	1971	RHONE	9.75	405.0	30000	6.25	93.8	N
SAUVETERRE	1973	RHONE	9.40	400.0	33000	6.90	93.8	N
AVIGNON	1973	RHONE	9.10	400.0	30000	6.25	93.8	N
CADEROUSSE	1975	RHONE	9.10	400.0	32500	6.25	93.8	N

Table A1 (continued)

B U L B   T U R B I N E S

POWER STATION	DATE OF COMMIS- SIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIA- METER (M)	RUNNING SPEED (RPM)	MANUFACTURER
CADEROUSSE	1975	RHONE	9.10	410.0	32500	6.90	93.8	N
PEAGE-DE-ROUSSILLON	1977	RHONE	12.00	400.0	40000	6.25	93.8	CL
VAUGRIS	1980	RHONE	5.65	350.0	18000	6.25	75.0	A
VAUGRIS	1980	RHONE	5.65	350.0	18000	6.90	75.0	A
ANGELEFORT	1980	RHONE	15.00	350.0	45000	6.40	107.0	A
BRENS	1981	RHONE	15.00	350.0	45000	6.40	107.0	A
BREGNIER-GORDON	1983	RHONE	11.40	350.0	35000	6.25	93.8	-
ABZAC	1958	ISLE	2.20	8.5	165.5	1.72	158.0	V-C
MARCKOLSHEIM	1957	RHINE	9.50	14.4	1205	1.60	333.3	V-C
RABODANGES	1959	ORNE	6.00	7.6	401	1.40	315.0	V-C
RHINAU	1960	RHINE	6.90	14.1	860	1.70	300.0	V-C
GERSTHEIM	1967	RHINE	11.10	235.5	23850	5.60	107.0	V-C
GERSTHEIM	1968	RHINE	9.00	14.0	1113	1.60	333.3	V-C
STRASBOURG	1970	RHINE	11.65	257.75	27100	5.60	100.0	V-C
STRASBOURG	1970	RHINE	14.50	-	29000	5.60	100.0	N
CASTET	1953	-	7.80	-	810	1.65	250.0	N
WADRINAU	1957	-	4.50	36.40	1487	3.05	107.0	N
SAINT-MALO	1959	-	4.80	-	9000	5.80	88.3	N
GERSTHRIM	1957	-	9.80	-	23000	5.60	107.0	N
BEAUCAIRE	1970	-	15.30	-	35000	6.25	93.8	N
GERVANS	1971	-	12.00	-	30000	6.52	93.8	N
AVIGNON	1973	-	10.50	-	30000	6.52	93.8	N
GAMBSHEIM	1974	-	13.20	-	24500	5.60	100.0	N
CHAUTAGNE	-	-	14.67	-	46600	6.40	107.0	N
BELLEY	-	-	14.70	-	46670	6.40	107.0	N

Table A1 (continued). B U L B   T U R B I N E S

POWER STATION	DATE OF COMMISSIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIAMETER (M)	RUNNING SPEED (RPM)	MANUFACTURER
GERMANY								
PALZEM	1964	MOSELLE	3.40	50.00	1500	3.60	78.0	MA
GREVENMACHER	1962	MOSELLE	5.50	59.00	2600	3.20	120.0	EW
TRIER(TREVES)	1958	MOSELLE	5.10	95.00	4400	4.60	78.0	EW
DETZAM	1959	MOSELLE	7.00	95.00	5800	4.20	92.5	EW
WINTRICH	1963	MOSELLE	5.60	95.00	4900	4.60	83.0	EW
ZELTINGEN	1964	MOSELLE	4.00	95.00	3300	4.80	67.0	MA
ENKIRCH	1965	MOSELLE	5.10	95.00	4300	4.60	79.0	MA
NEEF(ST.ALDEGUND)	1964	MOSELLE	5.50	95.00	4000	4.60	76.0	EW
FRANKEL	1962	MOSELLE	4.10	95.00	3700	4.60	77.0	V
MUDEN	1962	MOSELLE	4.10	95.00	3600	4.60	77.0	V
LEHMEN	1966	MOSELLE	5.30	95.00	4600	4.60	85.0	V
BUCKENHOFEN	1960	ILLER	5.20	35.00	1500	2.45	166.7	EW
FINSING	1961	-	10.60	35.00	3000	2.30	214.3	V
URSPRING	1963	LECH	8.10	52.00	3400	2.85	166.7	V
LECH 3	1963	LECH	9.20	47.50	4200	2.85	166.7	EW
SYLVENSTEIN	1960	ISAR	23.40	12.50	2500	1.46	452.0	V
IFFEZHEIM	1977	RHINE	11.70	267.50	27000	5.80	100.0	EW
LECHSTUFE 2	1968	LECH	15.20	52.30	7500	2.85	200.0	EW
LECHSTUFE 18	1973	LECH	12.80	47.50	6700	2.85	200.0	EW
LECHSTUF 23	1978	LECH	8.60	47.50	5000	2.85	187.5	EW
ISARWERK 3	1979	ISAR	4.50	32.50	1200	2.45	157.0	EW
LECHSTUFE 19	1980	LECH	8.70	47.50	4500	2.85	176.5	EW
LECHSTUFE 20	1984	LECH	9.40	47.50	4090	2.85	176.5	V
LECHSTUFE 22	-	LECH	9.77	47.50	-	2.85	176.5	V
GOTTFRIEDING	1977	ISAR	6.00	50.00	2710	2.92	135.0	V
REHLINGEN	1984	SAAR	7.60	30.00	2080	2.30	187.5	V
SCHODEN	1984	SAAR	5.70	30.00	1550	2.30	187.5	V

Table A1 (continued)

B U L B   T U R B I N E S

POWER STATION	DATE OF COMMIS-SIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIA-METER (M)	RUNNING SPEED (RPM)	MANUFACTURER
HUNGARY								
TISZA 2	1973	-	6.40	138.00	7200	4.30	107.0	GM
INDIA								
GANDAK	1966	-	6.10	112.00	5500	4.10	107.0	EW
KOSI	1984	-	7.70	-	5000	4.50	93.8	H
WESTERN YAMUNA	1982	-	-	-	-	-	-	
CANAL	1982	-	-	73.30	9080	3.15	187.5	FE
IRAK								
MOSUL 2	-	TIGRIS	10.50	16.00	-	5.00	115.4	V
ITALY								
FIORINO NUOVO	1966	PIAVE	16.50	62.00	9000	3.00	187.5	RA
IVORY COAST								
SAN PEDRO	1982	SAN PEDRO	9.80	30.00	2600	2.05	272.7	V-C

Table A1 (continued). BULB TURBINES

POWER STATION	DATE OF COMMISSIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIAMETER (M)	RUNNING SPEED (RPM)	MANUFACTURER
JAPAN								
HITOKITA	1959	NATCRI	12.00	12.50	1375	1.50	333.3	MI
KONAKAJIMA	1961	MABUCHI	9.20	29.00	2320	2.30	200.0	T
AKIRASHIMA	1964	TECORI	13.70	40.00	4800	2.30	240.0	MI
OMATA	1960	WACA	13.00	30.00	3350	2.20	200.0	FE
JOGANJIGAWA(NO.1,2,3,4)	1964	JOGANJI	15.10	40.00	5340	2.47	240.0	FE
TAGUCHI	1966	HIRCSE	12.40	58.20	6300	2.90	187.5	FE
KOIDE	1967	HIRCSE	12.90	78.10	8800	3.40	150.0	FE
YANAGIHARA	1967	HIRCSE	10.00	90.10	7850	4.00	125.0	T
HITOKITA	1959	NATORI	12.00	12.50	1375	1.50	333.0	MI
KOSHI	1959	SENDAI	8.00	22.00	1640	1.90	225.0	MI
SAIKAWA	1961	SAI	18.30	13.50	2216	1.43	450.0	FE
SHIMOAKA	1962	KITA	10.65	20.00	1840	1.84	240.0	FE
TAMAYODA 2	1964	ARA	16.80	30.00	4370	1.95	300.0	FE
MIZUKOSHI	1965	NISHIKI	12.12	12.00	1410	1.30	400.0	E/M
SEKINE	1967	HIRCSE	9.50	99.00	8200	4.00	125.0	T
KUROTORI	1968	NARIHA	10.21	26.00	2310	2.10	225.0	FE
ISHII	1975	CHIKUGO	13.74	10.00	1176	1.27	450.0	FE
KURCKAWA 2	1975	SHIRO	22.70	11.13	2194	1.27	600.0	FE
IKEDA	1976	YOSHINO	10.73	62.00	5200	3.13	150.0	E/M
AKAO	1978	SHC	17.40	220.00	34000	5.10	128.6	FE
FUTAKAWA	1979	SHIZUNAI	12.00	73.00	7300	3.40	150.0	T
ARAMAKI	1966	-	9.50	108.00	8200	-	125.0	T
SAKUMA 2	1982	TENRYU	12.30	122.00	16800	4.49	125.0	FE
KOREA								
NAM GANG	1972	-	8.70	93.00	6500	3.00	189.5	J
PALDANG	1972	-	11.80	200.00	21000	5.20	120.0	N

Table A1 (continued). BULB TURBINES

POWER STATION	DATE OF COMMIS-SIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIA-METER (M)	RUNNING SPEED (RPM)	MANUFACTURER
NORWAY								
GAMLEBROFOSS	1970	LAGEN	14.10	110.0	15610	4.20	150.0	KMW
KLOSTERFOSSEN	1969	SKIENSELVEN	5.03	119.0	5330	4.50	85.7	KHARKOV
ASMUDFOSS	1971	NAMSEN	10.00	135.0	12500	4.30	125.0	KB
FUNNEFOSS	1975	GLOMMA	10.30	220.0	20000	5.20	100.0	KB
KONGSVINGER	1975	GLOMMA	9.16	240.0	19100	5.50	93.8	KB
DOVIKFOSS	1975	DRAMMENSELVA	5.85	300.0	14700	6.40	75.0	KMW
O.FISKUMFOSS	1976	NAMSEN	6.20	130.0	6700	4.30	107.5	KB
BINGFOSS	1976	GLOMMA	5.00	250.0	10800	6.05	71.4	KB
BRASKEREIDFOSS	1978	GLOMMA	9.17	270.0	22200	5.80	88.2	KB
POLAND								
CIECHOCINEK	1984	LOWER	5.10	375.0	16800	7.10	65.2	
PORTUGAL								
CRESTUMA	1984	DOURO	10.25	423.0	39000	6.80	93.75	N
BELVER	1980	TAJO	14.20	267.5	35300	6.00	100.0	EW
RAIVA	1980	MONDEGO	16.00	75.0	12840	3.30	200.0	EW
ROMANIA								
IRON GATES 2	1984	DANUBE	7.40	425.0	28000	7.50	62.5	LMZ
SPAIN								
CHERTA	1984	-	11.00	296.0	26000	5.90	-	-
GARCIA	1984	-	8.00	270.0	17200	5.90	-	-
SANTIAGO-DEL-SIL	1965	SIL	12.00	86.0	8300	3.30	157.5	EW

Table A1 (continued).   B U L B   T U R B I N E S

POWER STATION	DATE OF COMMIS-SIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIA-METER (M)	RUNNING SPEED (RPM)	MANUFACTURER
SUDAN								
KHASM-EL-GIRBA	1967	ATBARA	7.00	50.0	2800	2.70	150.0	R
SWEDEN								
SKOGSFORSEN	1959	ATRAN	14.00	29.0	3700	2.18	250.0	KMW
HALLEFORS	1966	SVARTALVEN	7.50	32.0	2180	2.45	190.0	KMW
SPERLINGSHOLM	1967	LAGAN	3.70	25.0	800	2.45	125.0	KMW
PARKI	1970	LULEALVEN	11.00	168.0	21200	4.90	115.4	KMW
LOVCN	1973	FAXALVEN	13.80	160.0	19800	4.50	136.4	NO
GULLSPANG	1972	GULLSPANGSALVEN	21.00	6.0	1200	0.90	750.0	KMW
VITTJARV	1974	LULEALVEN	5.60	250.0	12300	5.80	75.0	KMW
GADDEDE	1973	STROMS	15.00	180.0	24300	4.50	136.4	KMW
BAGEDE	1974	VATTUDAL	9.30	160.0	13300	4.50	125.0	KMW
BODUM	1975	ANGERMANALVEN	6.50	225.0	13000	5.80	75.0	KMW
FJALLSJO	1976	ANGERMANALVEN	6.80	220.0	13200	5.80	79.0	KMW
SIL	1976	ANGERMANALVEN	6.40	225.0	12800	5.80	79.0	KMW
LANDAFORS	1976	LJUSNAN	5.30	350.0	16200	6.40	68.2	KMW
LJUSNEFORS	1976	LJUSNAN	6.70	340.0	19800	6.40	75.0	KMW
ASELE	1981	ANGERMANALVEN	10.10	320.0	28300	6.10	93.0	KMW
SODERFORS	1979	DALAVEN	4.50	220.0	9400	6.10	62.5	KMW
JUVELN	1978	INDALSALVEN	11.00	150.0	15700	4.20	136.0	KMW
TORRON	1978	DALSALVEN	19.00	165.0	31600	4.50	150.0	KMW
NAS 1	1979	DALALVEN	5.20	230.0	14700	5.80	75.0	KMW
AVESTALILLFORS	1982	DALALVEN	5.30	250.0	14300	6.10	68.2	KMW
MATFORS	-	-	9.45	250.0	23000	5.60	93.0	KMW
LILLA EDET 4	1982	GOTA ALV	6.50	280.0	18000	6.10	75.0	KMW
NAS 2	1980	DALALVEN	5.20	230.0	14700	5.80	75.0	KMW
GRANBOFORSEN	1980	INDALSALVEN	6.00	220.0	15200	5.80	75.0	KMW
WINZNAU	1962	AAR	5.50	4.85	235	1.06	383.0	V-C
TASJO	1978	FAJALLS-JOALV	12.00	125.0	13530	4.10	150.0	TAM
HOTING	1978	FAJALLS-JOALV	10.40	165.0	15340	4.60	125.0	TAM
VIFORSEN	1982	LJUNGAN	7.30	150.0	9715	4.60	107.0	TAM

Table A1 (continued).

B U L B   T U R B I N E S

POWER STATION	DATE OF COMMIS-SIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIA-METER (M)	RUNNING SPEED (RPM)	MANUFACTURER
SWITZERLAND								
RUCHLIG	1962	BUNZE	3.30	60.0	1600	3.70	75.0	EW
AUE	1963	LIMMAT	5.50	38.0	1700	2.70	136.4	EW
FLUMENTHAL	1965	AARE	7.50	133.0	8000	4.20	107.0	EW
NEU-BANNWIL	1965	AARE	8.10	116.7	8420	4.20	107.1	EW
ZUFIKON	1971	REUSS	10.93	100.0	10060	3.80	150.0	EW
	-	-						
USA								
ROCK ISLAND	1978	COLUMBIA	12.10	481.00	54000	7.40	85.7	CL
VACEBURG	-	-	9.40	360.00	24000	6.10	90.0	-
RACINE	1980	OHIO	6.23	443.50	24600	7.70	62.1	EW
MERCED MAIN CANAL	1981	-	-	43.20	2830	2.50	180.0	FE
IDAHO FALLS	1981	SNAKE	5.50	165.0	8300	4.85	94.7	VA
DAWSON	1982	-	5.50	96.3	4660	3.87	120.0	FF
LAWRENCE	1981	-	5.80	-	7600	4.00	128.6	AL
PELTON REREG.	1982	DESCHUTES	10.60	170.0	16000	4.85	112.5	VA
W. T. LOVE	1982	-	8.63	-	24300	6.10	90.0	N

Table A1 (continued). B U L B T U R B I N E S

POWER STATION	DATE OF COMMISSIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIAMETER (M)	RUNNING SPEED (RPM)	MANUFACTURER
USSR								
KISLAYAGUBSK	1961	-	2.50	19.10	400	3.30	92.0	N
KIEV	1966	DNIEPER	7.70	290.00	23000	6.00	85.7	KHARKOV
KISLOGUBSKAYA	1965	-	1.28	.	400	3.30	72.0	N
KAMA	1968	-	21.00	130.00	21800	4.50	125.0	LMZ
PEREPAD	1972	-	11.20	230.00	20600	5.50	93.8	LMZ
SARATOV	1972	VOLGA	10.60	528.00	47300	7.50	75.0	LMZ
KANIEV	1972	-	8.40	240.00	18200	6.00	85.7	KHARKOV
TCHEREPOVETZ	1967	-	15.00	175.00	21000	5.50	93.8	LMZ
YUGOSLAVIA								
IRON GATES 2	1984	DANUBE	7.40	425.00	28000	7.50	62.5	-
CAKOVEC	1979	DRAVA	18.55	250.00	42240	5.40	125.0	N & L

MANUFACTURERS:

ALLIS = ALLIS CHALMERS; A = ALSTHOM; AD = ANDRITZ; B = BATIGNOLLES; BR =BREGUET; CL = CREUSOT-LOIRE;

E/M = EBARA/MEIDENSHA; EW = ESCHER WYSS; FE = FUJI ELECTRIC; GM = GANZ MAVAG; H= HITACHI; J = JEUMONT;

JS = JEUMONT-SCHNEIDER; KB = KVAERNER BRUG; KMW = KARLSTADS MEKANISKA VERKSTAD;

LMZ = LENINGRAD METAL WORKS; MA = MAIER; MI = MITSUBISHI; S = SFAC (STE DES FORGHES ET ATELIERS DU CREUSOT);

N = NEYRPIC; NO = NOHAB; R = RIVA; SW = SCHNEIDER-WESTINGHOUSE; T =TOSHIBA; VAL = VOEST-ALPINE;

V = VOITH; V-C = VEVEY-CHARMILLES;

Table A2. TUBULAR TURBINE DATA

POWER STATION	DATE OF COMMIS-SIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIA-METER (M)	RUNNING SPEED (RPM)	MANUFACTURER
FINLAND								
OKSAVA	1975	KALAJOKI	10.5	28.0	2610	2.40	250.0	TAM
KALLIOKOSKI	1976	PYHAJOKI	6.0	13.0	633	1.65	222.0	TAM
KALAJARVI	1976	SEINAJOKI	13.5	15.0	1802	1.72	300.0	TAM
HERRFORS	1978	AHTAVANJOKI	4.0	12.0	410	1.72	167.0	TAM
FINNHOLM	1978	AHTAVANJOKI	6.0	12.0	635	1.72	222.0	TAM
PADINGINKOSKI	1979	KALAJOKI	4.0	30.0	1040	2.65	141.0	TAM
KATTILAKOSKI	1979	AHTAVANJOKI	10.5	27.0	2540	2.20	250.0	TAM
SOININKOSKI	1980	KOKEMAENJOKI	7.5	22.0	1433	2.20	200.0	TAM
HATTAR	1981	AHTAVANJOKI	6.1	20.0	1080	2.20	179.0	TAM
KANNUSKOSKI	1957	-	4.6	-	230	-	250.0	TAM
SIIKAKOSKI	1959	-	3.4	-	1015	-	105.0	TAM
KUSIANKOSKI	1962	-	8.8	-	250	-	500.0	TAM
HANHIKOSKI	1967	-	7.06	-	755	-	250.0	TAM
KLAGARO	1981	-	3.1	-	2215	-	88.0	TAM
NEW ZELAND								
MONTALTO	1980	RANGITATA	7.1	31.0	2000	2.65	159.0	TAM
NORWAY								
BLAFALLI	-	MATREFJORDEN	27.0	36.7	8750	2.09	333.3	V-C
FLATENFOSS	1981	NIDELV	10.0	60.0	5340	3.20	167.0	TAM
ROSTEFOSSEN	1969	-	9.5	-	1545	-	280.0	TAM
MAGO A	1984	ANDELVEN	7.2	12.0	770	1.72	214.0	TAM

Table A2 (continued). TUBULAR TURBINE DATA

POWER STATION	DATE OF COMMIS-SIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M^3/s)	RATED CAPACITY PER UNIT (KW)	RUNNER DIA-METER (M)	RUNNING SPEED (RPM)	MANUFACTURER
SWEDEN								
KALSATER	1976		6.8	-	500	-	306.0	TAM
HATTORP	1976		24.0	-	800	-	765.0	TAM
KNISLINGE	1976		4.0	-	310	-	273.0	TAM
SWITZELAND								
LESSCC	1973	SARINE	20.7	16.1	2940	1.7	432.0	V-C
KALLNACH	1980	AAR	17.5	45.6	7050	2.5	250.0	V-C
USA								
BAKER MILL	-		14.6	-	1491	1.5	306.0	ALLIS
SAWMILL	-		5.5	-	760	2.0	-	ALLIS
SAWMILL	-		5.3	-	827	2.0	-	ALLIS
CORNELL	1976	CHIPPEWA	11.0	-	10400	4.65	100.0	ALLIS
DOLBY	-		14.6	-	4400	2.29	212.0	ALLIS
TRAICAO	-		7.0	-	257	-	-	ALLIS
TRUMAN	-		13.0	138.0	31500	6.5	-	ALLIS
LOWER PAINT	-		6.1	-	116	0.75	533.0	ALLIS
WISCONSON	-		6.7	-	2000	2.7	150.0	ALLIS
TURNIP CHECK	-		5.0	-	420	1.5	218.0	ALLIS
SWIFT RAPID	-		14.3	-	2500	2.0	277.0	ALLIS
10TH STREET	-		4.7	-	1440	2.75	128.6	ALLIS
OZARK LOCK	-		9.8	-	25200	8.0	60.0	ALLIS
WEBBERS FALLS	-		8.1	-	30100	8.0	60.0	ALLIS
IMPERIAL VAL	-		6.9	-	2100	2.5	176.0	ALLIS
P.E.C.22.7	1981	COLUMBIA	15.8	50.0	6500	2.6	225.0	TAM
ASHOKAN	1982		21.3	12.7	2430	1.4	400.0	TAM
KENNEBUNK	1980		5.5	7.4	300	1.22	323.0	ALLIS
ARMY CORP	-		15.2	138.0	31500	13.0.	-	ALLIS

MANUFACTURER:

ALLIS = ALLIS CHALMERS; TAM = TAMPELLA; V-C = VEVEY-CHARMILLES;

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Idaho Water and Energy Resources Research Institute Completion Report, Moscow, December 1982. 104 p., 42 fig., 8 tab., 23 ref.

16. Abstract
This report contains the results of an extensive investigation of the characteristics of over two hundred low-head turbines manufactured all over the world that have been installed or are due to be installed in hydropower plants between 1953 and 1984. The research focused mainly on bulb turbines, with horizontal shaft arrangement, tubular turbines with their shafts either horizontal or inclined at an angle to the horizontal. The characteristics of the above mentioned type of turbines are presented in the form of statistical diagrams and regression equations suitable for preliminary design and feasibility studies of low-head hydro projects. Nomographs have been developed for displaying the relationships between the various turbine characteristics and comparing the important dimensions and parameters of turbines which have found common application in the hydropower technology. New simplified parametric ratio for selection of turbines have been developed that should expedite preliminary selection study for hydropower projects.

17a. Descriptors
Hydroelectric Plants, Turbines, Avial Flow Turbines, Bulb Turbines, Tube Turbines.

17c. COWRR Field & Group
08A and 06B

18. Availability
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19. Security Class. (Report)

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