Research Technical Completion Report

EXPERIENCE CURVES FOR MODERN LOW-HEAD HYDROELECTRIC TURBINES

by

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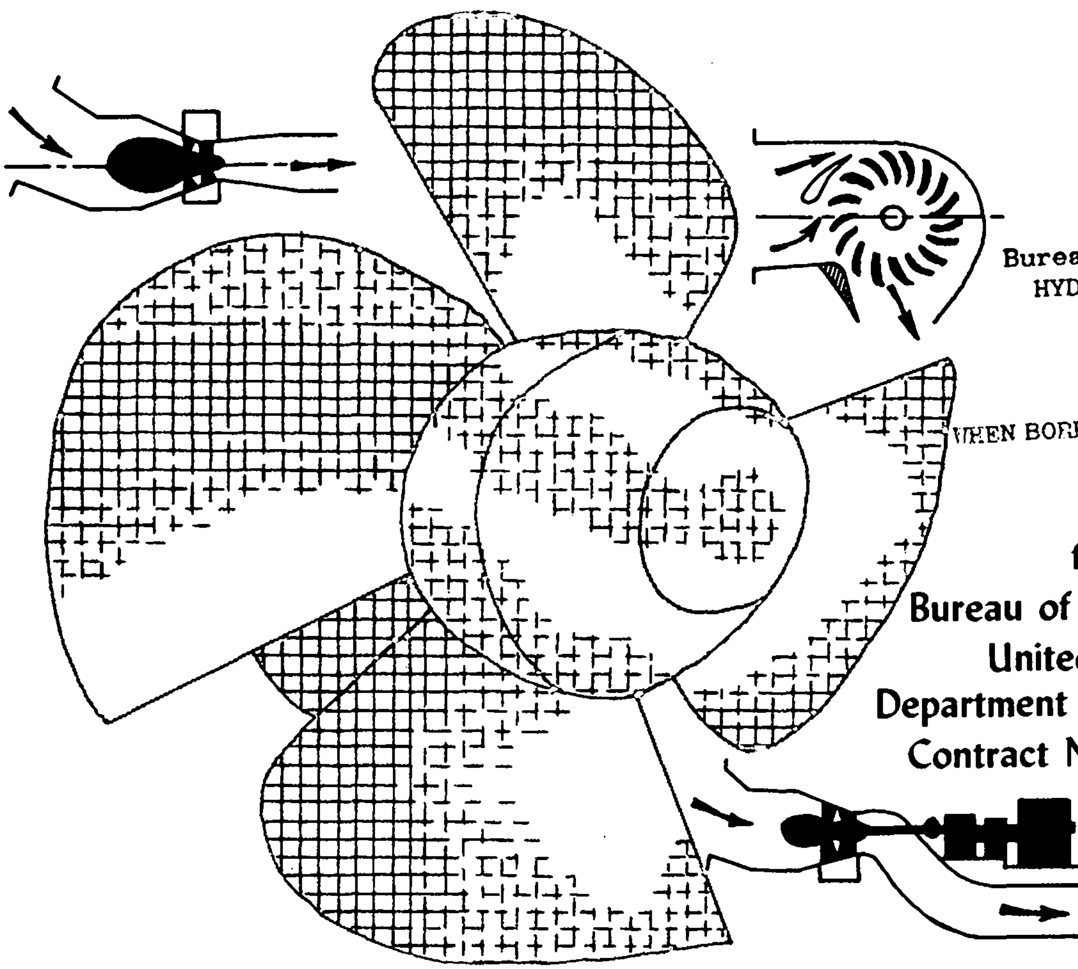

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Correction:

The "Correlation Coefficient" used in this report is r^2 instead of r which is shown on the nomographs and tables. r^2 as used measures how much variation in the dependent variable can be explained by the model. r^2 can range from 0 to 1, see page 11.

FOREWARD

This study of the characteristics of manufactured hydroelectric turbine equipment in the form of experience curves is presented to make available information and experience that can be used in planning and preliminary design of hydropower developments. It is intended to supplement material already available for the more conventional hydraulic turbines and therefore concentrates on information about low-head type turbines. In the tradition of the Idaho Water and Energy Resources Research Institute the report has been prepared to meet a need and desire of government agencies and practicing professional engineers involved in hydropower engineering.

ACKNOWLEDGEMENTS

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Most of the data used in this report came from a number of manufacturers of hydroelectric equipment. To name all who contributed data in this acknowledgement is not possible, however, a listing in the Appendix does give the names and addresses of all the manufacturers contacted in connection with the study. A very special thanks goes to all the firms that contributed, especially to representatives of several of the firms that took time to explain to the authors their approaches to selection of turbines.

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The report has been prepared under supervision of Dr. James H. Milligan as Chairman of the Department of Civil Engineering and Dr. John R. Busch as Director of the Idaho Water and Energy Resources Research Institute.

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ABSTRACT

This report contains the research findings of an extensive investigation of characteristics of over 300 low-head hydraulic turbines that have been manufactured all over the world. These results are presented in the form of experience curves and regression equations relating the traditional turbines constants of specific speed, speed ratio, unit power, and cavitation coefficient to such parameters as rated head, rated discharge, rated power output, runner speed, and runner diameter. Additional information on the characteristic dimension of the water passages is also presented. Traditional methods of estimating turbine diameter and turbine speed have been checked with actual practice and new simplified methods for estimating turbine diameter and turbine speed have been proposed and verified.

A comparison has been made as to how well the draft tube exit velocities on manufactured units are complying with recommended limits. Rather limited success was obtained in characterizing the turbine setting parameter and its relation to the specific speed. Excellent comparisons were possible with published regression relations and experience curves of conventional reaction turbines.

KEY WORDS

BT - Hydraulic Turbines, Power Plants, Turbines, Turbine Runners
NT - Axial Flow Turbines, Bulb Turbines, Tube Turbines, Impulse Turbines (cross-flow)
RT - Draft Tubes, Hydroelectric Plants

SUMMARY

This report presents information on experience curves and empirical relations useful in the preliminary planning of hydroelectric power plants and their components based on actual manufactured and operating units. The objectives of the study were to develop up-to-date relations for low-head hydropower turbines giving (1) relations of specific speed to design head, (2) relations of turbine runner diameter to design head, rotational speed, and velocity ratio, (3) draft head relations to specific speed and cavitation coefficient and (4) empirical relations of physical dimenions of flow passage dimensions of intake and draft tube areas to the turbine runner diameter.

Data for making the study were obtained by personal contact of the authors in visits to over twelve manufacturers of turbines, by careful review of existing technical literature, and by extensive correspondence with over thirty manufacturers of hydroelectric turbines. A careful assessment was also made of the literature on simulitude laws and turbine constants that have been extensively used in the hydraulic machinery field. Much reference and comparison have been made to the U.S. Bureau of Reclamation Monograph No. 20 which has wide acceptance and use in the planning and feasibility field by both public agency engineers and by consulting engineers. Contact with over 200 different consulting engineers by Professor Warnick has likewise been used as a basis for judging and determining the approaches that are currently used in professional practice. The ultimate goal of the study has been to present useful procedures that can be authoritatively accepted by the engineering profession and provide for a more uniform and consistent preliminary selection of hydraulic turbines.

The basic approach of the analytical portion of the study has been to make regression analyses of the data collected on various turbine characteristics used in hydropower planning. The regression approach used was that of relating one independent parameter to a dependent parameter, or to two parameters expressed as a single ratio. The curve fitting utilized a logarithmic equation of the form:

$$\log y = \log A + m \operatorname{Log} X.$$

Sets of data were analysed on a computer system known as Statistical Analysis System (SAS).

The study centered on three types of turbines, (1) the bulb type units, (2) the tubular type units, and (3) the cross-flow units (See Figures 1 and 2). The results are presented in four distinct contributions: (1) Experience curves and regression equations were developed for relating specific speed to rated head and similar regression equations were developed between the various standard turbine constants (see Tables 2, 3 and 4), (2) Relations were developed for determining a cavitation coefficient that is used in choosing the turbine setting (see Table 5), (3) Experience curves were developed for estimating water passage dimensions and referencing those dimensions to the nominal diameter of the turbine (see Figures 48 to 69), and (4) speed and diameter selection procedures were assessed and compared with published information on propeller turbines and new procedures developed for making speed and diameter selection at the feasibility stage of planning.

The new selection procedures are presented in the form of nomographs and comparative experience curves beginning with Figure 71 and continuing to Figure 77. Sample calculations on how to apply the experience curves are presented in Appendix 2. The conclusion is made that these procedures are simpler and more direct than conventional procedures now in use and appear to offer more consistent results. The compilation of data on manufactured low-head turbines should offer an excellent reference in itself for designers and planners to use in preliminary design and feasibility studies.

Because this study applied to only low-head turbines and also because new data on manufactured units are now available on conventional Kaplan, Francis and Pelton type turbines, it is recommended that the new methodology developed on this study be used to update experience curves and selection procedures for those types of turbines used in higher head applications.

INTRODUCTION

In planning and design of hydroelectric plants much advantage is gained by utilizing the experience gained from the various installations that have already been made. Publications like Engineering Monograph No. 20 of the U.S. Bureau of Reclamation (1976) entitled, "Selecting Hydraulic Reaction Turbines" have been developed for this purpose. Records of experience have been analysed and various experience curves and empirical equations developed that provide a convenient way to proceed in planning for new hydropower developments. Experience curves provide a way of making visual comparison easily and with engineering judgement help the engineer in proceeding through the complex task of planning and designing a hydropower development. These do not substitute for the design selection that a turbine manufacturer must make to proceed to final design. Experience curves however, do provide the planning engineer with useful information to proceed with feasibility and preliminary design studies.

Modern low-head hydroelectric turbines such as tubular turbines, bulb type installations, and cross-flow turbines have now been in production long enough to provide enough operating units from which experience curves can be generated. The work of de Siervo and de Leva (1976 and 1977) and de Siervo and Lugaresi (1978) treating conventional Francis turbines, vertical Kaplan turbines, and Pelton turbines did not consider the more modern low-head type turbines, neither did the Engineering Monograph No. 20.

OBJECTIVE

The objective of this report is to provide experience curves and practical empirical equations useful in planning and preliminary design of hydroelectric developments for modern low-head type turbines. Specifically, to provide information on bulb type turbines, tubular type turbines, and cross-flow turbines that have been manufactured in the past thirty years. Particular relationships to be developed would provide information on the following:

1. Specific speed relation to design head.
2. Turbine runner diameter relation to design head, rotational speed, and velocity ratio.
3. Draft head relation to specific speed and cavitation coefficient.
4. Physical dimensions of flow passages (intake and draft tube) relations to turbine runner diameter.

EXPERIENCE CURVES AND TURBINE CONSTANTS

Historically a series of turbine constants have been developed by using similarity laws of hydraulics and fundamental hydraulic equations to characterize the performance of hydraulic turbines. Mathematical development of the various constants is covered in texts by Barrows (1927), Doland (1954), Csanady (1964), Warnick (in press), and in an M.S. thesis by Kpordze (1982). A worthwhile discussion on different expressions for turbine constants is presented by Barr (1966). Recently international manufacturers have suggested an approach that reports the various constants in dimensionless form (Allis Chalmers, no date). Table 1 presents expressions for different forms of the various turbine constants in use and the new dimensionless system of expressing the turbine constants. This table contains a list of terms used in the report along with appropriate units in which the terms are expressed. The American system reports the constants in units of power output as horsepower, diameter of runner in inches, turbine discharge in ft^3/sec, head in feet, and rotational speed in rpm. The European system reports the constants in units of power output in kilowatts, diameter of runner in millimeters, turbine discharge in cubic meters per second, head in meters, and rotational speed in rpm. The European system has been used throughout this report because so much of the manufacturer's literature and experience curves that have been reported have been published in the European system. Conversions and relationships between the different forms of the turbine constants are provided in Table 1 and in an example in the Appendix demonstrating the use of the conversions.

Manufacturers who have worked with these constants and model tests have further utilized the constants to develop multiparameter relations termed "Hill Curves." These hill curves are proprietary information and therefore are not available to practicing engineers for use in selection and design. In practice many engineering firms develop their own experience curves and once developed the curves are made proprietary information of the firm. In this effort the experience curves and empirical equations are being proposed as a way to achieve more consistency in the planning studies and to provide a better and more uniform base for proceeding with engineering design. In a sense it does provide a check as to the recommendations and quotations of performance that are put forth by the manufacturers who may be asked to bid on and supply hydraulic turbines.

The types of turbines studied are of two general types, reaction turbines and impulse turbines. Three reaction type turbines were studied: bulb type units, tubular type units and rim-generator units. Typical representation of these units are shown in Figure 1. The

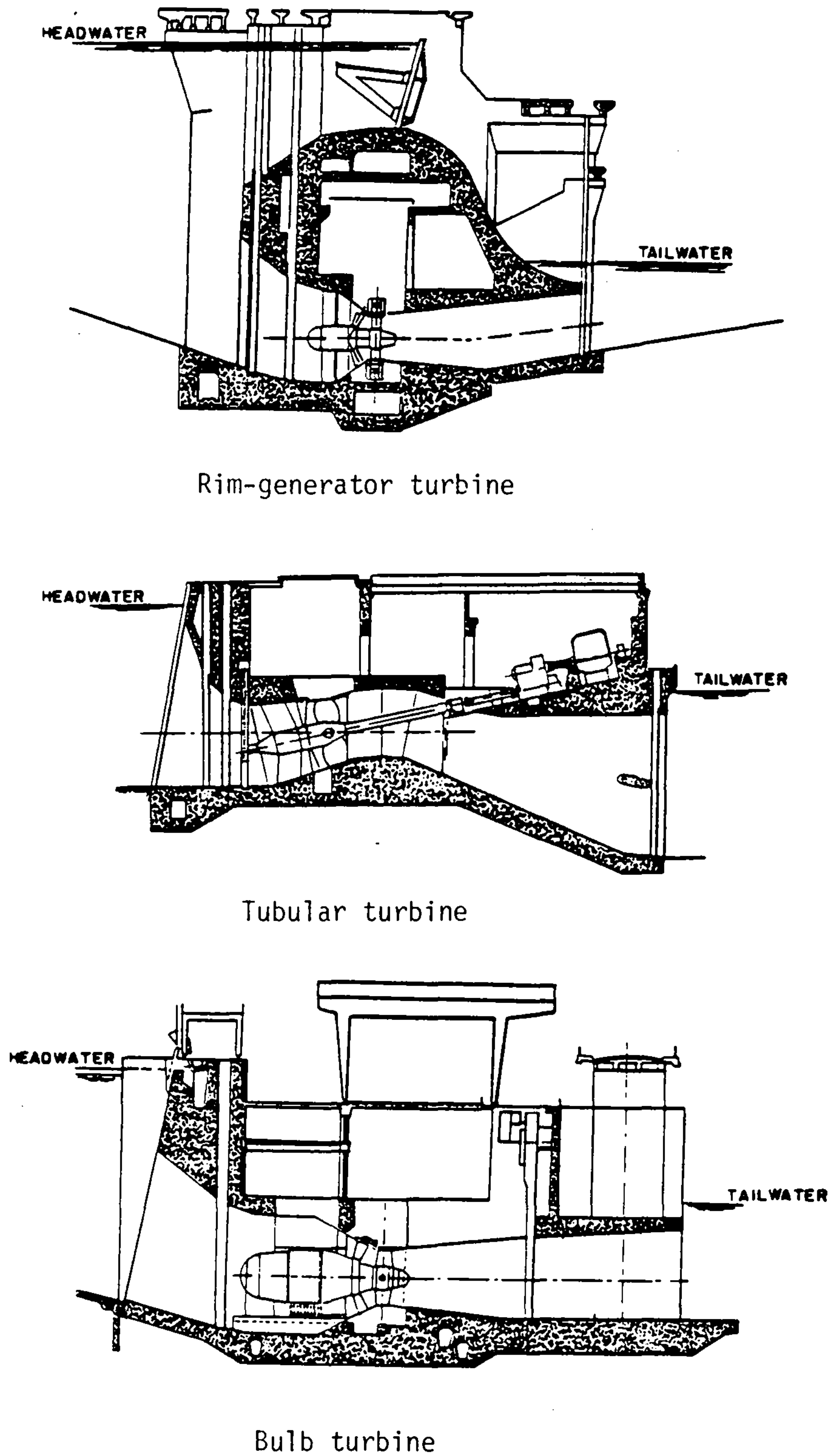

Figure 1. Schematic drawings of three types of low-head turbines of the reaction type.

impulse turbine studied was a cross-flow turbine. Figure 2 is a line drawing representation of the cross-flow type turbine.

COLLECTION AND ORGANIZATION OF DATA

DATA COLLECTION

Collection of data was initiated first on this project when one of the authors, Professor Warnick, contacted numerous turbine manufacturers in connection with preparation of a new textbook on hydropower engineering. This included reference lists and characteristics of turbines manufactured by various turbine manufacturers. These personal contacts have continued since that time and during the course of the present research contract, several manufacturers were visited. A table in the Appendix gives the list of manufacturers visited, a contact name, and the address and the then active telephone number. On these visits company literature particularly concerned with selection of turbines was collected. A complete set of this manufacturer's information has been assembled for the Bureau of Reclamation as a reference document. Much of this document includes nomographs published by the companies for use in selecting turbines and for providing preliminary data on dimensions of standard turbines and water passages of the civil works portion of hydropower installations.

The technical literature was searched for data on turbines and representative of this is the technical articles like that of de Siervo and de Leva (1977 and 1978) and also a listing of information prepared by Cottillon (1977, 1979, and 1981).

Subsequent to the literature search and the initial personal visits of Professor Warnick, considerable correspondence was carried on to complete the collection of data. In some cases there were no

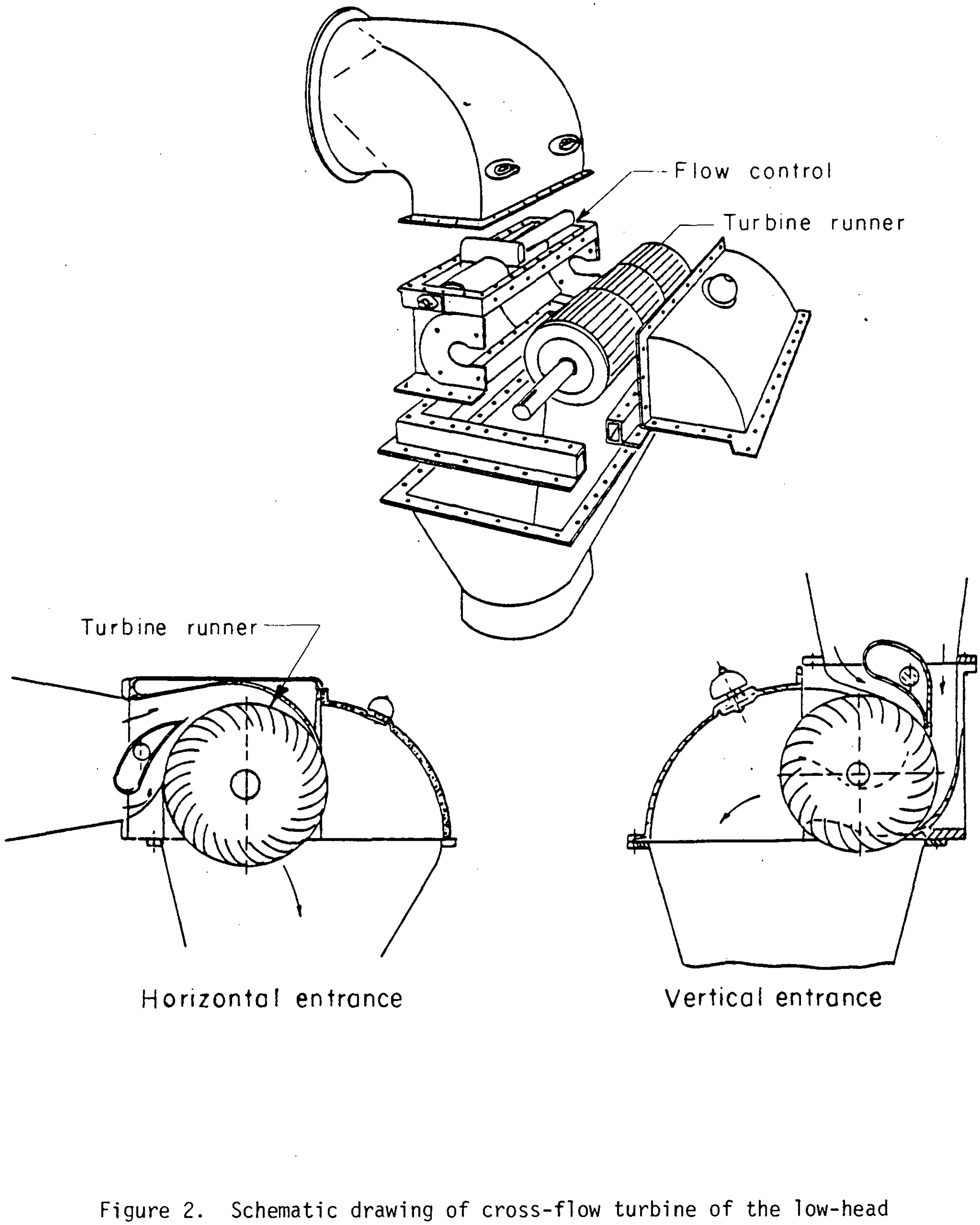

Figure 2. Schematic drawing of cross-flow turbine of the low-head impulse turbine type.

replies but in general good response was obtained in acquiring missing data and clarifying information that was obtained in personal contacts or from published reference lists.

ORGANIZATION OF DATA

All information that was received was first checked to verify consistency and identify appropriate measurement units. Transformation of all units were made to make all units compatible with the European system of reporting turbine constants. Data were then entered in a computer file that would permit easy access for analysis. This information included type of turbine, name of manufacturer, name of power station, date of commissioning, rated head, rated flow, rate capacity per unit, runner diameter, unit rotational or running speed and specific water passage dimensions designated by letters of identification. A complete list of all the data used or obtained during the study is reproduced as tabular material in the Appendix 3.

Once a standardized file of the various data was prepared then computer programs were developed to extract the data in various stratifications as to a particular type of turbine, a particular manufacturer, or a particular year of commissioning. These computer programs are filed in the Appendix 4 to permit future researchers to proceed with analyses of additional data.

METHODS OF ANALYSIS

The study basically entailed classifying and analysing different sets of data from various manufacturers and data reported by the numerous companies. Different statistical procedures were used in proceeding with the analysis. One such statistical procedure is cluster analysis.

The cluster analysis is a means of classifying observation (in this case turbine characteristics) on the basis of similarity (Anderberg, 1973). Cluster analysis in this research was used to group the turbine data into periods of similar turbine design characteristics. 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 221 bulb turbines manufactured all over the world, was treated as a 4 x 221 matrix. The four turbine characteristics used were: specific speed, rated head, unit discharge and unit power. Using a computer, the 4 x 221 matrix was partitioned 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 221 - 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 nearest previously computed values. Then the first period of analysis was taken as the sample period corresponding to the highest value of correlation coefficient. 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 221 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.

STATISTICAL METHOD OF DATA ANALYSIS

The data used in developing the experience curves resulted from the measurement of a number of variables and came from different sources and were collected under a variety of conditions. In order to describe the relationship existing between such variables, the standard procedure is to formulate a statistical hypothesis setting forth the explicit mathematical form of the relationship between the variables. A common assumption is that the relationship between two variables, for example, X and Y or the transformations of X and Y is linear. Having assumed linearity, our objective then is to specify a rule by which the "best" straight line fitting X and Y is to be determined. The "line of best fit" is said to be that which minimizes the sum of the squared deviations of the points of the graph from the points of the straight line (with distances measured vertically). The general method of finding equations for approximating curves which fit given sets of data points plotted on a rectangular coordinate is known as curve fitting. One of the main purposes of curve fitting is regression which is the process of estimating the variable Y (dependent variable) from the variable X (independent variable). If Y is to be estimated from X by means of some equation, the equation is called the regression curve of Y on X. The degree of relationship between variables is known as correlation. 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) and (Pindyck and Rubinfeld 1981). Sometimes it helps to plot the scatter diagrams in terms of transformed variables. For example if Log Y leads to a straight line, $\log Y = a + bX$ will be used as an equation for the approximation curve. The type of equations used in this study are:

Linear regression: $Y = a + bX$

Exponential curve fit: $Y = ae^{bx}$

Power curve fit: $Y = aX^b$

Logarithmic curve fit: $Y = a + \log_{10} X$

Where a, b and e are constants.

The degree to which numerical data tend to spread about an average value is called the variation or dispersion of the data. One of the most common measures of dispersion is the standard deviation, s. The standard deviation of a set of N numbers x_1, x_2, ...x_j is defined by the expression:

$$s = \left(\sum_{j=1}^{N} (x_j - \overline{x})^2 / N \right)^{0.50}$$

which is the root square mean deviation and x is the arithmetic mean. In the graphical representation of the curve, if parallel lines to the regression line of Y on X are constructed at respective vertical distances s, 2s, and 3s from the regression line, statistical theory states that there would be included between these lines 68%, 95% and 99.7% of the sample points, respectively. This is true only if the numbers of data points, N, is large enough. The symbols with the s, 2s, and 3s lines are referred to as one-, two-, and three standard deviations respectively.

The measure of how well a straight line explains the relationship between two variables X and Y is the correlation coefficient, r and it is expressed as the square root of the ratio of the explained variation to the total variation. $\left(\Sigma(\hat{Y} - \overline{Y})^2 / \Sigma(Y - \overline{Y})^2 \right)^{0.50}$ where $\hat{Y}$ is the estimated value of Y from the regression equation and Y is the arithemetic mean value. Values of r = 1 or r = -1 denote perfect correlation. The above defined statistical concepts have been used in the data analysis and were embodied in the computer system used in the studies and plotting the resulting experience curves.

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.

Traditionally the turbine constants specific speed, N_s, and the speed ratio, $\emptyset$, are used to select the appropriate type of turbine and with developed empirical equations estimates are made of turbine runner diameter and turbine speed. These turbine constant terms of N_s and $\emptyset$ are defined mathematically in Table 1 and procedures for using the constants in preliminary design and feasibility studies are illustrated in sample calculations in Appendix 2. Among the procedures illustrated in the sample calculations is the method used in the U.S.B.R. Monograph No. 20 for estimating turbine runner diameter and turbine speed. Other turbine constants such as unit speed, unit power, and unit discharge, that are used to report turbine test data were also calculated for the manufactured units and analyses were made to develop regression

Table 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	$k_u = \frac{D_3 N}{60(2gH)^{0.50}}$	ω_{ed}	$\omega_{ed} = \frac{\omega D}{(gH)^{0.5}}$
Unit speed	n_1	$n_1 = \frac{dn}{h^{0.5}}$	N_{11}	$N_{11} = \frac{DN}{H^{0.5}}$	ω_{ed}	$\omega_{ed} = \frac{\omega D}{(gH)^{0.5}}$
Unit discharge	q_1	$q_1 = \frac{q}{d^2\ h^{0.5}}$	Q_{11}	$Q_{11} = \frac{Q}{D^2 H^{0.5}}$	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_{11}	$P_{11} = \frac{P}{D^2 H^{.5}}$	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		$n_s = 0.262\ N_s$		$N_s = 166.\ \omega_s\ \eta^{0.5}$		$\omega_s = \frac{n_s}{43.5\ \eta^{0.5}}$

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 η = efficiency.

relations between the different constants and the basic parameters of rated head, rated power output, rated discharge, turbine speed, and turbine diameter.

In this study emphasis was directed toward relations of specific speed to rated head, speed ratio to specific speed, and the relation of these constants to actual runner diameter and actual runner speed the same as was used in the approach defined in the U.S.B.R. Monograph No. 20.

RESULTS

The results are presented in three main classifications and further subdivided into subclassifications. The first classification presents results relating to characteristics of the turbines and the turbine diameter in relation to parameters of rated head, rated discharge, rated output, and rotational speed of the turbine. This treats relationships and interelationships concerned with the turbine constants, specific speed, unit speed, unit power, velocity ratio, unit discharge, and some new alternative ratios as parameters.

The second classification presents information on draft head, suction head, specific speed, and cavitation coefficient. The third classification is concerned with turbine constants and the characteristic dimensions of the water passages of the civil works portions of the hydropower installations. This includes relating dimensions of the entrance works leading up to the turbine and dimensions of the draft tube to the turbine constants.

Under each of these classifications subclassification information is presented on the three different types of turbines: (1) bulb type units, (2) tubular type units, and (3) cross-flow type units. Information on rim-generator type units was insufficient to make any meaningful analyses.

TURBINES CHARACTERISTICS

The most common experience curve is obtained by relating the specific speed, N_S, to the rated head, H. Cluster analyses was performed and the data stratified according to the time of commissioning.

Bulb Turbines

For bulb type turbines the N_s vs H relation is shown in Figure 3, where three different curves representing three different time periods of manufacturing are given by the following regression equations:

$$N_s = 1155.937\, H^{-0.346} \qquad (1953\text{-}1960) \qquad \text{Eq. (1)}$$

$$N_s = 964.130\, H^{-0.1631} \qquad (1961\text{-}1970) \qquad \text{Eq. (2)}$$

$$N_s = 1520.256\, H^{-0.2837} \qquad (1971\text{-}1984) \qquad \text{Eq. (3)}$$

where $N_s = \dfrac{N\, P^{0.5}}{H^{1.25}}$ Eq. (4)

N = rotational speed in rpm

P = rated power output in KW

H = rated head in m.

A further stratification of the N_s vs H relationship showing the variation of the relation for various turbine manufacturers is presented in Figure 4 for all bulb turbines for which data were obtained. Summaries of the data from individual manufacturers is presented in Appendix 3 along with the specific regression equations.

Figure 5 presents the relation between specific speed, N_s, and unit power, P_{11}, for all bulb turbines for which data were obtained where the regression equation is given as:

$$N_s = 62.021\, P_{11}^{0.8361} \qquad \text{Eq. (5)}$$

where $P_{11} = \dfrac{P}{D^2 H^{1.5}}$ Eq. (6)

and D = turbine runner diameter in m.

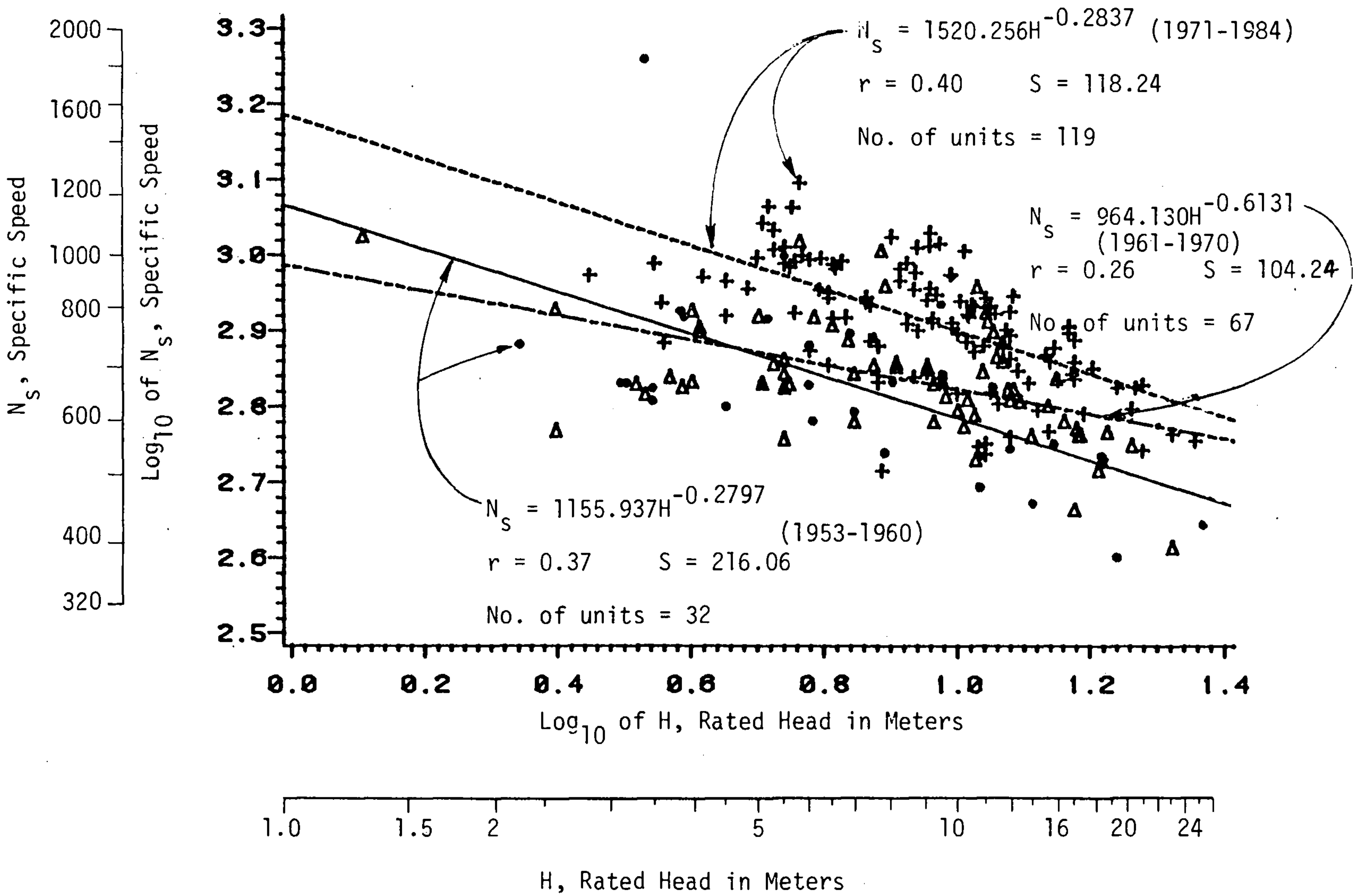

Figure 3. Specific speed versus rated head for bulb turbines.

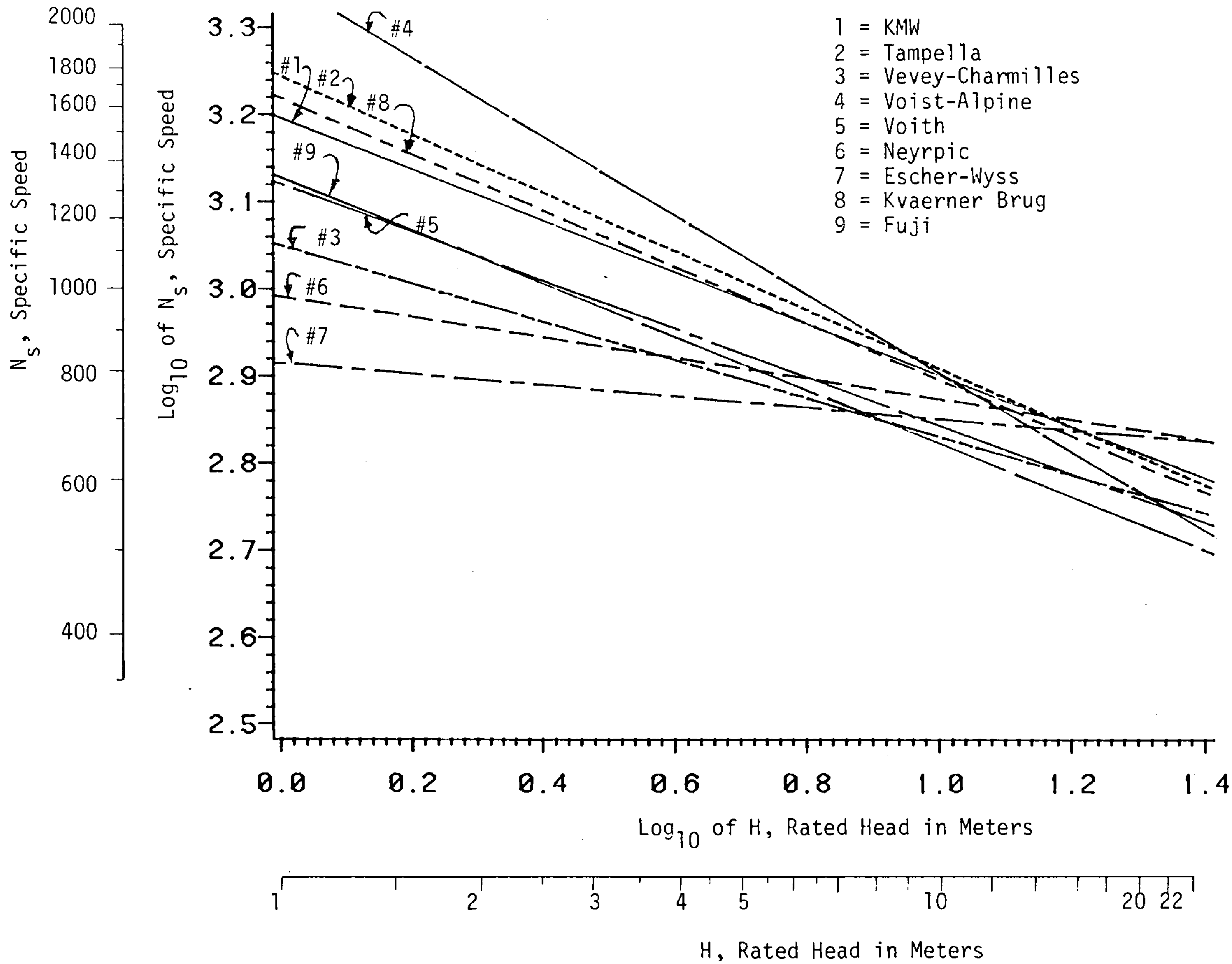

Figure 4. Specific speed versus rated head for bulb turbines for different manufacturers.

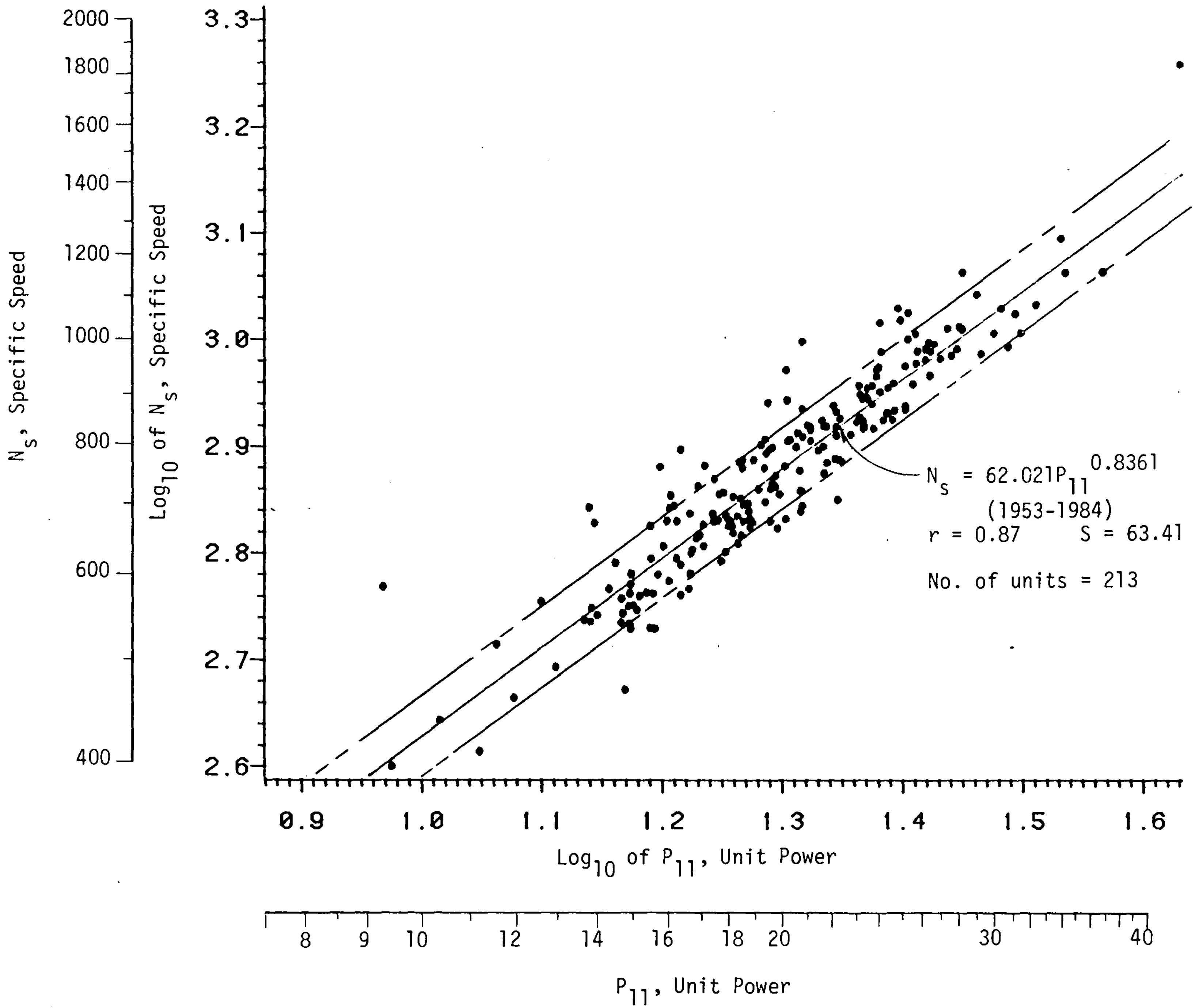

Figure 5 . Specific speed versus unit power for bulb turbines.

Figure 6 presents the relation between specific speed, N_S, and unit discharge Q_{11} for all bulb units for which data were obtained where the regression equations are given as:

$$N_s = 383.117\ Q_{11}^{0.8045} \qquad (1953\text{-}1965) \qquad \text{Eq. (7)}$$

$$N_s = 390.591\ Q_{11}^{0.8206} \qquad (1966\text{-}1984) \qquad \text{Eq. (8)}$$

where $Q_{11} = \dfrac{Q}{D^2 H^{0.5}}$ Eq. (9)

and Q = rated discharge in m^3/sec.

Figure 7 presents the relation between specific speed, N_S, and unit speed, N_{11}, for all bulb units for which data were obtained where the regression equations are given as:

$$N_{11} = 4.565\ N_s^{0.5478} \qquad (1953\text{-}1965) \qquad \text{Eq. (10)}$$

$$N_{11} = 7.987\ N_s^{0.4605} \qquad (1966\text{-}1984) \qquad \text{Eq. (11)}$$

where $N_{11} = \dfrac{ND}{H^{0.5}}$ Eq. (12)

Figure 8 presents the relation between unit power, P_{11}, and unit discharge, Q_{11}, for bulb turbines studied and the resulting regression equations are:

$$P_{11} = 9.027\ Q_{11}^{0.9347} \qquad (1953\text{-}1965) \qquad \text{Eq. (13)}$$

$$P_{11} = 9.345\ Q_{11}^{0.9445} \qquad (1966\text{-}1984) \qquad \text{Eq. (14)}$$

Figure 9 presents the relation between unit speed, N_{11}, and unit power, P_{11}, for bulb turbines studied and the resulting regression equation is:

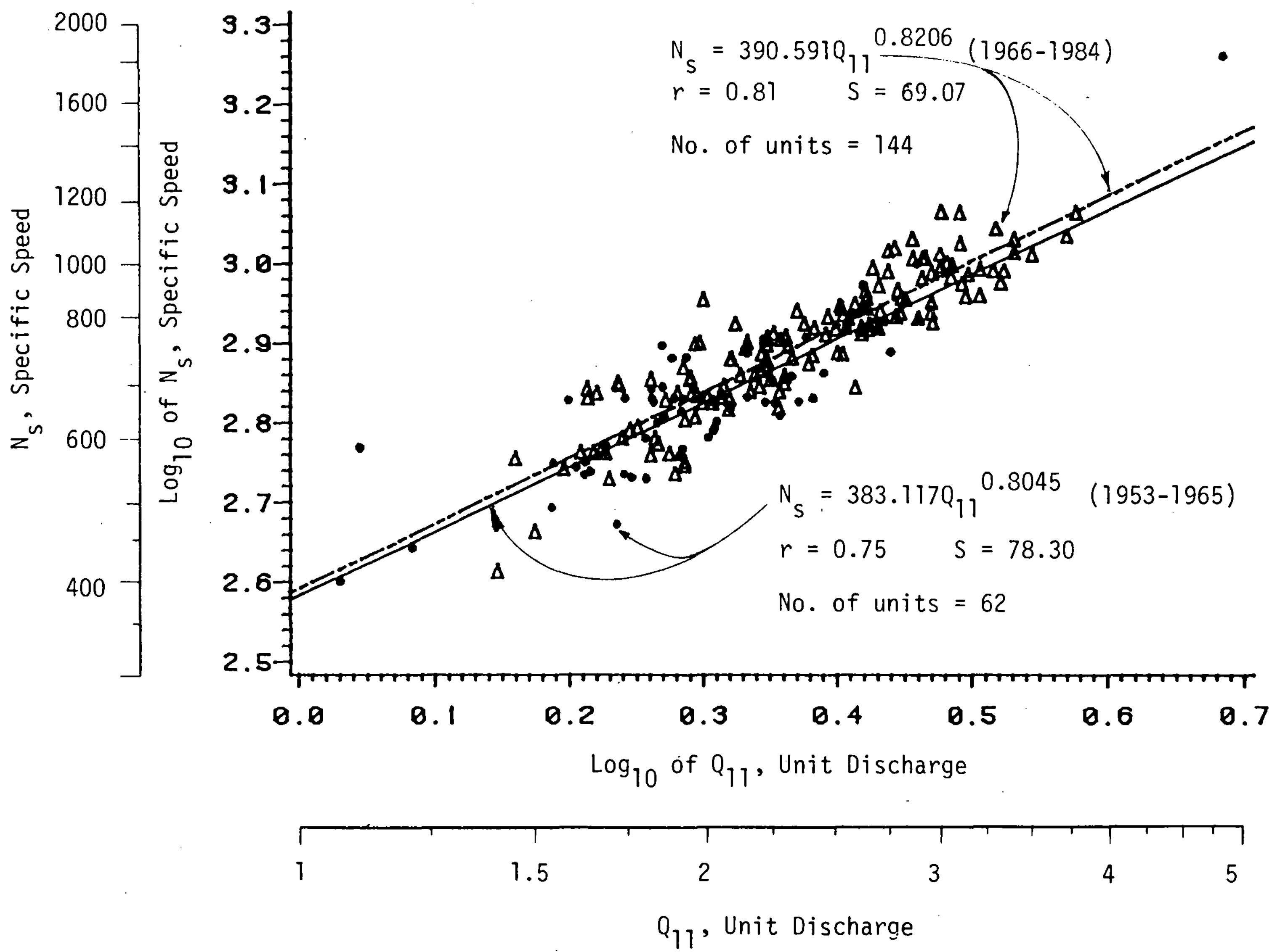

Figure 6. Specific speed versus unit discharge for bulb turbines.

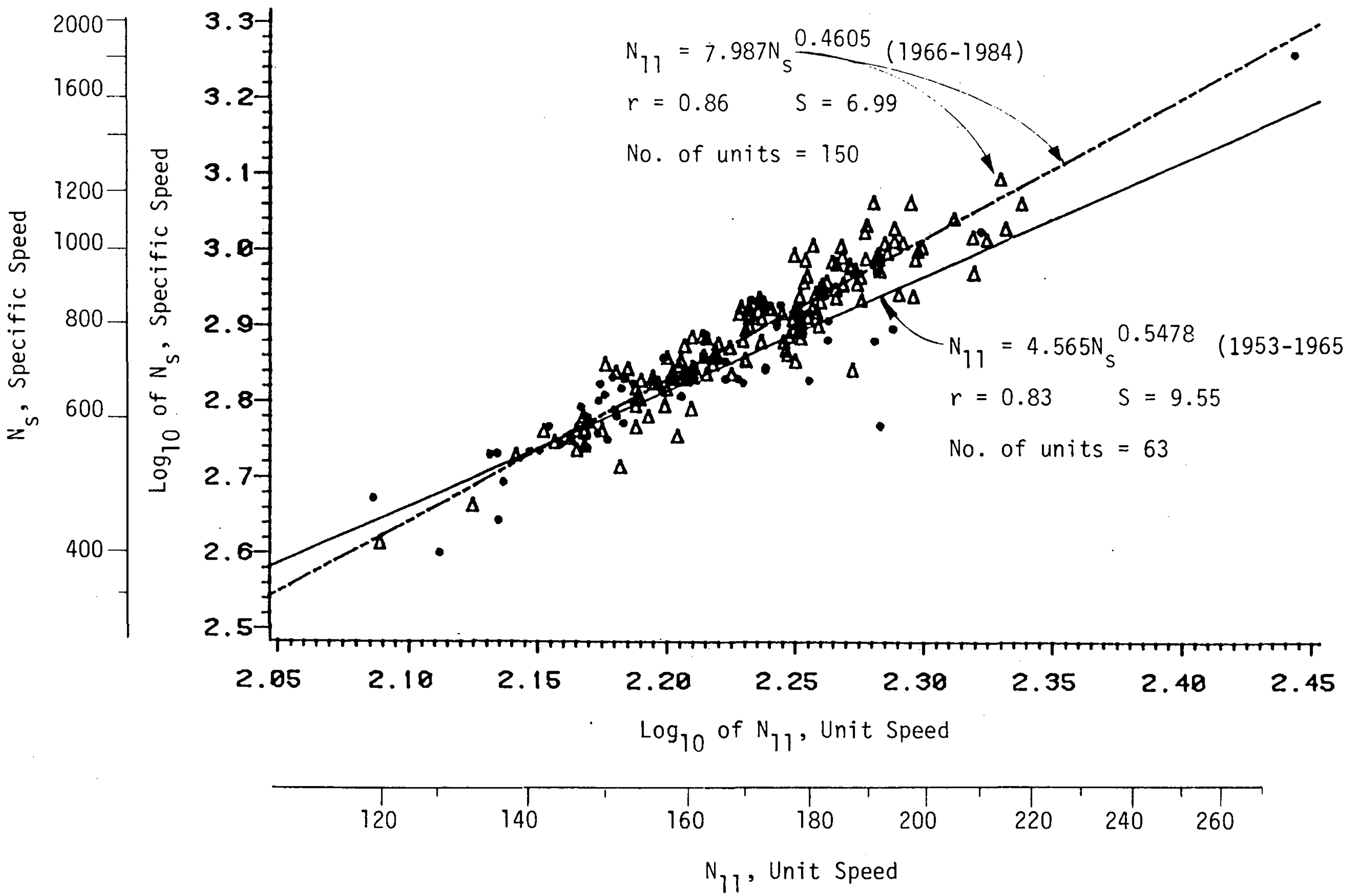

Figure 7. Unit speed versus specific speed for bulb turbines.

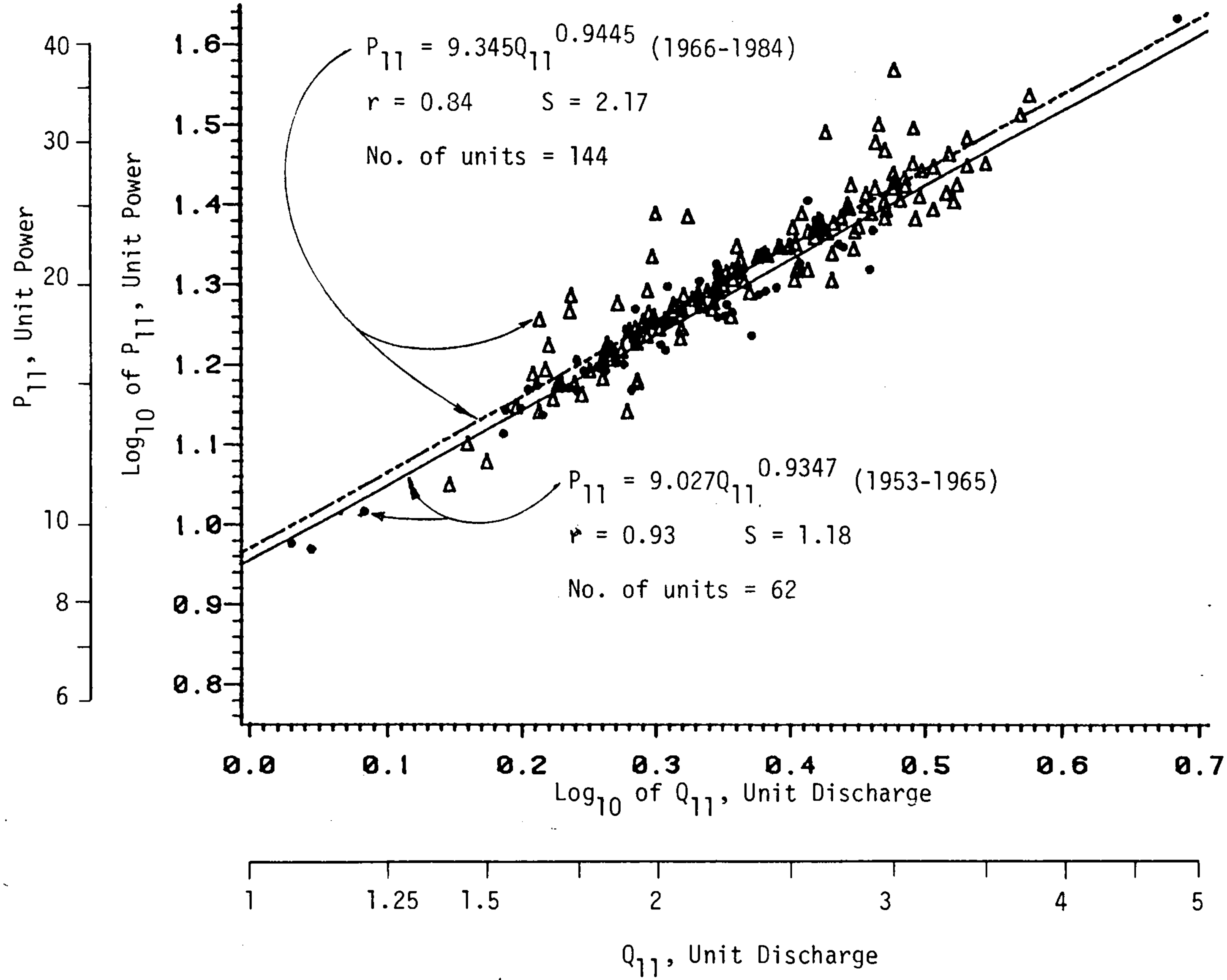

Figure 8. Unit power versus unit discharge for bulb turbines.

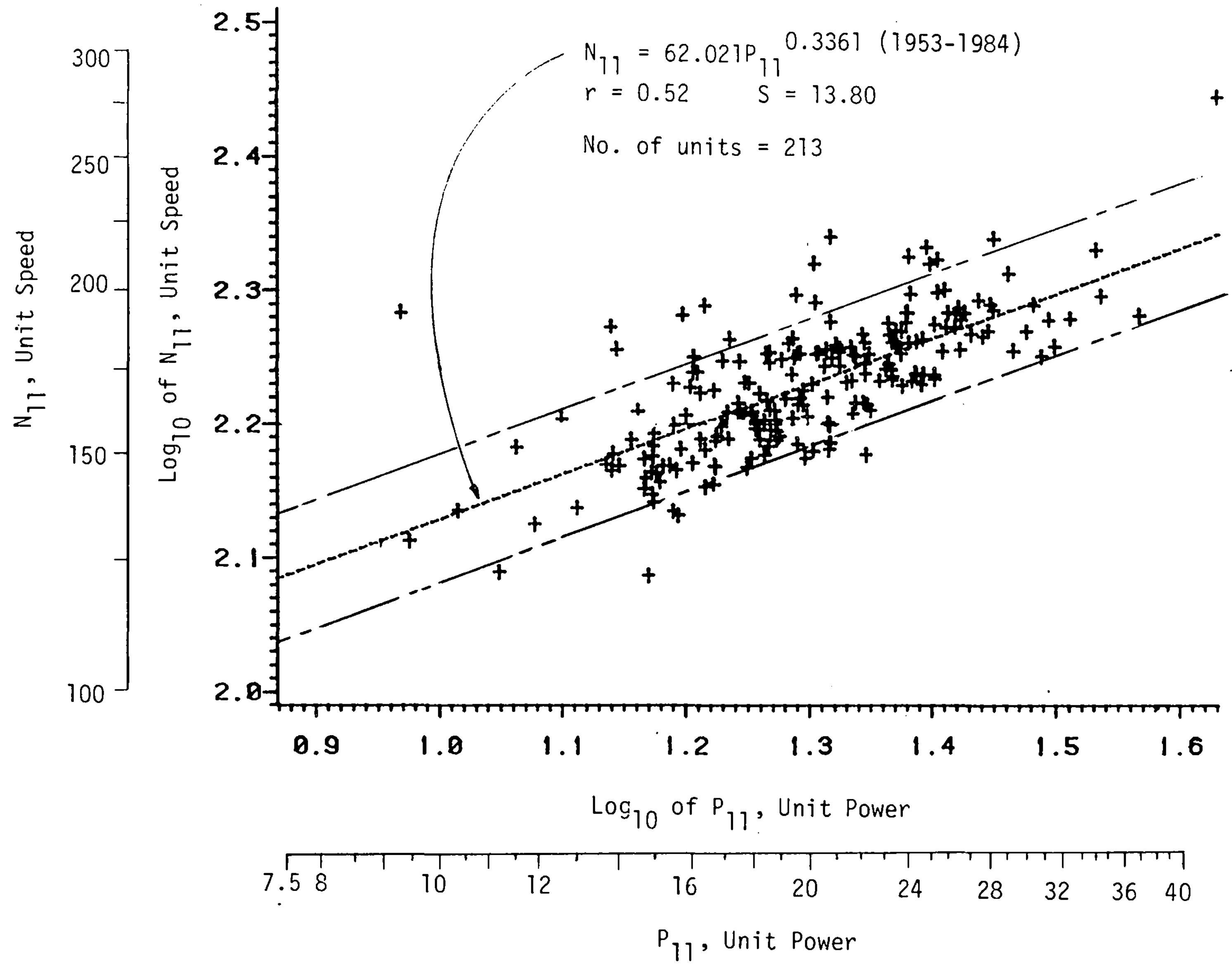

Figure 9. Unit speed versus unit power for bulb turbines.

$$N_{11} = 62.021\, P_{11}^{0.3361} \qquad (1953\text{-}1984) \qquad \text{Eq. (15)}$$

Figure 10 presents the relation between unit speed, N_{11}, and unit discharge Q_{11} for bulb turbines studied and the resulting regression equation is:

$$N_{11} = 127.119\, Q_{11}^{0.3513} \qquad (1953\text{-}1984) \qquad \text{Eq. (16)}$$

In many engineering offices and in some manufacturer's comparisons, the speed ratio or velocity ratio is used instead of the term unit speed, N_{11}, by practice and mathematically speed ratio is:

$$\emptyset = \frac{D\,\pi N}{60\sqrt{2gH}} = 11.82086 \times 10^{-3}\, N_{11} \quad * \qquad \text{Eq. (17)}$$

where g = acceleration of gravity in m/sec^2

D = turbine diameter in m.

Using the speed ratio, $\emptyset$, as a characteristic turbine parameter relations were developed for manufactured bulb type turbines as follows:

$$\emptyset = 0.0540\, N_s^{0.5478} \qquad (1953\text{-}1965) \qquad \text{Eq. (18)}$$

$$\emptyset = 0.0944\, N_s^{0.4605} \qquad (1966\text{-}1984) \qquad \text{Eq. (19)}$$

$$\emptyset = 0.1232\, P_{11}^{0.9615} \qquad (1953\text{-}1965) \qquad \text{Eq. (20)}$$

$$\emptyset = 0.3518\, P_{11}^{0.5772} \qquad (1966\text{-}1984) \qquad \text{Eq. (21)}$$

$$D = 1.554\, \emptyset^{0.7640} \qquad (1953\text{-}1965) \qquad \text{Eq. (22)}$$

$$D = 1.393\, \emptyset^{1.4780} \qquad (1966\text{-}1984) \qquad \text{Eq. (23)}$$

* Sometimes the speed ratio is expressed in the American system of units and the D is expressed in inches and the H in feet.

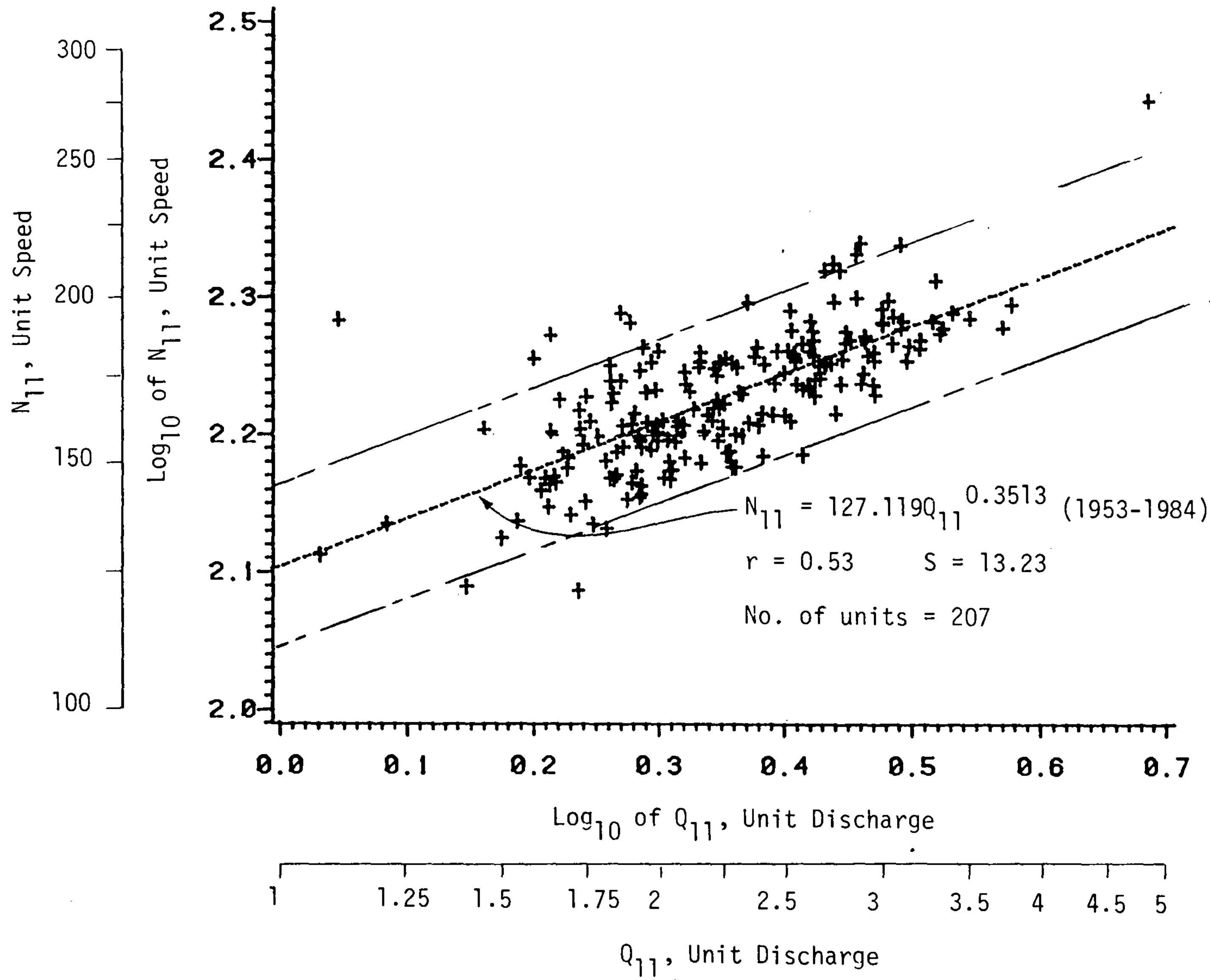

Figure 10. Unit speed versus unit discharge for bulb turbines.

The graphical relations for these three regression equations are shown in Figures 11, 12, and 13. In seeking a simplification for use of experience curves it was recognized that relating diameter to the basic well known parameters of rated head and rated power would be most useful because in preliminary planning the parameters of rated head and rated power are most generally estimated early in the planning of projects based on the physical elevation situation of the water and the power available from the estimated flows. On this basis a new regression analysis was made relating turbine diameter to the ratio of P/H where P is the rated power output and H is the design head or rated head. Figure 14 presents for manufactured bulb type turbines the relation between turbine diameter and the ratio of rated power to rated head and the resulting regression equations are:

$$D = 0.2119(P/H)^{0.4374} \qquad (1953\text{-}1965) \qquad \text{Eq. (24)}$$

$$D = 0.1826(P/H)^{0.4462} \qquad (1966\text{-}1984) \qquad \text{Eq. (25)}$$

A similar new relation was developed relating turbine diameter to the ratio of rated discharge, Q, to the operating speed, N. This relationship is shown in Figure 15 and the resulting regression equation is:

$$D = 4.181\ (Q/N)^{0.3175} \qquad \text{Eq. (26)}$$

This again recognizes that in early planning stages the rated discharge is known from the hydrologic analysis of power or energy potential at a site and the choices of operating speeds are rather limited because there are a limited number of available synchronous speeds at which bulb turbines can operate if directly connected to the generator.

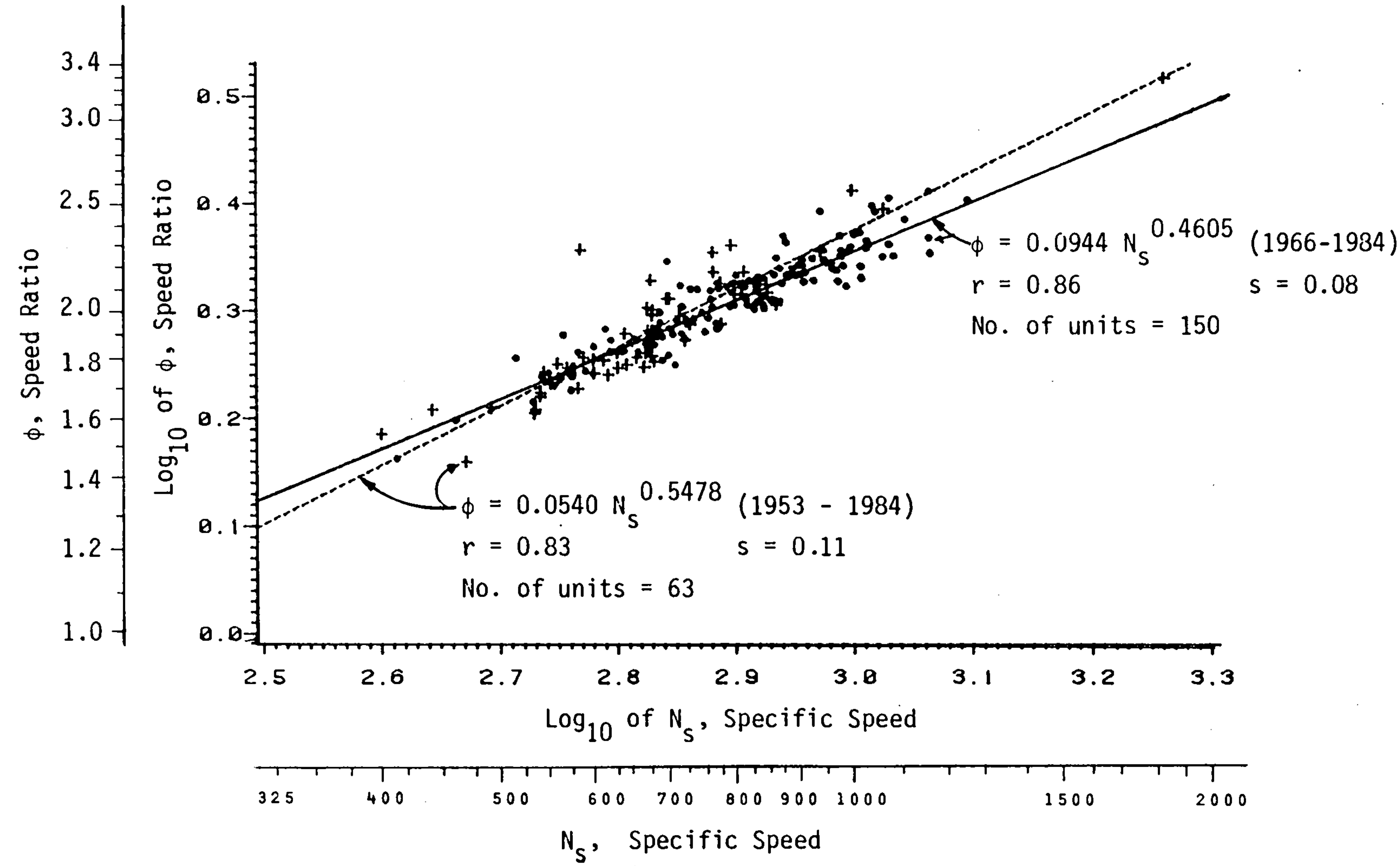

Figure 11. Speed ratio versus specific speed for bulb turbines.

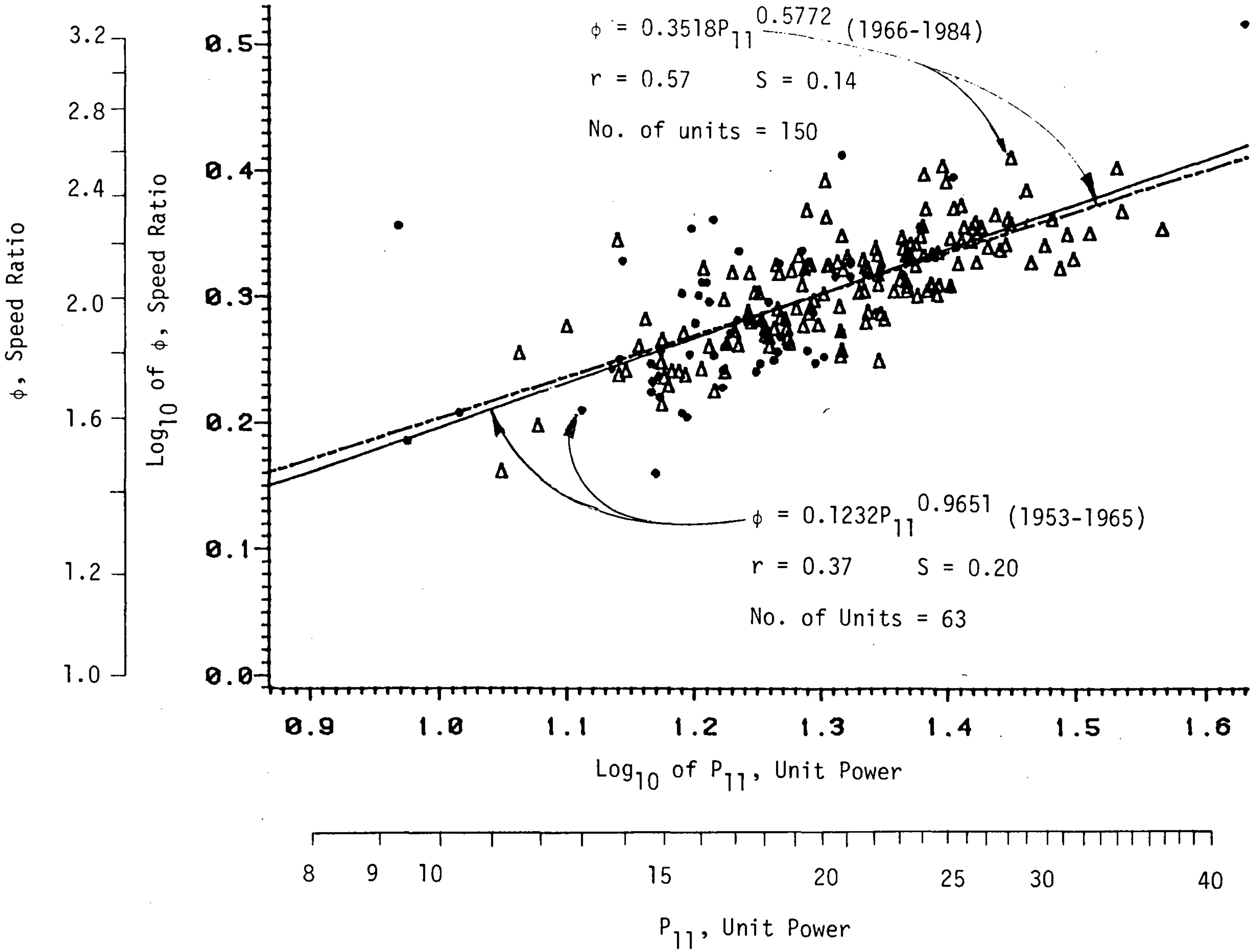

Figure 12. Speed ratio versus unit power for bulb turbines.

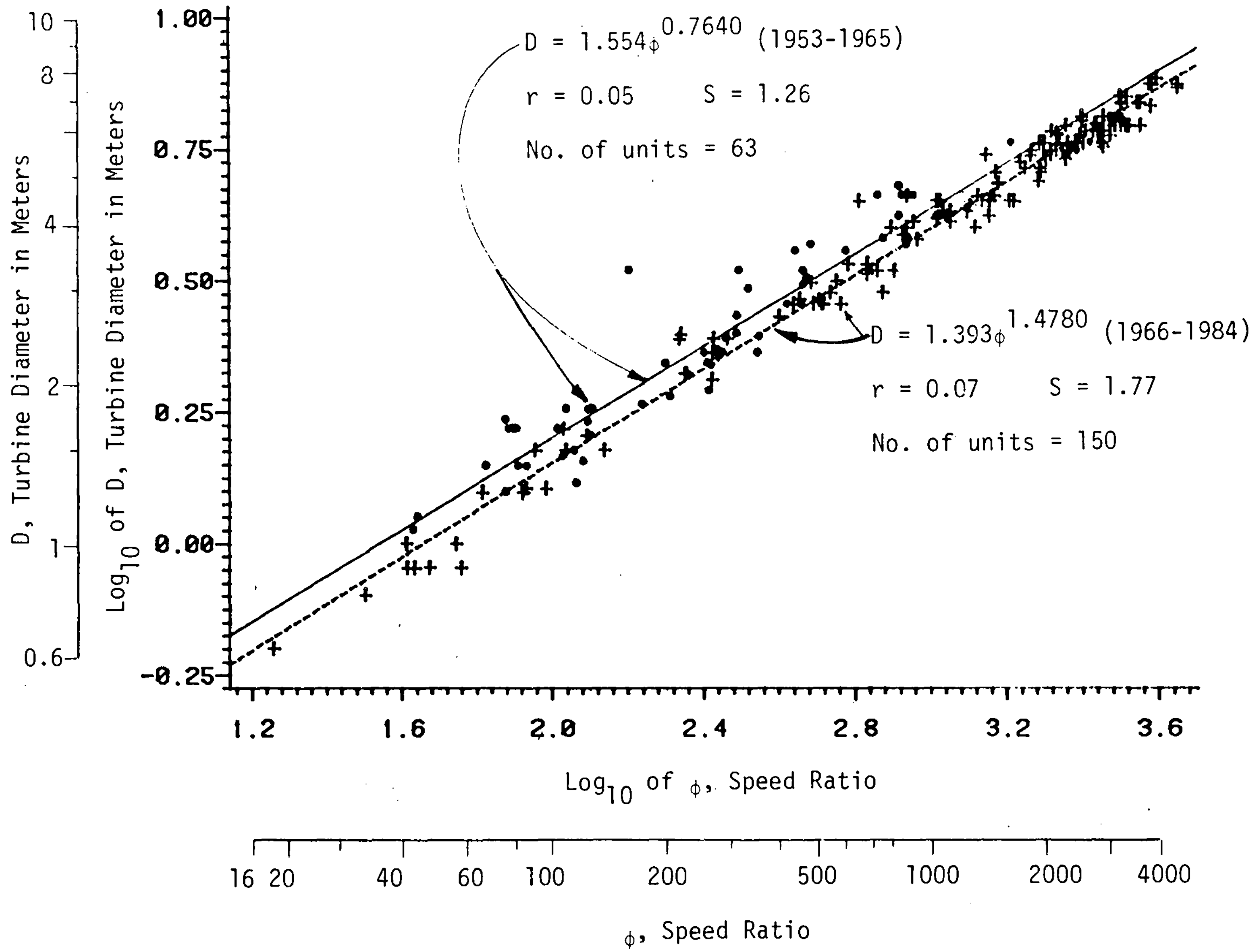

Figure 13. Turbine diameter versus speed ratio for bulb turbines.

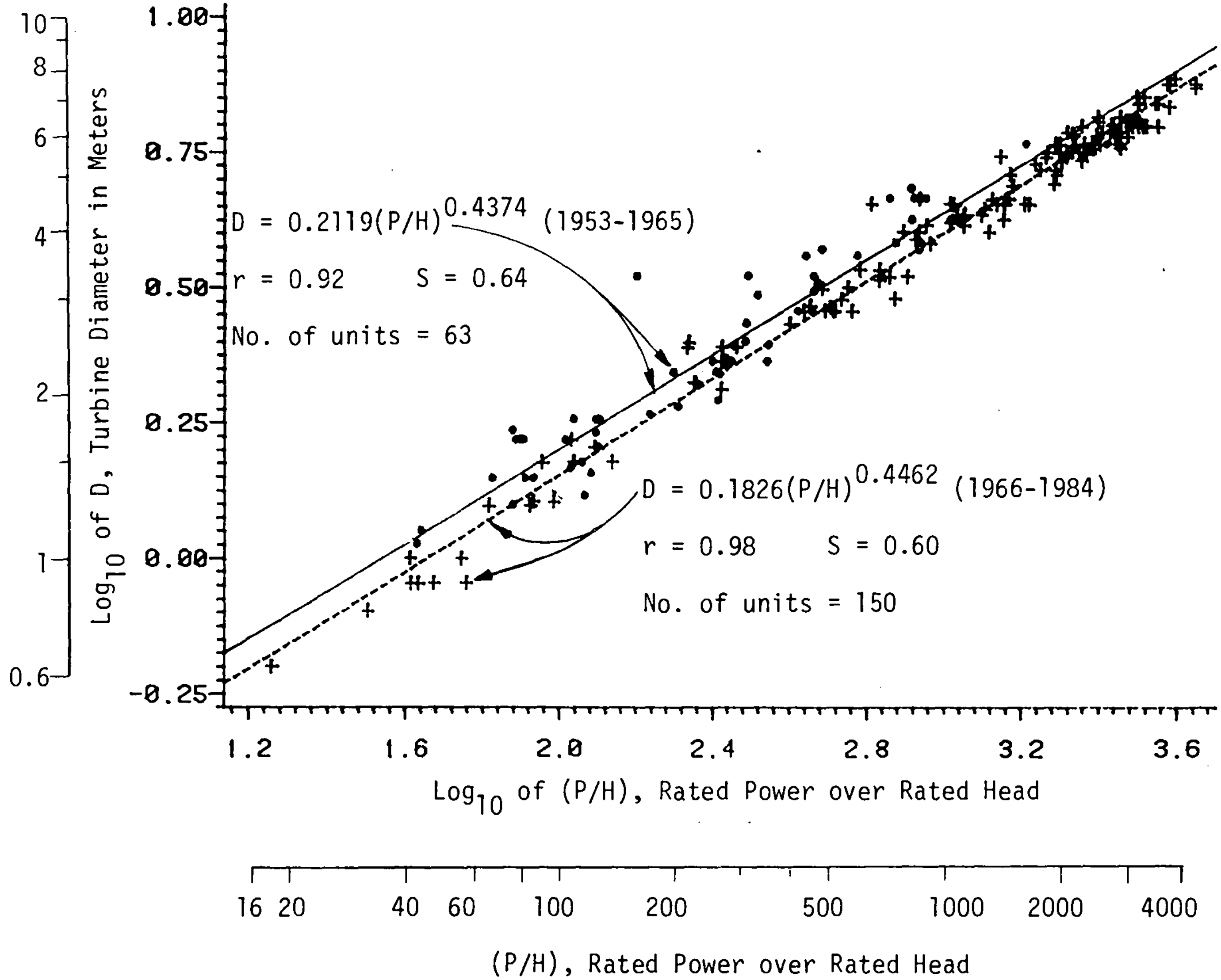

Figure 14. Turbine diameter versus (P/H) ratio for bulb turbines.

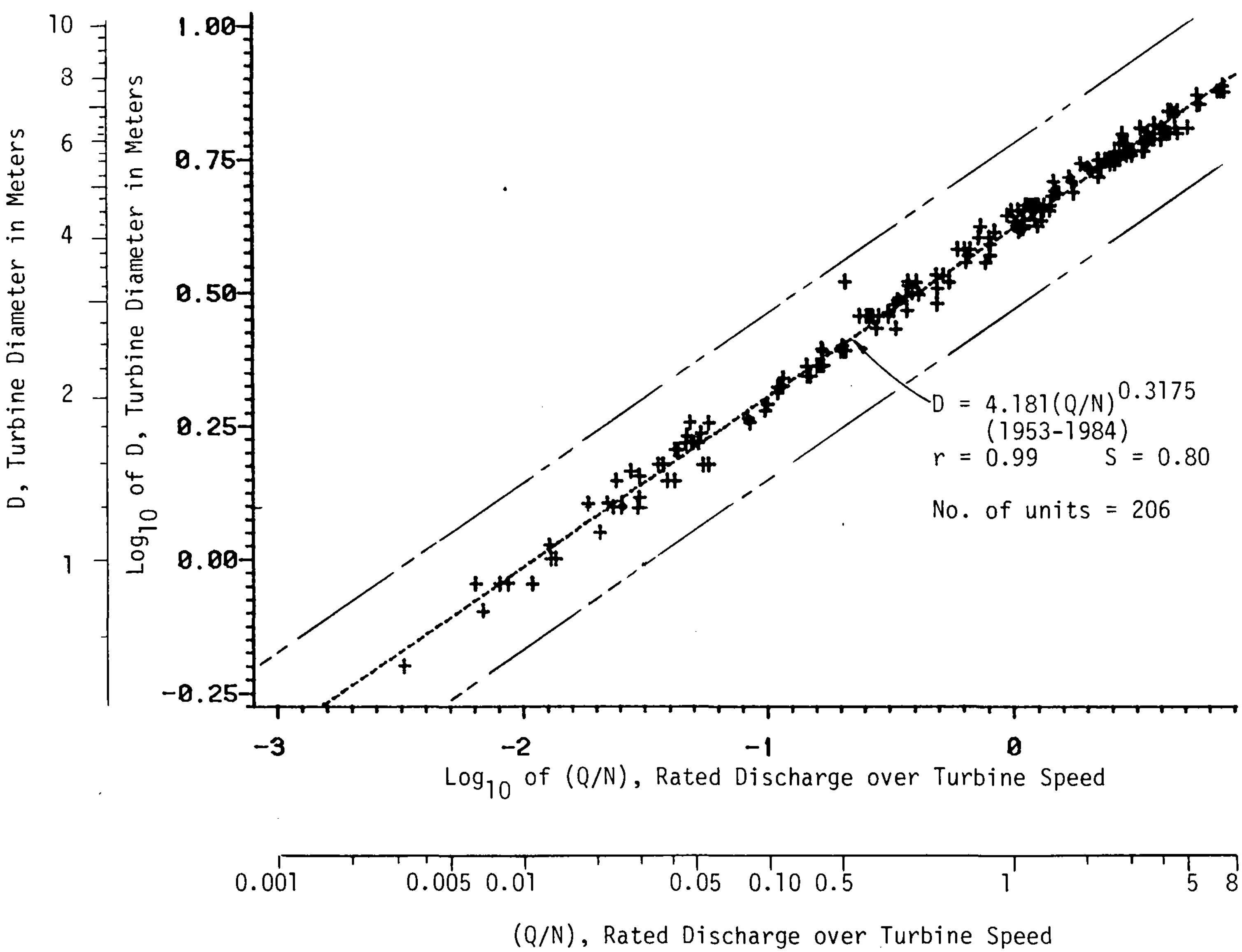

Figure 15. Turbine diameter versus (Q/N) ratio for bulb turbines.

An additional regression was developed between the turbine speed and the ratio of rated power to rated head and the resulting regression equations are

$$N = 1810.648\ (P/H)^{-0.4176} \qquad (1953 - 1965) \qquad \text{Eq. (27)}$$

$$N = 2152.857\ (P/H)^{-0.4062} \qquad (1966 - 1984) \qquad \text{Eq. (28)}$$

Figure 16 presents the graphical representation of N vs P/H.

As a result of inspection of an Escher Wyss nomograph for standard tubular turbines a regression relation was developed between turbine speed and the ratio, $\sqrt{H}/D$. The regression equations for bulb turbines for that relation between turbine speed, N, and the ratio $\sqrt{H}/D$ are as follows:

$$N = 162.103\ (\sqrt{H}/D)^{0.8912} \qquad (1953\text{-}1965) \qquad \text{Eq. (29)}$$

$$N = 169.119\ (\sqrt{H}/D)^{0.9260} \qquad (1966\text{-}1984) \qquad \text{Eq. (30)}$$

Figure 17 presents the graphical representation of N vs $\sqrt{H}/D$.

Table 2 summarizes all the regression relations that were developed for manufactured bulb type turbines. In the table are shown all the equations that were developed, the regression correlation coefficient for each particular regression, the corresponding standard deviation, the sample period and the number of different units used in developing a particular relation.

In the Appendix an example is given showing how these turbine constants and regression equations can be used to make a diameter selection utilizing the analysis system used in Monograph No. 20 of the U.S. Bureau of Reclamation and parallel calculations show selection of turbine diameter using newly developed experience curves involving directly a P/H ratio and a Q/N ratio and the resulting regression equations.

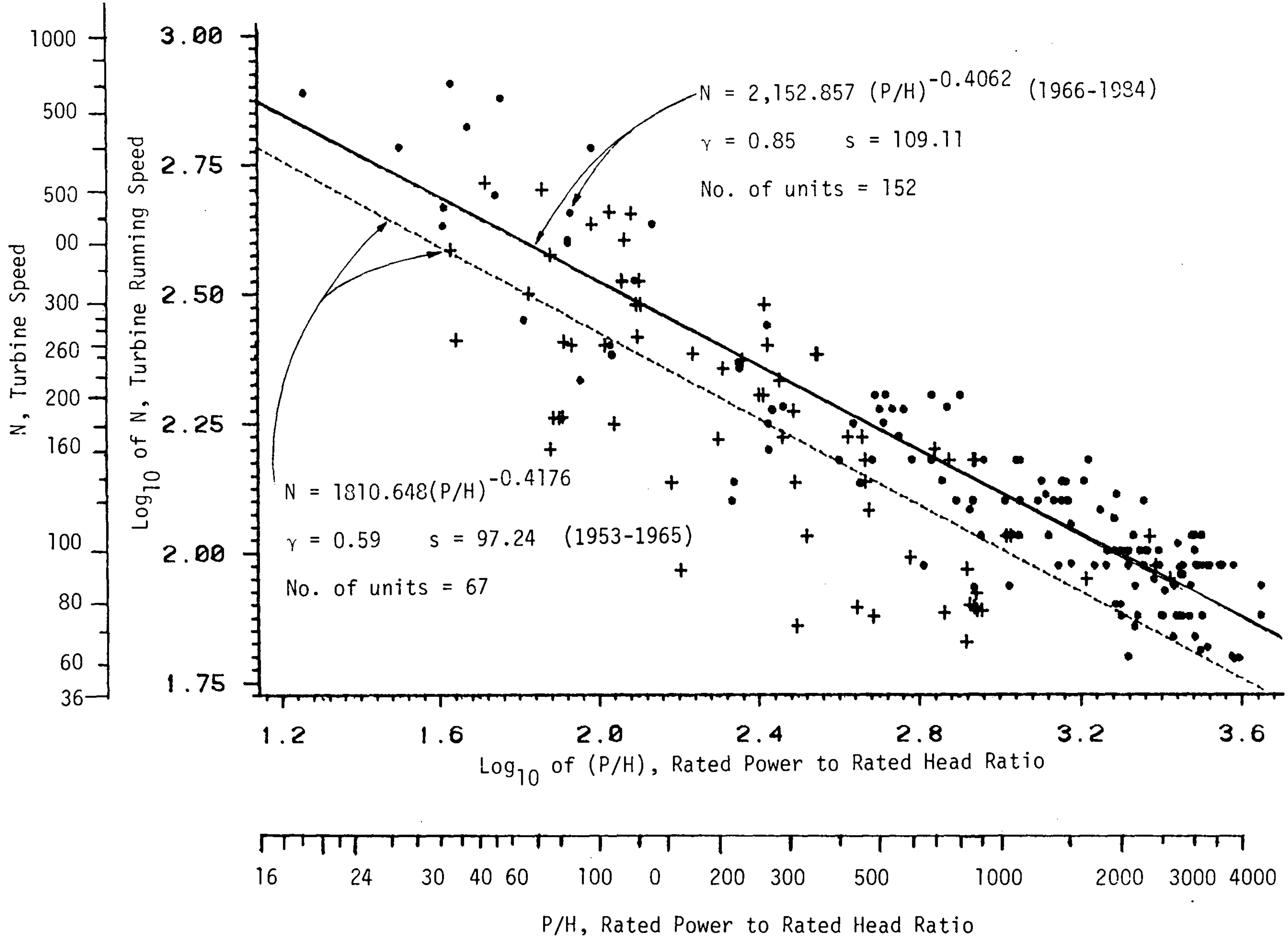

Figure 16. Turbine speed N, versus P/H ratio for bulb turbines.

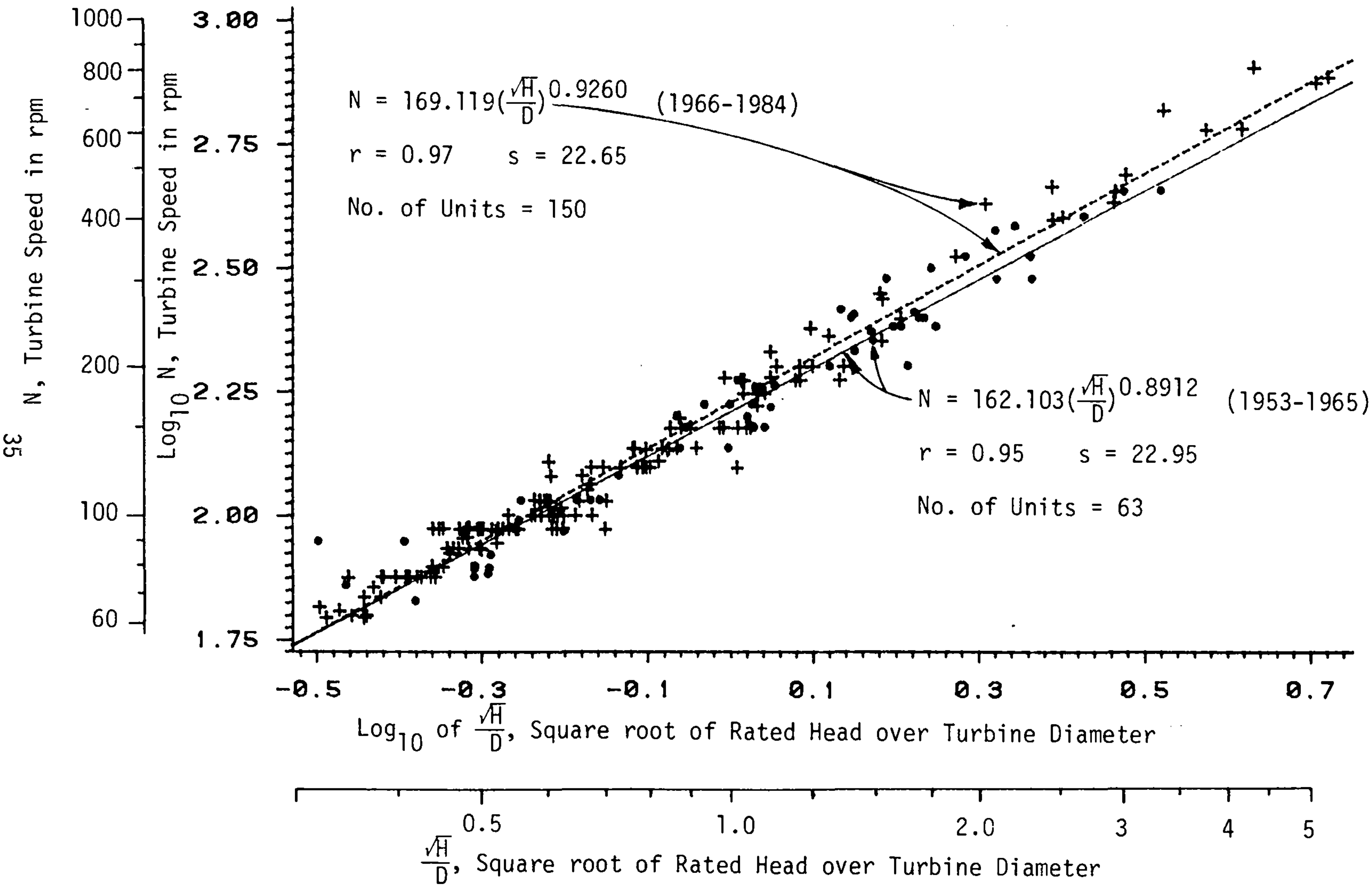

Figure 17. Turbine speed versus $\frac{\sqrt{H}}{D}$ ratio for bulb turbines.

TABLE 2

SUMMARY LISTING OF REGRESSION INFORMATION AND EQUATIONS RELATING TURBINE CHARACTERISTICS TO VARIOUS TURBINE CONSTANTS FOR BULB TURBINES

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
1	N_s	$N_s = 1155.937\ H^{-0.2797}$	0.37	216.06	1953-1960	32
2	N_s	$N_s = 964.130\ H^{-0.1631}$	0.26	104.24	1961-1970	67
3	N_s	$N_s = 1520.256\ H^{-0.2837}$	0.40	118.24	1971-1984	119
5	N_s	$N_s = 62.021\ P_{11}^{0.8361}$	0.87	63.41	1953-1984	213
7	N_s	$N_s = 383.117\ Q_{11}^{0.8045}$	0.75	78.30	1953-1965	62
8	N_s	$N_s = 390.591\ Q_{11}^{0.8206}$	0.81	69.07	1966-1984	144
10	N_{11}	$N_{11} = 4.565\ N_s^{0.5478}$	0.83	9.55	1953-1965	63
11	N_{11}	$N_{11} = 7.987\ N_s^{0.4605}$	0.86	6.99	1966-1984	150

TABLE 2 CONTINUED

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
13	P_{11}	$P_{11} = 9.027\ Q_{11}^{0.9347}$	0.93	1.18	1953-1965	62
14	P_{11}	$P_{11} = 9.345\ Q_{11}^{0.9445}$	0.84	2.17	1966-1984	144
15	N_{11}	$N_{11} = 62.021\ P_{11}^{0.3361}$	0.52	13.80	1953-1984	213
16	N_{11}	$N_{11} = 127.119\ Q_{11}^{0.3513}$	0.53	13.23	1953-1984	207
18	ϕ	$\phi = 0.0540\ N_s^{0.5478}$	0.83	0.11	1953-1965	63
19	ϕ	$\phi = 0.0944\ N_s^{0.4605}$	0.86	0.08	1966-1984	150
20	ϕ	$\phi = 0.1232\ P_{11}^{0.9615}$	0.37	0.20	1953-1965	63
21	ϕ	$\phi = 0.3518\ P_{11}^{0.5772}$	0.57	0.14	1966-1984	150
22	D	$D = 1.554\ \phi^{0.7640}$	0.05	1.26	1953-1965	63
23	D	$D = 1.393\ \phi^{1.4780}$	0.07	1.77	1966-1984	150
24	D	$D = 0.2119(P/H)^{0.4374}$	0.92	0.64	1953-1965	63

TABLE 2 CONTINUED

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
25	D	$D = 0.1826(P/H)^{0.4462}$	0.98	0.60	1966-1984	150
26	D	$D = 4.181(Q/N)^{0.3175}$	0.99	0.80	1953-1984	206
27	N	$N = 1810.648(P/H)^{-0.4176}$	0.59	97.24	1953-1965	67
28	N	$N = 2152.857(P/H)^{-0.4062}$	0.85	109.11	1966-1984	152
29	N	$N = 162.103(\frac{\sqrt{H}}{D})^{0.8912}$	0.95	22.95	1953-1965	63
30	N	$N = 169.119(\frac{\sqrt{H}}{D})^{0.9260}$	0.97	22.65	1966-1984	150

Tubular Turbines

For tubular type turbines the N_s vs H relation is shown in Figure 18 and the regression relation is given as:

$$N_s = 1107.303\ H^{-0.2998} \qquad \text{Eq. (31)}$$

Stratification of the N_S vs H relationship showing the variation of the relation for various turbine manufacturers is presented in Figure 19. A summary of the data for individual manufacturers is presented in Appendix 3 along with the specific regression equations.

Figure 20 presents the relation between specific speed, N_s, and unit power, P_{11}, for tubular turbines and the resulting regression equation is given as:

$$N_s = 52.96\ P_{11}^{0.8882} \qquad \text{Eq. (32)}$$

Figure 21 presents the relation between specific speed, N_s, and unit discharge, Q_{11}, for all tubular turbines and the resulting regression equation is given as:

$$N_s = 357.294\ Q_{11}^{0.9029} \qquad \text{Eq. (33)}$$

Figure 22 presents the relation between specific speed, N_s, and unit speed, N_{11}, for tubular type turbines for which data were obtained where the regression equation is given as:

$$N_s = 0.497\ N_{11}^{1.4080} \qquad \text{Eq. (34)}$$

Figure 23 presents the relation between unit power, P_{11}, and unit discharge, Q_{11}, for tubular type turbines studied and the resulting regression equation is:

$$P_{11} = 10.133\ Q_{11}^{0.7315} \qquad \text{Eq. (35)}$$

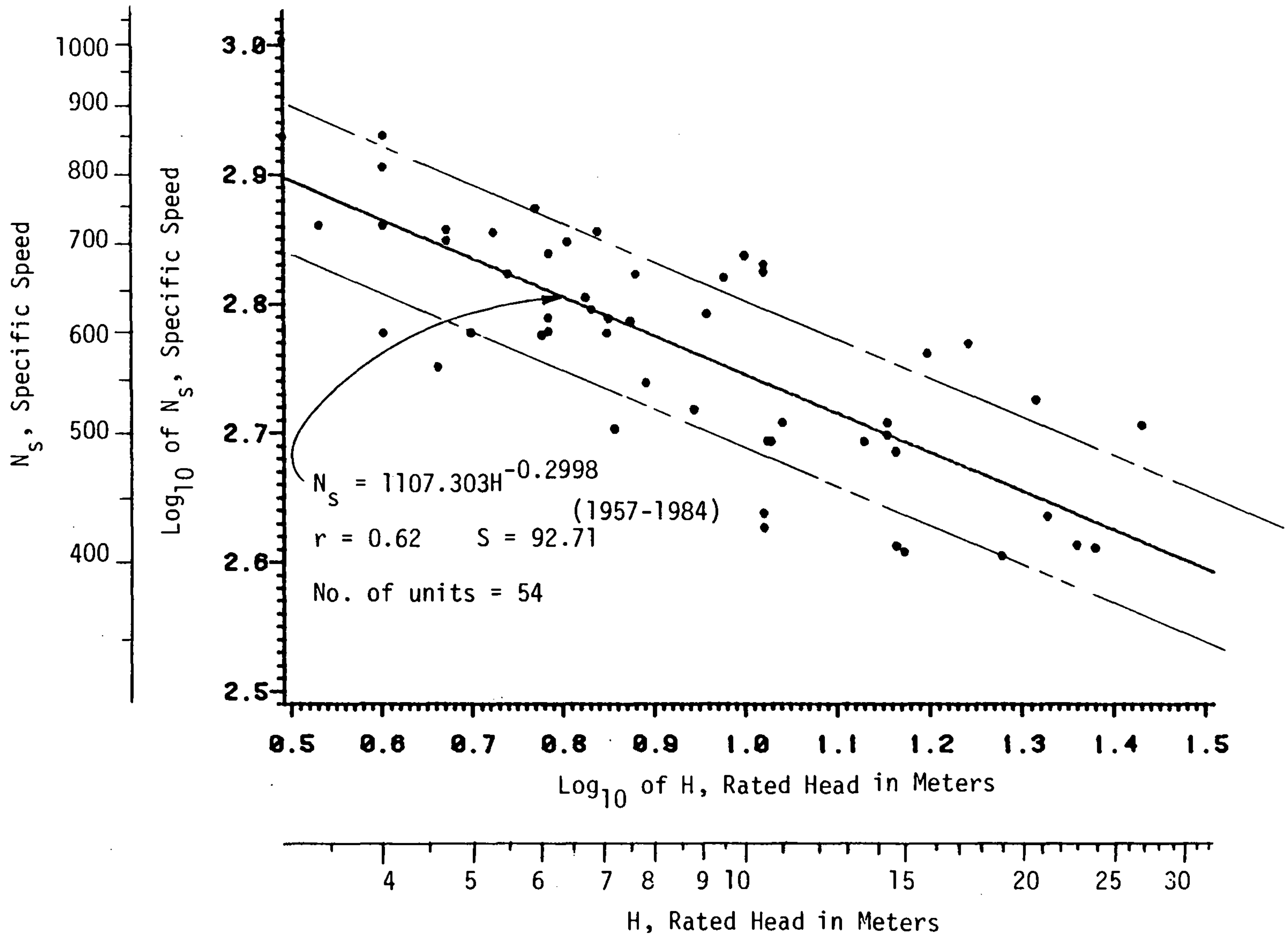

Figure 18. Specific speed versus rated head for tubular turbines.

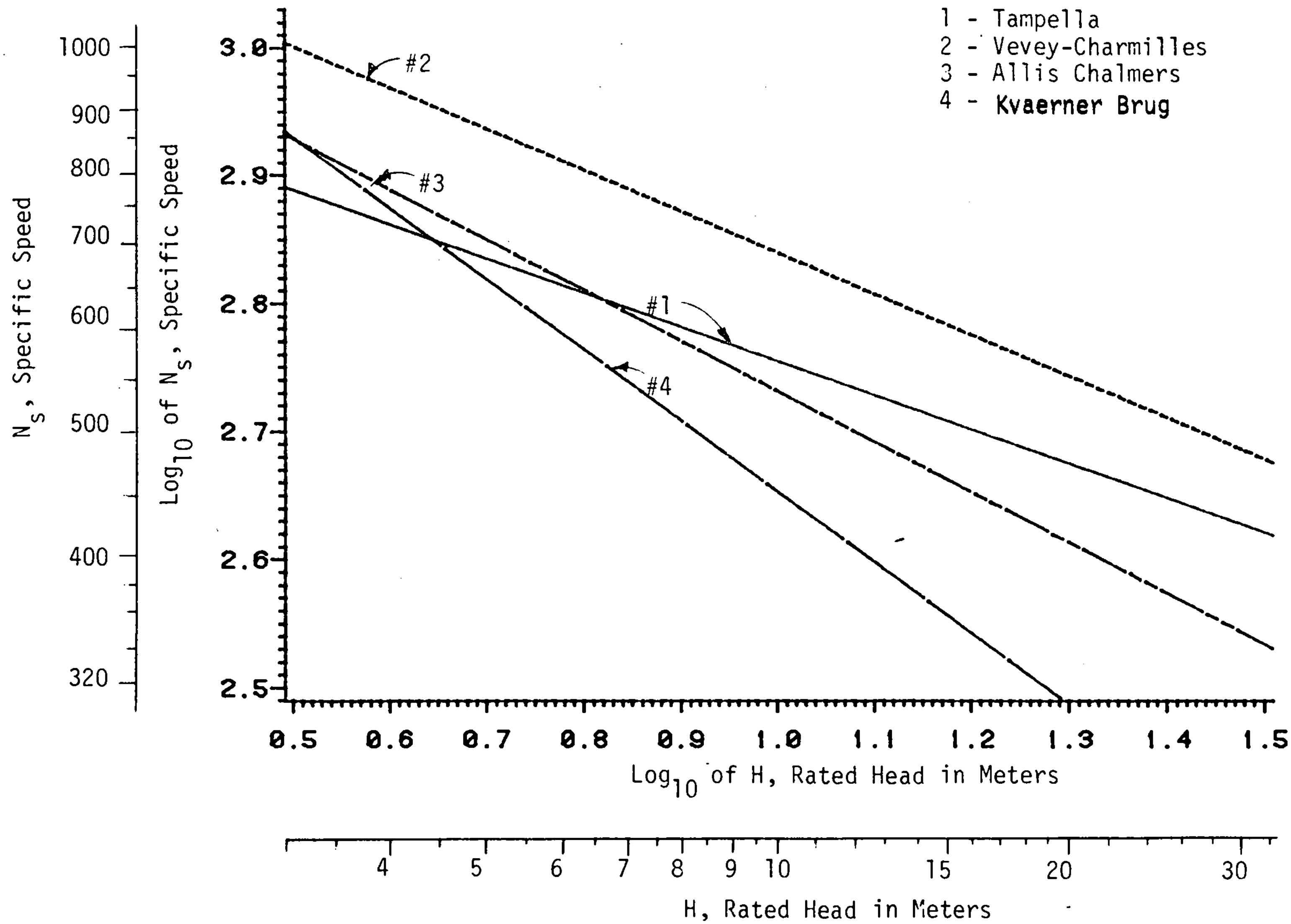

Figure 19. Specific speed versus rated head for tubular turbines from different turbine manufactures.

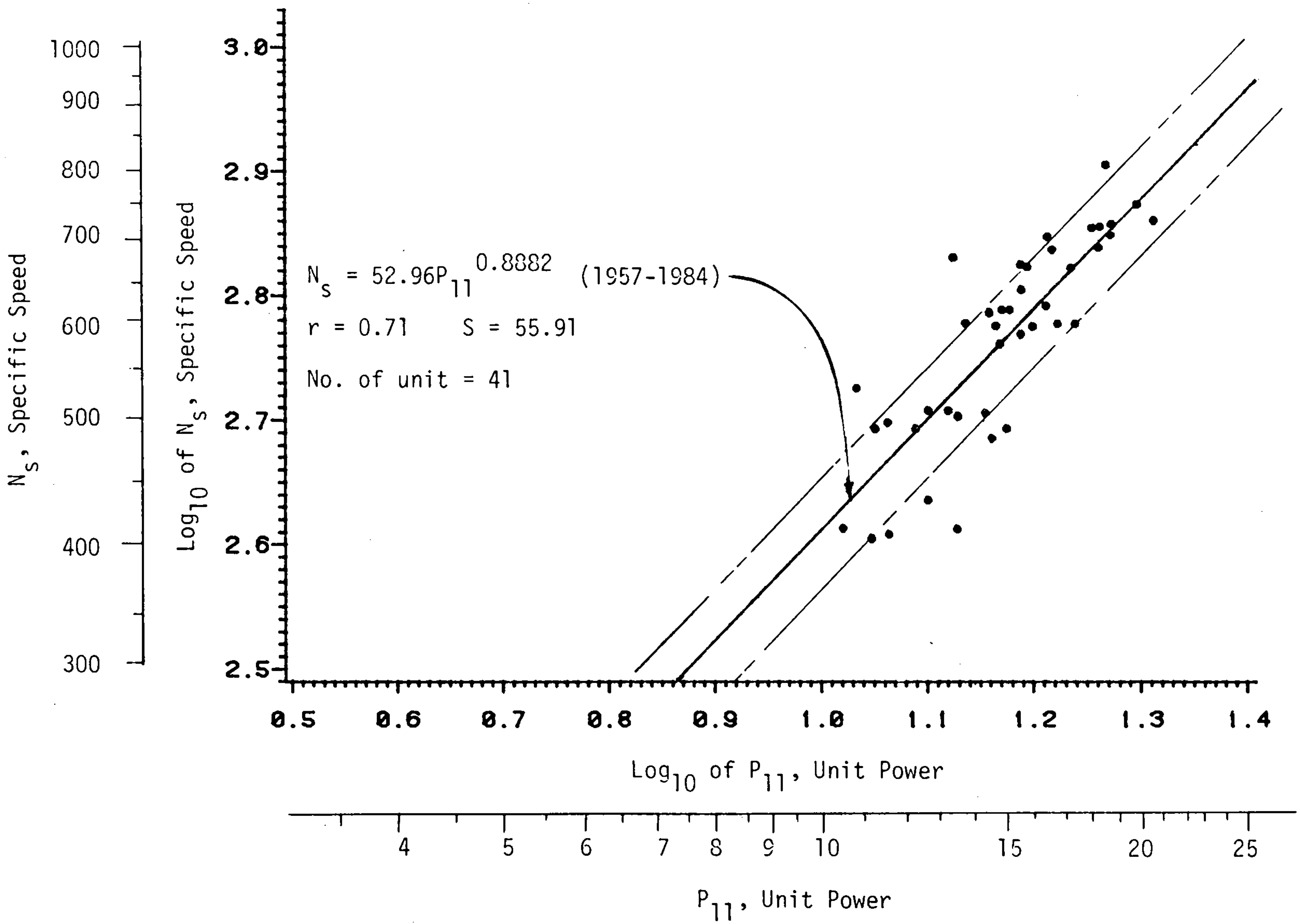

Figure 20. Specific speed versus unit power for tubular turbines.

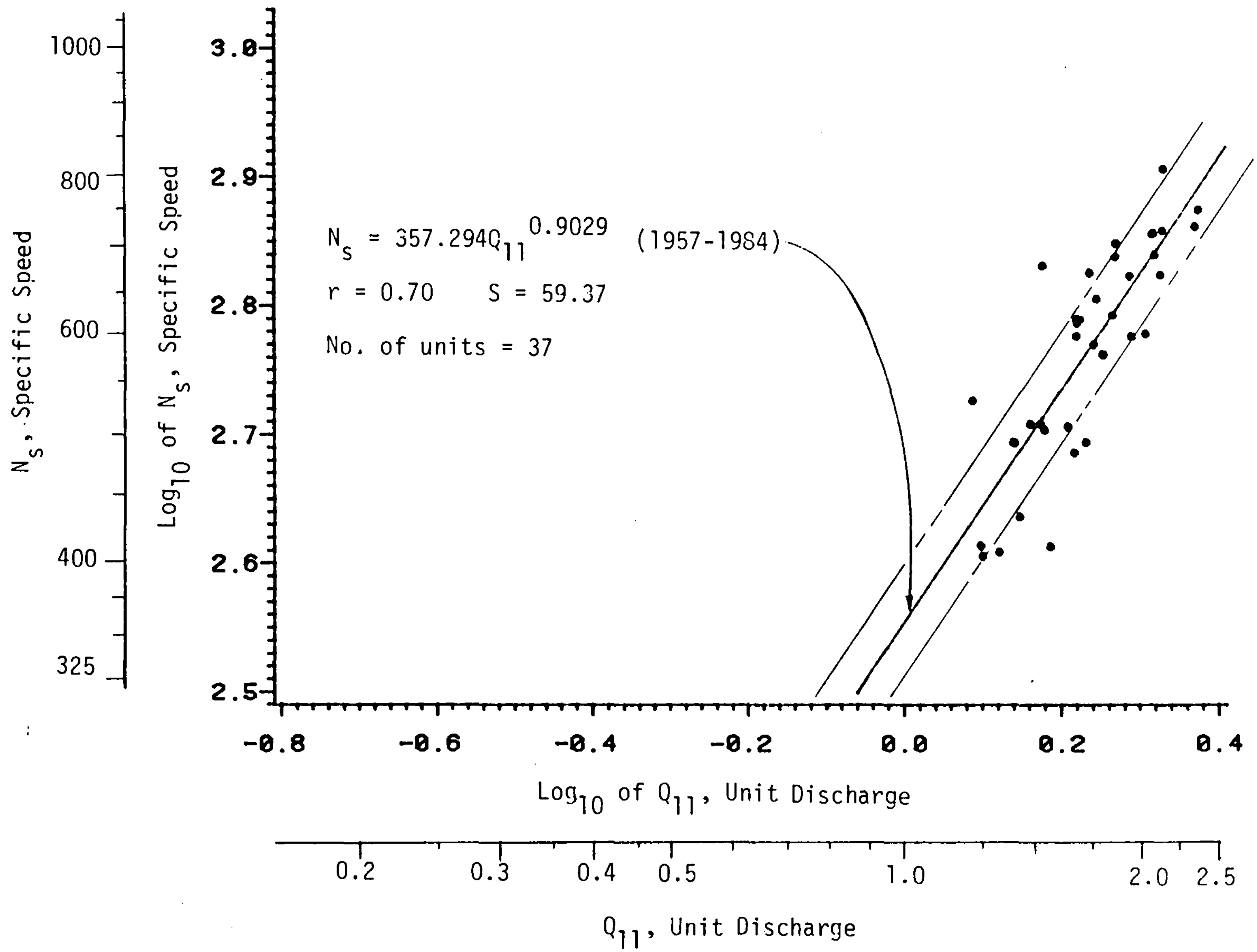

Figure 21. Specific speed versus unit discharge for tubular turbines.

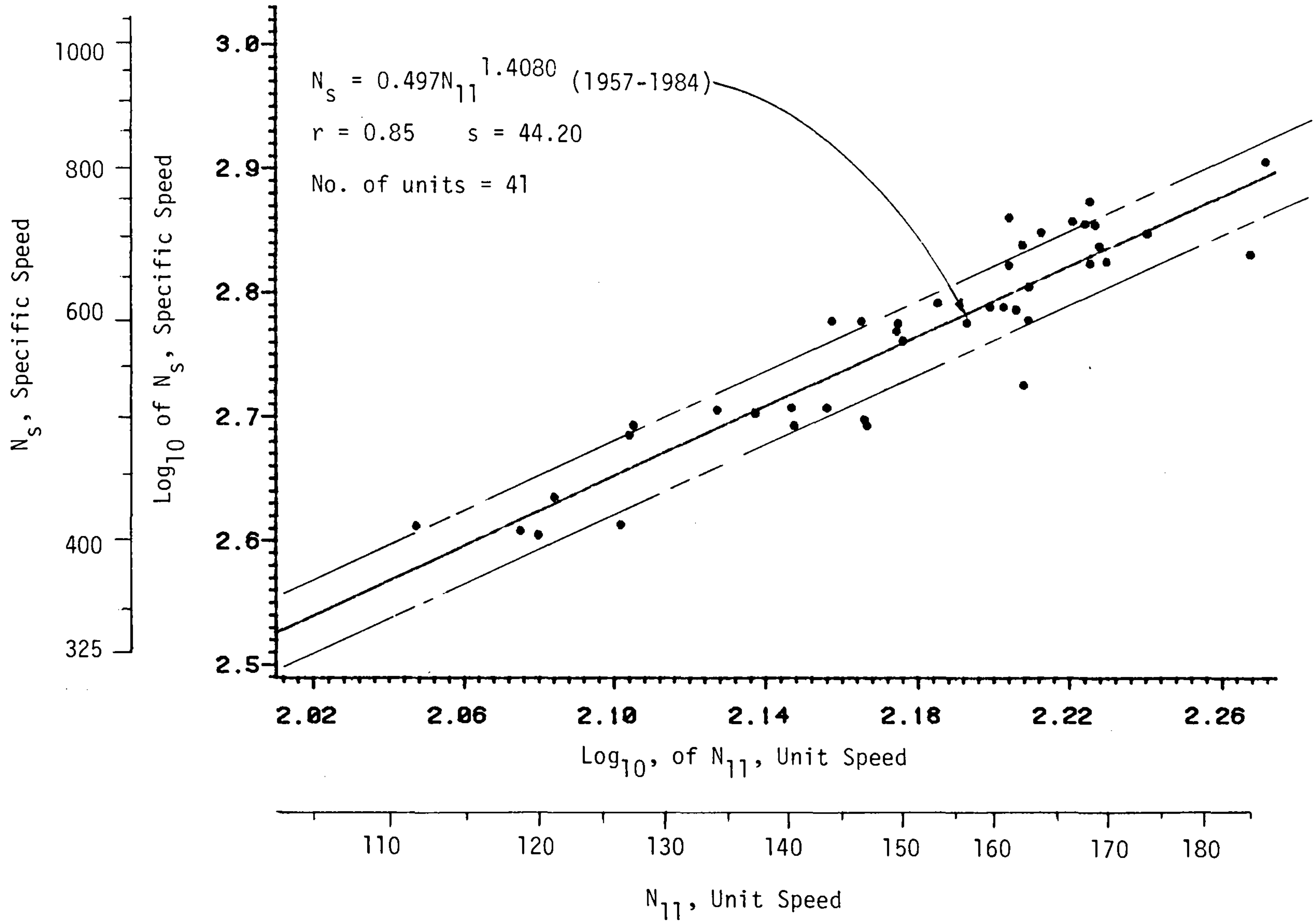

Figure 22. Specific speed versus unit speed for tubular turbines.

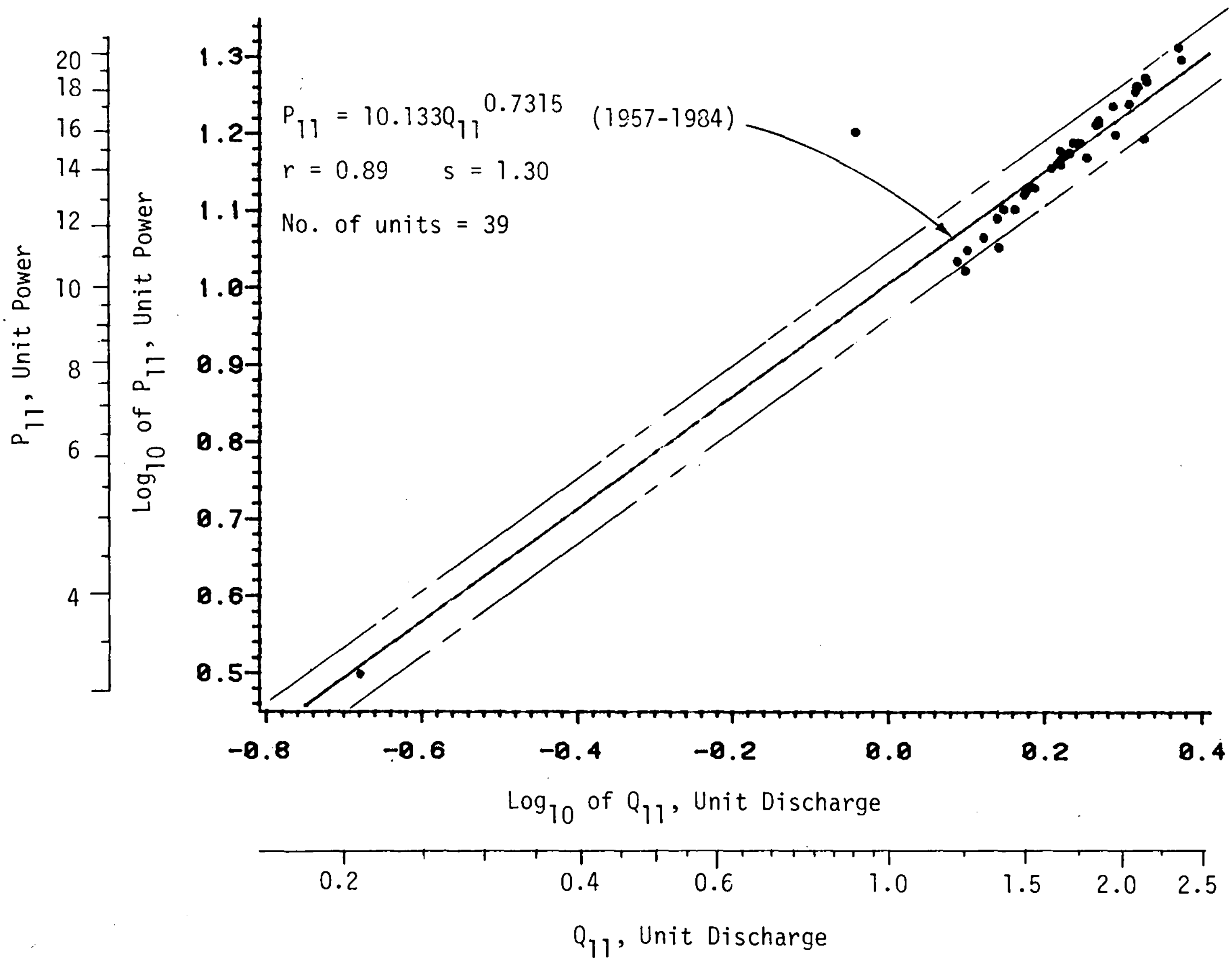

Figure 23. Unit power versus unit discharge for tubular turbines.

Figure 24 presents the relation between unit speed, N_{11}, and unit power, P_{11}, for tubular type turbines studied and the resulting regression equation is:

$$N_{11} = 52.96 \ P_{11}^{0.3882} \qquad \text{Eq. (36)}$$

Figure 25 presents the relation between unit speed, N_{11}, and unit discharge, Q_{11}, for tubular type turbines studied and the resulting regression equation is:

$$N_{11} = 120.144 \ Q_{11}^{0.4210} \qquad \text{Eq. (37)}$$

Using the speed ratio, Ø as the dependent term of characteristic turbine parameter, empirical relations were developed for manufactured tubular type turbines as follows:

$$\emptyset = 0.0389 \ N_s^{\ 0.6013} \qquad \text{Eq. (38)}$$

$$\emptyset = 0.626 \ P_{11}^{\ 0.3882} \qquad \text{Eq. (39)}$$

With the turbine diameter, D, as the dependent term of the empirical relations for manufactured tubular type turbines the following regression equation was developed:

$$D = 1.5424 \ \emptyset^{\ 0.5767} \qquad \text{Eq. (40)}$$

The graphical relations involving the speed ratio, Ø , and the specific speed, N_S, unit power, P_{11}, and tubular turbine diameter, D, are presented in Figures 26, 27 and 28.

The graphical relations relating the tubular turbine diameter, D, to the P/H ratio is presented in Figure 29 and the relation between tubular turbine diameter, D, and Q/N ratio is presented in Figure 30.

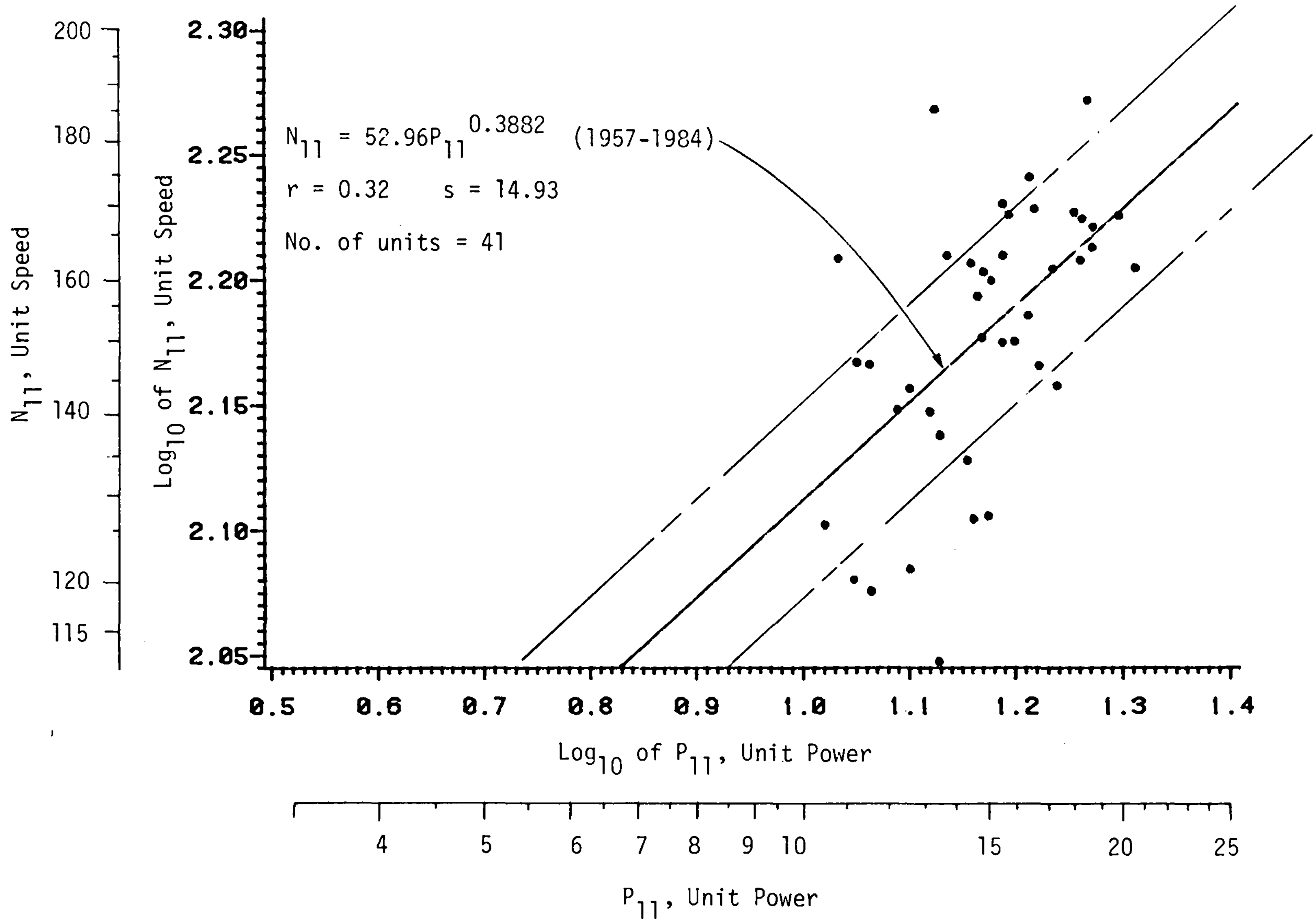

Figure 24. Unit speed versus unit power for tubular turbines.

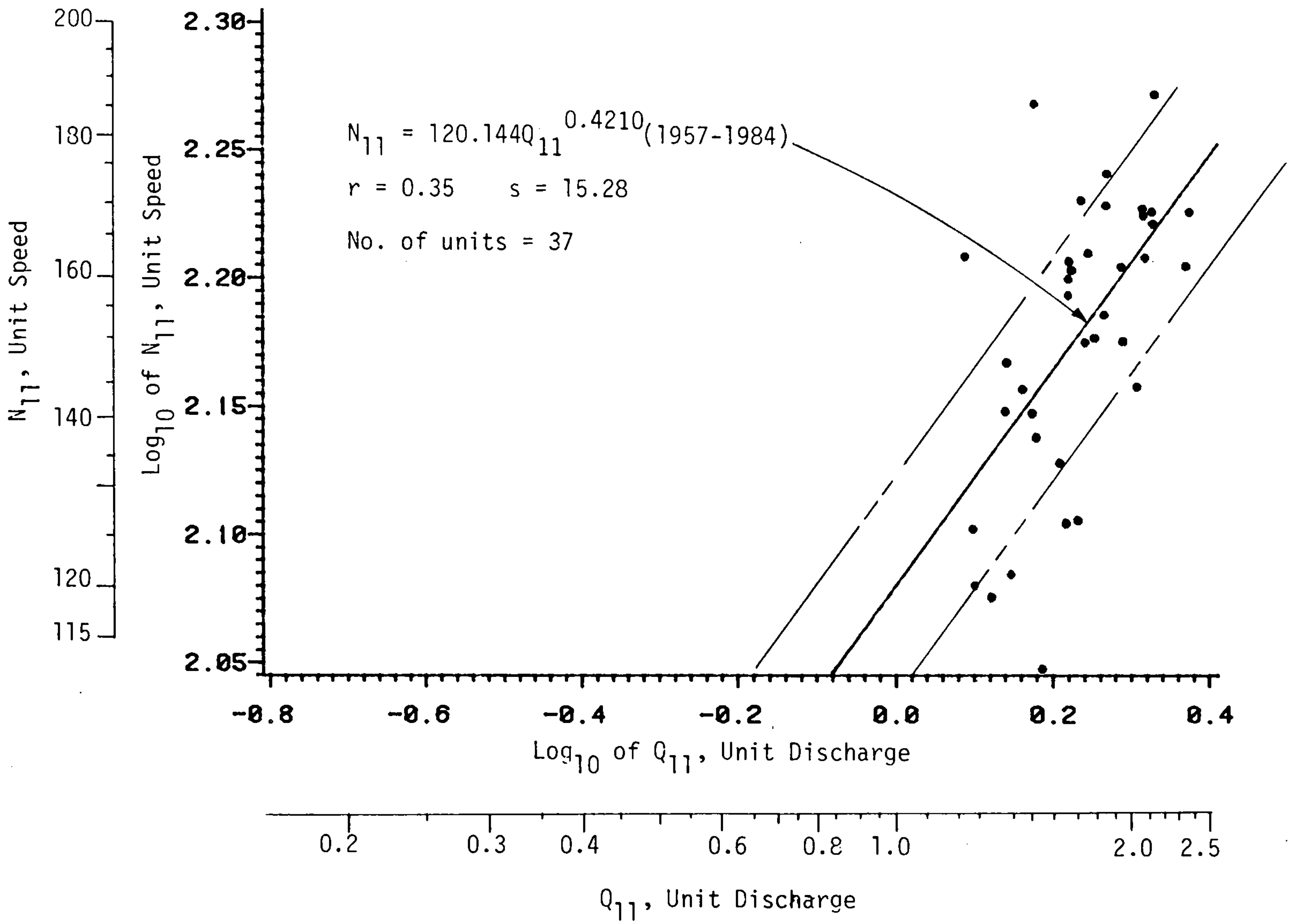

Figure 25. Unit speed versus unit discharge for tubular turbines.

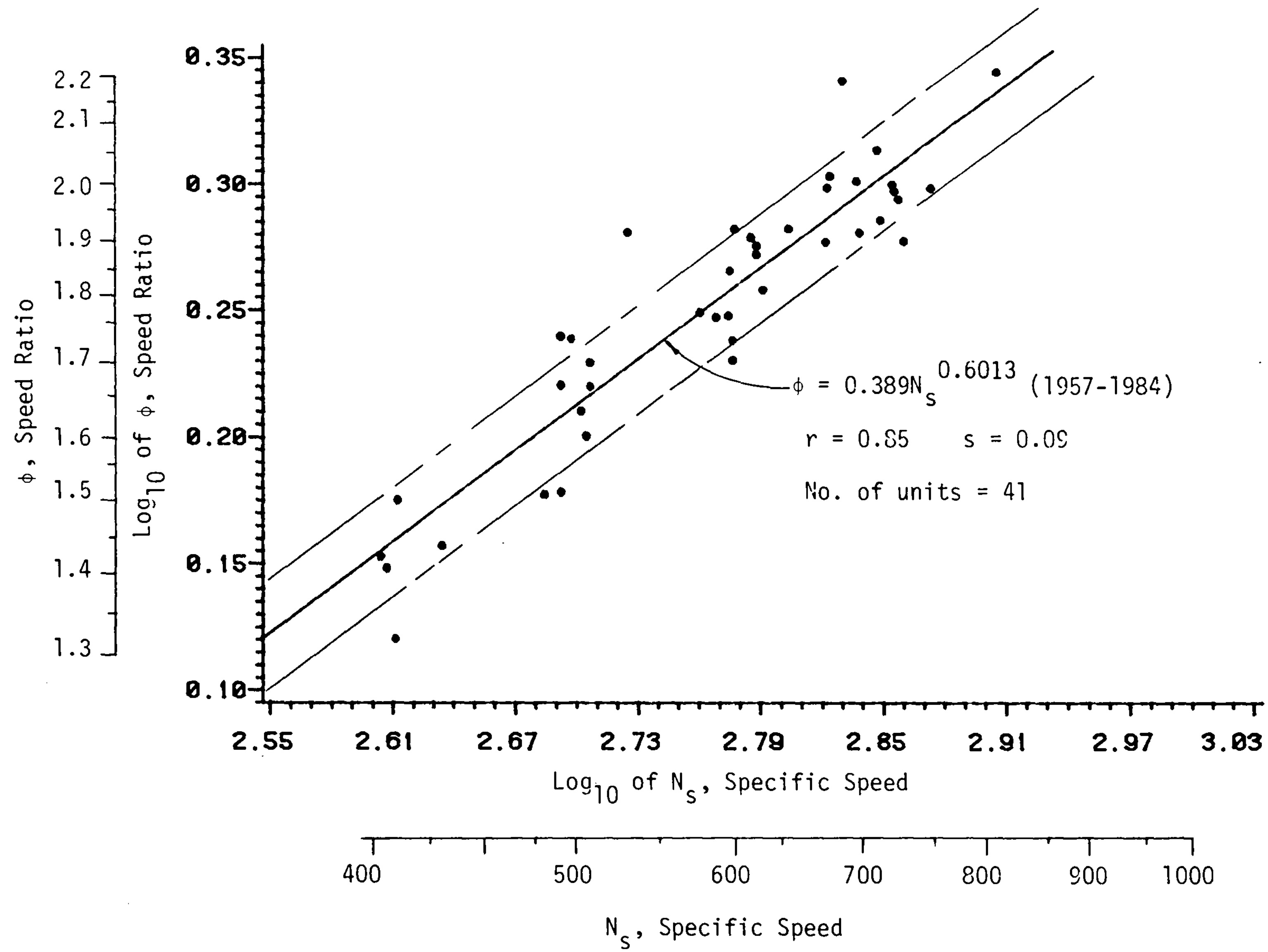

Figure 26. Speed ratio versus specific speed for tubular turbines.

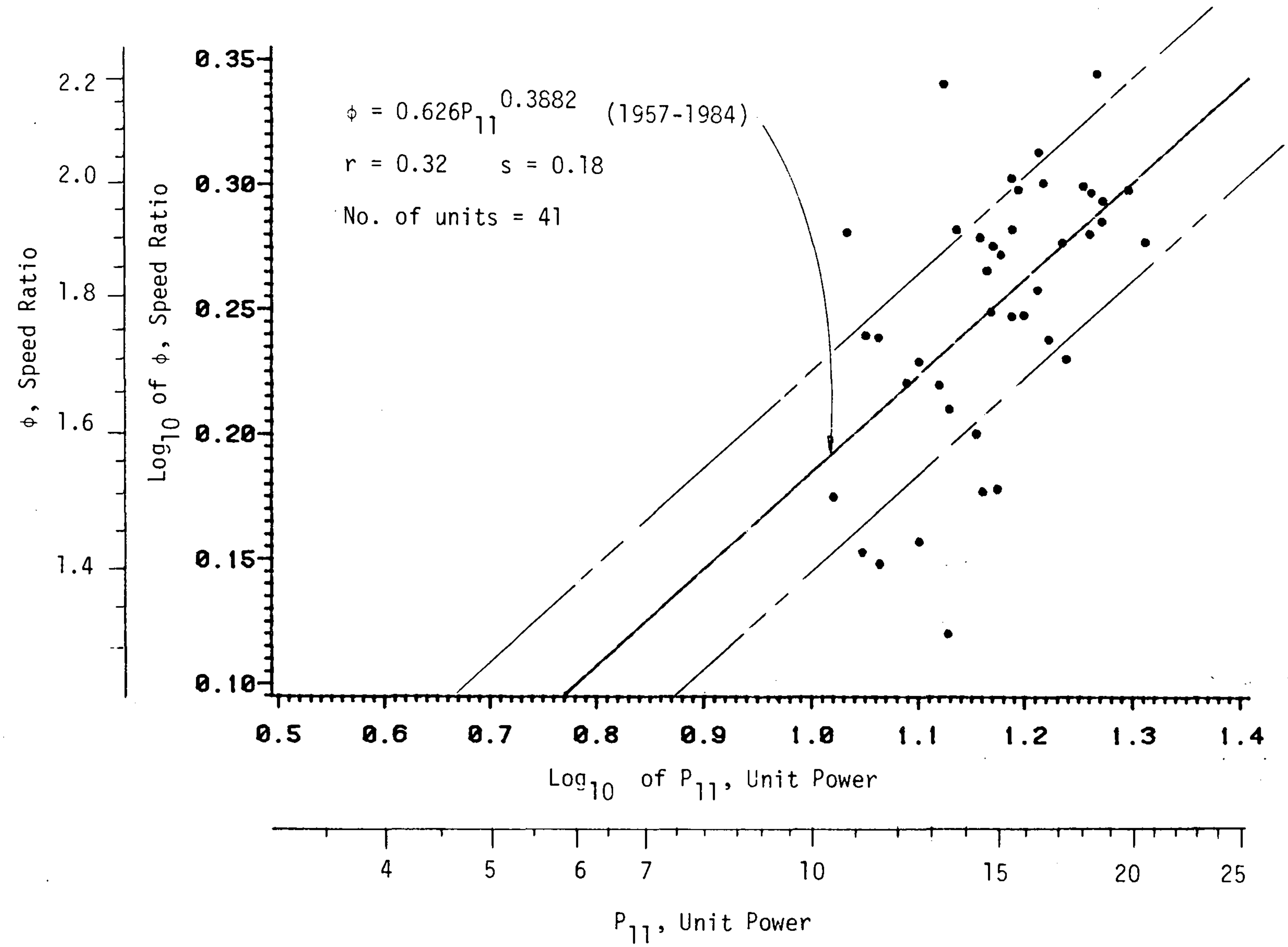

Figure 27. Speed ratio versus unit power for tubular turbines.

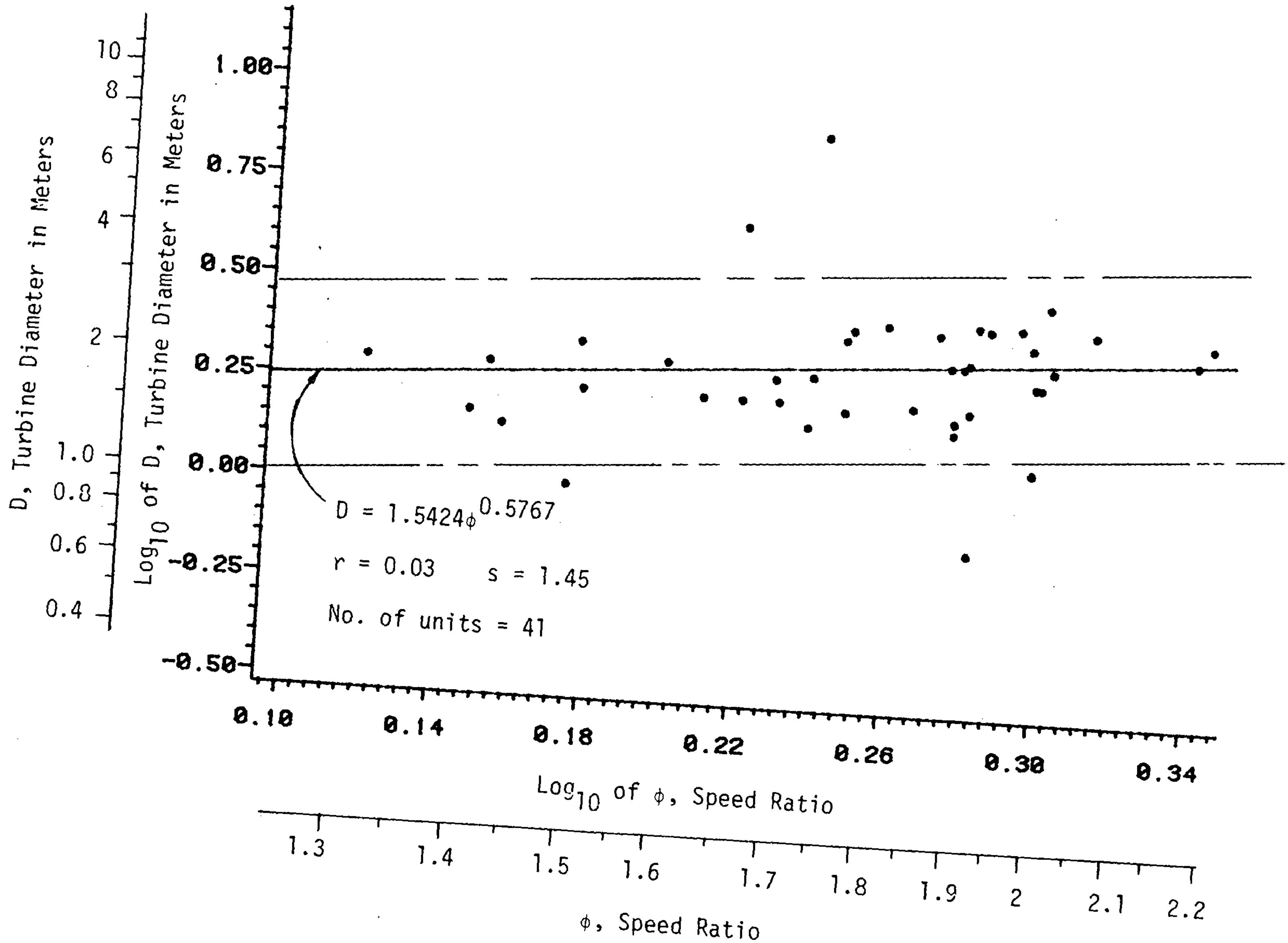

Figure 28. Turbine diamater versus speed ratio for tubular turbines.

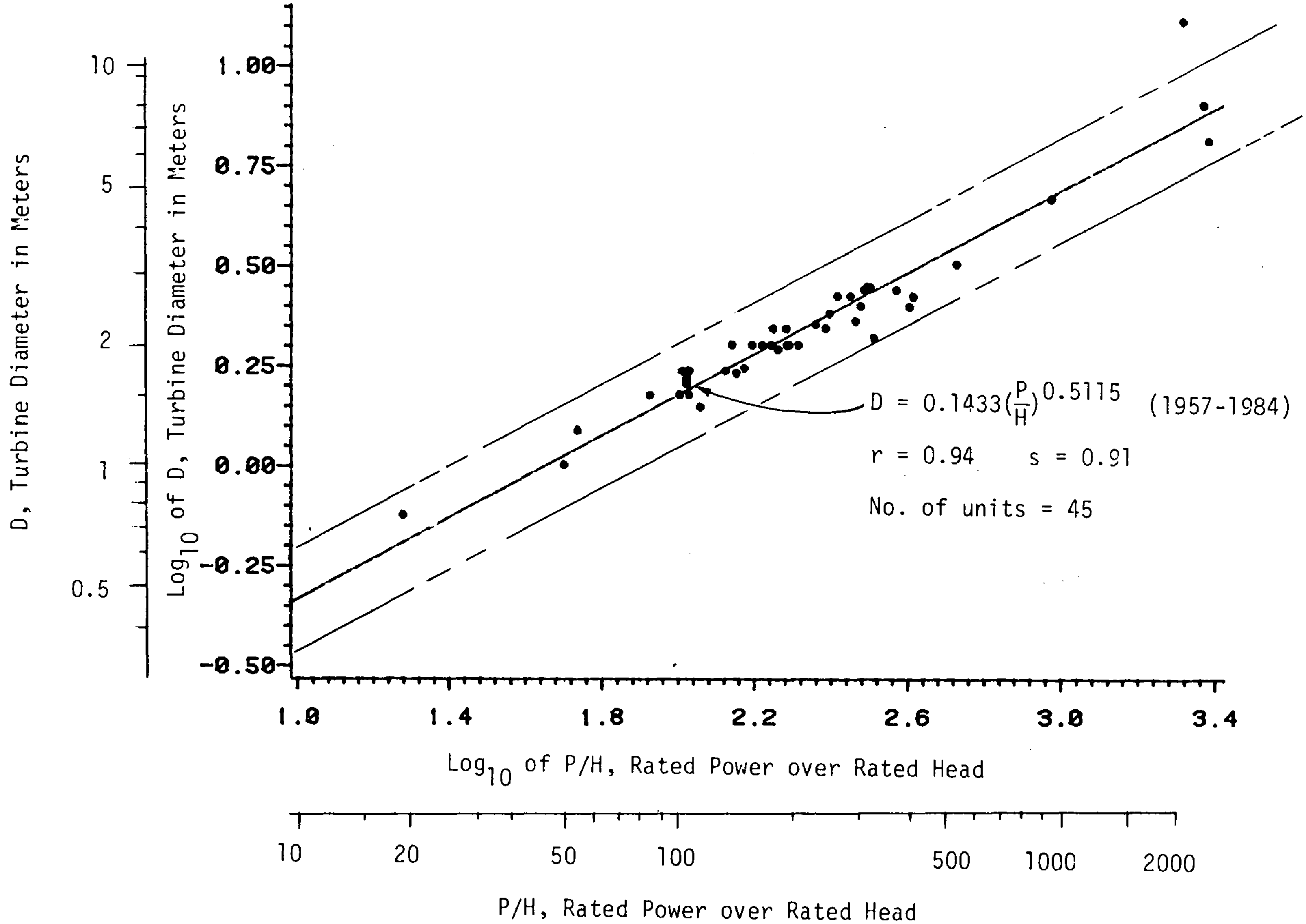

Figure 29. Turbine diameter versus P/H ratio for tubular turbines.

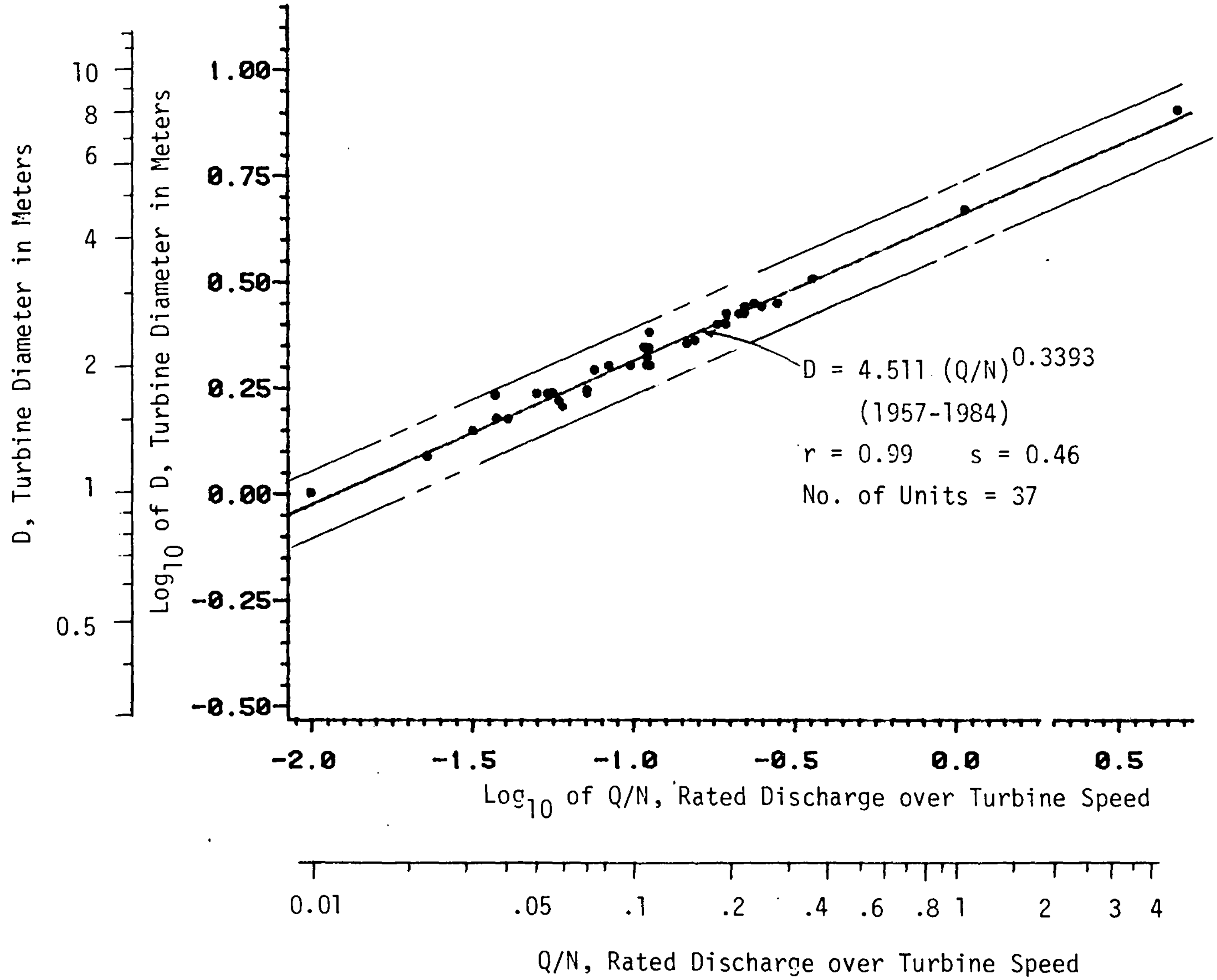

Figure 30. Turbine diameter versus Q/N ratio for tubular turbines.

The empirical relation as a regression equation relating tubular turbine diameter D, to the P/H ratio is given as:

$$D = 0.1433\ (P/H)^{0.5115} \qquad \text{Eq. (41)}$$

The corresponding empirical relation as a regression equation relating tubular turbine diameter, D, to the Q/N ratio is given as:

$$D = 4.511\ (Q/N)^{0.3393} \qquad \text{Eq. (42)}$$

The additional new relation relating turbine speed, N, to the ratio of rated power output, P, to the rated head, H, is given by the following regression equation:

$$N = 2044.395\ (P/H)^{-0.4329} \qquad \text{Eq. (43)}$$

This relation is shown graphically in Figure 31.

The regression equation for tubular turbines relating turbine speed to the ratio $\sqrt{H}/D$ is given as:

$$N = 156.193\ (\sqrt{H}/D)^{0.8895} \qquad \text{Eq. (44)}$$

This relation is shown graphically in Figure 32.

Table 3 summaries all the regression relations that were developed for manufactured tubular type turbines. In the table are shown all the equations that were developed, the regression correlation coefficient for each particular regression, the corresponding standard deviation, the sample period and the number of different manufactured units used in developing a particular relation.

<u>Cross-Flow Turbines</u>

For cross-flow type turbines the specific speed, N_s, vs rated head, H, relation is shown in Figure 33 and the resulting regression equation is given as:

$$N_s = 513.846\ H^{-0.5047} \qquad \text{Eq. (45)}$$

TABLE 3

SUMMARY LISTING OF REGRESSION INFORMATION AND EQUATIONS RELATING TURBINE CHARACTERISTICS TO VARIOUS TURBINE CONSTANTS FOR TUBULAR TURBINES

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
31	N_s	$N_s = 1107.303\ H^{-0.2998}$	0.62	92.71	1957-1984	54
32	N_s	$N_s = 52.96\ P_{11}^{0.8882}$	0.71	55.91	1957-1984	41
33	N_s	$N_s = 357.294\ Q_{11}^{0.9029}$	0.70	59.37	1957-1984	37
34	N_s	$N_s = 0.497\ N_{11}^{1.4080}$	0.85	44.20	1957-1984	41
35	P_{11}	$P_{11} = 10.133\ Q_{11}^{0.7315}$	0.89	1.30	1957-1984	39
36	N_{11}	$N_{11} = 52.96\ P_{11}^{0.3882}$	0.32	14.93	1957-1984	41
37	N_{11}	$N_{11} = 120.144\ Q_{11}^{0.4210}$	0.35	15.28	1957-1984	37
38	ϕ	$\phi = 0.0389\ N_s^{0.6013}$	0.85	0.09	1957-1984	41

TABLE 3 CONTINUED

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
39	ϕ	$\phi = 0.626\ P_{11}^{0.3882}$	0.32	0.18	1957-1984	41
40	D	$D = 1.5424\ \phi^{0.5767}$	0.03	1.45	1957-1984	41
41	D	$D = 0.1433(\frac{P}{H})^{0.5115}$	0.94	0.91	1957-1984	45
42	D	$D = 4.511(Q/N)^{0.3393}$	0.99	0.46	1957-1984	37
43	N	$N = 2044.395(P/H)^{-0.4329}$	0.69	114.60	1957-1984	54
44	N	$N = 156.193(\frac{\sqrt{H}}{D})^{0.8895}$	0.95	29.47	1957-1984	41

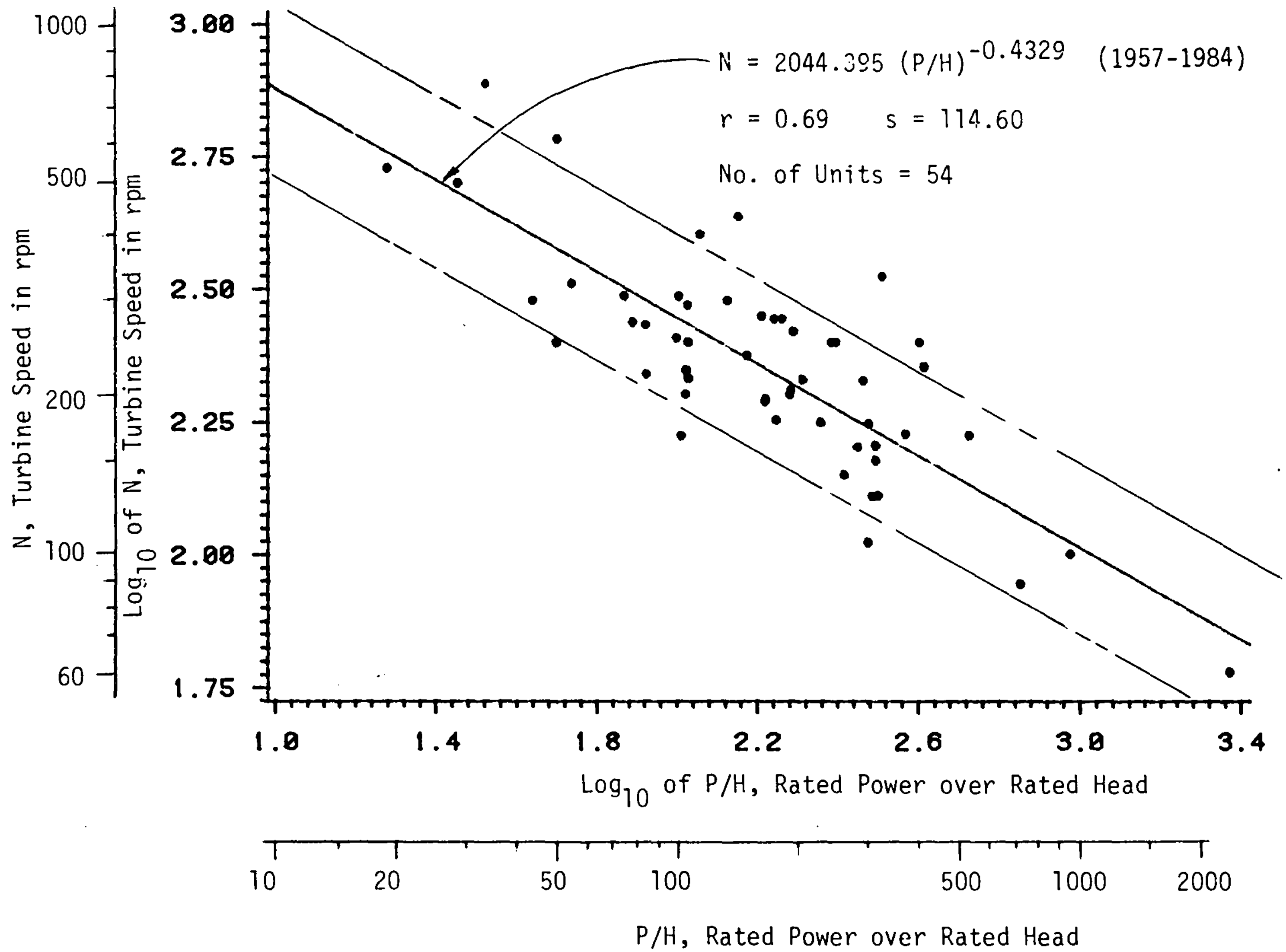

Figure 31. Turbine Speed versus P/H ratio for tubular turbines.

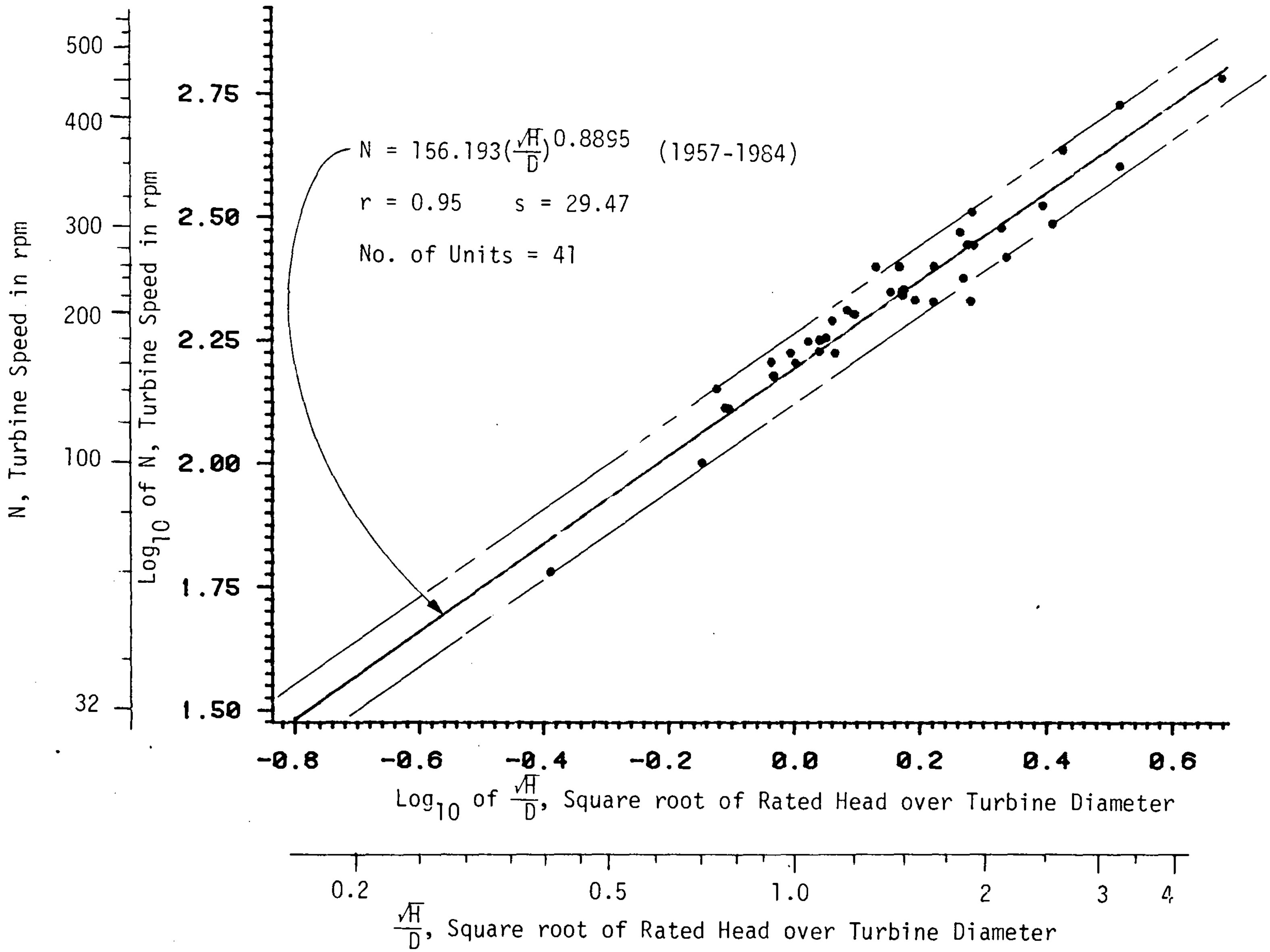

Figure 32. Turbine speed versus $\frac{\sqrt{H}}{D}$ ratio for tubular turbines.

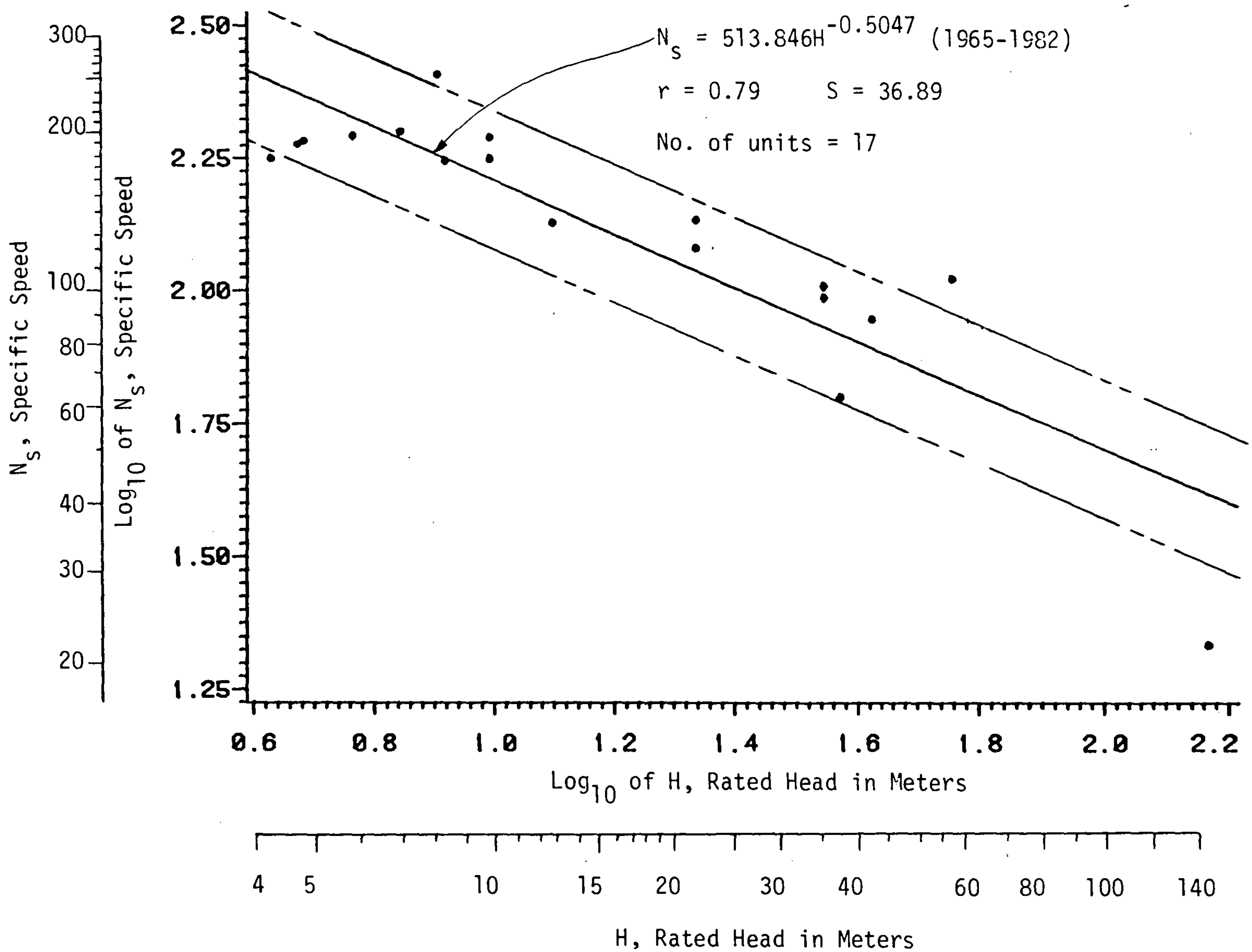

Figure 33. Specific speed versus rated head for cross-flow turbines.

Here again only one manufacturer's equipment was studied and no stratification of experience data was attempted for the modern units that have been manufactured. Figure 34 presents the relation between specific speed, N_s, and unit power, P_{11}, for cross-flow turbines studied and the resultant regression equation is given as:

$$N_s = 41.989\ P_{11}^{0.5049} \qquad \text{Eq. (46)}$$

Figure 35 presents the relation between specific speed, N_s, and unit discharge, Q_{11}, for cross-flow turbines studied and the resultant regression equation is given as:

$$N_s = 120.605\ Q_{11}^{0.4958} \qquad \text{Eq. (47)}$$

Figure 36 presents the relation between specific speed, N_s, and unit speed, N_{11}, for cross-flow turbines studied and the resultant regression equation is given as:

$$N_s = 1.249\ N_{11}^{1.2379} \qquad \text{Eq. (48)}$$

Figure 37 presents the relation between unit power, P_{11}, and unit discharge, Q_{11}, for cross-flow turbines studied and the resultant regression equation is given as:

$$P_{11} = 8.0743\ Q_{11}^{0.9905} \qquad \text{Eq. (49)}$$

Figure 38 presents the relation between unit speed, N_{11}, and unit power, P_{11}, for cross-flow turbines studied and the resultant regression equation is given as:

$$N_{11} = 41.989\ P_{11}^{0.0049} \qquad \text{Eq. (50)}$$

Figure 39 presents the relation between unit speed, N_{11}, and unit discharge, Q_{11}, for cross-flow turbines studied and the resultant regression equation is given as:

$$N_{11} = 42.444\ Q_{11}^{0.0005} \qquad \text{Eq. (51)}$$

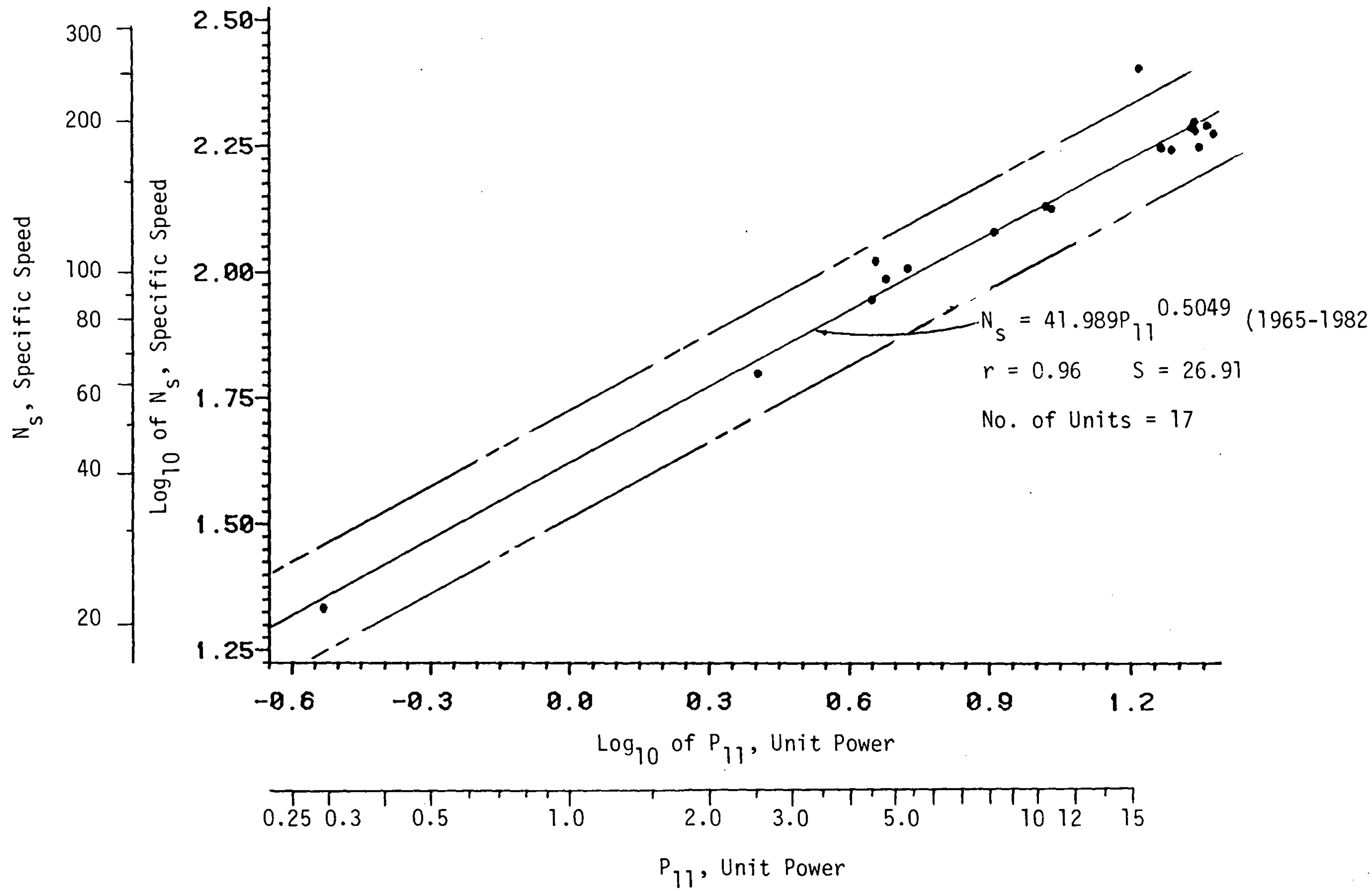

Figure 34. Specific speed versus unit power for cross-flow turbines.

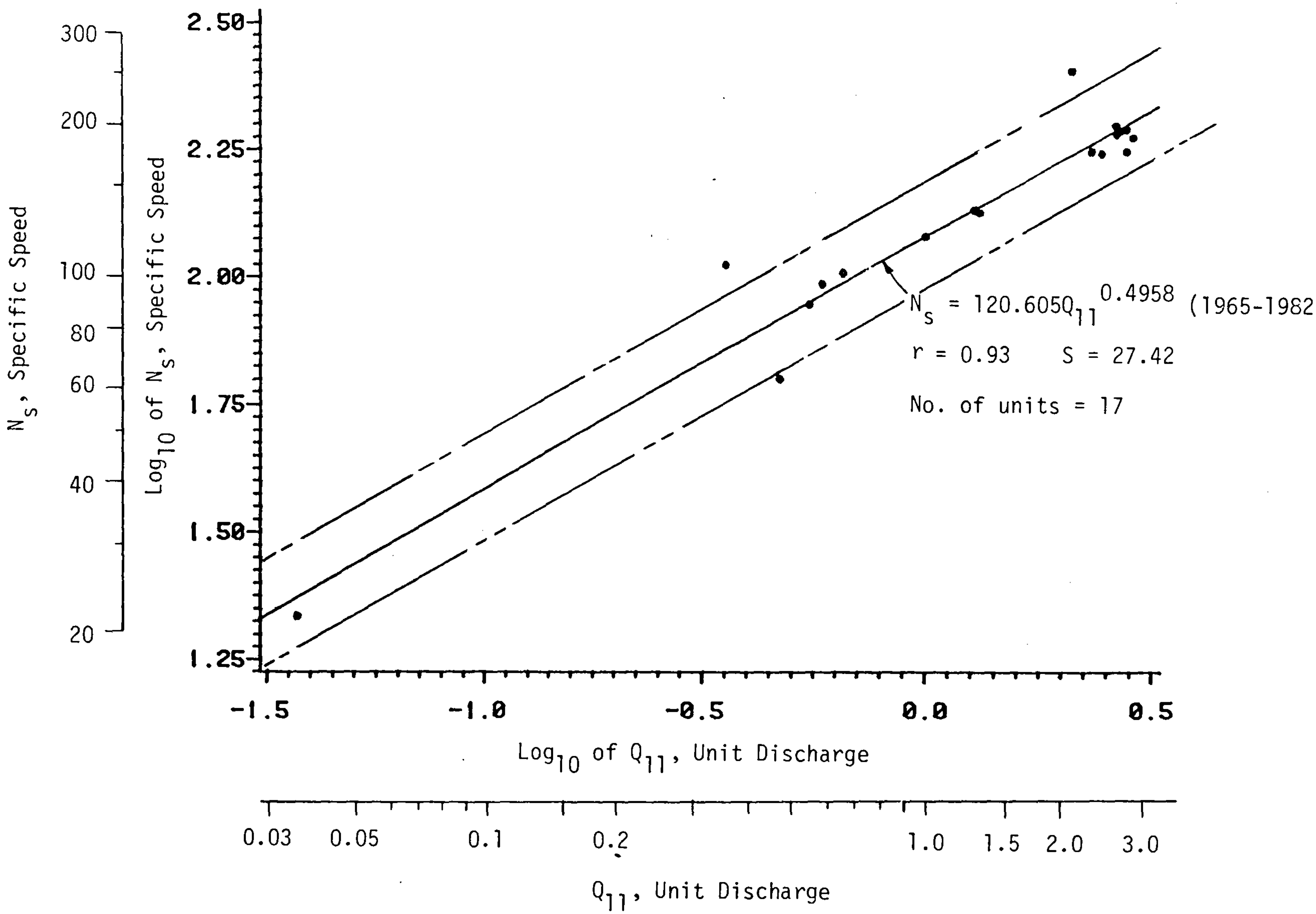

Figure 35. Specific speed versus unit discharge for cross-flow turbines.

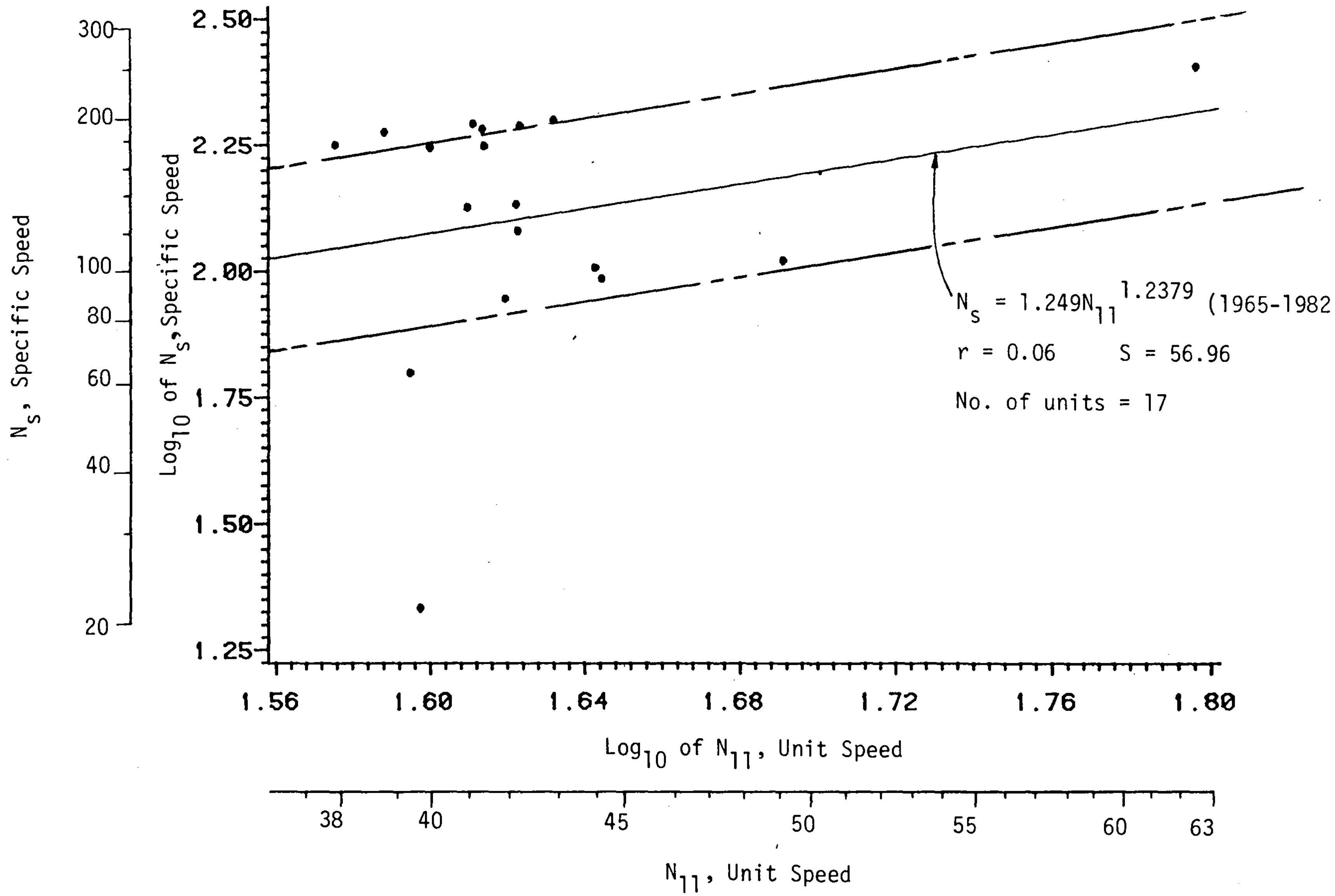

Figure 36. Specific speed versus unit speed for cross-flow turbines.

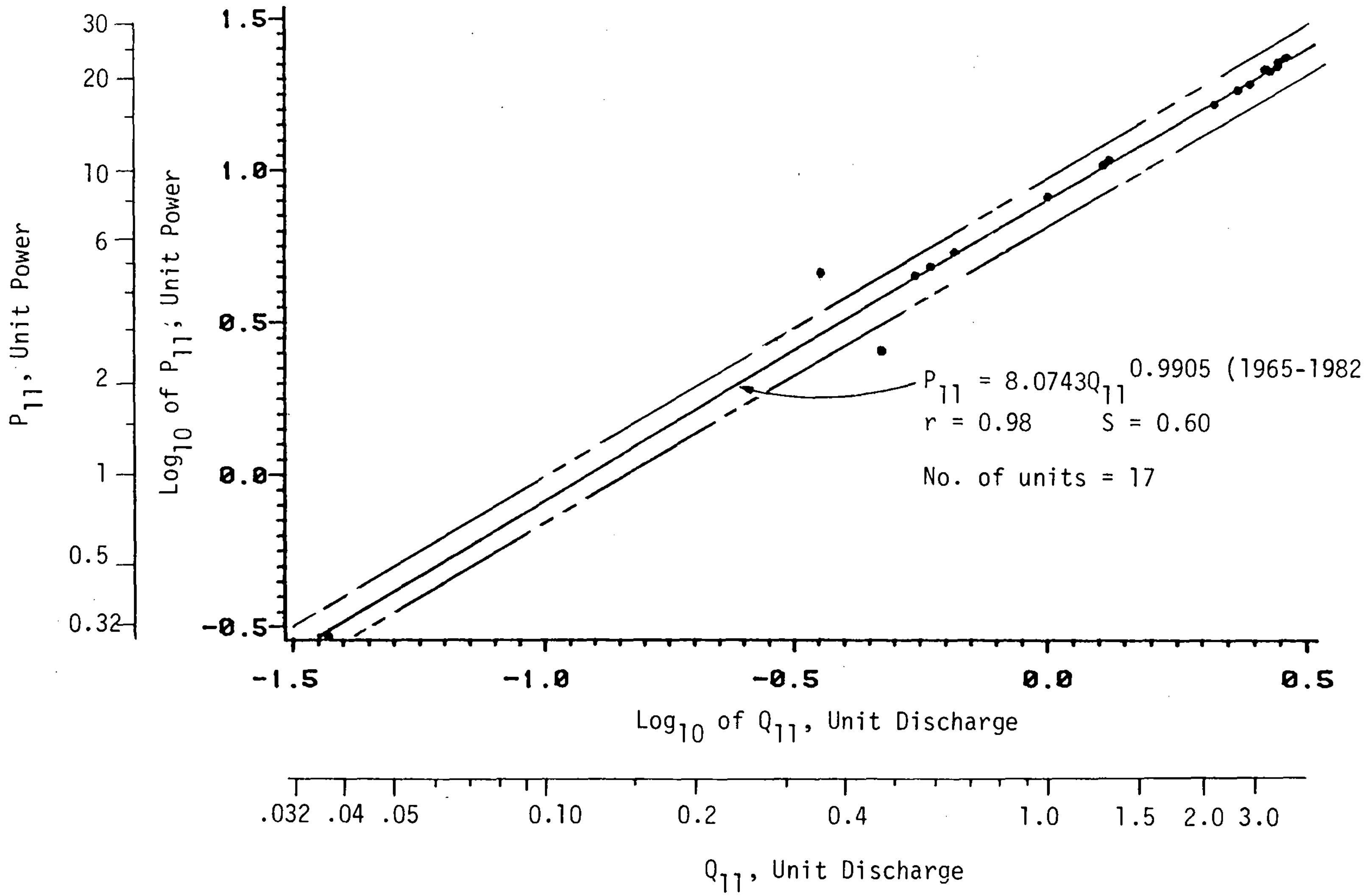

Figure 37. Unit power versus unit discharge for cross-flow turbines.

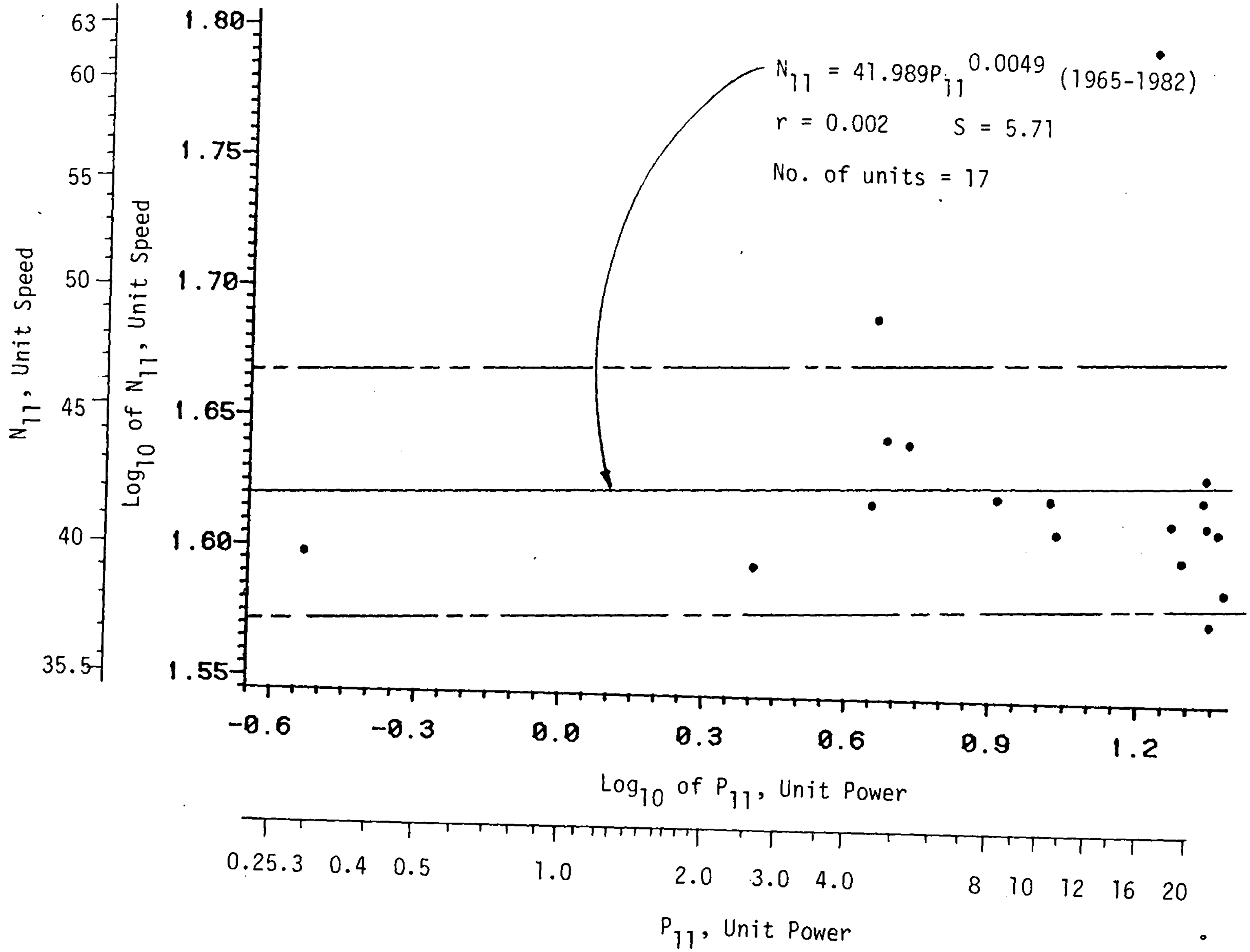

Figure 38. Unit speed versus unit power for cross-flow turbines.

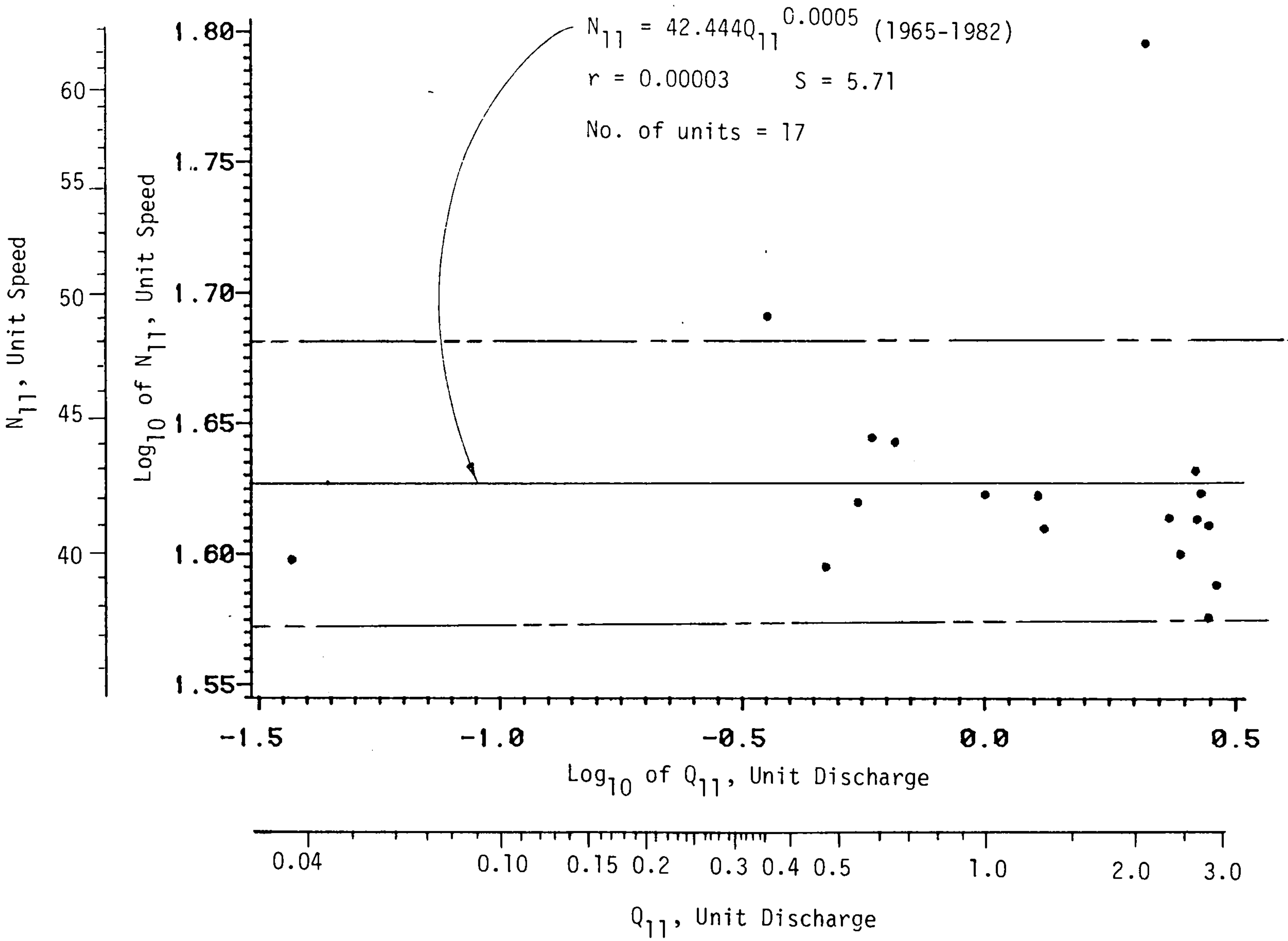

Figure 39. Unit speed versus unit discharge for cross-flow turbines.

Using the speed ratio, $\emptyset$, as a dependent term of characteristic turbine parameters empirical relations were developed for cross-flow type turbines studied as follows:

$$\emptyset = 0.3977\ N_s^{0.0478} \qquad \text{Eq. (52)}$$

$$\emptyset = 0.4963\ P_{11}^{0.005} \qquad \text{Eq. (53)}$$

The regression equation relating the cross-flow turbine diameter D, to the speed ratio, $\emptyset$, is given as:

$$D = 1.2151\ \emptyset^{0.6254} \qquad \text{Eq. (54)}$$

The graphical relations involving the speed ratio, $\emptyset$ and the specific speed, N_s, unit power, P_{11} and cross-flow turbine diameter, D, are presented in Figure 40, 41 and 42.

The graphical relations relating the cross-flow turbine diameter, D, to the P/H ratio is presented in Figure 43 and the relation between cross-flow turbine diameter, D, and the Q/N ratio is presented in Figure 44. The empirical relation as a regression equation relating cross-flow turbine diameter, D, to the P/H ratio is given as:

$$D = 0.354\ (P/H)^{0.2571} \qquad \text{Eq. (55)}$$

The corresponding empirical relation as a regression equation relating cross-flow turbine diameter, D, to the Q/N ratio is given as:

$$D = 1.5848\ (Q/N)^{0.1615} \qquad \text{Eq. (56)}$$

The additional empirical relation as a regression equation relating cross-flow turbine speed, N, to the P/H ratio is given as:

$$N = 1126.25\ (P/H)^{-0.5367} \qquad \text{Eq. (57)}$$

The regression equation for cross-flow turbines relating turbine speed, N, to the ratio $\sqrt{H}/D$, is given as:

$$N = 42.866(\sqrt{H}/D)^{0.9939} \qquad \text{Eq. (58)}$$

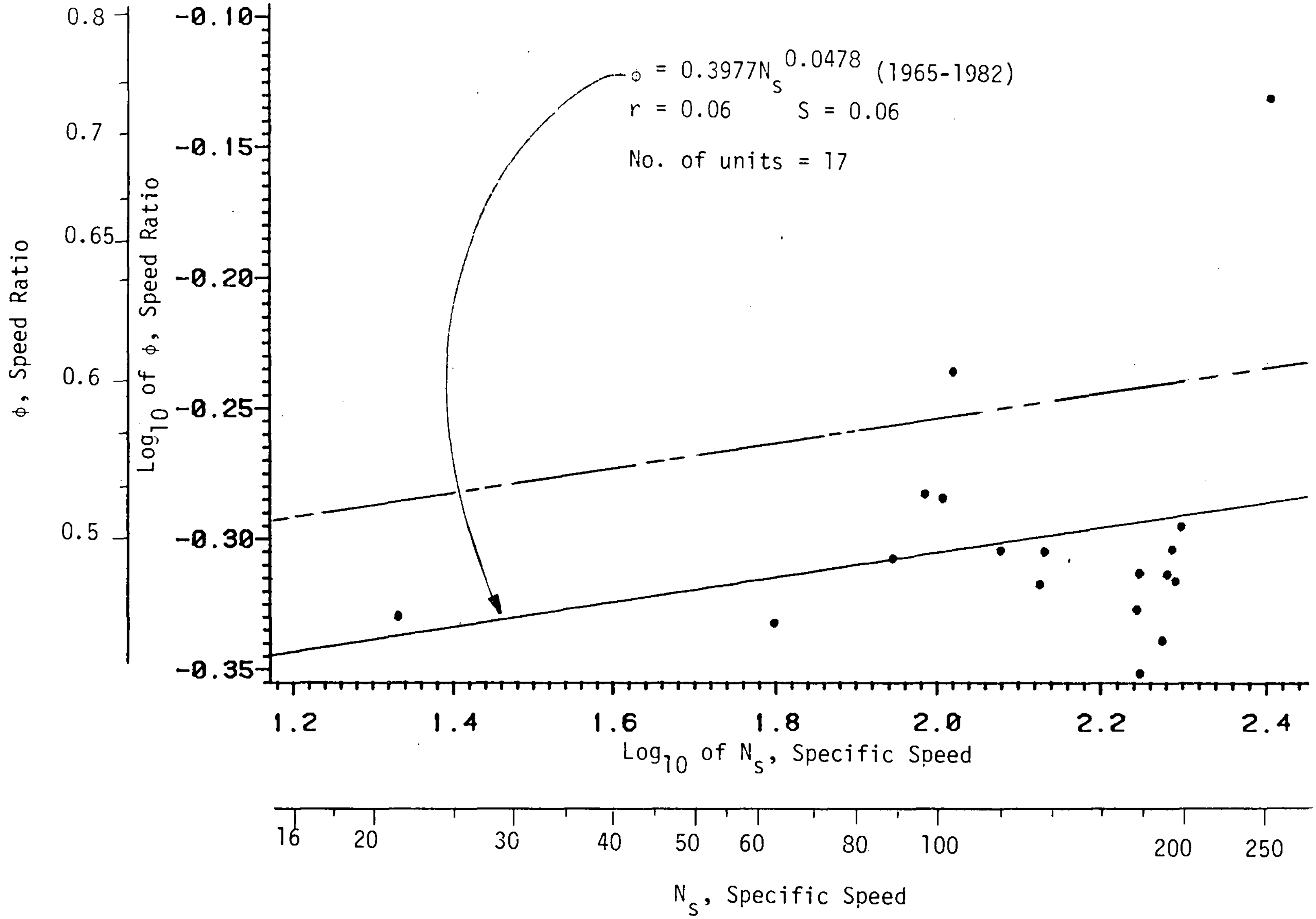

Figure 40. Speed ratio versus specific speed for cross-flow turbines.

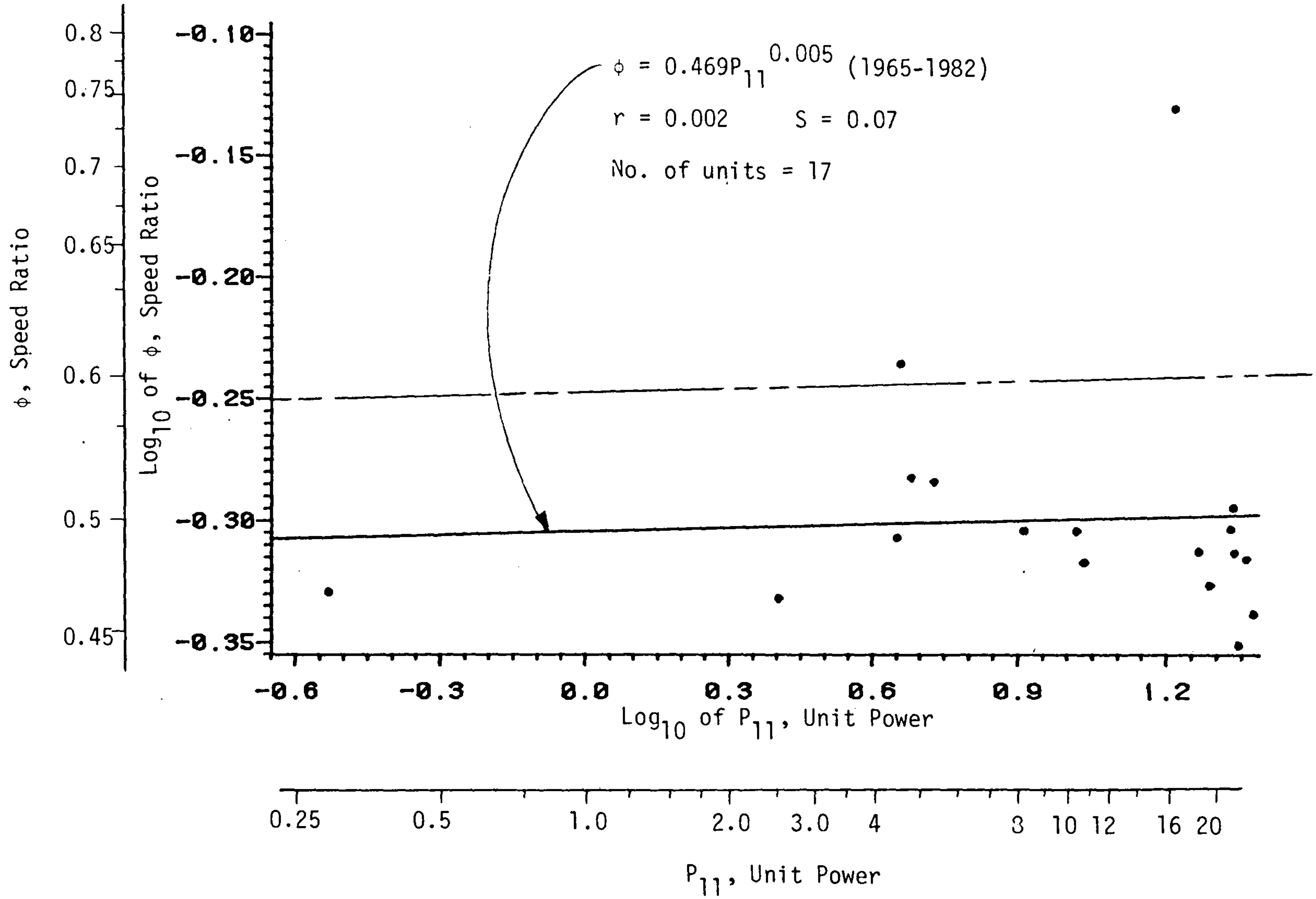

Figure 41. Speed ratio versus unit power for cross-flow turbines.

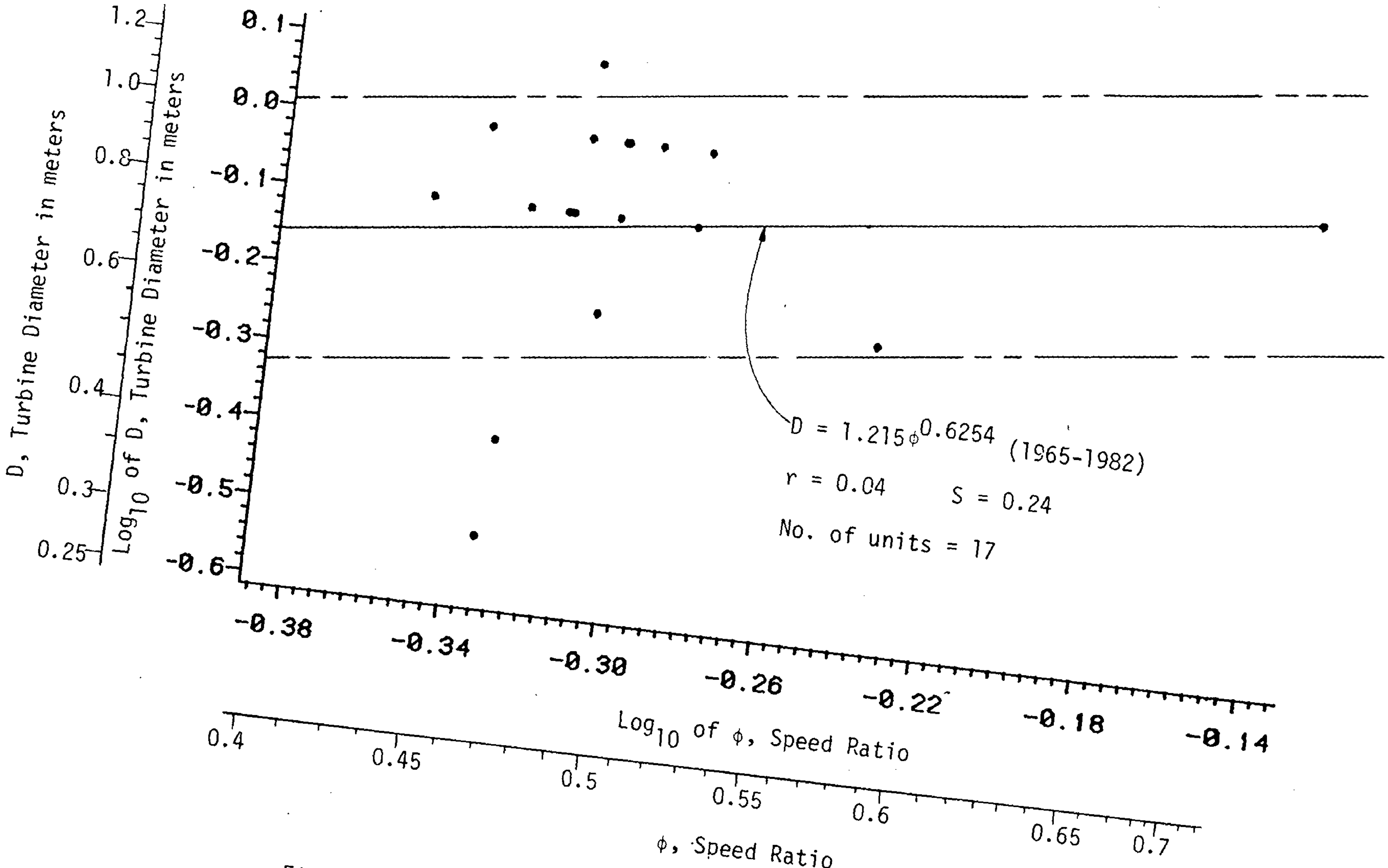

Figure 42. Turbine diameter versus speed ratio for cross-flow turbines.

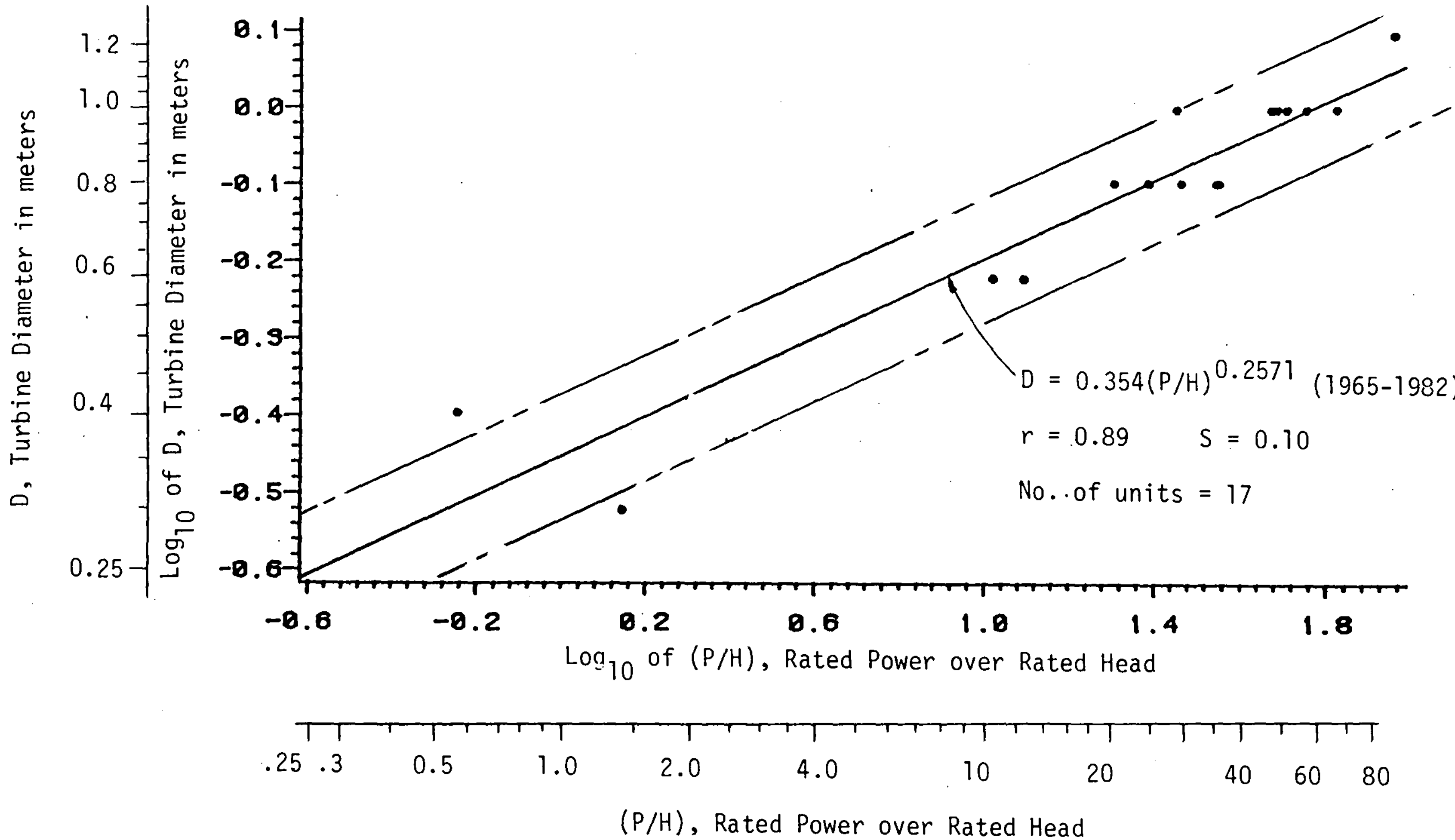

Figure 43. Turbine diameter versus (P/H) ratio for cross-flow turbines.

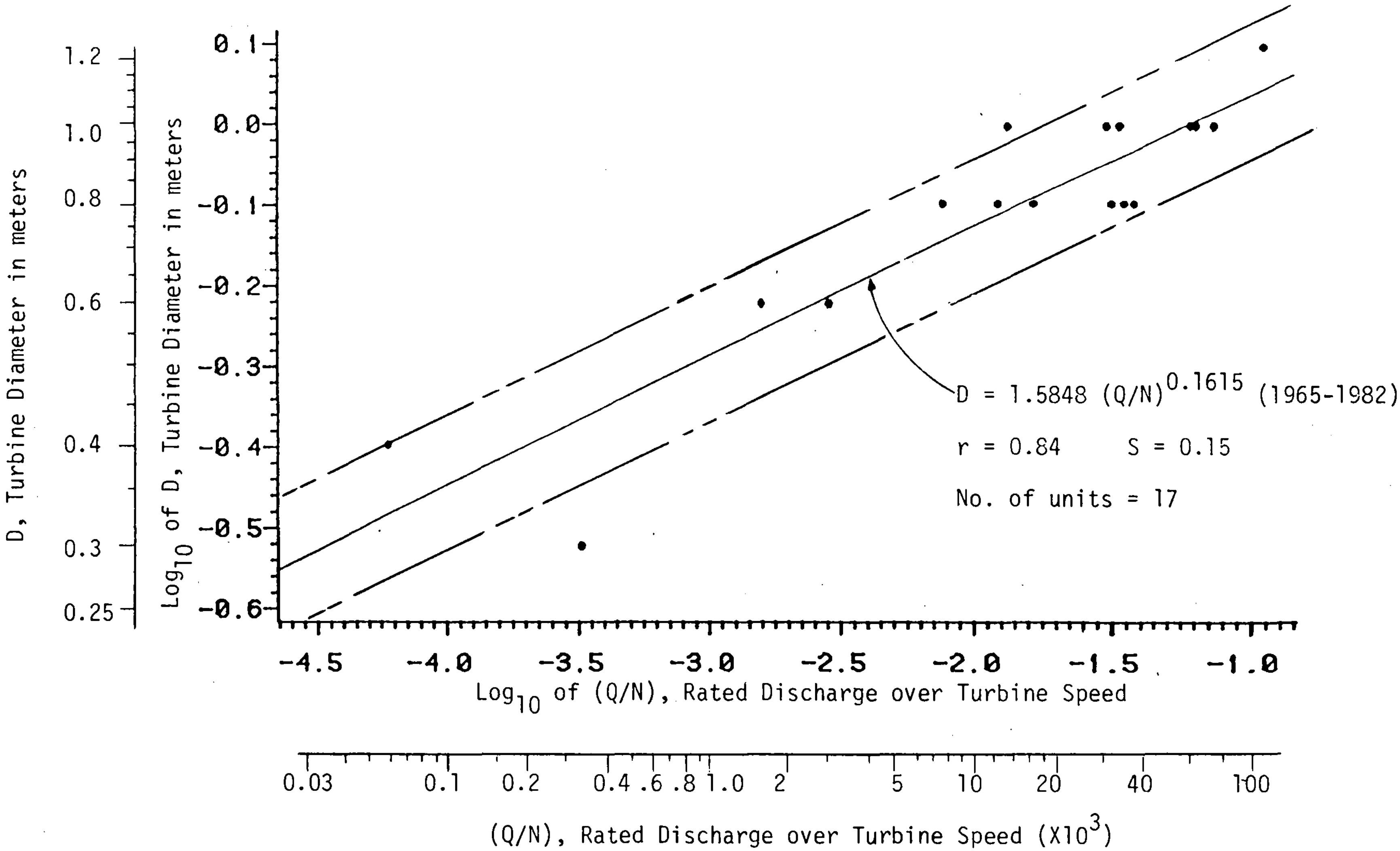

Figure 44. Turbine diameter versus (Q/N) ratio for cross-flow turbines.

Table 4 summarizes all the regression relations that were developed for manufactured cross-flow type turbines. In the table are shown all the equations that were developed, the regressions correlation coefficient for each particular regression, the particular standard deviation, and the number of different manufactured units used in developing a particular relation.

TURBINE SETTING CHARACTERISTICS

It is common practice to relate a turbine constant known as the cavitation coefficient or plant sigma to the specific speed for experience curves. The equation for the plant sigma is given as follows:

$$\sigma = \frac{H_a - H_v - H_s}{H} \qquad \text{Eq. (59)}$$

where σ = plant sigma, dimensionless

H_a = atmospheric pressure head in ft or meters

H_v = vapor pressure head at temperature of water issuing from turbine in ft or meters

H_s = difference in elevation between minimum tailwater level and the cavitation reference point at the outflow from the turbine in ft or meters

H = net effective head in feet or meters

The term, H_s, is referred to as suction head and it has slightly different designation depending on the type of turbine, the location of the tailwater and the orientation of the turbine and turbine shaft. A related term is, z, the draft head the difference in elevation between the tailwater level and the centerline of the distributor or the centerline of the turbine runner. Figure 45 shows diagramatically what these two terms are for different types of reaction turbines having

TABLE 4

SUMMARY LISTING OF REGRESSION INFORMATION AND EQUATIONS RELATING

TURBINE CHARACTERISTICS TO VARIOUS TURBINE CONSTANTS FOR CROSS-FLOW TURBINE

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
45	N_s	$N_s = 513.846\ H^{-0.5047}$	0.79	36.89	1965-1982	17
46	N_s	$N_s = 41.989\ P_{11}^{0.5049}$	0.96	26.91	1965-1982	17
47	N_s	$N_s = 120.605\ Q_{11}^{0.4958}$	0.93	27.42	1965-1982	17
48	N_s	$N_s = 1.249\ N_{11}^{1.2379}$	0.06	56.96	1965-1982	17
49	P_{11}	$P_{11} = 8.0743\ Q_{11}^{0.9905}$	0.98	0.60	1965-1982	17
50	N_{11}	$N_{11} = 41.989\ P_{11}^{0.0049}$	0.002	5.71	1965-1982	17
51	N_{11}	$N_{11} = 42.444\ Q_{11}^{0.0005}$	0.00003	5.71	1965-1982	17
52	ϕ	$\phi = 0.3977\ N_s^{0.0478}$	0.06	0.06	1965-1982	17

TABLE 4 CONTINUED

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
53	ϕ	$\phi = 0.4963\ P_{11}^{0.005}$	0.002	0.07	1965-1982	17
54	D	$D = 1.2151\ \phi^{0.6254}$	0.04	0.24	1965-1982	17
55	D	$D = 0.354\ (P/H)^{0.2571}$	0.89	0.10	1965-1982	17
56	D	$D = 1.5848(Q/N)^{0.1615}$	0.84	0.15	1965-1982	17
57	N	$N = 1126.25(P/H)^{-0.5367}$	0.79	213.95	1965-1982	17
58	N	$N = 42.866(\frac{\sqrt{H}}{D})^{0.9939}$	0.98	31.55	1965-1982	17

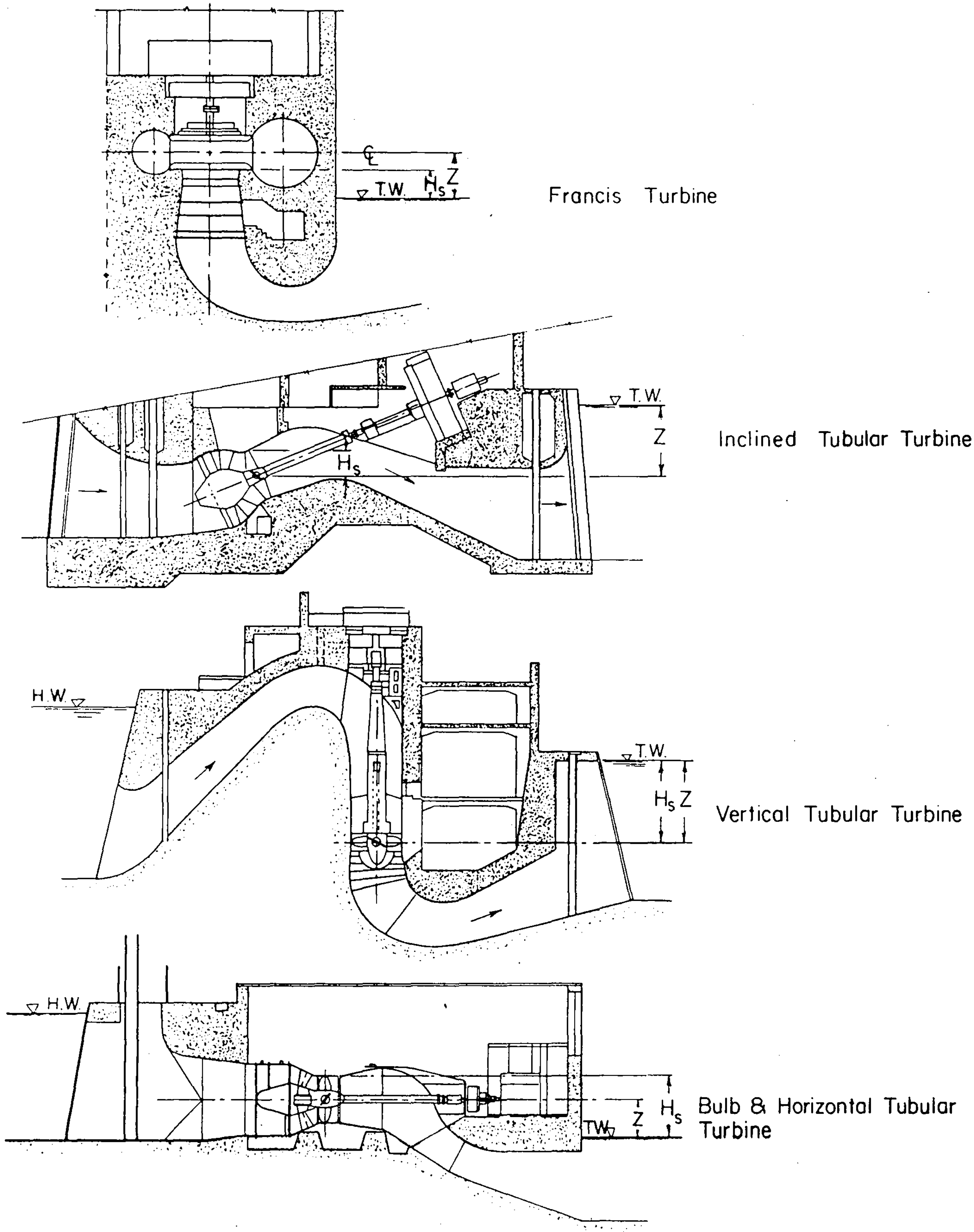

Figure 45. Definition diagram for suction head, H_s and draft head, Z, for different types of turbines.

different shaft orientations. Sometimes difficulty is experienced in relating the plant sigma to other turbine characteristics because the cavitation reference point is not always consistently defined. In this study for the axial flow units which includes bulb type units, the tubular type units, and the rim-generator units the cavitation reference point was taken as the highest point on the propeller blade above the tailwater level. In the case of cross-flow turbines the pressure in the runner zone is essentially atmospheric pressure and is therefore not subject to cavitation. No turbine setting and plant sigma analysis was done on the cross-flow turbines.

Bulb Turbines

Figure 46 presents stratification of the relation between the plant sigma, σ, and the specific speed, N_s, for six different turbine companies' manufactured bulb type turbines. It is interesting to note that the correlation coefficient for different companies varies quite markedly. The empirical equations for the relation between plant sigma, σ, and specific speed, N_s, for the respective manufacturer's units are indicated below:

	Source	
$\sigma = 4.549 \times 10^{-6} N_s^{1.908}$ *	KMW	Eq. (60)
$\sigma = 313.332 \times 10^{-6} N_s^{1.274}$ *	NO-KMW	Eq. (61)
$\sigma = 0.097 \times 10^{-6} N_s^{2.479}$ *	TAMP	Eq. (62)
$\sigma = 111.435 \times 10^{-6} N_s^{1.423}$ *	VOITH	Eq. (63)
$\sigma = 80.774 \times 10^{-6} N_s^{1.491}$ *	VEVEY	Eq. (64)
$\sigma = 1541.62 \times 10^{-6} N_s^{1.015}$ *	VOEST ALPINE	Eq. (65)

*The values of σ are based on the definition of plant sigma used in this study.

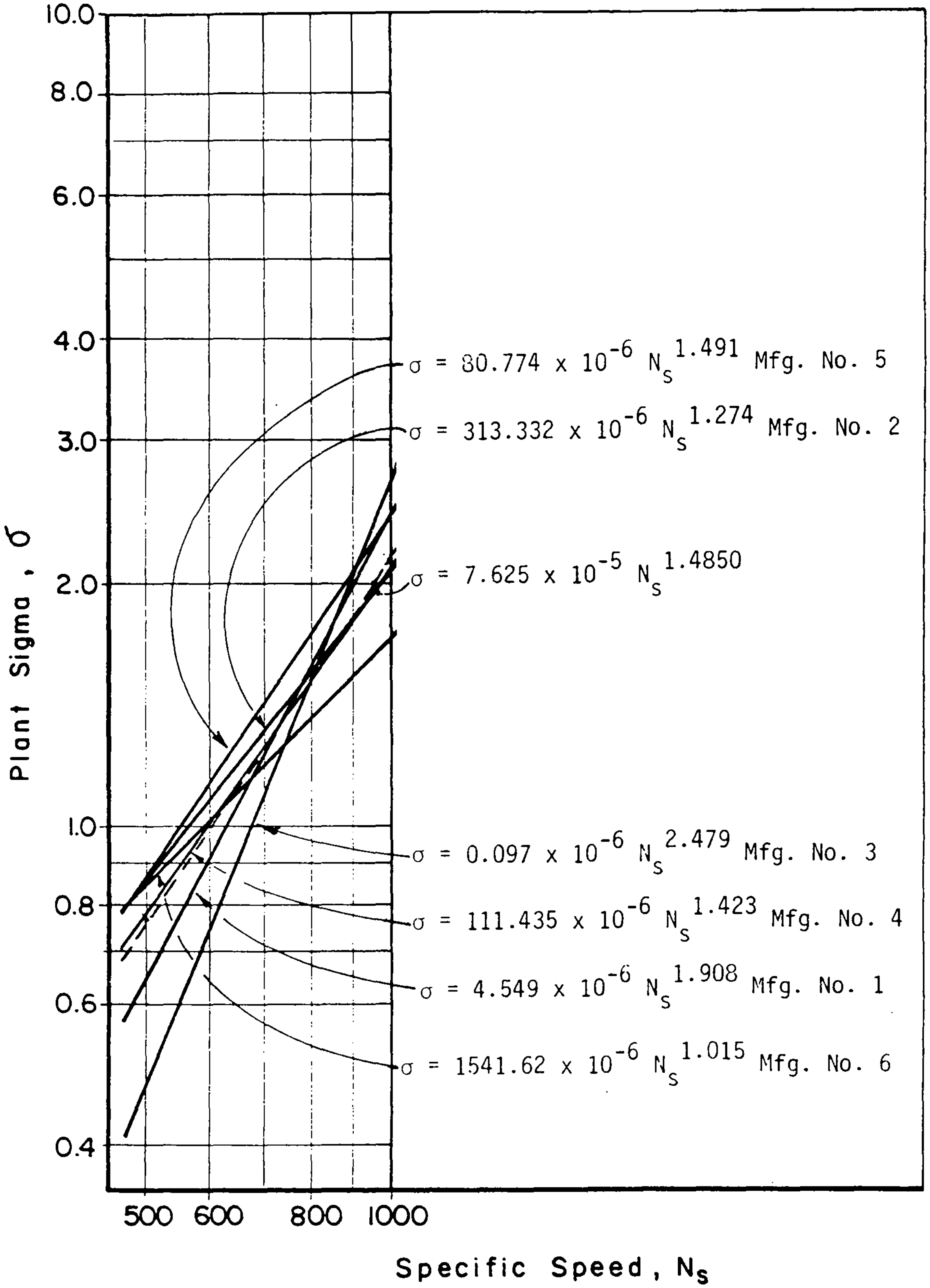

Figure 46. Stratification of relation between plant sigma and specific speed for different manufacturers.

Figure 46 also presents a composite experience curve of the relation between plant sigma, σ, and specific speed, N_S, for all manufactured bulb turbines for which turbine setting data were obtained. The regression equation for this composite experience curve is given by the following regression equation.

$$\sigma = 7.625 \times 10^{-5} N_s^{1.485} \qquad \text{Eq. (66)}$$

The correlation coefficient for this regression is not very high and it shows that such an experience curve is not expected to be very reliable. Using a regression relation suggested by Khanna and Bansal (1979) a relation was developed between plant sigma, σ, and unit discharge, Q. The regression equation developed for bulb turbines studied on this project is:

$$\sigma = 0.5750\ Q_{11}^{1.1937} \qquad \text{Eq. (67)}$$

Table 5 summarizes all the regression information on turbine setting for manufactured bulb-type turbines that was obtained and gives the respective correlation coefficients and the number of units used in each regression relation that was developed. The information source or manufacturer is also indicated in Table 5.

Tubular Turbines

Figure 47 presents the relation between plant sigma, σ , and the specific speed, N_S, for all manufactured tubular turbines studied. The empirical equation for the relation between the plant sigma, σ , and specific speed, N_S, for the manufactured tubular turbines is indicated below:

$$\sigma = 3.987 \times 10^{-5} N_s^{1.579} \qquad \text{Eq. (68)}$$

TABLE 5

SUMMARY LISTING OF REGRESSION INFORMATION RELATING TO TURBINE

SETTING FOR BULB AND TUBULAR TURBINES

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units	Type of Turbine
60	σ	$\sigma = 4.549 \times 10^{-6} N_s^{1.9080}$	0.58	0.84	1953-1984	12	Bulb
61	σ	$\sigma = 313.332 \times 10^{-6} N_s^{1.2740}$	0.92	0.11	1953-1984	10	Bulb
62	σ	$\sigma = 0.097 \times 10^{-6} N_s^{2.4790}$	0.92	0.15	1953-1984	4	Bulb
63	σ	$\sigma = 111.435 \times 10^{-6} N_s^{1.4230}$	0.47	0.47	1953-1984	15	Bulb
64	σ	$\sigma = 80.774 \times 10^{-6} N_s^{1.4910}$	0.44	1.02	1953-1984	11	Bulb
65	σ	$\sigma = 1541.62 \times 10^{-6} N_s^{1.1050}$	0.84	0.20	1953-1984	3	Bulb
66	σ	$\sigma = 7.625 \times 10^{-5} N_s^{1.4850}$	0.53	0.64	1953-1984	61	Bulb
67	σ	$\sigma = 0.575\ Q_{11}^{1.1937}$	0.43	0.68	1953-1984	61	Bulb

TABLE 5 CONTINUED

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units	Type of Turbine
68	σ	$\sigma = 3.987 \times 10^{-5} N_s^{1.579}$	0.53	0.33	1957-1984	31	Tubular
69	σ	$\sigma = 0.3074\ Q_{11}^{2.066}$	0.77	0.24	1957-1984	31	Tubular

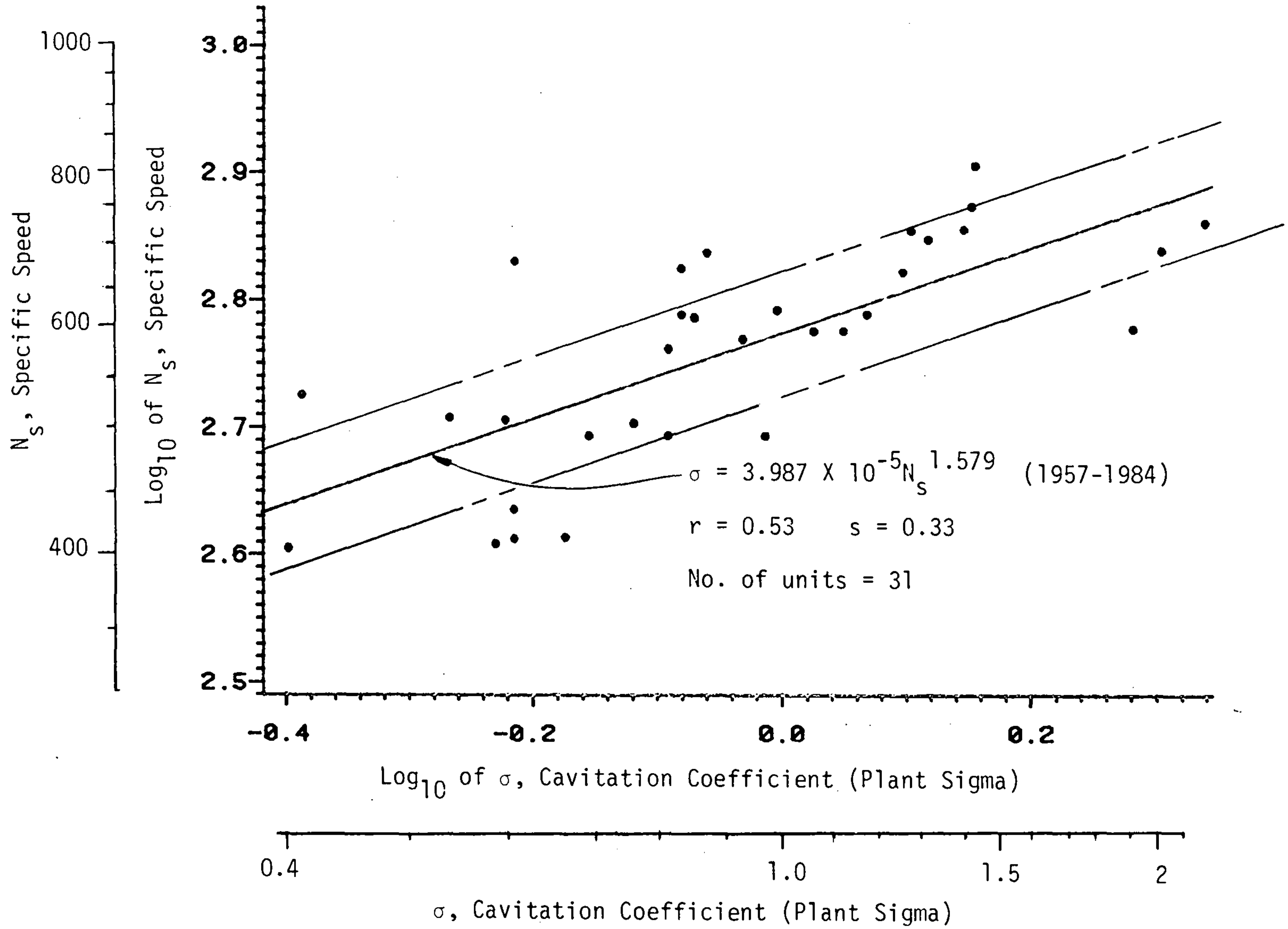

Figure 47. Specific speed versus cavitation coefficient for tubular turbines.

As for bulb turbines the correlation coefficient for this composite regression for tubular turbines is not very high and it shows that such an experience curve is not expected to be very reliable.

The relation between sigma, σ, and unit discharge, Q_{11}, for tubular turbines is given by the regression equation:

$$\sigma = 0.3074\ Q_{11}^{\ 2.066} \qquad \text{Eq. (69)}$$

The summary of regression information on turbine setting characteristics for tubular turbines is presented along with regression information on bulb turbines in Table 5.

WATER PASSAGE CHARACTERISTICS

The water passages of low-head turbines are quite different from conventional Francis and vertical shaft Kaplan propeller turbines and as such the dimensioning of the water passages is different for different types. Significant in feasibility and preliminary design are the entrance dimensions, the draft tube outlet dimensions or area, the maximum diameter of the water passage surrounding the turbine, the total length from entrance to draft tube outlet, and the length from the centerline of the turbine to entrance. These data are useful in layout design of the civil works and power house arrangement planning as well as helpful in cost estimating. In this study it was possible to obtain only enough different sets of data on manufactured bulb type units to make regression analyses and develop experience curves.

In seeking the water passage information it was found that most turbine manufacturers prefer to consider the various dimensions proprietary information so that this phase of the research had to be scaled to what could be collected under public disclosure allowances.

In the manufacturer contacts it was possible in several cases to get recommended dimensions related back to a common turbine parameter such as turbine runner diameter. This information has been grouped and organized to be useful for design and also compared with different manufacturers performance data to provide representative dimensions that can be related to plant capacities.

During the study several companies provided standardized selection information that gives considerable detail on different sized units. These water passage dimensions have been analysed and comparisons between different company's unit made and where possible regression studies were conducted. In general there was insufficient information on the possible standardized units to develop experience curves. Following the earlier pattern the specific information on water passage dimensions is presented systematically according to different turbine types, beginning with bulb type turbines.

Bulb Turbines

To present the water passage information it is necessary to show schematically the various water passage dimensions that were analysed. Figure 48 shows a simplified dimensioning sketch with dimensions labeled with letters that were used in the regression analyses and the comparisons. All dimensions have been related back to the design diameter of the turbine runner as obtained from the manufacturer. Since the rated power is frequently an estimated value that is obtained early in the feasibility study, water passage dimensions were also related to rated power, P, and in some cases relations were sought with the rated discharge, Q. In certain cases like the entrance to the turbine and the exit from the draft tube the dimensions actually represent areas.

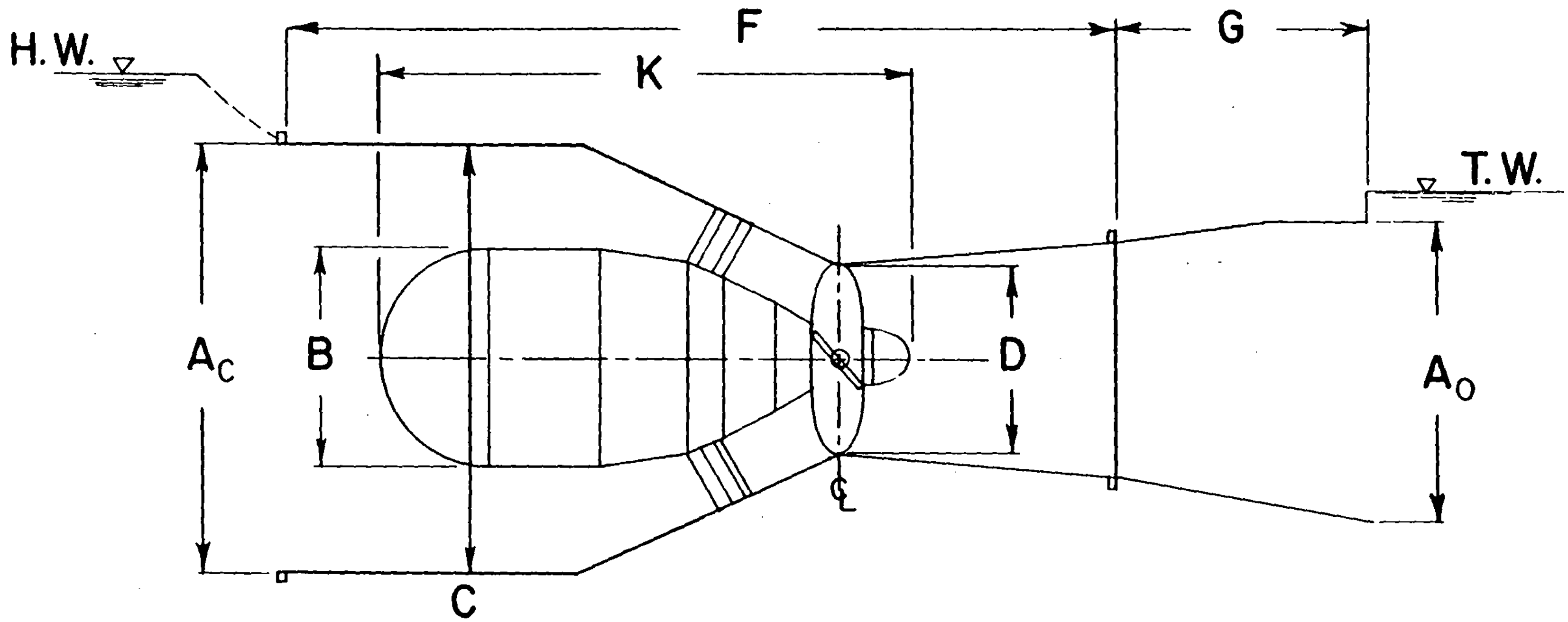

Figure 48. Simplified dimensioning sketch for water passages of bulb turbines.

These areas are sometimes circular, square, or rectangular in cross section.

Figure 49 presents the relation of the distance from turbine entrance to the exit of the draft tube outlet (F + G), to the rated power and the resulting regression equation for bulb turbines is given as:

$$(F + G) = 0.6744\ P^{0.4188} \qquad \text{Eq. (70)}$$

Figure 50 presents the relation of the distance from the turbine entrance to the exit of the draft tube outlet, (F + G) to the runner diameter, D, and the resulting regression equation for bulb turbines is given as:

$$(F + G) = 8.2075\ D^{0.9801} \qquad \text{Eq. (71)}$$

Figure 51 presents the relation of the length of the bulb, K, including the turbine runner to the rated power, P, and resulting regression equation for bulb turbines is given as:

$$K = 0.580\ P^{0.3268} \qquad \text{Eq. (72)}$$

Figure 52 presents the relation of the length of the bulb including the turbine runner to the turbine diameter, D, and the resulting regression equation for bulb turbines is given as:

$$K = 3.1994\ D^{0.8744} \qquad \text{Eq. (73)}$$

Figure 53 presents the relation of the entrance area. A_e, to the rated power, P, and the resulting regression equation for bulb turbines is given as:

$$A_e = 0.1465\ P^{0.6503} \qquad \text{Eq. (74)}$$

Figure 54 presents the relation of the entrance area, A_e, to the turbine diameter, D, and the resulting regression equation for bulb

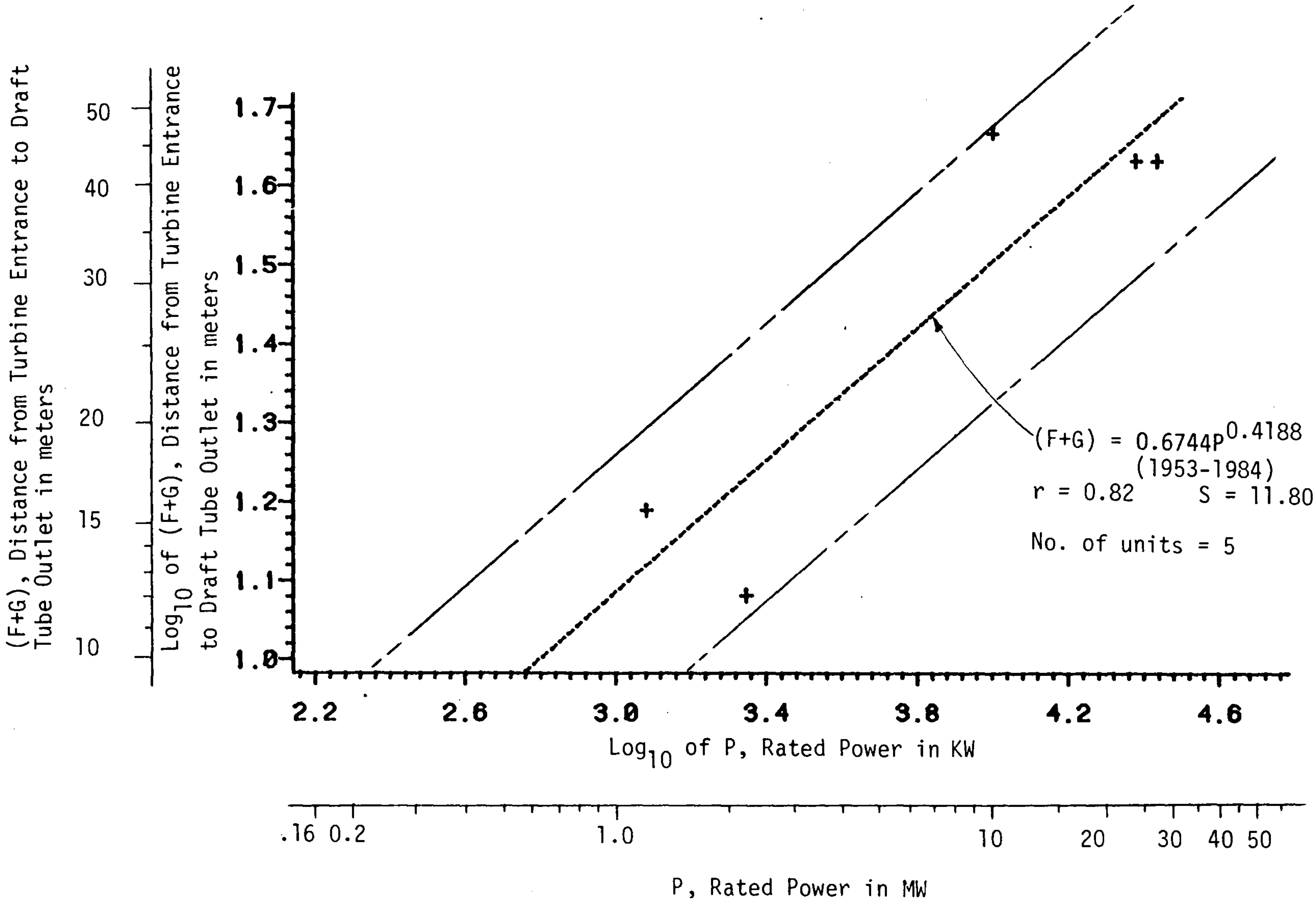

Figure 49. Distance from turbine entrance to draft tube outlet versus rated power output for bulb turbines.

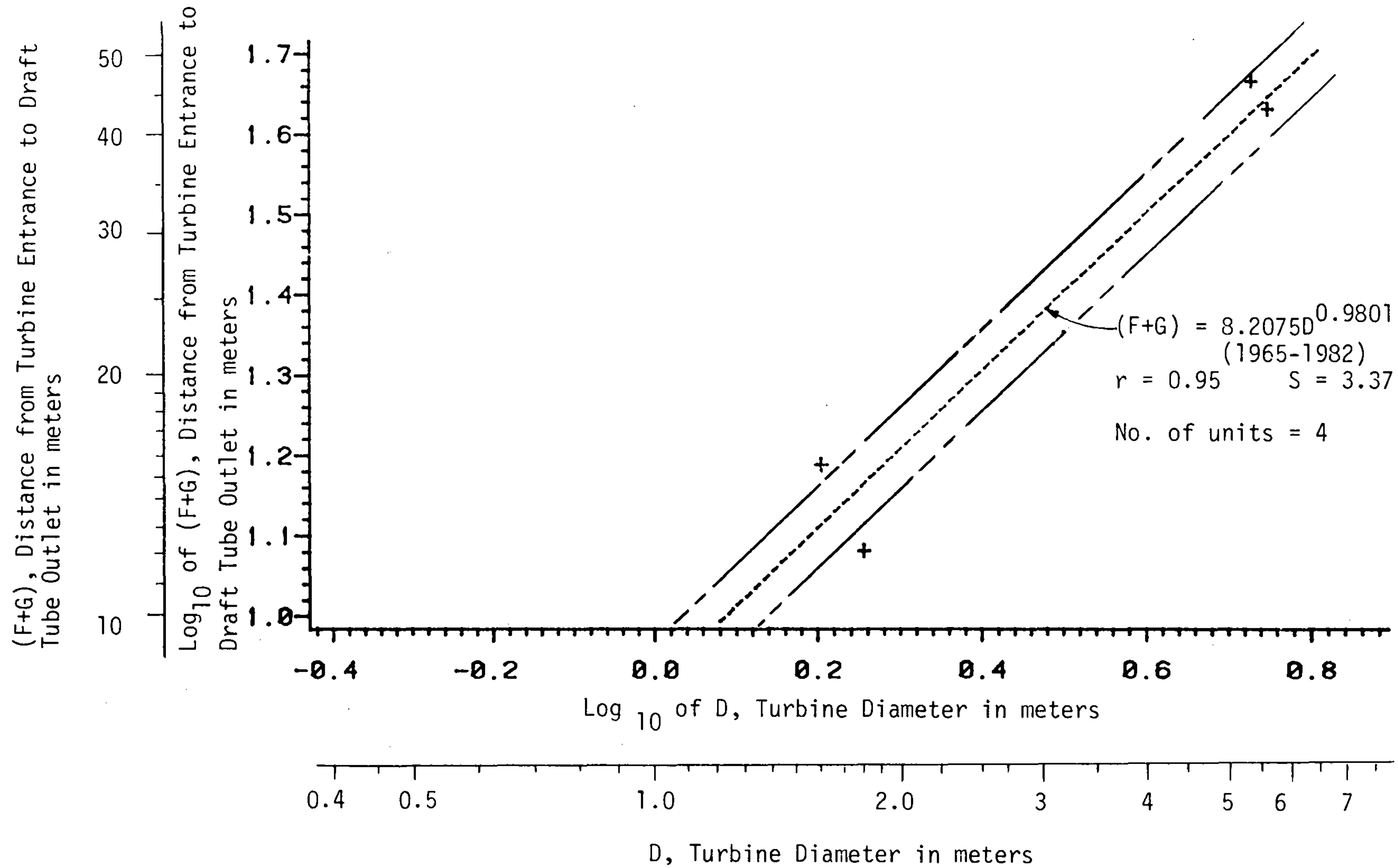

Figure 50. Distance from turbine entrance to draft tube outlet versus turbine diameter for bulb turbines.

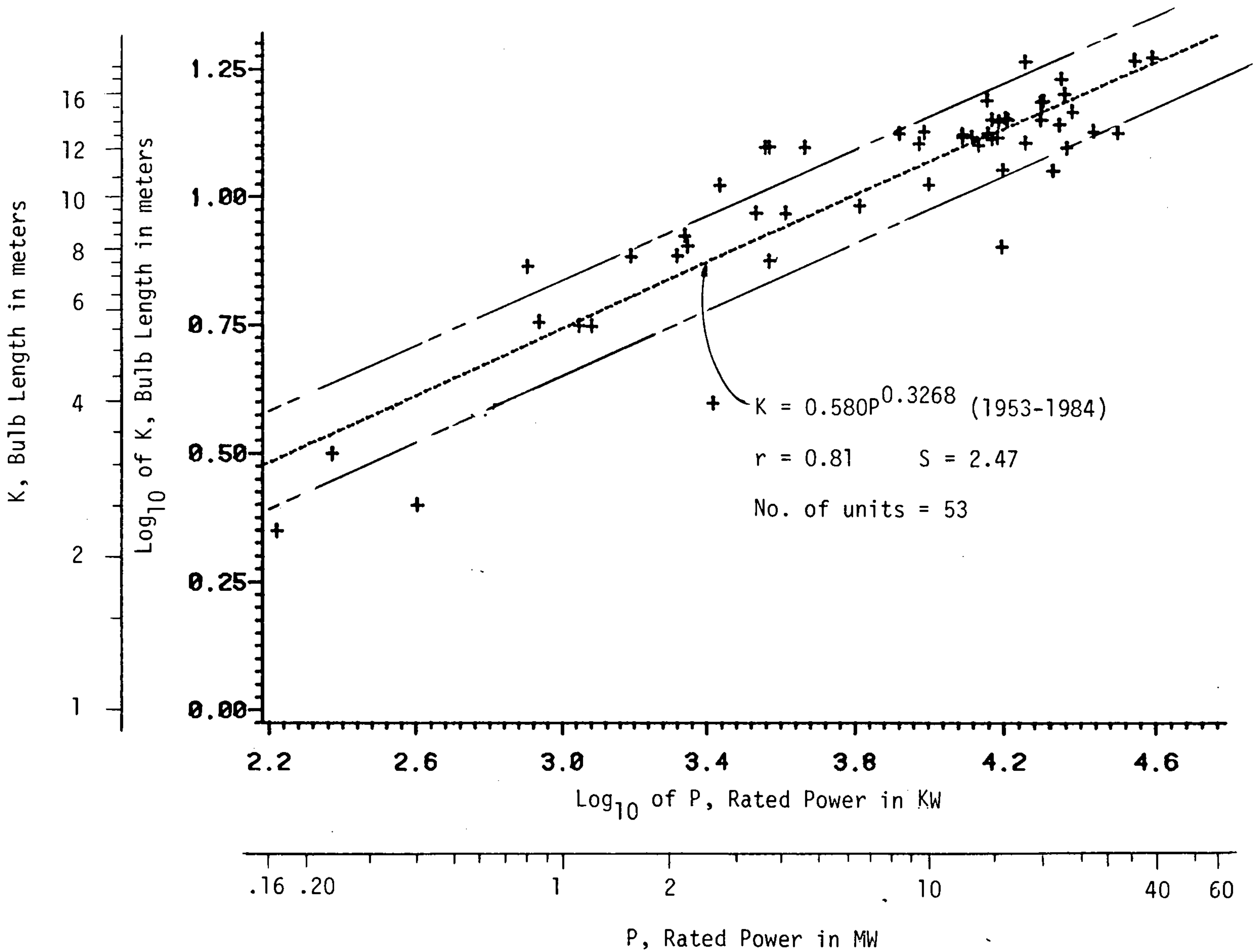

Figure 51. Length of bulb versus rated power for bulb turbines.

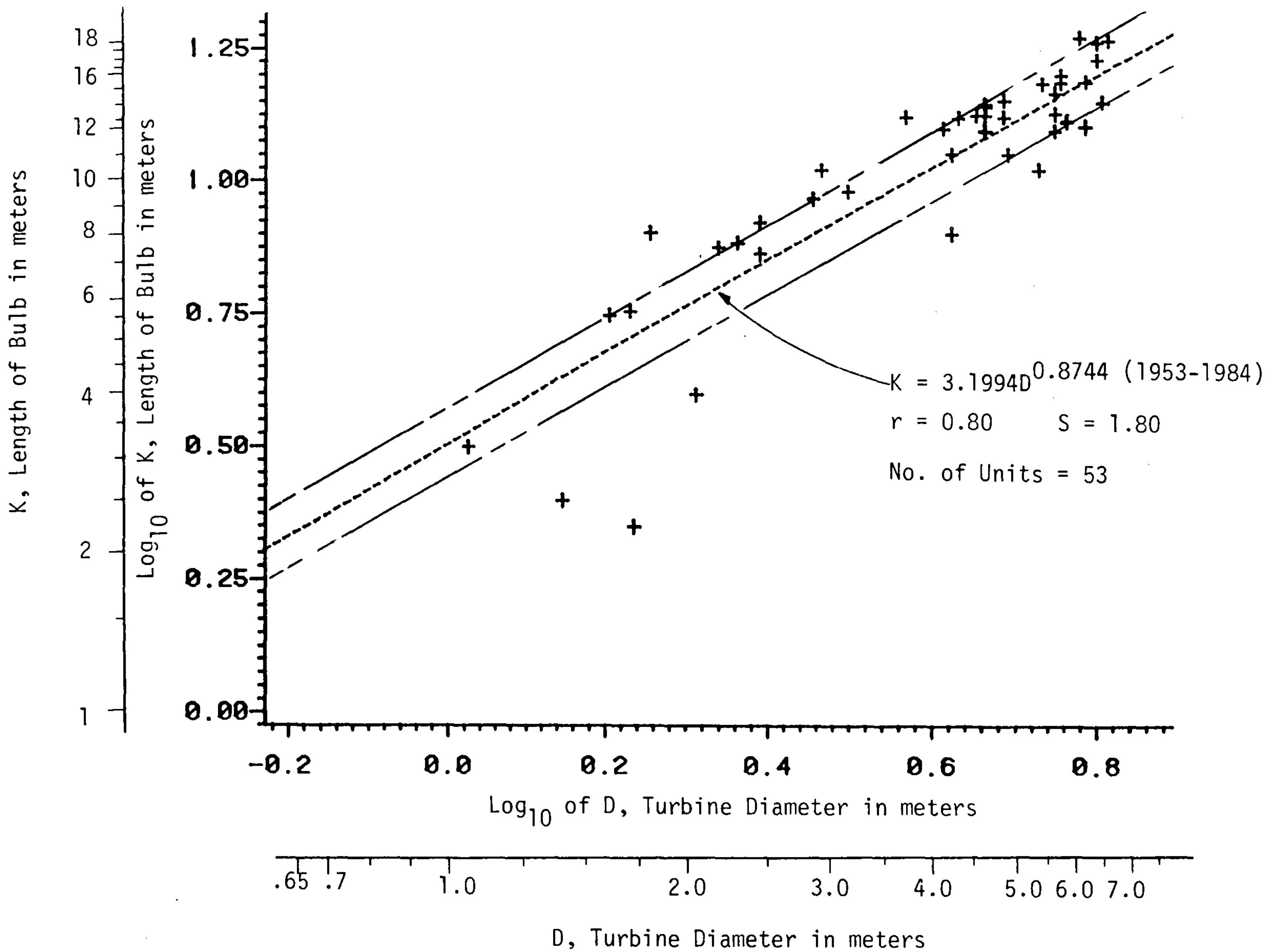

Figure 52. Length of bulb turbine versus turbine diameter.

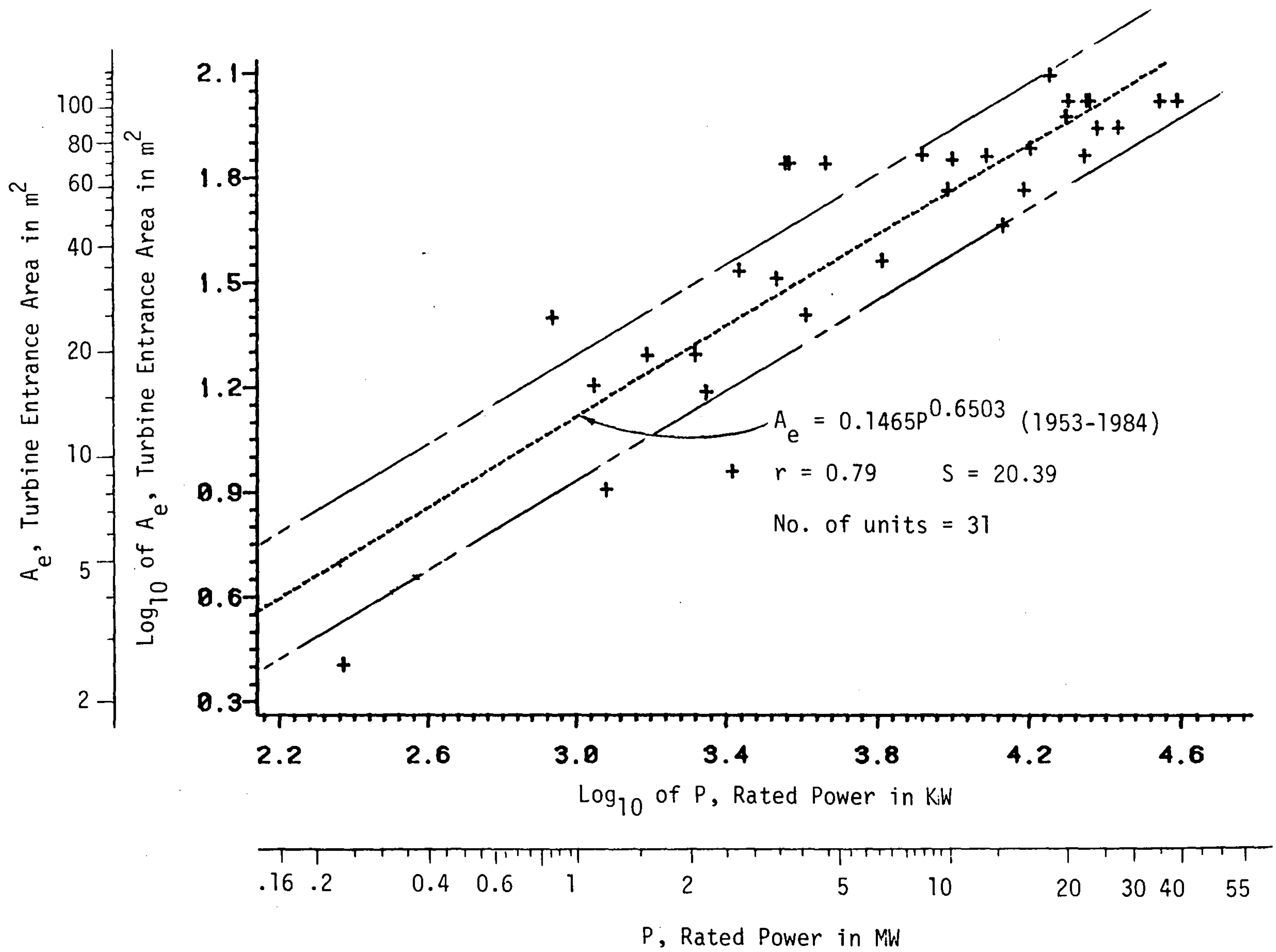

Figure 53. Turbine entrance area versus rated power output for bulb turbines.

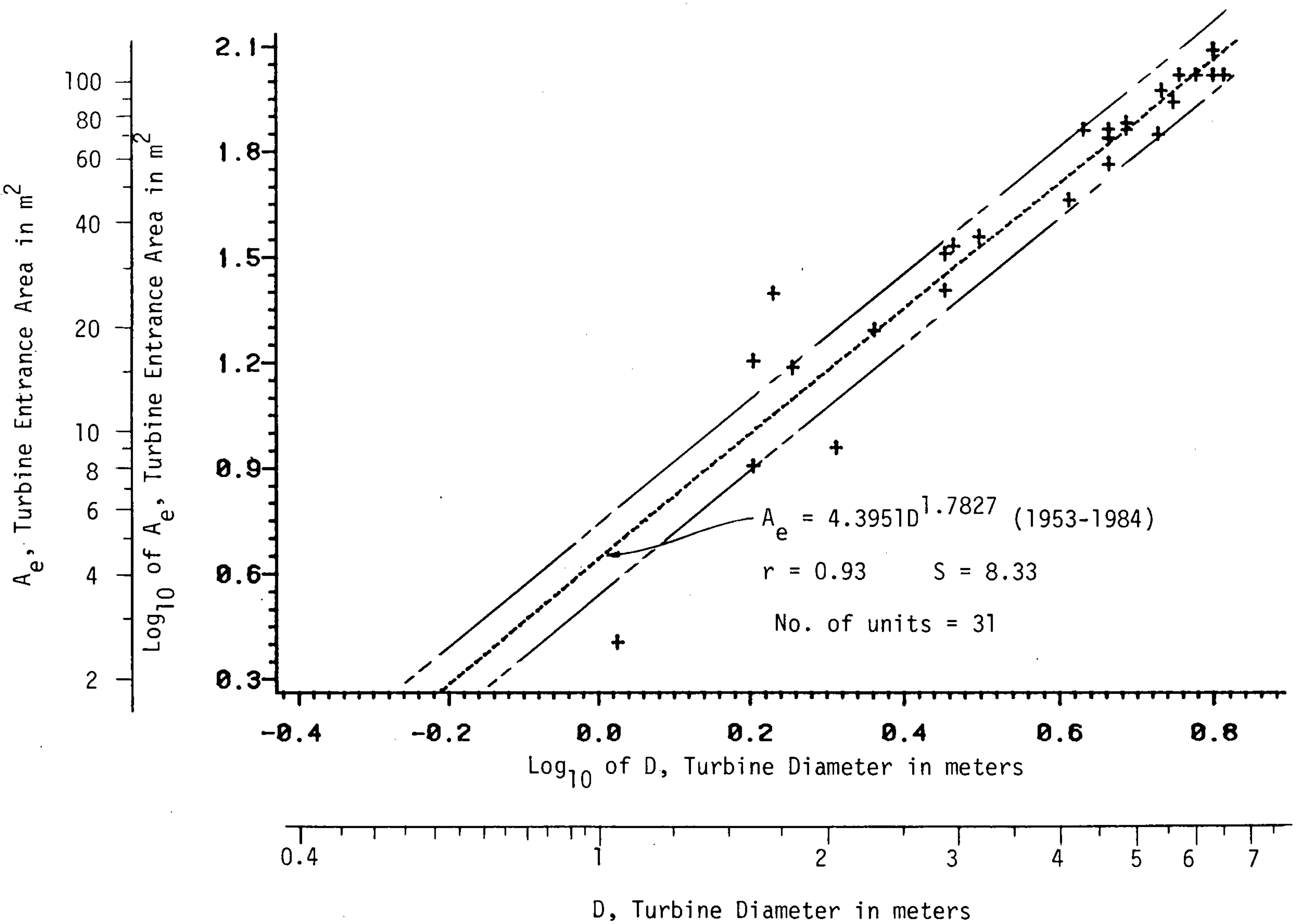

Figure 54. Turbine entrance area versus turbine diameter for bulb turbines.

turbines is given as:

$$A_e = 4.3951\ D^{1.7827} \quad \text{Eq. (75)}$$

Figure 55 presents the relation of the bulb diameter, B, to the rated power, P, and the resulting regression equation for bulb turbines is given as:

$$B = 0.1887\ P^{0.3526} \quad \text{Eq. (76)}$$

Figure 56 presents the relation of the bulb diameter, B, to the turbine diameter, D, and the resulting regression equation for bulb turbines is given as:

$$B = 1.1745\ D^{0.9546} \quad \text{Eq. (77)}$$

Figure 57 presents the relation of the draft tube exit area, A_o, to the rated power, P, and the resulting regression equation for bulb turbines is given as:

$$A_o = 0.0978\ P^{0.6846} \quad \text{Eq. (78)}$$

Figure 58 presents the relation of the draft tube exit area, A_o, to the turbine diameter, D, and the resulting regression equation for bulb turbines is given as:

$$A_o = 2.8686\ D^{2.0047} \quad \text{Eq. (79)}$$

Figure 59 presents the relation of the ratio, K/A_e, to the rated power, P, and the resulting regression equation for bulb turbines is given as:

$$K/A_e = 4.335\ P^{-0.3278} \quad \text{Eq. (80)}$$

Figure 60 presents the relation of the velocity at turbine entrance, V_e, to the rated power, P, and the resulting regression equation for bulb turbines is given as:

$$V_e = 0.2690\ P^{0.2254} \quad \text{Eq. (81)}$$

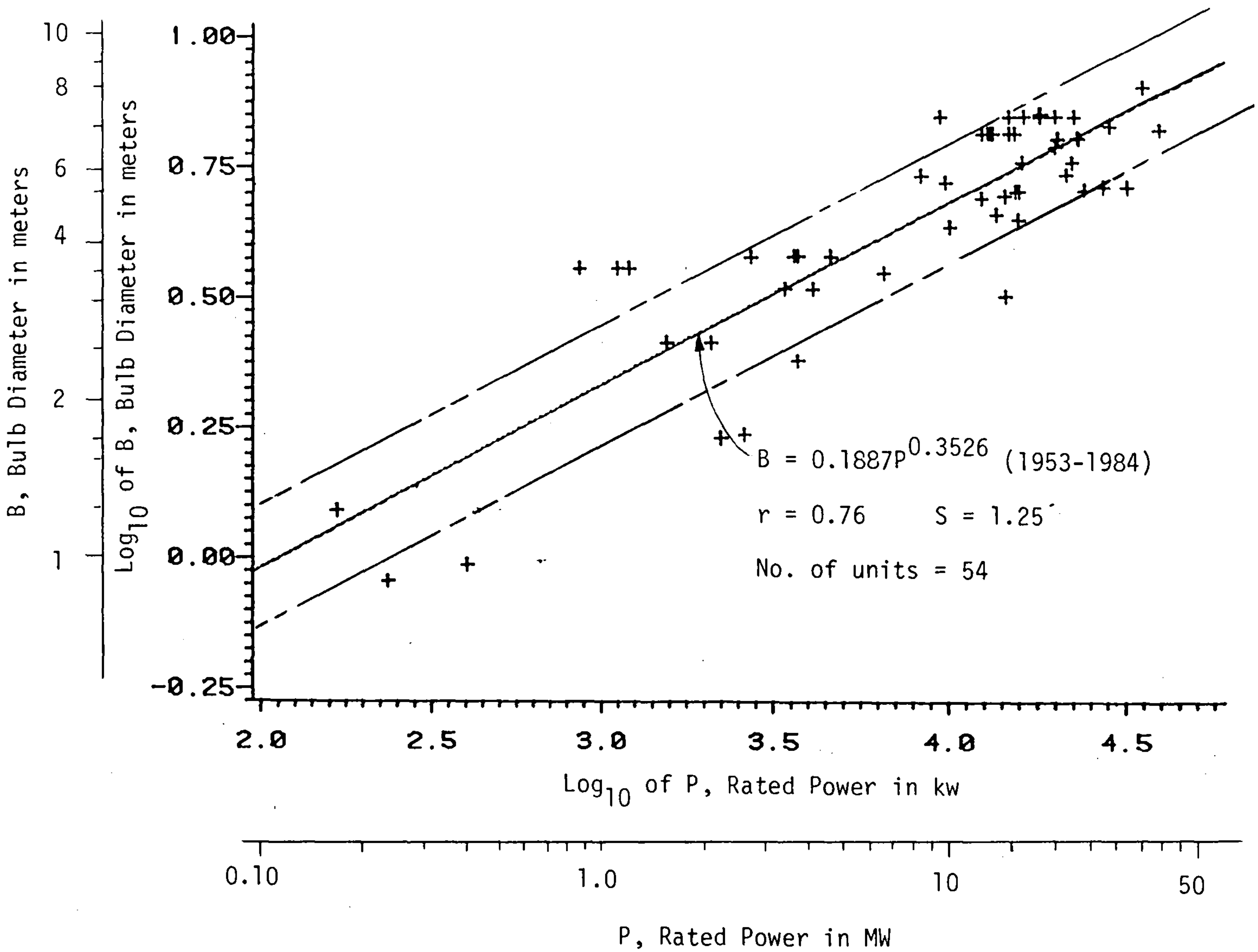

Figure 55. Bulb diameter versus rated power for bulb turbines.

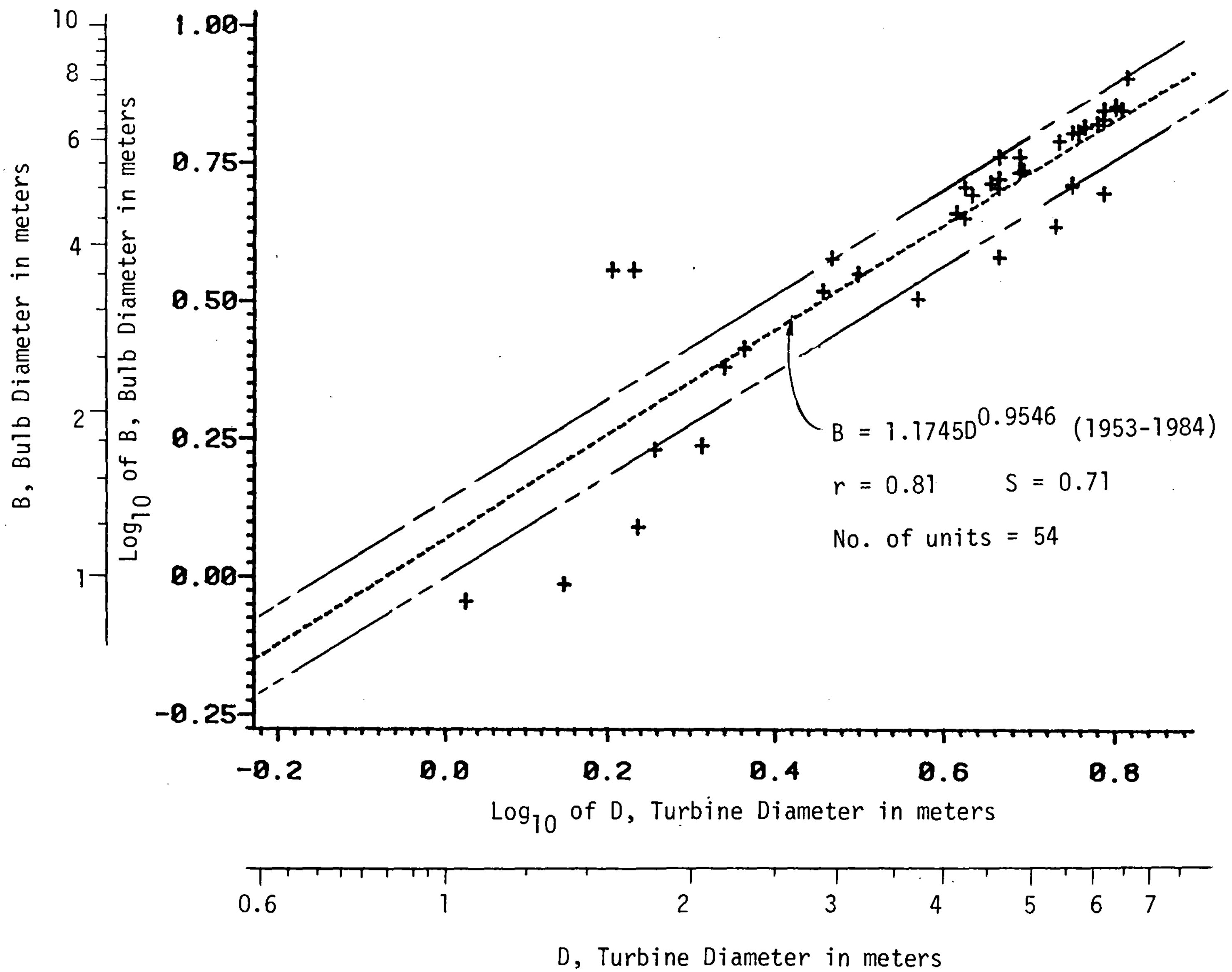

Figure 56. Bulb diameter versus turbine diameter.

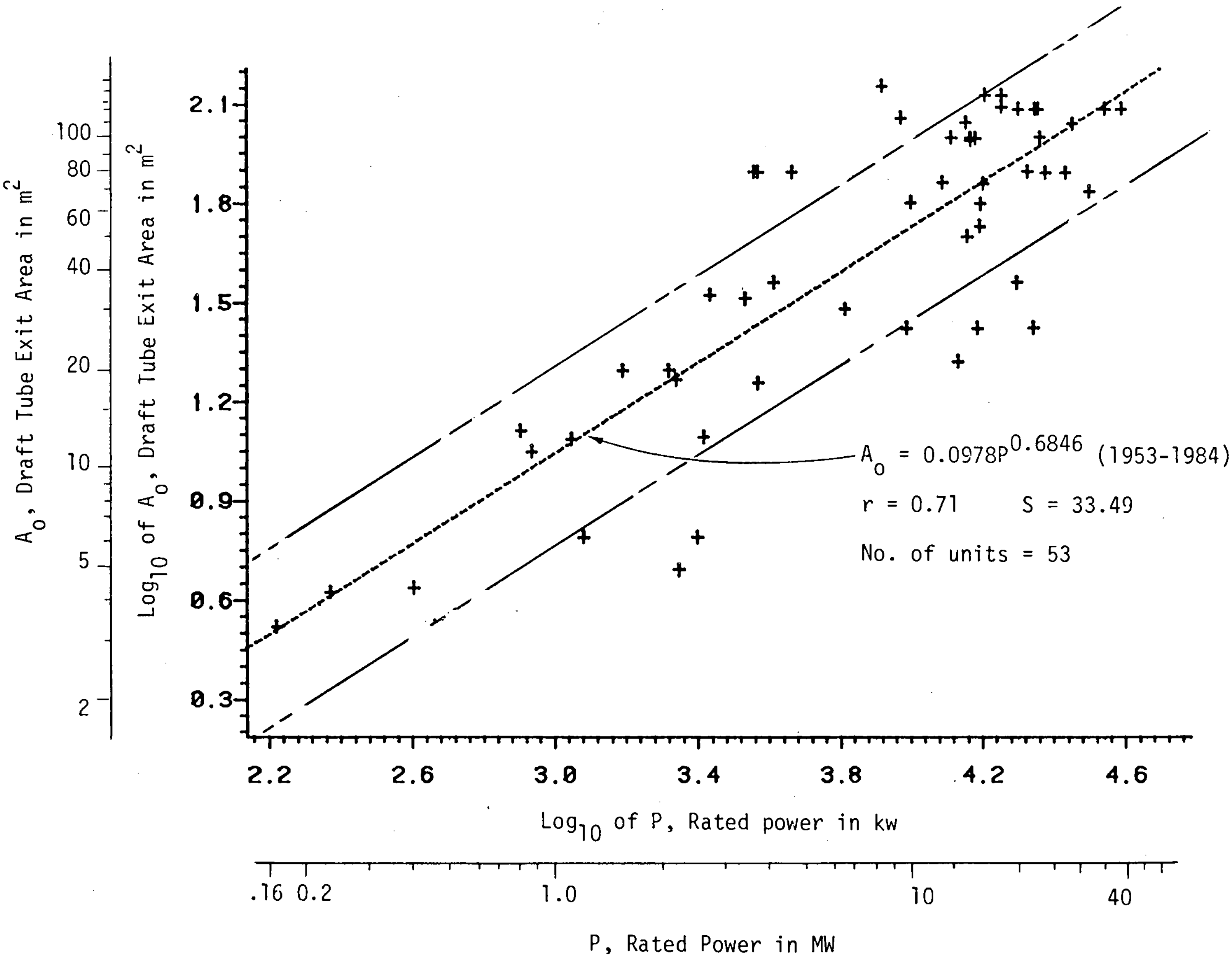

Figure 57. Draft tube exit area versus rated power for bulb turbine.

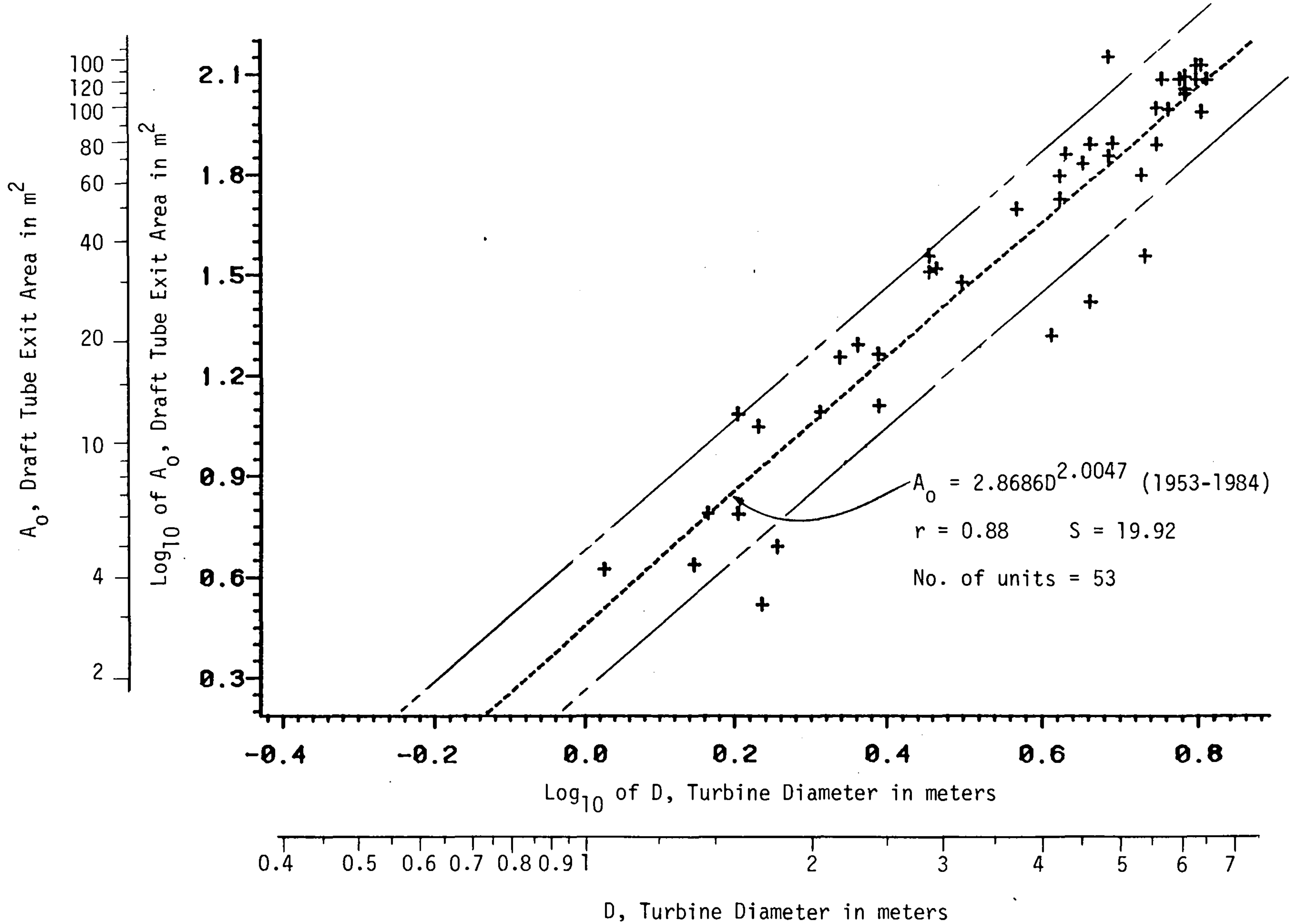

Figure 58. Draft tube exit area versus turbine diameter for bulb turbine.

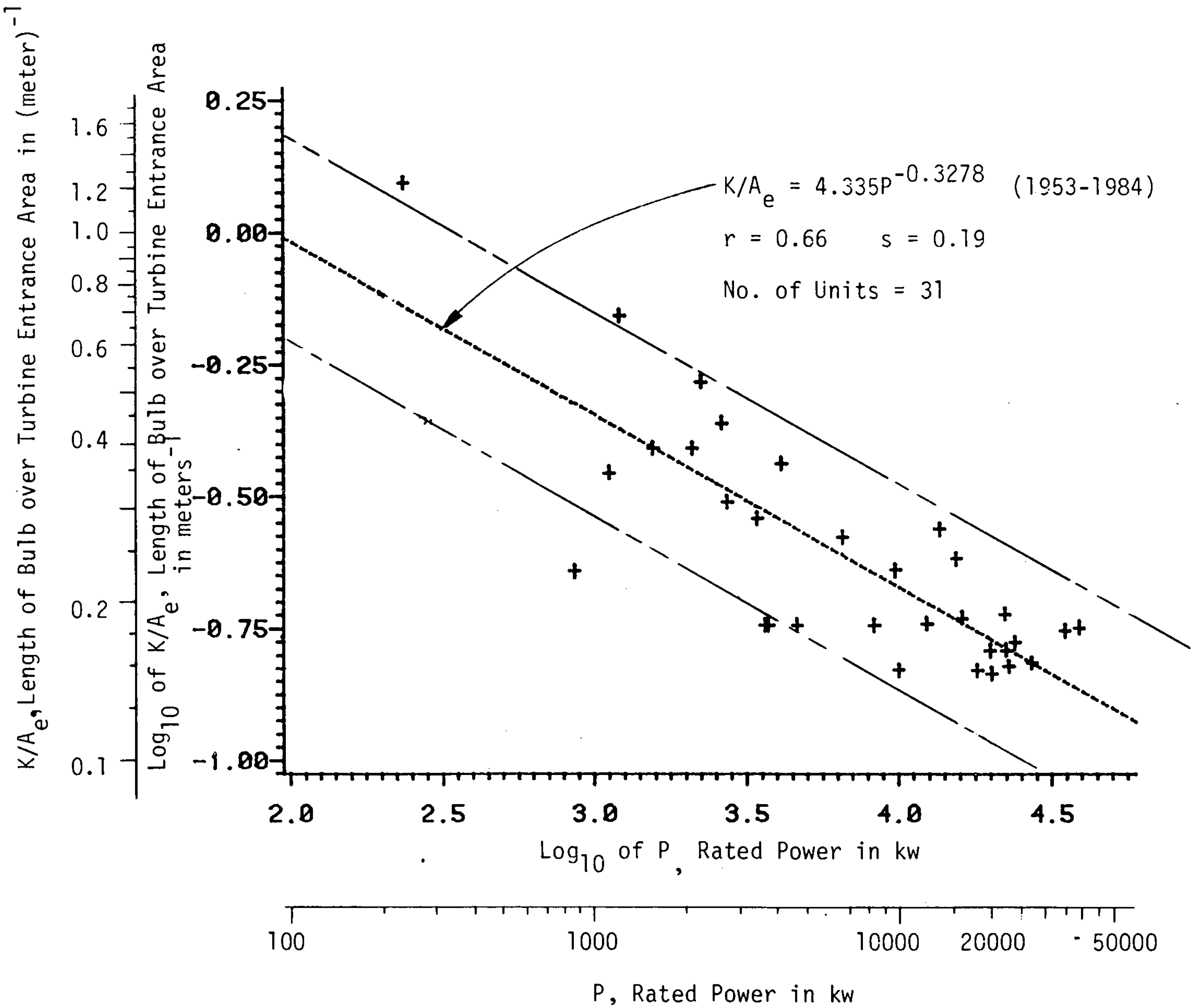

Figure 59. K/A_e ratio versus rated power output for bulb turbines.

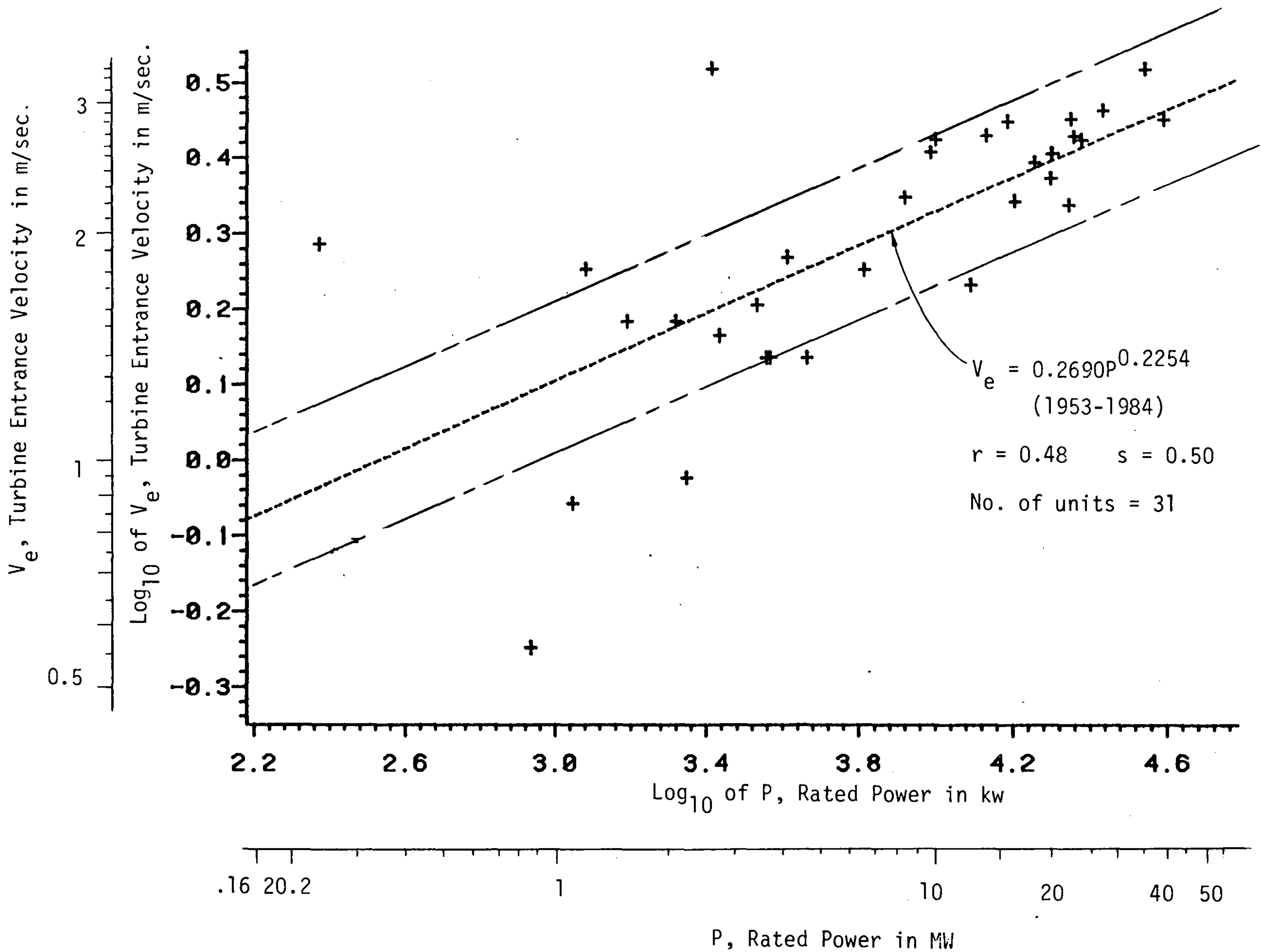

Figure 60. Turbine entrance velocity versus rated power for bulb turbines.

Figure 61 presents the relation of the velocity at turbine entrance, V_e, to the turbine diameter, D and the resulting regression equation for bulb turbines is given as:

$$V_e = 1.0133\ D^{0.5043} \qquad \text{Eq. (82)}$$

Figure 62 presents the relation of the turbine entrance area, A_e, to the rated turbine discharge, Q, and the resulting regression equation for bulb turbines is given as:

$$A_e = 1.01\ Q^{0.848} \qquad \text{Eq. (83)}$$

Figure 63 presents the relation of the draft tube exit area, A_o, to the rated turbine discharge, Q, and the resulting regression equation for bulb turbines is given as:

$$A_o = 0.5045\ Q^{0.9743} \qquad \text{Eq. (84)}$$

Table 6 summarizes all the regression relations that were developed for water passage dimensions of manufactured bulb turbines. In the table are shown the equations that were developed, the regression correlation coefficient, for each dependent parameter studied, the corresponding standard deviation, the period of analysis for which the manufactured turbines were designated for commissioning, and the number of different units used in developing a particular relation.

Tubular Turbines

Insufficient manufacturer's data on actual manufactured turbines were obtained to develop a useful regression equation for tubular turbines water passage dimension. However, information was obtained from certain manufacturers that gave recommended relations between the sizes of certain water passage locations and the diameters of the propeller runners. Figure 64 gives the recommendations for preliminary

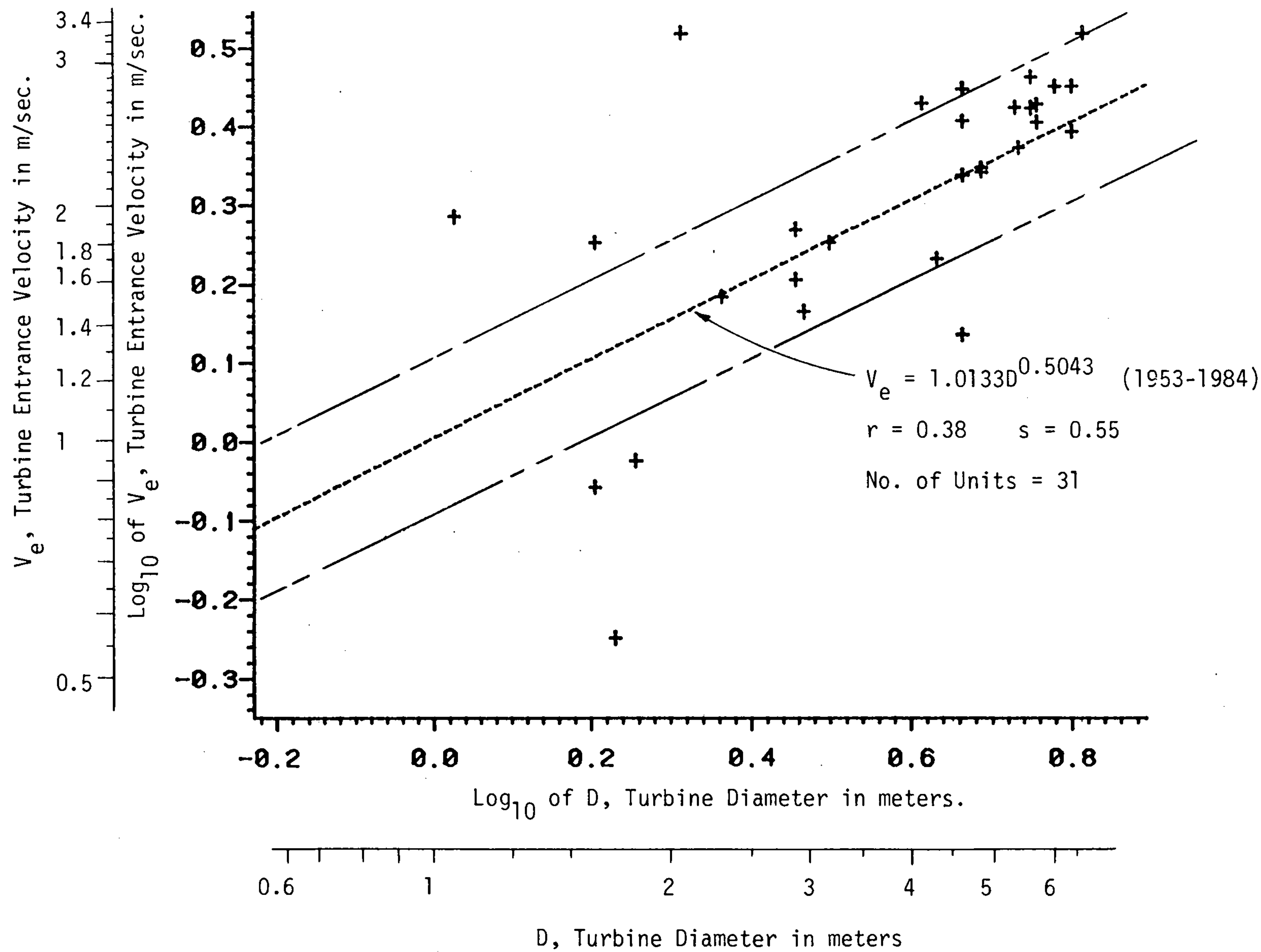

Figure 61. Turbine entrance velocity versus turbine diameter for bulb turbines.

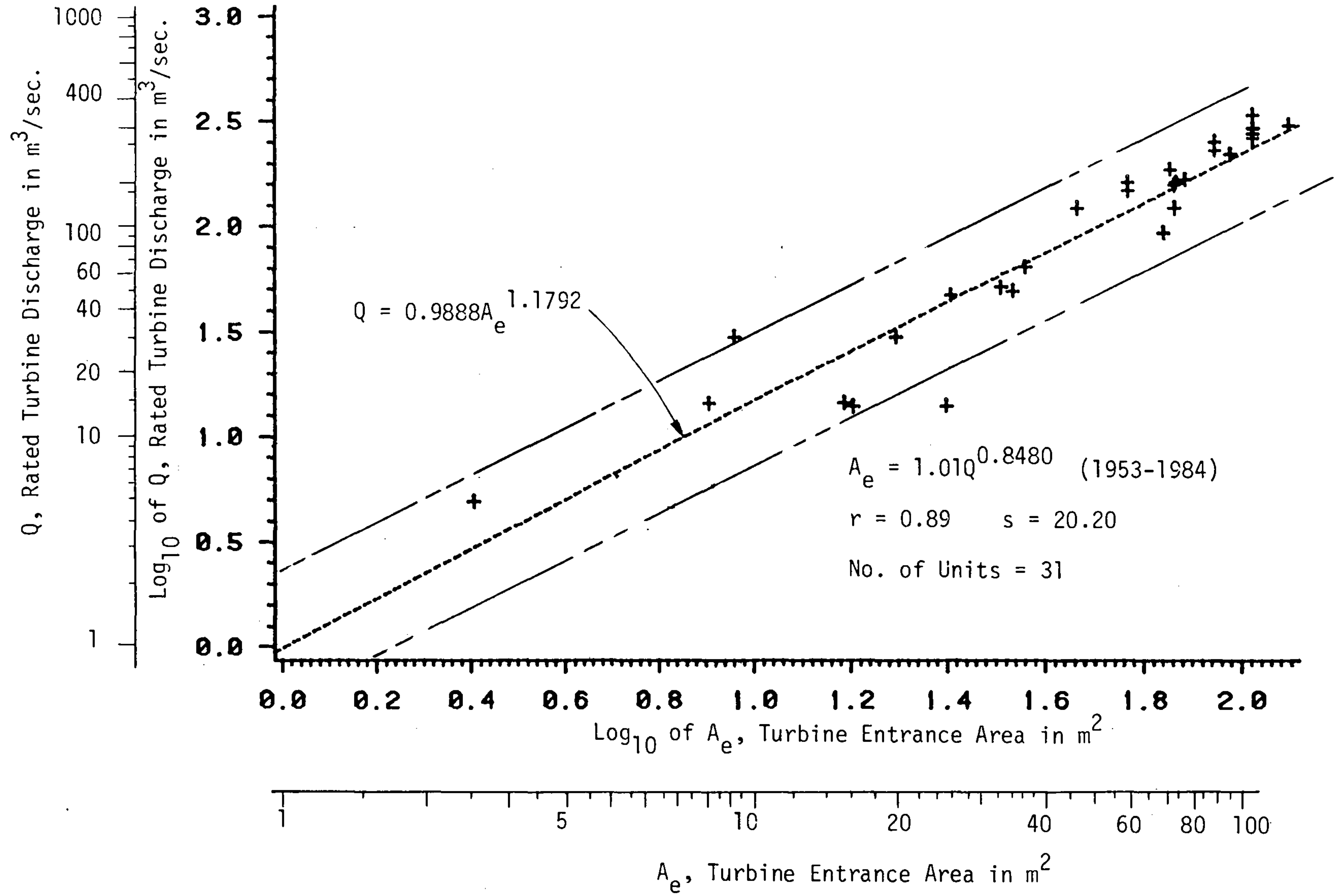

Figure 62. Turbine entrance area versus rated turbine discharge for bulb turbines.

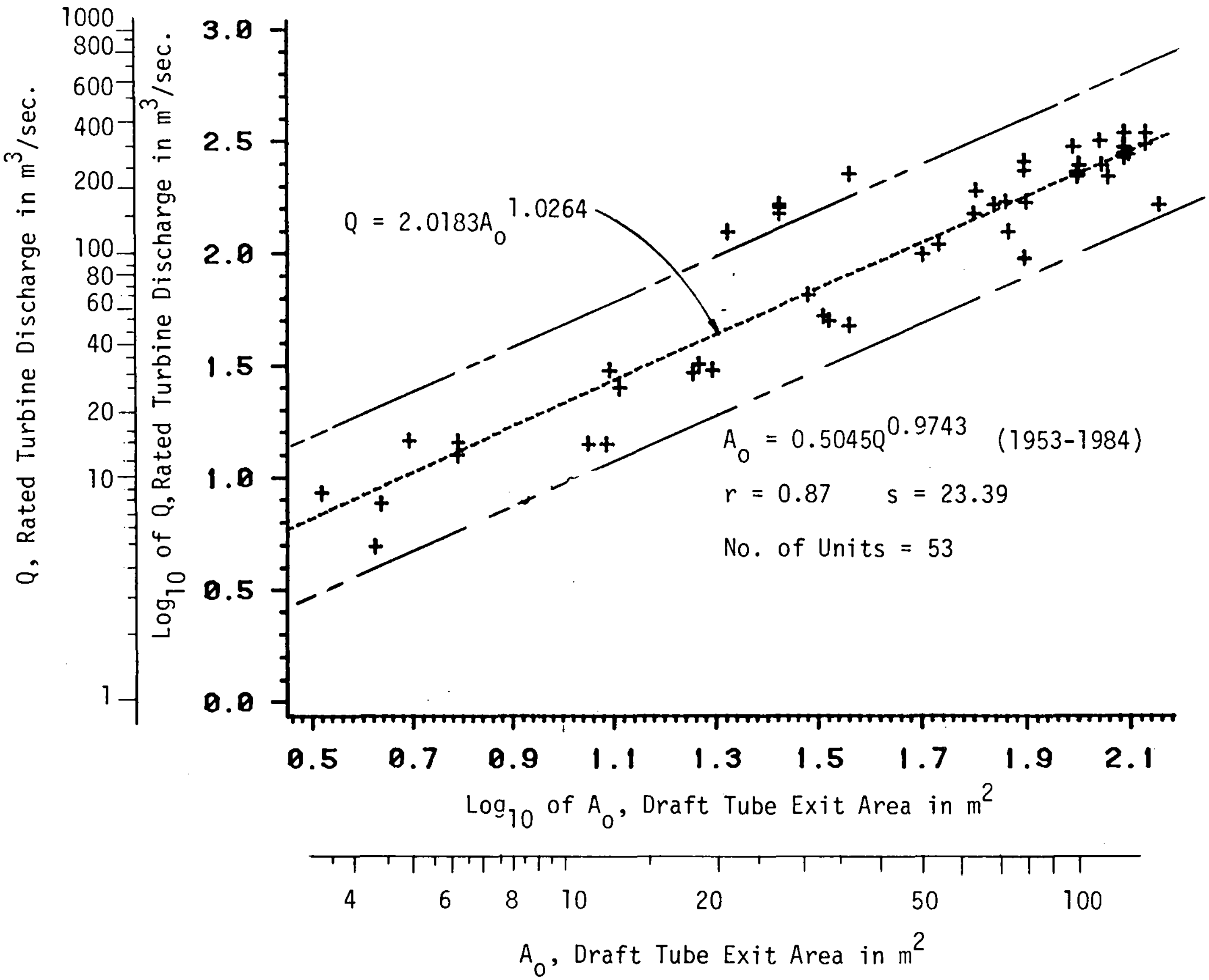

Figure 63. Draft tube exit area versus rated turbine discharge for bulb turbines.

TABLE 6

SUMMARY LISTING OF REGRESSION INFORMATION AND EQUATIONS

RELATING TO WATER PASSAGE DIMENSIONS FOR BULB TURBINES

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
70	(F + G)	$(F + G) = 0.6744\ P^{0.4188}$	0.82	11.80	1953-1984	5
71	(F + G)	$(F + G) = 8.2075\ D^{0.9801}$	0.95	3.31	1953-1984	4
72	K	$K = 0.580\ P^{0.3268}$	0.81	2.47	1953-1984	53
73	K	$K = 3.1994\ D^{0.8744}$	0.80	1.80	1953-1984	53
74	A_e	$A_e = 0.1465\ P^{0.6503}$	0.79	20.39	1953-1984	31
75	A_e	$A_e = 4.3951\ D^{1.7827}$	0.93	8.33	1953-1984	31
76	B	$B = 0.1887\ P^{0.3526}$	0.76	1.25	1953-1984	54
77	B	$B = 1.1745\ D^{0.9546}$	0.81	0.71	1953-1984	54
78	A_o	$A_o = 0.0978\ P^{0.6846}$	0.71	33.49	1953-1984	53

TABLE 6 CONTINUED

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
79	A_o	$A_o = 2.8686\ D^{2.0047}$	0.88	19.92	1953-1984	53
80	K/A_e	$K/A_e = 4.335\ P^{-0.3278}$	0.66	0.19	1953-1984	31
81	V_e	$V_e = 0.2690\ P^{0.2254}$	0.48	0.50	1953-1984	31
82	V_e	$V_e = 1.0133\ D^{0.5043}$	0.38	0.55	1953-1984	31
83	A_e	$A_e = 1.01\ Q^{0.8480}$	0.89	20.20	1953-1984	31
84	A_o	$A_o = 0.5045\ Q^{0.9743}$	0.87	23.39	1953-1984	53

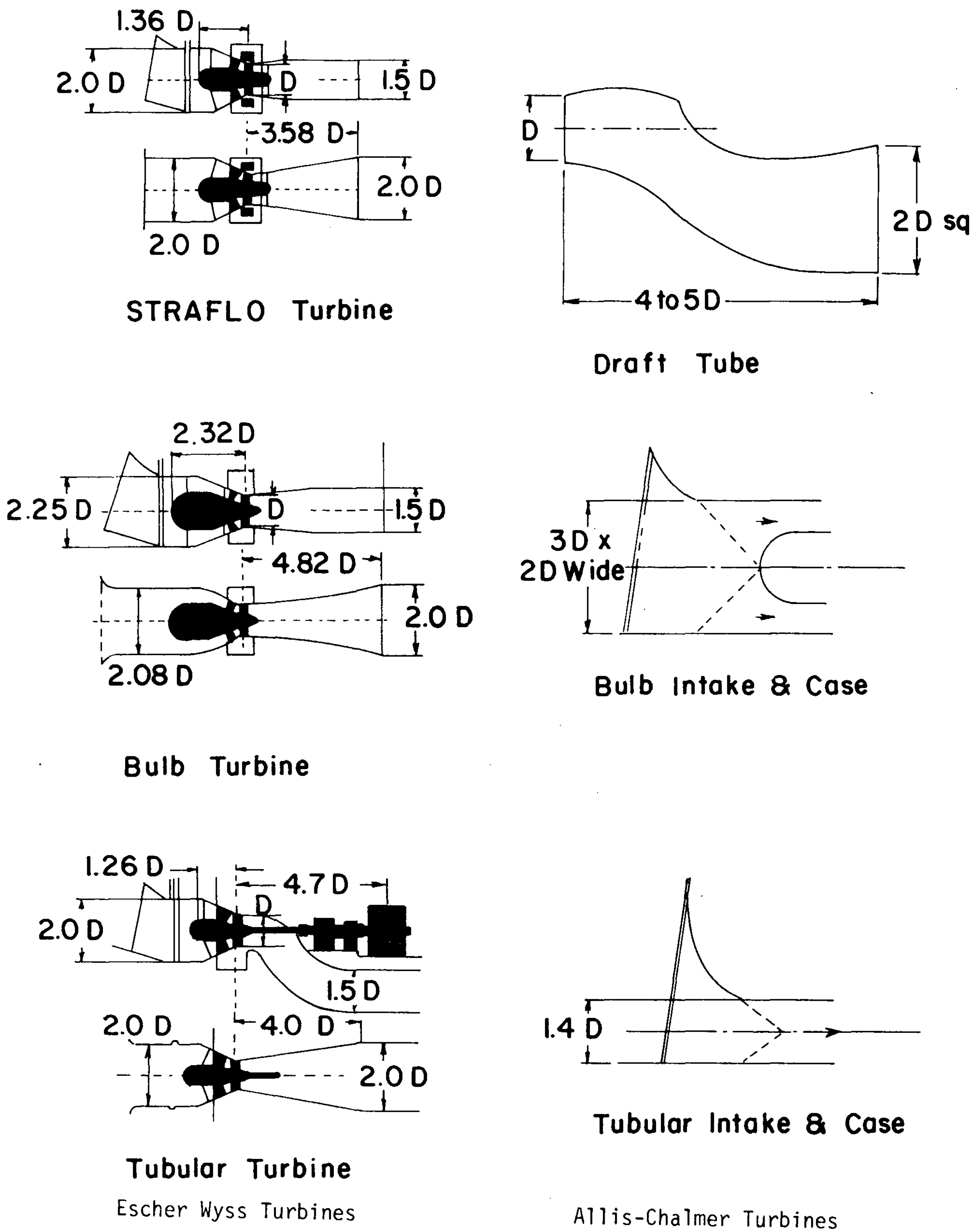

Figure 64. Dimensioning recommendations for low-head reaction turbines.

sizing of tubular turbines as suggested by Allis-Chalmers Corporation. Figure 64 also gives similar recommendations for preliminary sizing of tubular turbines as suggested by Escher-Wyss of Switzerland.

A few of the manufacturers have developed recommended dimensions for standard tubular turbines and published these data. Copies of the information was furnished to the U.S. Bureau of Reclamation. Table 7 gives the standard tubular recommendation information and the source from which the data were taken. These respective tables of recommended dimensions were used to develop experience curves relating water passage dimensions for tubular turbines to the propeller diameter. The information presented in each company's tubular material apparently was developed by the companies from their own model tests. The water passage dimensions A_e, A_o, L_1, and M used in the regression equations are defined on Figure 65.

Figure 66 presents the relation between turbine entrance area, A_e, and the turbine diameter, D, and the resulting regression equation for tubular turbines is given as:

$$A_e = 2.345 \ D^{1.1067} \qquad \text{Eq. (85)}$$

Figure 67 presents the relation between draft tube exit area, A_o, and the turbine diameter, D, and the resulting regression equation for tubular turbines is given as:

$$A_o = 3.330 \ D^{1.5605} \qquad \text{Eq. (86)}$$

Figure 68 presents the relation between the distance, L_1, from the runner blade centerline to the turbine entrance where, A_e, is measured and the turbine diameter, D, and the resulting regression equation for tubular turbines is given as:

$$L_1 = 2.5408 \ D^{0.1522} \qquad \text{Eq. (87)}$$

Table 7. REFERENCE INFORMATION AND SOURCE FOR STANDARD TUBULAR TURBINE WATER PASSAGE DIMENSIONS

Company	Address	Publication Title	Publication Code No.	Page
Allis-Chalmers	Hydro-Turbine Div. York, PA	"Stnadardized Hydroelectric Generating Units"	54B1241-03	6
Tampella-Leffel	426 East Street Springfield, OH	"Standard Tubular Turbines"	None	None
Neyripic	Box 3834 969 High Ridge Rd. Stamford, CT	"Standardized Hydroelectic Turbine for Low Heads"	None	None
Kvaerner Moss	800 Third Ave. New York, NY	"Mini Hydro Turbines" Sørumsand Verstsad A/S N-1920 Sørumsand, Norway	None	8
Other Standard Turbine Literature with Dimensioning but not Used in the Study.				
Barber Hydraulic	Barber Point, Box 346, Port - Colborne, Ontario Canada, L3K 5W1	"Standard Turbine Arrangement No. 5" Single Horizontal Open Bulkhead	SHOB No. 5	- 1978
This is not a true tubular turbine, it has spiral casing for entrance.				
Bell Engineering Escher Wyss	Sulzer Bros. Inc. Western District Office 1255 Post St. Suite 911 San Francisco	"Standard S-turbines"	None	None
KMW	Fach S-68101 Kristinehamn, Sweden	"KMW Miniturbines"	T178-E	

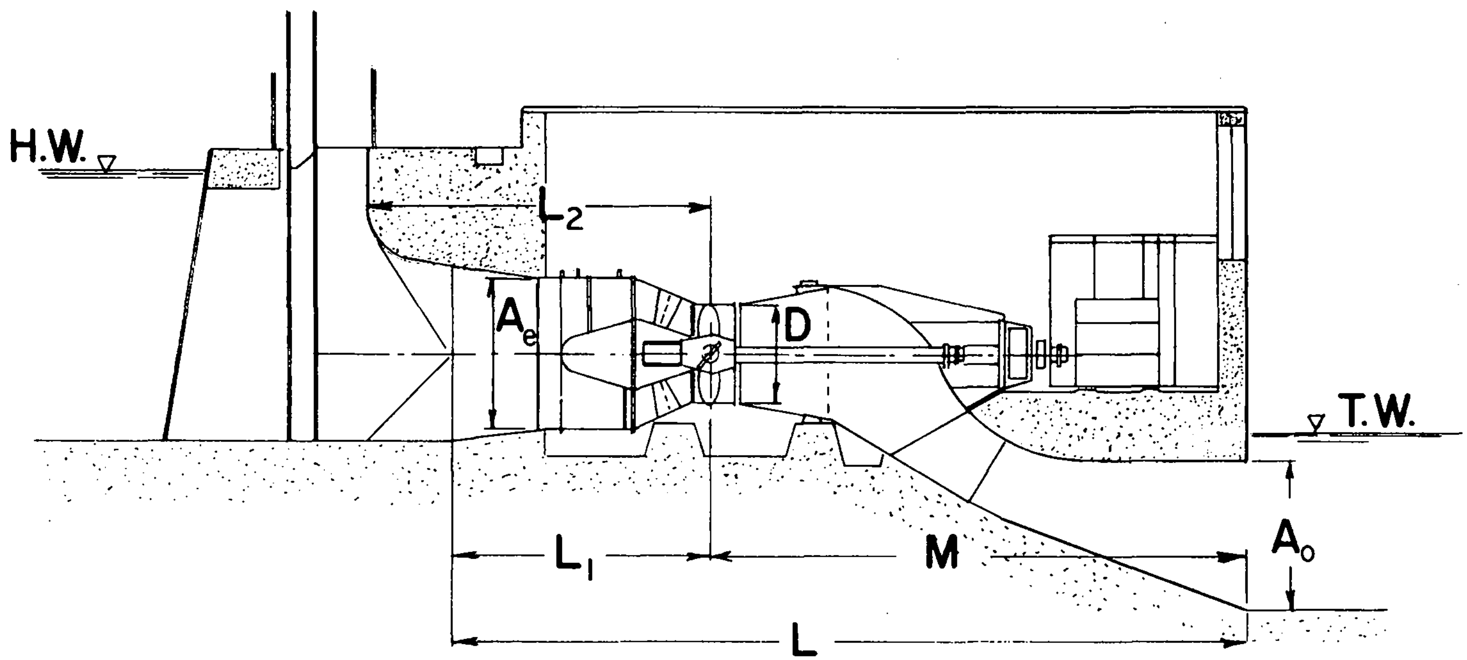

Figure 65. Schematic drawing defining dimensions used in study of standard tubular turbines.

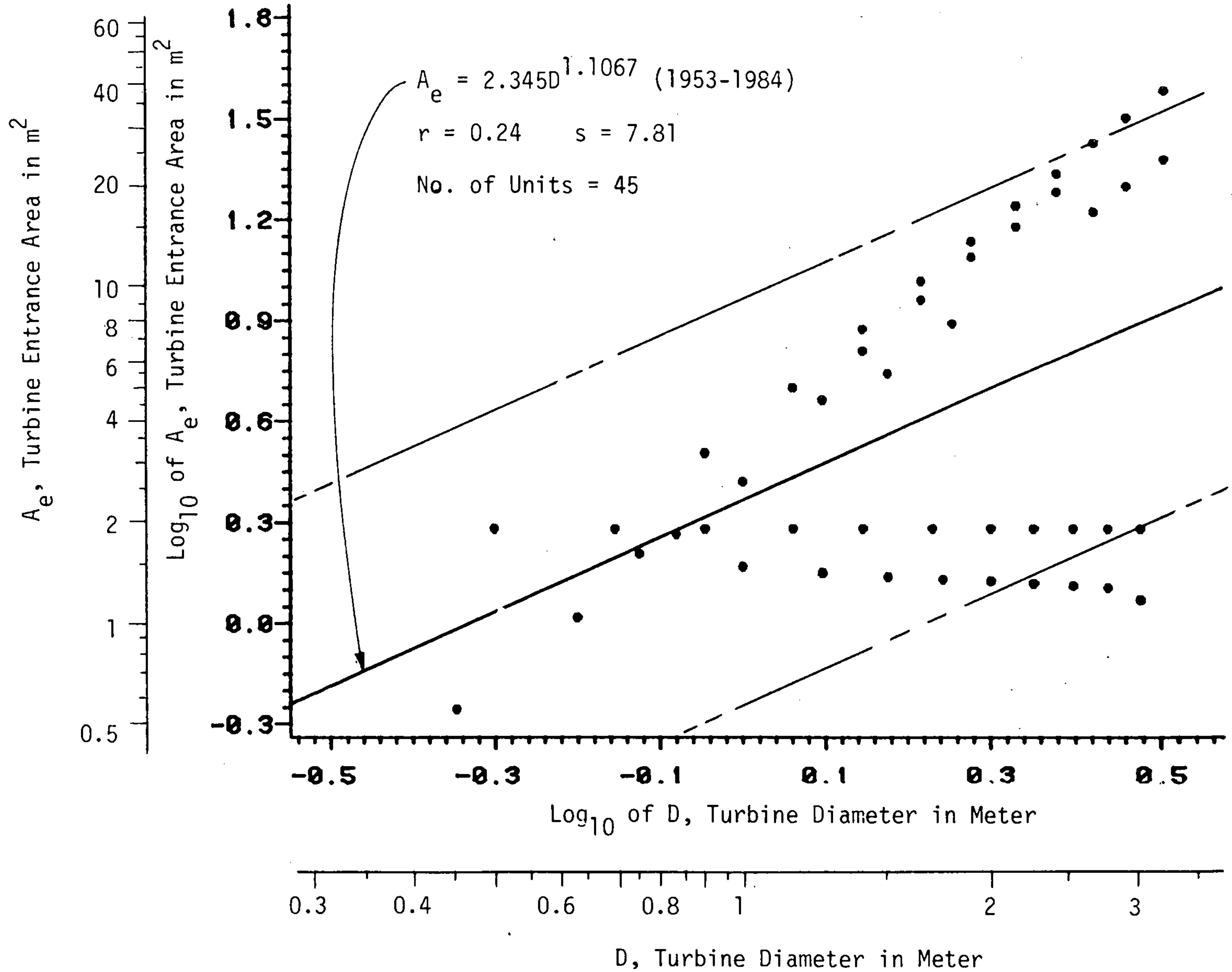

Figure 66. Turbine entrance area versus turbine diameter for standard tubular turbines.

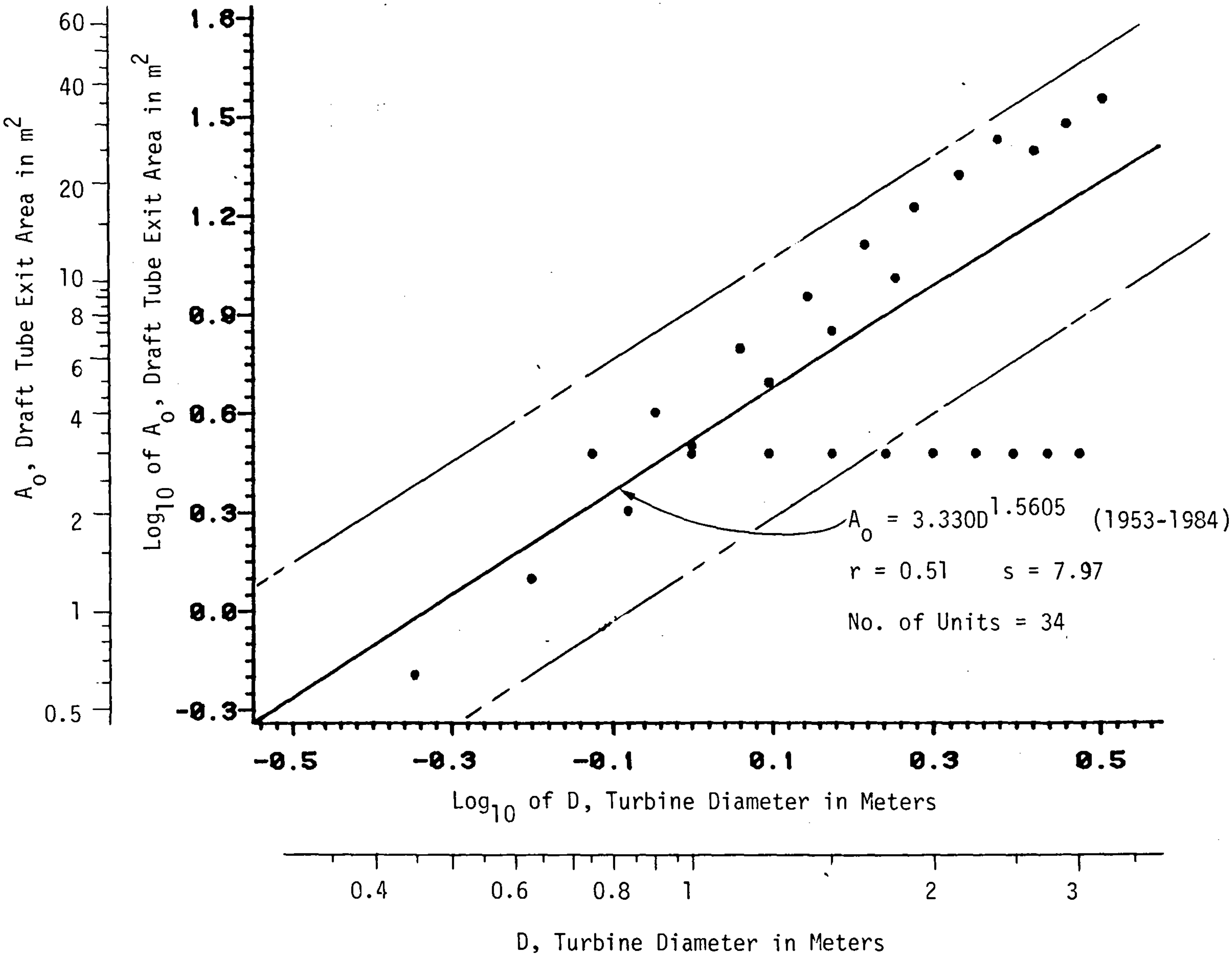

Figure 67. Draft tube exit area versus turbine diameter for standard tubular turbines.

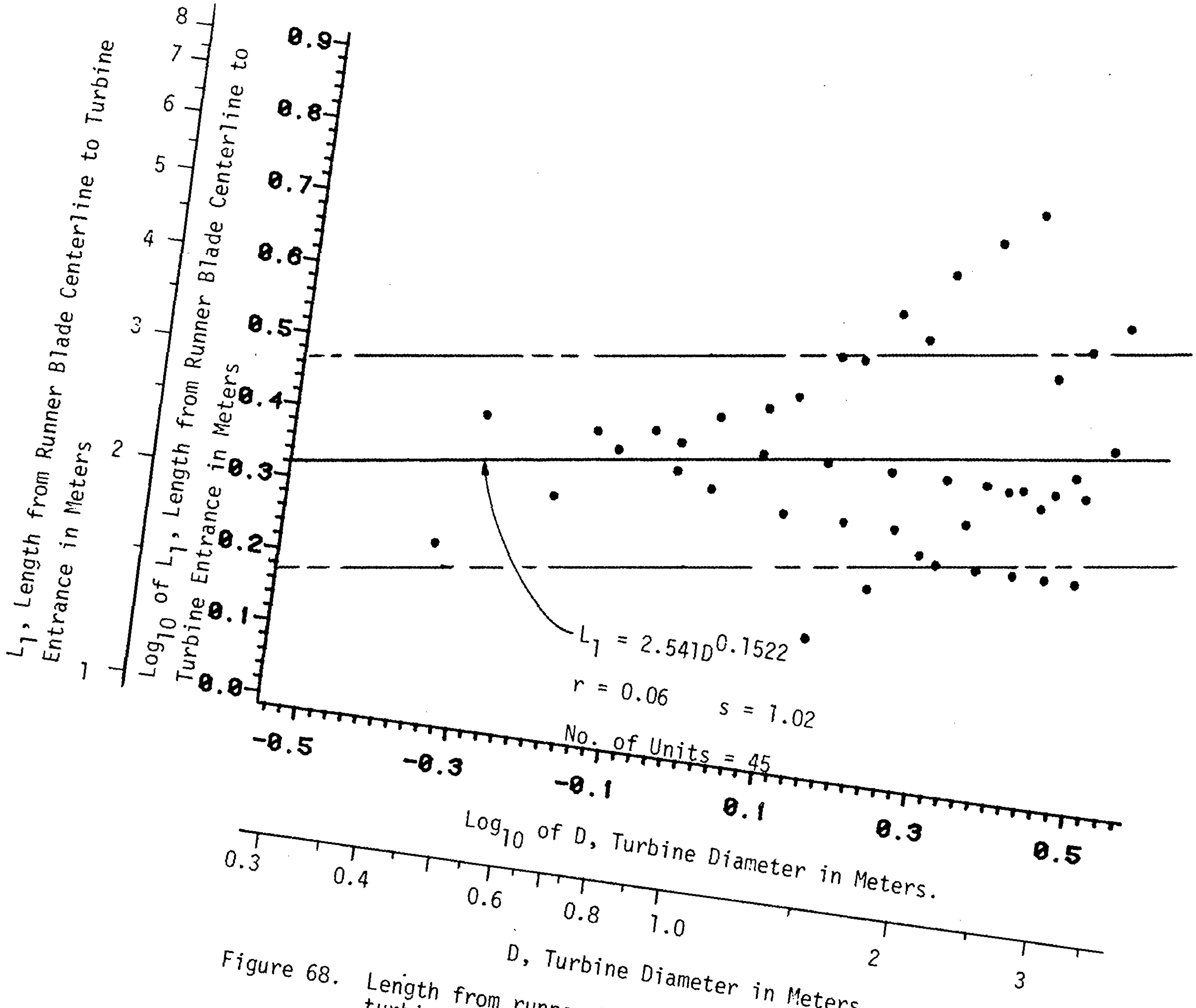

Figure 68. Length from runner blade centerline to turbine entrance versus turbine diameter for tubular turbines.

Figure 69 presents the relation between the distance, M, from the runner blade centerline to the draft tube exit where A_0 is measured, and the turbine diameter, D, and the resulting regression equation for tubular turbines is given as:

$$M = 5.939\ D^{0.5560} \qquad \text{Eq. (88)}$$

Table 8 summarizes the regression information and equations developed for relating water passage dimensions to the turbine diameter for standard tubular turbines.

The actual data used in this regression analysis of standard tubular turbines is presented in the Appendix 3.

Cross-Flow Turbines

No information was obtained on sizes of water passage dimensions for cross-flow turbines.

ANALYSIS AND USE OF RESULTS

The basic purpose of the research was to present simplified methods for making preliminary selection of diameter and speed of low-head turbines. A review of the work of Lindestrom (no date) of the Swedish firm KMW presented a simplified nomograph for making that selection. Figure 70 is a reproduction of the nomograph from Lindestron (no date) for bulb turbines. Because the basic parameters used were the same as those involved in the regression developed as Eqs. (24) and (25) that is D = F (P/H), it was simple to construct a similar nomograph from the regression equations developed on this project. To check the validity of the KMW nomograph, the basic data for bulb turbines manufactured by only KMW were subjected to a seperate regression analysis the same as with all the bulb units. Table 9

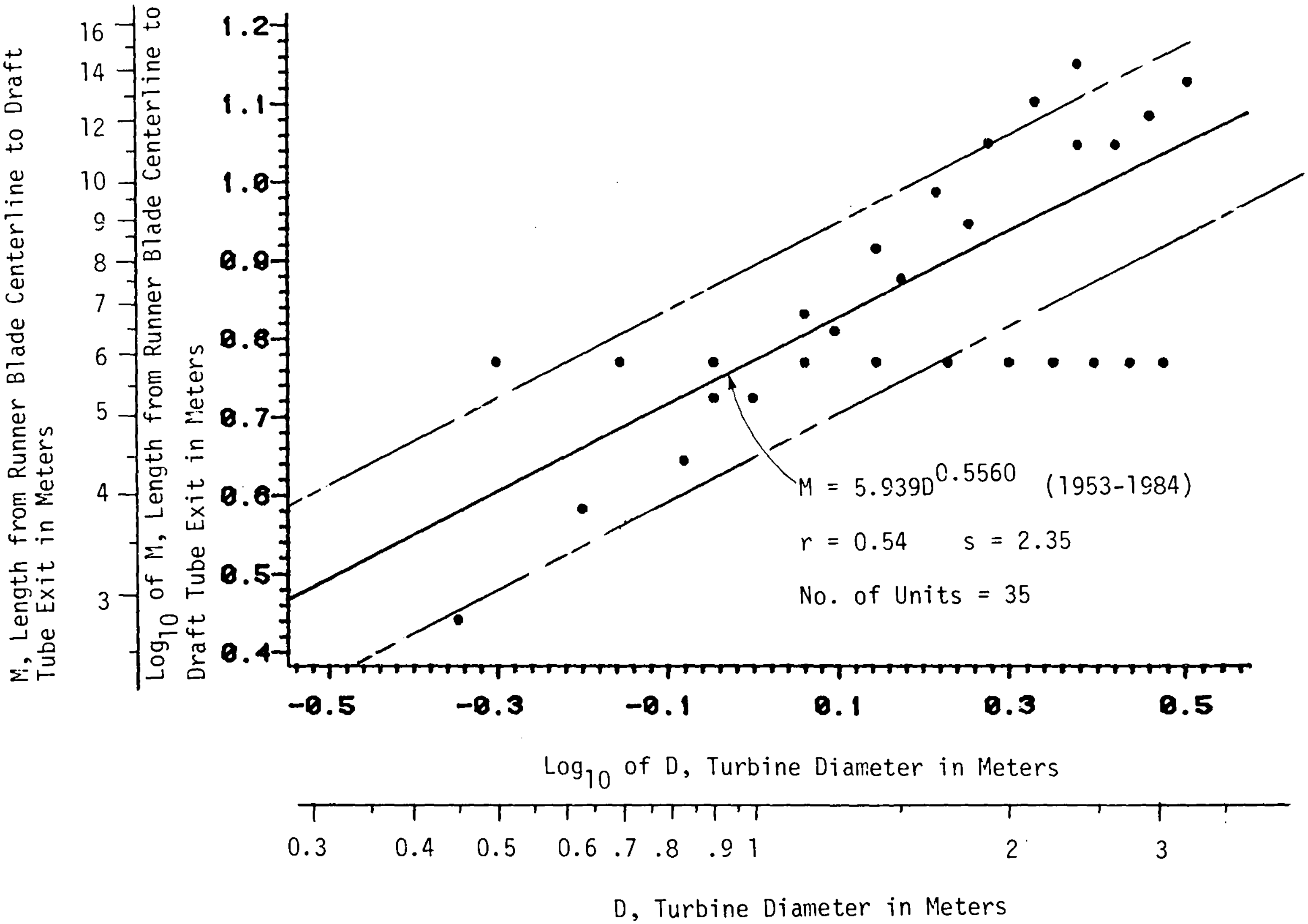

Figure 69. Length from runner blade centerline to draft tube exit versus turbine diameter for standard tubular turbines.

TABLE 8

SUMMARY LISTING OF REGRESSION INFORMATION AND EQUATIONS

RELATING TO WATER PASSAGE DIMENSIONS FOR STANDARD TUBULAR TURBINES

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
85	A_e	$A_e = 2.345\ D^{1.1067}$	0.24	7.81		45
86	A_o	$A_o = 3.330\ D^{1.5605}$	0.51	7.97		34
87	L_1	$L_1 = 2.5408\ D^{0.1522}$	0.06	1.02		45
88	M	$M = 5.939\ D^{0.5560}$	0.54	2.35		35

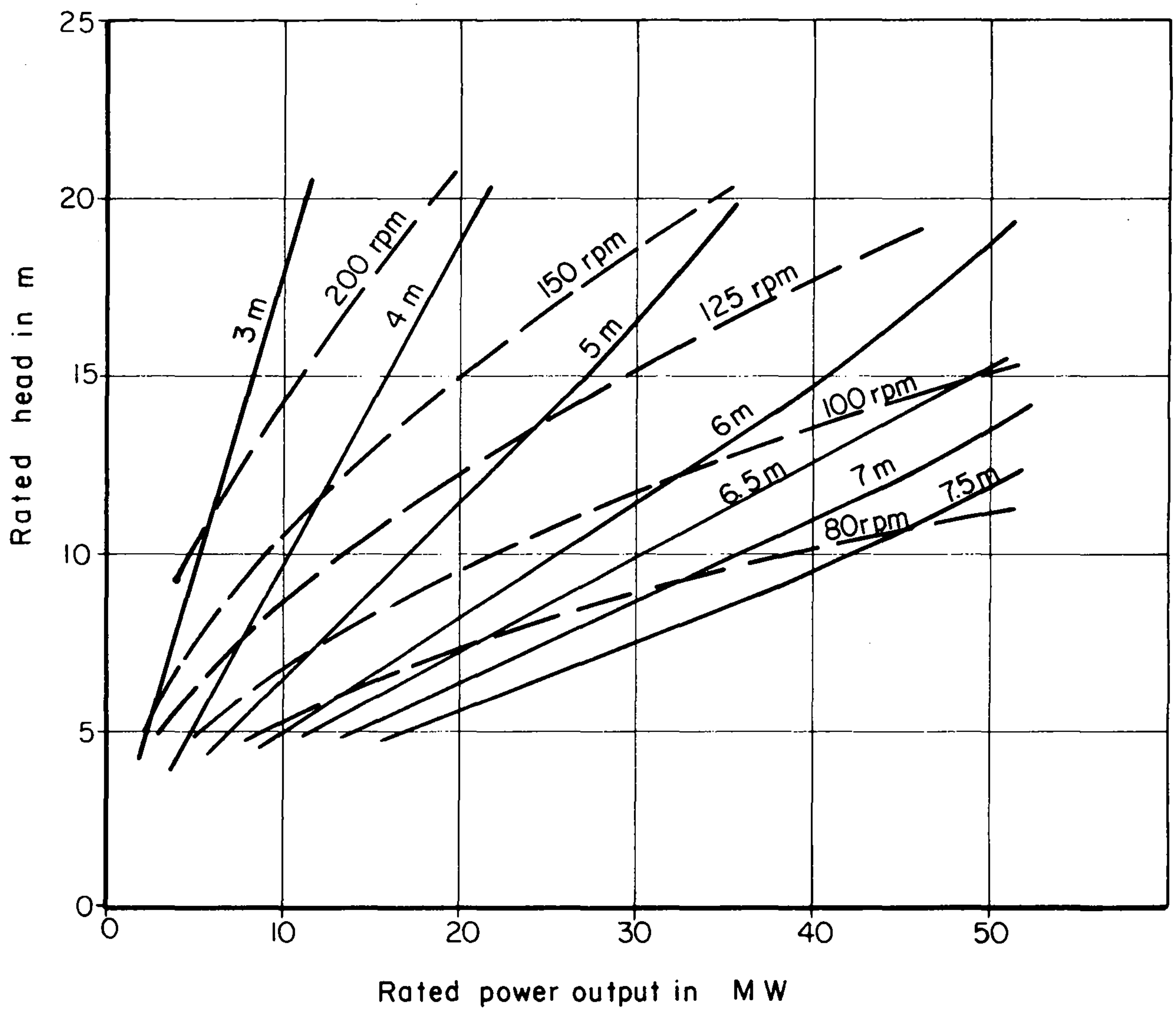

Figure 70. Reproduction of KMW nomograph for selection of turbine diameter and turbine speed for bulb turbines.

TABLE 9

SUMMARY LISTING OF REGRESSION INFORMATION AND EQUATIONS

FOR SPECIAL CASE OF MANUFACTURED KMW BULB TURBINES

Equation Number	Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Sample Period	Number of Units
	N_s	$N_s = 1553.445\ H^{-0.2918}$	0.50	112.23	1959-1984	25
	ϕ	$\phi = 0.1660\ N_s^{0.3728}$	0.86	0.07	1959-1984	25
	ϕ	$\phi = 0.9205\ P_{11}^{0.2522}$	0.65	0.10	1959-1984	25
	D	$D = 0.2917\ \phi^{3.8367}$	0.52	1.00	1959-1984	26
	D	$D = 0.1763\ (P/H)^{0.4489}$	0.97	0.48	1959-1984	25
	D	$D = 4.1604\ (Q/N)^{0.3064}$	0.99	0.64	1959-1984	26
	N	$N = 3583.987\ (P/H)^{-0.4833}$	0.78	104.66	1959-1984	25
	N	$N = 164.706\ (\sqrt{H}/D)^{0.8876}$	0.99	5.58	1959-1984	26
	σ	$\sigma = 1.786 \times 10^{-5}\ N_s^{1.7023}$	0.60	0.61	1959-1984	24
	σ	$\sigma = 0.422\ Q_{11}^{1.5486}$	0.64	0.64	1959-1984	24

presents the summary of the results of that special regression analysis of KMW manufactured bulb units, giving the empirical equation, correlation coefficient, standard deviation, sample period and the number of units involved. A check of using the regression from the authors special study confirmed the individual curves of the nomograph that had been presented in Lindestrom (no date).

Figure 71 gives a nomograph for estimating bulb turbine diameters based on rated head and rated power output. This nomograph was developed by using the regression equation, Eq. 25. A similar nomograph for tubular turbines is presented in Figure 72 which utilizes regression equation, Eq. 41. The corresponding nomograph for cross-flow turbines is presented in Figure 73 which utilizes regression equation, Eq. 57.

An estimation of turbine speed can be made in several ways. One way is to use the same parameters of rated head and rated power output as used for bulb turbines the regression equation, Eq. 27. Another method is to use the estimated diameter as found from the nomograph Figure 71 or Eq. 25 and substitute that in regression equation, Eq. 26. An additional approach is to take the estimated diameter as found from nomograph Figure 71 or Eq. 25 and substitute that value of diameter into the regression equation, Eq. 30.

The more conventional approach for estimating turbine diameter and speed has been that explained in U.S.B.R. Monograph No. 20 and is to first find a trial value of specific speed, N_S, from a curve like Figure 3. Then proceed to find a trial speed, N', from the specific speed equation.

$$N_S = \frac{N \sqrt{P}}{H^{1.25}} \qquad \text{From Eq. (4)}$$

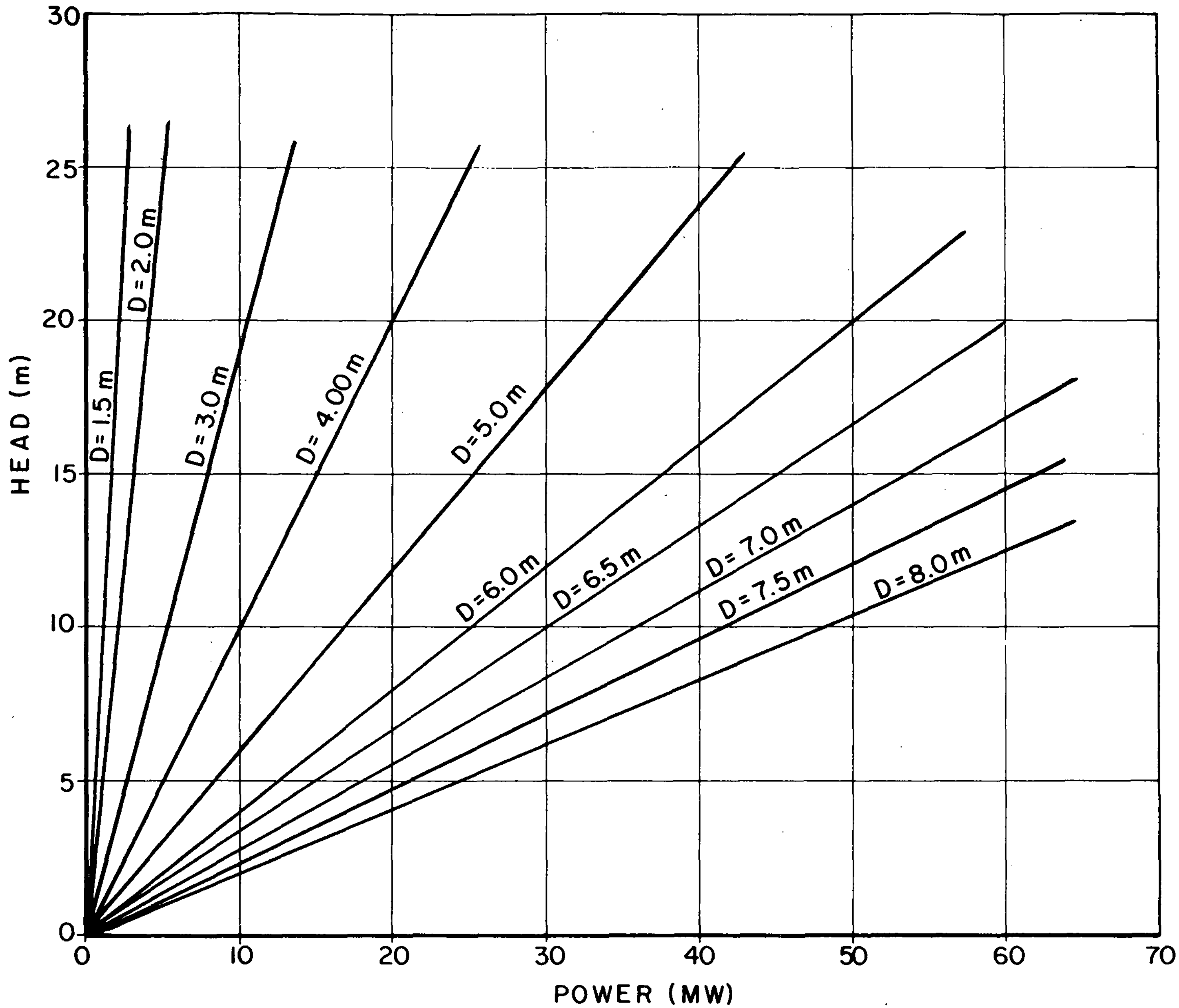

Figure 71. Nomograph for estimating turbine diameter from rated head and rated power output for bulb turbines.

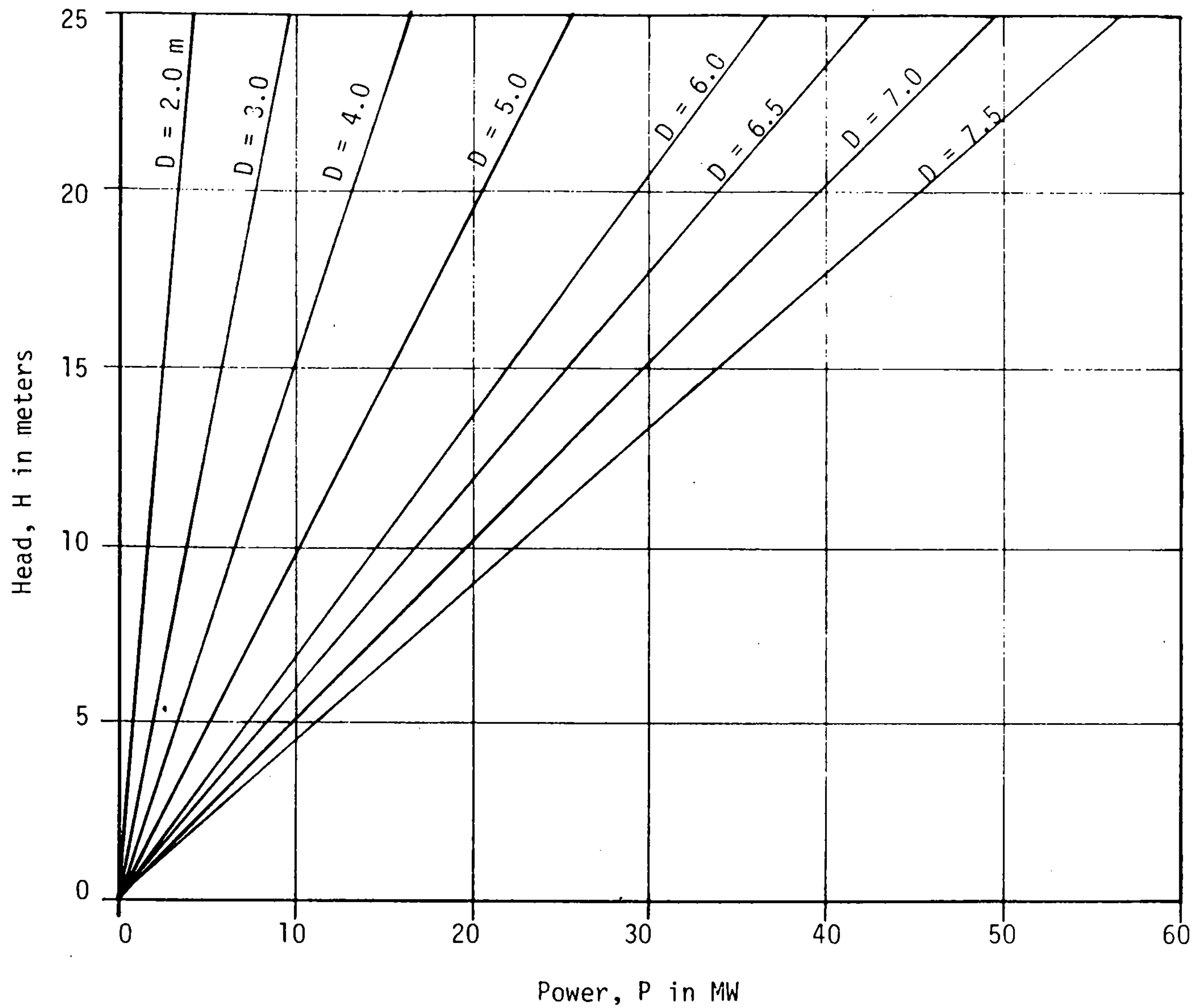

Figure 72. Nomograph for estimating turbine diameter from rated head and rated power output for tubular turbines.

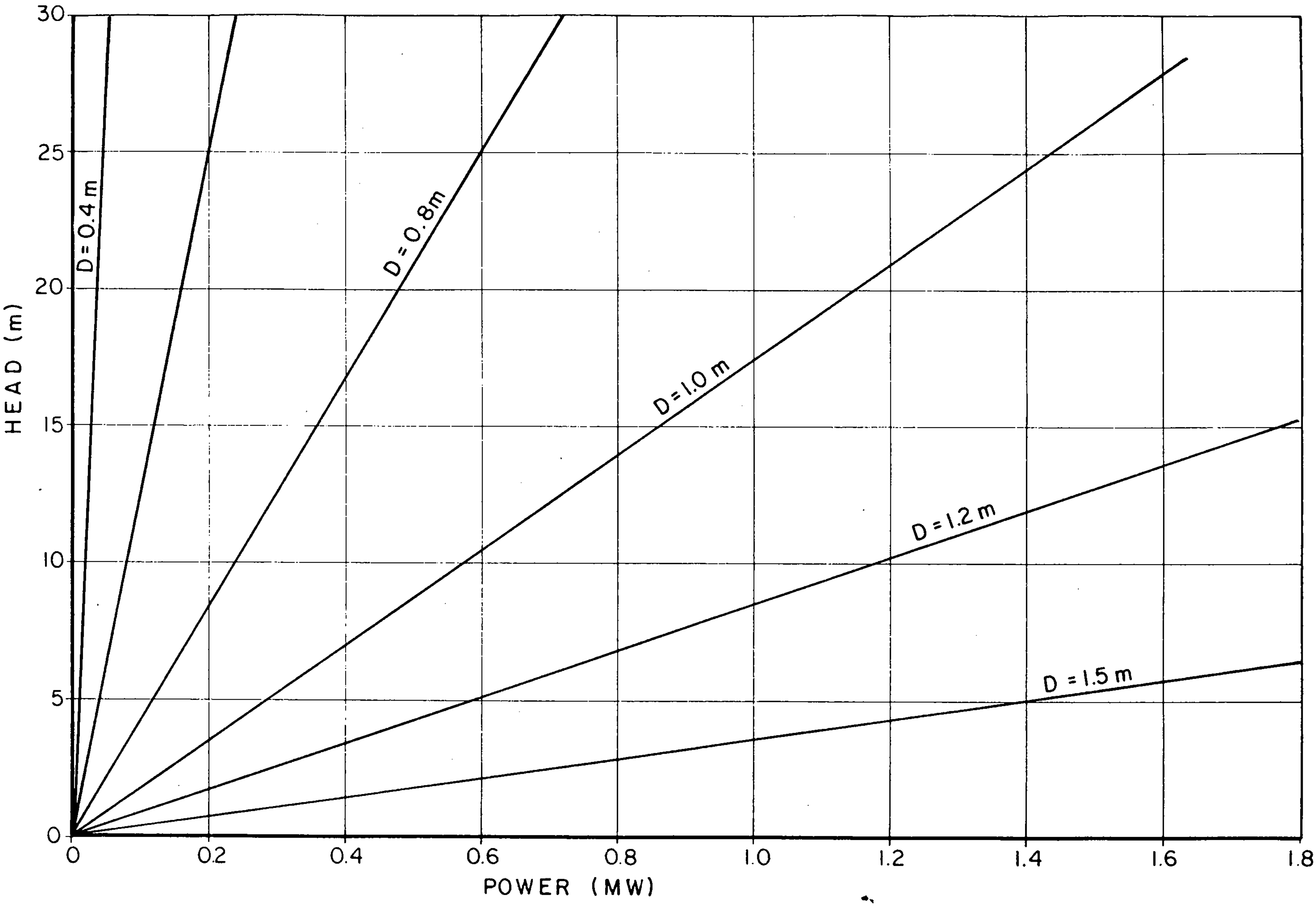

Figure 73. Nomograph for estimating turbine diameter from rated head and rated power output for cross-flow turbines.

A synchronous speed must then be chosen utilizing the relation.

$$Np = \frac{120 \times f}{N'} \qquad \text{Eq. (89)}$$

where Np = number of generator poles

f = electrical frequency in H_Z.

The number of poles, Np, must be in multiples of two or four, usually in multiples of four. Once a synchronous speed is chosen then the actual specific speed, N_S, is calculated using, Eq. 4. The next step is to use the actual, N_S, in an empirical equation to determine the speed ratio, Ø. For bulb turbines this would utilize regression equation, Eq. 18. For propeller units the U.S. Bureau of Reclamation Monograph No. 20 (1976) gives the following:

$$\emptyset = 0.0233\, N_s^{2/3} \qquad \text{Eq. (90)}$$

As a final step the estimated turbine diameter can be determined using selected turbine speed, N, the rated head, H, and the empirically determined value of speed ratio, Ø, in the following form of the speed-ratio equation:

$$D = 84.58\, \emptyset\, \frac{H^{0.5}}{N} \qquad \text{Eq. (91)}$$

This equation comes from the basic definition of speed ratio. To illustrate the procedure for this selection process for estimating turbine diameter and turbine speed sample calculations have been presented in the Appendix. The sample calculations have been performed for a manufactured unit at a plant in Europe known as Isawerk 3.

Additional comments are presented on the advantages of different approaches to diameter estimation following a presentation of comparisons.

COMPARISONS

With the various different regression that were performed it is informative to make a few simple comparisons. Figure 74 is a comparison of several different experience curves relating specific speed, N_S, to the rated head, H, for different kinds of low-head turbines studies on this project as well as results from other published studies. The curves include two experience curves taken from the Figure 11 of the U.S. Bureau of Reclamation Monograph No. 20 (1976), the work of de Siervo and de Leva (1977), the work of Lindestrom (no date), and the experience curves for the three different types of turbines (bulb, tubular, and cross-flow turbines) studied on this project. Table 10 summarizes the information on the specific speed versus rated head relations for low-head type turbines.

Because the U.S. Bureau of Reclamation Monograph 20 gives an empirical equation relating the speed ratio, $\emptyset$, to the specific speed, N_S, that is used in preliminary speed and diameter selection a comparison was made with similar relations developed in this study. Figure 75 shows this comparison. The data gathered on this project were used to develop a regression equation with the same exponential power of the N_S as was reported in the U.S.B.R. Monograph 20, that is, N_S raised to two thirds power. The regression equations for the different types of turbines developed are indicated below:

$$\emptyset = 0.6374 + 0.164\ N_s^{2/3} \quad \text{(Bulb)} \qquad \text{Eq. (92)}$$

$$\emptyset = 0.2036 + 0.0227\ N_s^{2/3} \quad \text{(Tubular)} \qquad \text{Eq. (93)}$$

$$\emptyset = 0.4356 + 0.0026\ N_s^{2/3} \quad \text{(Cross-flow)} \qquad \text{Eq. (94)}$$

It should be noted that the plotting of Equation 19 developed by Kpordze-Warnick for bulb turbines shows a slight deviation from

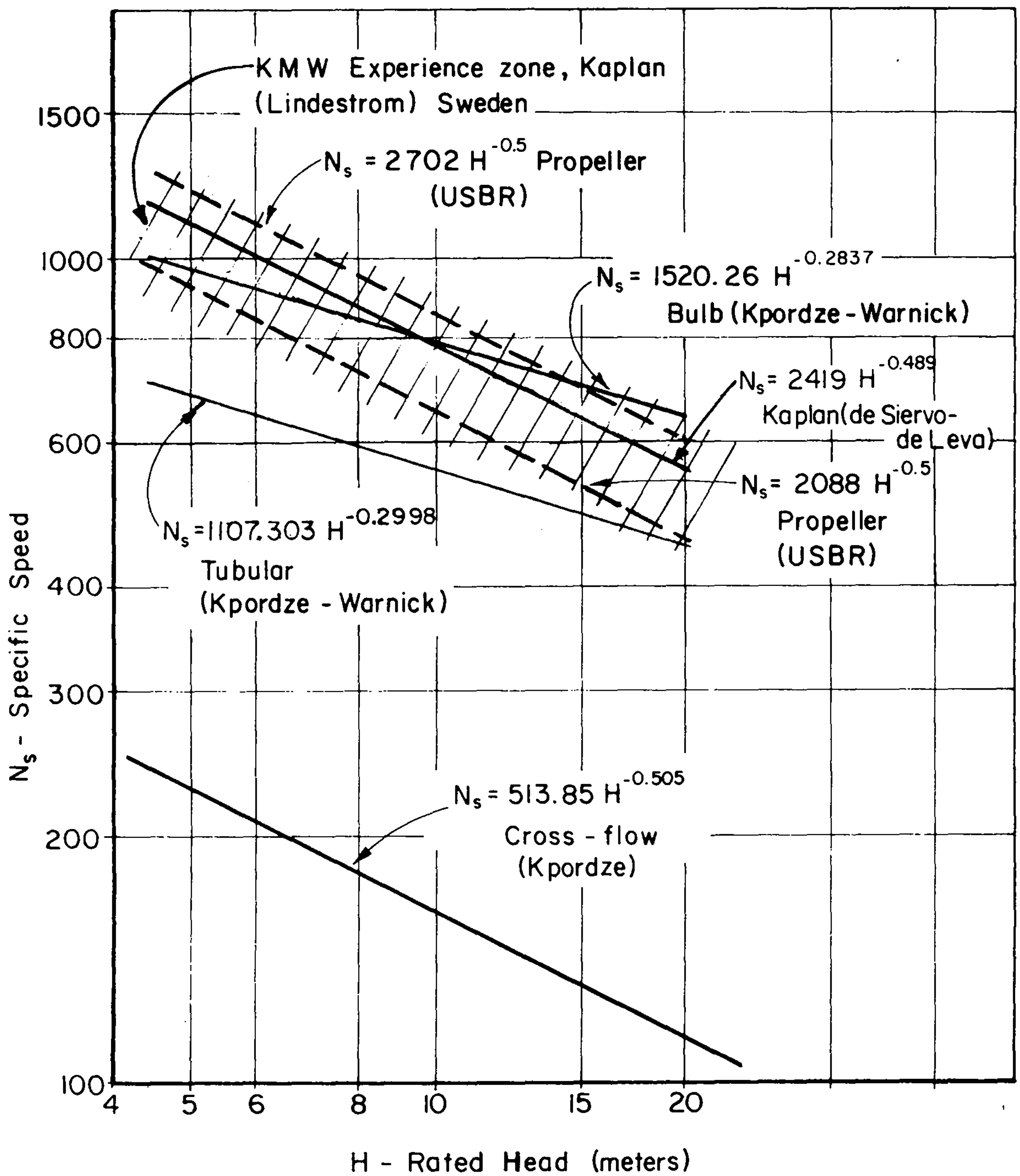

Figure 74. Comparison of experience curves of specific speed versus rated head for different types of low-head turbines.

TABLE 10. COMPARISON INFORMATION OF REGRESSION EQUATIONS FOR N_s VERSUS H FOR DIFFERENT TYPES OF LOW-HEAD TYPE TURBINES

Type of Turbine	Regression Equation	Correlation Coefficient	Standard Deviation	Number of Units	Period of Manufacture	Authors
Propeller	$N_s = 2702H^{-0.5}$	---	---	---	prior to 1976	U.S.B.R.
Propeller	$N_s = 2088H^{-0.5}$	---	---	---	prior to 1976	U.S.B.R.
Kaplan	$N_s = 2419H^{-0.489}$	0.89	47.6	N.A.	1970-76	de Siervo
Bulb	$N_s = 1520.256H^{-0.2837}$	0.40	118.24	119	1971-84	Kpordze-Warnick
Tubular	$N_s = 1107.303H^{-0.2998}$	0.62	92.71	54	1957-84	Kpordze-Warnick
Cross-flow	$N_s = 513.846H^{-0.5047}$	0.79	36.89	17	1966-82	Kpordze-Warnick
Kaplan	$^*N_s = 2400H^{-0.5}$	---	---	---	N.A.	Lindestrom

*Median line as interpolated from Fig. 11 of report by Lindestrom

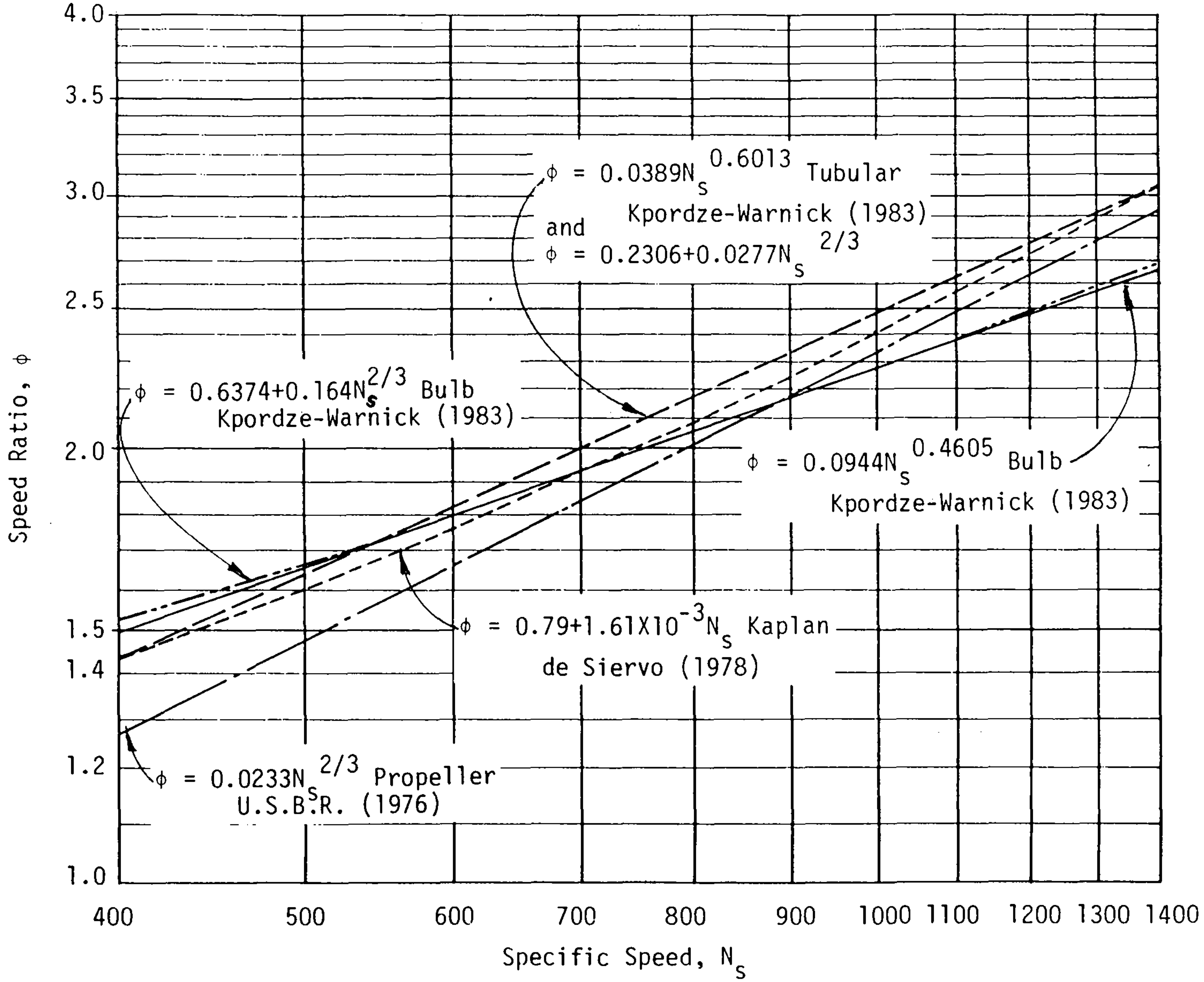

Figure 75. Comparison of experience curves of speed ratio versus specific speed for different types of axial-flow turbines.

Equation 92 at the two extremities of the plotted lines. The Kpordze-Warnick form of the relationship plots as a straight line on logarithmic paper and has N_S raised to the exponential power value of 0.4605. The correlation coefficient is slightly better for the Kpordze-Warnick form than with the N_S raised to the two-thirds power. There is essentially the same margin of error in the two forms of the equation as indicated by the values of the standard deviation found in the development of the two equations.

The plotting of Equation 38 developed by Kpordze-Warnick for tubular turbine and the Equation 93 utilizing N_S raised to the two thirds power for tubular turbine are so nearly the same it is not possible to distinguish between the two lines on the scale shown in Figure 75.

Brief trial comparisons of using these different experience curves shown in Figure 75 would indicate that in the middle range of situations calling for turbine selection for N_S in the range from 700 to 900, reasonably similar results can be expected using de Siervo empirical relations, the U.S.B.R. empirical equation for propeller units, and the empirical equations for bulb turbine units developed in this study. In ranges of N_S values outside the range 700 to 900 traditional empirical equations should not give good results.

An additional comparison was made of the regression analysis involving the plant sigma, σ, and the specific speed, N_S. Figure 76 gives the comparison that includes σ versus N_S for bulb turbines, σ versus N_S for tubular turbines and a reproduction of a KMW relation between σ versus N_S for all turbines manufactured by that company, Lindestrom (no date). Plotted on Figure 76 is the empirical equation for σ versus N_S as taken from U.S. Bureau of Reclamation Monograph 20 (1976).

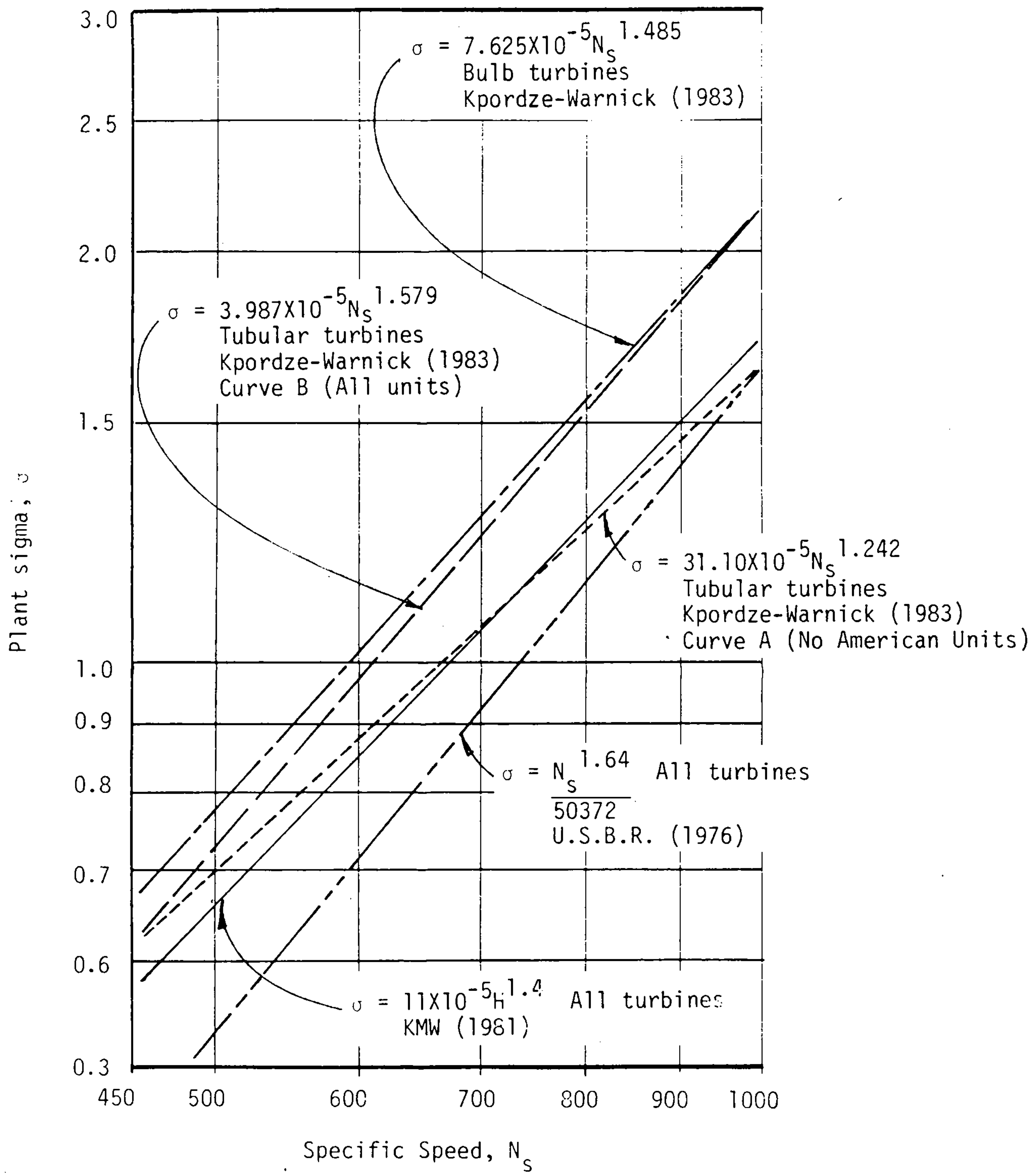

Figure 76. Comparative of experience curves of plant sigma versus specific speed for different low-head turbines.

The comparison shown in Figure 76 includes a stratification of tubular turbine data (Curves A and Curves B) of those tubular turbine manufactured outside the United States. The σ versus N_s curve for just the units manufactured outside the United States (Curve A) does show that lower values of σ will be predicted for corresponding values of N_s. Curve B is for all tubular turbines studied including American manufactured units and some European units and a few Japanese units. This indicates that if units are submerged below tailwater (as they usually are for bulb and tubular turbines) greater submergence has been required on American manufactured tubular turbines. Likewise, it would indicate that the experience curves show bulb turbines have been submerged less than tubular units.

Review of an article by Khanna and Bansal (1979) revealed an experience curve relating plant sigma, σ, to the unit discharge, Q_{11}, for bulb turbines. With the regression analyses performed on this project involving the plant sigma, σ, and the unit discharge, Q_{11}, for bulb turbines, Eq. 66 and for tubular turbines Eq. 68 it was possible to make a comparison. The comparison is shown in Figure 77.

The equation listed for the reproduction of experience curves from Khanna and Bansal (1979) were developed using curve fitting by the authors of this report. The work of Khanna and Bansal (1979) also included an experience curve for Kaplan turbines. It has also been reproduced on Figure 77 for comparison purposes.

An analysis for comparative purposes was made of the characteristics of the draft tube exit velocities of 54 bulb units for which data were available. Purdy (1979) reported that the exit velocity should

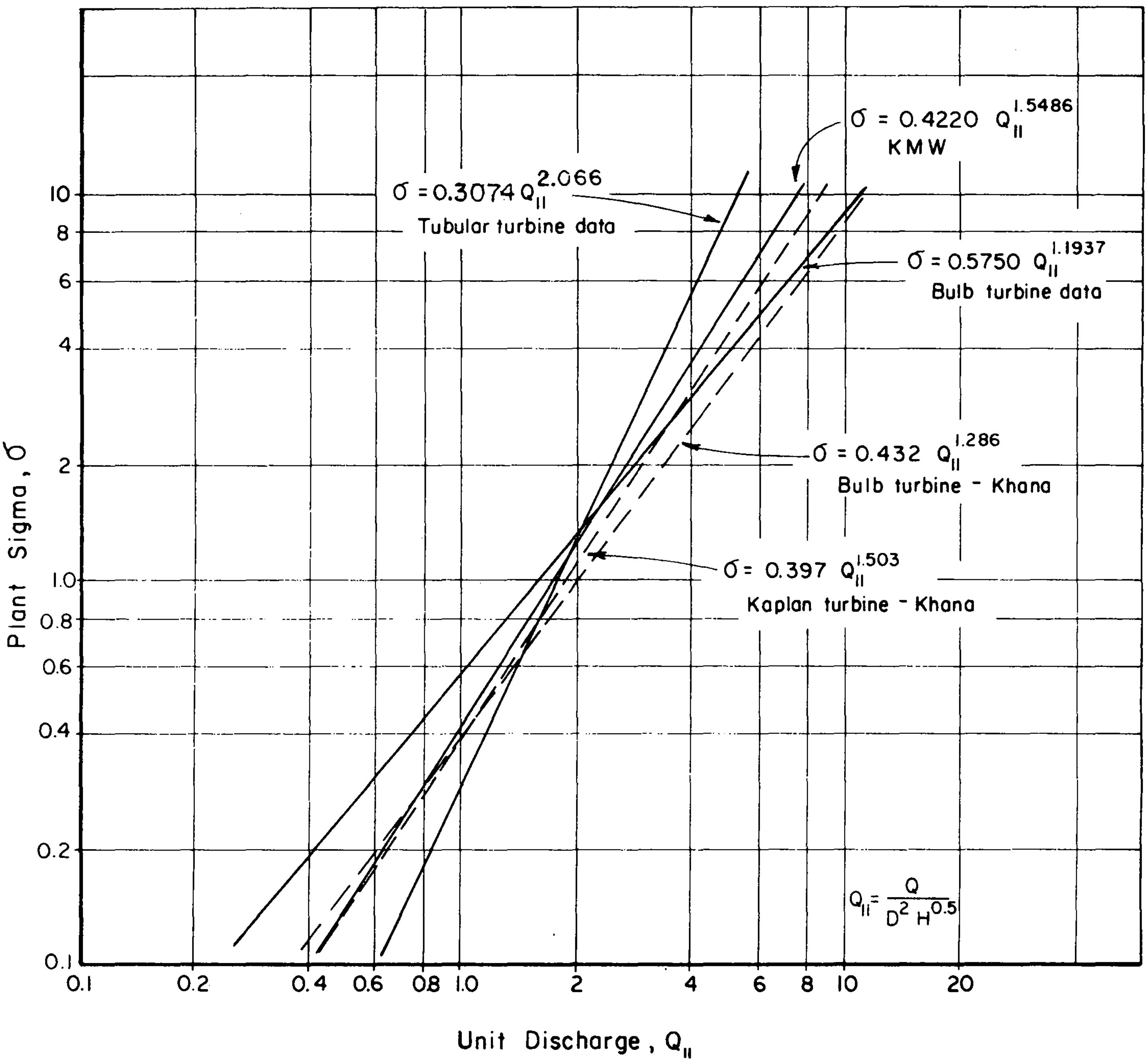

Figure 77. Comparison of experience curves for plant sigma versus unit discharge for different low-head turbines.

not exceed 0.8 $\sqrt{H}$ for rated heads, H, of low-head turbines up to 17 m. Table 11 shows how exit velocity compares with the value of 0.8 $\sqrt{H}$ for each turbine. The recommendation of Purdy was based on the fact that if higher velocities were permitted considerable power was lost but not often considered in the real overall performance. This comparison shows that many of the manufactured turbines have exit velocities that exceed the Purdy recommendations.

To assess the difference that might be expected in using different methods of estimating turbine diameter and turbine speed a comparative study was made of eight hydro power plants that had data on rated head, rated discharge, and rated power output. The data on the eight plants also included the actual manufactured diameter and actual turbine speed used at each plant. Five different methods were used in the assessment: (1) using the traditional approach as presented in U.S. Bureau of Reclamation Monograph No. 20 for propeller turbines, (2) using the regression equations developed by de Siervo and de Leva (1977 and 1978) for Kaplan turbines, (3) using the nomograph from Lindestrom (no date), (4) using the regression equation developed in the special study of KMW manufactured units, and (5) using the regression relations developed in this study using all the bulb turbines. Sample calculations showing how the comparative numerical values for turbine diameter, D, and turbine speed, N, were obtained are presented in the Appendix 2. Table 12 presents the results of the assessment.

The results would indicate that the simplified selection procedures suggested by the authors of this report have several advantages. The procedures are simple and require only two parameters, rated head and rated power, that are normally available early in feasibility studies. A review and comparison of the correlation coefficients of the various

Table 11. COMPARISON OF DRAFT TUBE EXIT VELOCITY WITH PURDY'S RECOMMENDED LIMIT FOR MANUFACTURED BULB TURBINES

OBS	STATION	MANU-FACTURER	YEAR OF COMMIS-SIONING	DRAFT TUBE EXIT VELOCITY (M/SEC)	PURDY SUGGESTED VELOCITY
1	URSTEIN	V	1969	1.70905	2.64121
2	ALTENWORTH	V	1976	2.44459	2.99333
3	ABWINDEN-AS	V	1979	2.31421	2.25708
4	ABWINDEN-AS	VA	1979	2.20013	2.29085
5	MELK	V	1982	2.44459	2.29085
6	GREIFENSTEI	VA	1984	2.85202	2.67731
7	KLEINMUENCHEN	VA	1978	2.15303	2.71293
8	MA JI TANG	V	1984	2.30004	2.04900
9	ANKKAPURHA	TAM	1983	6.19493	2.50440
10	VAJUKOSKI	TAM	1984	6.05602	3.09839
11	ARGENTAT	V-C	1957	1.95942	3.25945
12	ARGENTAT	V-C	1958	2.95316	3.33706
13	LA RANCE	V-C	1966	3.00220	1.92666
14	ABZAC	V-C	1958	2.57576	1.18659
15	MARCKOLSHEIM	V-C	1957	2.33766	2.46577
16	RABODANGES	V-C	1959	1.75520	1.95959
17	RHINAU	V-C	1960	1.25893	2.10143
18	GERSTHEIM	V-C	1967	2.99847	2.66533
19	GERSTHEIM	V-C	1968	1.14943	2.40000
20	STRASBOURG	V-C	1970	3.28240	2.73057
21	FANKEL	V	1962	1.20957	1.61988
22	MUDEN	V	1962	1.20957	1.61988
23	LEHMEN	V	1966	1.20957	1.84174
24	URSPRING	V	1963	1.60643	2.27684
25	SYLVENSTEIN	V	1960	2.02922	3.86988
26	LECHSTUFE20	V	1984	1.30782	2.45275
27	GOTTFRIEDING	V	1977	1.50693	1.95959
28	REHLINGEN	V	1984	1.52827	2.20545
29	SCHODEN	V	1984	1.52827	1.90997
30	SAN PEDRO	V-C	1982	2.42915	2.50440
31	GAMLEBROFOSS	KMW	1970	2.04082	3.00400
32	DUVIKFOSS	KMW	1975	3.06122	1.93494
33	SKOGSFORSEN	KMW	1959	1.61111	2.99333
34	HALLEFORS	KMW	1966	1.73442	2.19089
35	SPERLINGSHOLM	KMW	1967	1.93798	1.53883
36	PAKKI	KMW	1970	2.12094	2.65330
37	BODUM	KMW	1975	2.24775	2.03961
38	LANDAFORS	KMW	1976	2.59259	1.84174
39	ASELE	KMW	1981	2.90276	2.54244
40	SODERFORS	KMW	1979	1.92157	1.69706
41	JUVELN	KMW	1978	2.38095	2.65330
42	TORRON	KMW	1978	2.39756	3.48712
43	NASI	KMW	1979	2.31063	1.82428
44	AVESTA-	KMW	1982	2.24618	1.84174
45	MATFORS	KMW	-	2.48830	2.45927
46	LILLA EDET 4	KMW	1982	2.24359	2.03961
47	NAS2	KMW	1980	2.31063	1.82428
48	GRANBOFORSEN	KMW	1980	2.21017	1.95959
49	WINZNAU	V-C	1962	1.16667	1.87617
50	TASJO	TAM	1978	5.95522	2.77128
51	HOTING	TAM	1978	6.24527	2.57992
52	VIFORSEN	TAM	1982	5.67752	2.16148
53	IDAHO FALLS	VA	1981	1.15272	1.87617
54	PELTON REREG.	VA	1982	2.34872	2.60461

TABLE 12. COMPARATIVE RESULTS OF DIFFERENT METHODS OF ESTIMATING TURBINE DIAMETER AND TURBINE SPEED

Name of Plant	Isarwerk 3 (EW)		Gerstheim (VC)		Brashereidfoss (KB)		Koide (Fugi)		Cakovec (N)		Lechstufe 20 (V)		Idaho Falls (VA)		Lachine (AC)		Granboforsen (KMW)	
Parameters	D(m)	N(rpm)	D(m)	N(rpm)	D(m)	N(rpm)	D(m)	N(rpm)	D(m)	N(rpm)	D(m)	N(rpm)	D(m)	N(rpm)	D(m)	N(rpm)	D(m)	N(rpm)
Actual Parameter Values	2.45	157	1.60	333.33	5.80	88.20	3.40	150	5.40	125	2.85	176.50	4.85	94.70	6.90	93.80	5.80	75
USBR Equation																		
$N_s = 2702H^{-0.5}$																		
$\phi = 0.0233N_s^{2/3}$																		
$D = 84.47\phi H^{0.5}/N$	2.01	250	1.36	375	5.83	93.75	3.03	187.5	5.33	115.38	2.50	214.29	4.77	106.52	6.52	88.24	6.16	88.33
deSiervo Equations																		
$N_s = 2419H^{-0.489}$																		
$\phi = 0.79+1.61X10^{-3}N_s$																		
$D = 84.5\phi H^{0.5}/N$	2.19	214.29	1.41	375	6.14	88.24	3.15	187.5	5.81	107.14	2.58	214.29	5.00	100.00	7.02	78.95	6.56	75.00
KMW Graph	-	200.00	-	-	5.91	91.76	3.23	194	5.71	128.20	-	-	5.14	86.36	6.57	98.92	6.39	70.71
KMW Equations																		
D = F(P/H)	2.17		1.53		5.83		3.30		5.67		2.70		4.71		6.59		5.95	
D = F(Q/N)	2.23		1.22		5.86		3.41		5.14		2.62		5.04		6.73		5.87	
$N_s = 1553.495H^{-0.2918}$																		
$\phi = 0.166N_s^{0.3728}$																		
$D = 84.6\phi H^{015}/N$	2.08		1.36		5.89		3.12		5.47		2.50		4.82		6.31		6.18	
N = F(P/H)		250		299.41		83.33		150		93.75		187.5		107.14		71.43		83.33
$N = F(N_s)$		187.5		375.00		88.24		187.5		125.00		214.29		88.24		83.33		71.43
$N = F(\sqrt{H}/D)$		166.7		375.00		93.75		187.5		125.00		187.50		88.24		88.24		75.00
K-W Equations																		
D = F(P/H)	2.21		1.57		5.91		3.36		5.75		2.75		4.78		6.67		6.03	
D = F(Q/N)	2.30		1.47		6.07		3.40		5.21		2.59		5.10		6.88		5.98	
$N_s = 1520.256H^{-0.2837}$																		
$\phi = 0.0944N_s^{0.4605}$																		
$D = 84.6\phi\sqrt{H}/N$	2.16		1.39		6.03		3.16		5.50		2.53		5.0		6.82		6.40	
N = (P/H)		214.3		300.00		88.23		150		88.24		187.50		107.14		83.33		
N = (Q/N)		250		375.00		88.24		150		125.00		214.29		88.24		83.33		
$N = (\sqrt{H}/D)$		166.7		300.00		88.24		187.5		125.00		187.50		106.52		88.24		

regression equations used in the selection prodecures is revealing. Table 13 shows the various regression relations used and the value of the correlation coefficient for each relation for the various different kinds of low-head turbines. This shows that for the functions involving $D = F(P/H)$, and $N = F(\frac{\sqrt{H}}{D})$ the regression correlation coefficients are higher than the functions involving N_S and $\emptyset$. The author's suggested approach to estimation of turbine diameter and turbine speed appears to give greater accuracy and consistency.

CONCLUSIONS AND RECOMMENDATIONS

This study of experience curves has collected data on rated head, rated discharge, rated power output, turbine speed, and turbine diameter on more than 300 manufactured low-head turbines produced throughout the world since 1953. Additional information on turbine water passage dimensions and on particular characteristic sizes of turbine intakes and draft tube exits has been compiled. The data have been subjected to an intensive mathematical analysis by regression techniques in an attempt to develop useful predictive methods for feasibility and preliminary design purposes. The following conclusions are made.

The information on rated head, rated discharge, rated power output, turbine speed and turbine diameter along with water passage dimensions has been catalogued in a convenient computer format (see Appendix 3). The catalogue in itself should be a valuable reference from which comparisons could be made when choosing preliminary features of turbine installations for a new hydro power sites.

A comprehensive collection of experience curves for the conventional turbine constants and turbine selection approaches has been developed for bulb turbines, tubular turbines and cross-flow turbines.

Table 13. Comparison of value of correlation coefficients for the important regression equations.

	Separate Study of KMW Turbines	Bulb Turbines	Tubular Turbines	Rim-Generator Turbines	Cross-flow Turbines
Number of Units	26	150	28	-	17
Regression Relation		Values of Correlation Coefficient			
N_s vs H	0.50	0.40	0.52	-	0.79
ϕ vs N_s	0.86	0.86	0.82	-	0.06
D vs P/H	0.97	0.98	0.96	-	0.89
D vs Q/N	0.99	0.99	0.96	-	0.84
N vs P/H	0.77	0.76	0.69	-	0.79
N vs $\frac{\sqrt{H}}{D}$	0.99	0.97	0.96	-	0.98

The experience curves have been developed using conventional hydropower terms and turbine constants that have been applied to Kaplan turbines, Francis turbines and Pelton turbines of the impulse type. The results have been presented in easy-to-use equation form and are also presented graphically to show the scatter of the data in the various relations that were developed.

The results of the study of cavitation characteristics of low-head turbines using the relation between plant sigma, σ, and specific speed, N_S, did not show as good a correlation as expected. There is considerable variation in the relation between plant sigma and specific speed from company to company and the correlation coefficients of the regression are not very high. Caution should be used in applying the experience curves of plant sigma versus specific speed developed in this study. Because the use of this cavitation coefficient in turbine setting elevation determination is highly dependent on cost of excavation for the draft tube this becomes a difficult item to make authoritative guidelines for preliminary design purposes.

The results of the study of dimensions of water passage, and their relation to turbine diameter are reasonably good for the bulb turbines. Insufficient data were obtained on tubular turbines to make regression analysis of relations between turbine diameter and water passage dimensions. However, the latest recommendation of manufacturers with regard to sizing water passages has been catalogued and presented in a useful form for tubular turbines.

A significant and very simplified procedure for estimating turbine runner diameter and turbine speed has been developed. This new procedure was tested and compared with the procedure presented in the

U.S.B.R. Monograph No. 20 and with other approaches. Results of the comparison shown in Table 12 indicates that the new simplified procedures give more consistent estimates of turbine diameter and speed than other methods and are easier to apply using data that are readily available early in the planning stage of a hydropower investigation. A careful documentation of steps in the selection process for estimation of turbine diameter and turbine speed has been presented in sample calculations shown in Appendix 2.

Because these regression equations developed in this study are from a much larger sampling of manufactured units that was used in development of the empirical equations in U.S.B.R. Monograph No. 20 and because the study is for specific types of low-head turbines, the empirical equations developed in this study should be relied on more than using the older more traditional equations. It should always be remembered that final design and confirmation of size of runner and runner speed should be worked out with the individual manufacturers and the estimation developed from experience curves should be used as a check on manufacturers recommendations.

In general good response from turbine manufacturers was obtained but no data were obtained from Chinese and Indian manufacturers and only limited data were obtained from Japanese firms.

Recommendations

The writers recommend that this information be incorporated in a revised edition of the U.S. Bureau of Reclamation Monograph No. 20. To make Monograph No. 20 most useful, the data on more conventional turbines such as Pelton turbines, Frances turbines and vertical Kalpan turbines should be updated and subjected to the same type of regression analysis as was done in this study of low-head type turbines.

If desirable a nomograph for easy selection of each type of low-head turbine could be developed similar to that given in the work of Lindestrom (no date). This nomograph could include further development of the turbine setting restraint as limited by the plant sigma. A recommendation here would be to develop some kind of standardized safety factor that could be agreed to by a team of authorities. The result could be developed as a family of curves of suction head superimposed on an experience curve for selecting diameters given rated head and rated power output. It is recommended that more careful appraisal be made of the exit velocity from draft tubes in manufactured units of low-head turbines to see if reductions in velocities could improve future hydropower installations.

The new procedures developed for estimating of turbine runner diameter and runner speed are recommended for use in preliminary design and feasibility studies for low-head turbines because of the simplicity and the evidence presented in this report of giving consistent results when compared with other more involved procedures.

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TABLE 14

SUMMARY LISTING OF REGRESSION INFORMATION AND EQUATIONS

RELATING TURBINE SPECIFIC SPEED TO RATED HEAD FOR BULB AND TUBULAR TURBINES

FROM DIFFERENT TURBINE MANUFACTURERS

Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Source	# of Units	Type of Unit
N_s	$N_s = 1570.183\ H^{-0.2954}$	0.49	114.92	KMW	24	Bulb
N_s	$N_s = 1752.508\ H^{-0.3353}$	0.90	17.0	TAMP	4	Bulb
N_s	$N_s = 1119.621\ H^{-0.2191}$	0.27	125.63	V-C	11	Bulb
N_s	$N_s = 2263.884\ H^{-0.4520}$	0.75	101.17	VA	5	Bulb
N_s	$N_s = 1316.418\ H^{-0.2770}$	0.38	119.08	V	15	Bulb
N_s	$N_s = 977.618\ H^{-0.1176}$	0.10	194.69	N	59	Bulb
N_s	$N_s = 820.288\ H^{-0.0642}$	0.04	96.13	EW	27	Bulb

TABLE 14 CONTINUED

Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Source	# of Units	Type of Unit
N_s	$N_s = 1653.119\ H^{-0.3230}$	0.98	17.86	KB	5	Bulb
N_s	$N_s = 1340.564\ H^{-0.3053}$	0.38	107.43	FE	12	Bulb
N_s	$N_s = 1053.040\ H^{-.02679}$	0.53	103.57	TAMP	22	Tubular
N_s	$N_s = 1452.099\ H^{-0.3229}$	0.89	23.30	V-C	2	Tubular
N_s	$N_s = 1335.510\ H^{-0.3948}$	0.84	56.52	ALLIS	23	Tubular
N_s	$N_s = 1607.067\ H^{-0.5533}$	0.98	22.02	KB	3	Tubular

Dependent Parameter	Regression Equation	Correlation Coefficient	Standard Deviation	Source	# of Units	Type of Unit
σ	$\sigma = 2.527 \times 10^{-3} N_s^{0.9224}$	0.20	0.34	Tampella	13	Tubular
σ	$\sigma = 1.1529 \times 10^{-5} N_s^{1.7918}$	0.80	0.29	Allis Chalmers	14	Tubular
σ	$\sigma = 2.135 \times 10^{-11} N_s^{3.8269}$	0.49	0.23	Vevey Chamille	2	Tubular
σ	$\sigma = 4.549 \times 10^{-6} N_s^{1.9082}$	0.58	0.84	KMW	12	Bulb
σ	$\sigma = 9.723 \times 10^{-8} N_s^{2.4794}$	0.92	0.15	Tamp	4	Bulb
σ	$\sigma = 8.077 \times 10^{-5} N_s^{1.4907}$	0.44	1.02	V-C	11	Bulb
σ	$\sigma = 1.5416 \times 10^{-3} N_s^{1.0153}$	0.84	0.20	VA	3	Bulb
σ	$\sigma = 1.1143 \times 10^{-4} N_s^{1.4233}$	0.47	0.47	V	15	Bulb

APPENDIX 1

SAMPLE CALCULATIONS FOR TURBINE CONSTANTS CONVERSIONS

A series of sample calculations are presented using actual data from the Rock Island power plant on the Columbia River. Different forms of turbine constants are used in both the American system of units and also the metric system of units. This is presented in case engineers desire to use different forms of the turbine constants and desire to work in different measurement units.

SAMPLE CALCULATIONS FOR TURBINE CONSTANT CONVERSION

Given: Rock Island plant data as example

H = rated head		= 12.1 m
Q = Rated discharge		= 481.0 m^3/sec
P = Rated power output		= 54,000
D = Turbine diameter		= 7.40 m
N = Turbine speed		= 85.7 rpm

Required: To show conversion example calculations.

Analysis and Solution:

From general power equation.

$$P_{theoretical} = \frac{QH\rho g}{1000} = \frac{(481)(12.1)(1000)(9181)}{1000}$$

$$= \underline{\underline{57,095 \text{ kw}}} \longleftarrow \text{answer}$$

$$\eta = \frac{P_{rated}}{P_{th}} = \frac{54,000}{57,095} \text{ X } 100 = \underline{\underline{94.6\%}} \longleftarrow \text{answer}$$

Using Eq. (4) N_s (metric) $= \frac{N\sqrt{P}}{H^{5/4}} = \frac{85.7\sqrt{54,000}}{(12.1)^{1.25}} = \underline{\underline{882.5}}$

$$N_s \text{ American} = 0.262\ N_s \text{ metric}$$

$$= 0.262(882.5) = \underline{\underline{231.2}} \longleftarrow \text{answer}$$

or N_s American $= \frac{N\sqrt{P_{horse\ power}}}{(H\ ft)^{1.25}}$

$P_{kip} = P_{kw}/0.746$ h $= H_{ft} = H_m/0.3048$

$P_{kip} = 54,000/0.746 = 72.386$ hp $H_{ft} = 12.1/0.3048 = 39.7$ ft

$$N_s \text{ American} = \frac{85.7\sqrt{72,386}}{(39.7)^{1.25}} = \underline{\underline{231.4}} \longleftarrow \text{Answer Check}$$

Using Eq. (105)

$$D = 84.58\ \phi \frac{\sqrt{H}}{N}$$

Solve for speed ratio

$$\phi_{metric} = \frac{ND}{\sqrt{H}} \frac{1}{84.58} = \frac{85.7\ (7.40)}{\sqrt{12.1}} \frac{1}{84.58} = \underline{\underline{2.16}} \longleftarrow \text{answer}$$

This noted as K_u in Table 1 and deSiervo (1977) in the American system with diameter expressed in inches from Table 1.

$$\phi_{American} = \frac{dn}{43.368(h_{ft})}\ 0.5$$

$$D = 7.4 \text{ om} \quad d = \frac{7.40}{0.3048} \text{ X } 12 = 291.3 \text{ in.}$$

$$\phi_{American} = \frac{2913\ (85.7)}{43.368(39.7^{0.5}} = \underline{\underline{1.06}} \longleftarrow \text{answer}$$

The dimensionless specific speed is computed from

$$\omega_s = \frac{N_{s\ American}}{43.5\ \sqrt{n}} = \frac{231.2}{43.5\ \sqrt{0.946}} = \underline{\underline{5.46}} \longleftarrow \text{answer}$$

Recognizing that the basic equation for dimensionless specific speed is from Table 1

$$\omega_s = \frac{\omega Q^{1/2}}{(gH)^{3/4}} = \frac{2\pi 85.7(481)^{1/2}}{60\ [(9.81)(12.1)]^{3/4}} = \underline{\underline{5.47}} \longleftarrow \text{Answer Check}$$

APPENDIX 2

SAMPLE CALCULATIONS FOR DETERMINING TURBINE DIAMETER AND TURBINE SPEED BY DIFFERENT METHODS

These sample calculations are executed to illustrate different methods of estimating preliminary values of turbine speed and turbine runner diameter. The traditional method as put forth in the U.S. Bureau of Reclamation Monograph No. 20 (1976) is compared with published results of deSiervo, the work and methodology of Lindestrom of KMW in Sweden and different approaches developed on this research project. This illustrates the variability that can be obtained. Each method and the appropriate equations require at least one empirical equation that is based on experience curves based on performance of manufactured units or from studies of model test data. Documentation as to where each empirical equation came from is presented in these sample calculations.

SAMPLE CALCULATIONS

Given: Isarwerk 3 plant as an example

H = Rated head = 4.5 m

Q = Rated discharge = 32.5 m^3/sec.

P = Rated power = 1200 kW

Other assumption

Speed to be based on the nearest possible synchronous speed using multiples of 4-pole generators and 50 Hz frequency because the Isarwerk 3 unit was manufactured for that frequency.

Required:

To make preliminary estimates of turbine speed and diameter using different methods.

Analysis and Solution

A. <u>U.S. Bureau of Reclamation Monograph No. 20 Procedure</u>

Using the Equation

$N_s = 2702\ H^{-0.5}$ from Fig. 11, p. 15 (U.S.B.R.-M20)
Note: USBR-M20 = U.S.B.R. Monograph No. 20.

determine trial N_s'

$$N_s' = 2702\ (4.5)^{-0.5} = 1273.7$$

Using the specific speed equation:

$$N_s = \frac{N\sqrt{P}}{H^{5/4}}$$ from Table 2 and p. 14; (USBR-M20)

determine a trial speed N' by solving for N in above equation

$$N' = \frac{(4.5)^{5/4}\ 1273.7}{\sqrt{1200}} = 241.0$$

Recognizing $N_p = 6000/N$

Where N_p = number of poles at 50 Hz

Then $N_p = 6000/241 = 24.9$ poles

Therefore the nearest multiple of four poles would be $N_p = 24$

Synchronous speed N = 6000/24 = 250 rpm ⟵ ANSWER

Calculate the actual N_s from

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{250\sqrt{1200}}{(4.5)^{1.25}} = 1321.3$$

Now determine speed ratio from empirical Equation

$$\phi = 0.0233\, N_s^{2/3} \text{ from p. 14 (USBR-M20)}$$

$$\phi = 0.0233\, (1321.4)^{2/3} = 2.806$$

Note, this equation is for propeller turbines

Now determine turbine diameter from Equation

$$D = \frac{84.47\,\phi\sqrt{H}}{N} \text{ from p. 14, (USBR-M20)}$$

$$D = \frac{84.47\,(2.806)\sqrt{4.5}}{250} = 2.01 \text{ m} \longleftarrow \qquad \text{ANSWER}$$

B. <u>deSiervo and deLeva Equations</u>

Using the equation

$$N_s = 2419\, H^{-0.489} \text{ from p. 52 [deSiervo and deLeva(1977)]}$$

$$N_s = 2419\, (4.5)^{-0.489} = 1159.4$$

Using the specific speed equation

$$N_s = \frac{N\sqrt{P}}{H^{5/4}}$$

determine a trial speed N' by solving for N in above equation,

$$N' = \frac{(4.5)^{1.25}\ (1159.4)}{\sqrt{1200}} = 219.4$$

Recognizing $N_p' = 6000/N$

then $N_p = 6000/219.4 = 27.4$ poles

Therefore nearest multiple of four poles would be $N_p = 28$

Synchronous speed N = 6000/28 = 214.3 rpm <— ANSWER

Calculate the actual N_s from

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{214.3\sqrt{1200}}{(4.5)^{1.25}} = 1132.7$$

Now determine speed ratio from Equation:

$$\phi = 0.79 + 1.61 \times 10^{-3}\ N_s$$ from p. 56 [deSierve & deLeva (1977)]

$$\phi = 0.79 + 1.61 \times 10^{-3}\ (1132.7) = 2.614$$

Now determine turbine diameter from Equation

$$D = \frac{84.5\ \sqrt{H}}{N}$$ from p. 14 (USBR-M20)

$$D = \frac{84.5\ (2.614)\ \sqrt{4.5}}{214.3} = 2.19 \text{ m}$$ <— ANSWER

C. KMW Graphical Solution

From the KMW nomograph reproduced as Figure 70 as taken from [Lindestrom (n.d.)]

N = 200 this really falls off the scale of the nomograph

D = less than 3

D. Special study of KMW Bulb Units Using Techniques and Regressions Developed by Kpordze - Warnick

1. Determine turbine diameter by Equation:

$$D = F(P/H) = 0.17633 \left(\frac{P}{H}\right)^{0.449}$$

$$D = 0.17633 \left(\frac{1200}{4.5}\right)^{0.449} = 2.17 \text{ m} \longleftarrow \quad \text{ANSWER}$$

Then using this value of D determine a trial value of N from Equation

$$N = F\left(\frac{\sqrt{H}}{D}\right) = 164.706 \left(\frac{\sqrt{H}}{D}\right)^{0.8876} \text{ from Table 9}$$

$$N' = 164.706 \left(\frac{\sqrt{4.5}}{2.17}\right)^{0.8876} = 161.42 \text{ rpm}$$

For synchronous speed $N_p = 6000/N = 37.2$ poles

choose 36 poles

Therefore N = 6000/36 = 166.7 rpm ⟵ ANSWER

2. Using D from above (1) and using empirical equation:

$$D = F\left(\frac{Q}{N}\right) = 4.1604 \left(\frac{Q}{N}\right)^{0.3064} \text{ from Table 9}$$

and transposing solve for N

$$N = \left(\frac{4.1604}{D}\right)^{3.264} Q$$

$$N' = \left(\frac{4.1604}{2.17}\right)^{3.264} (32.5) = 272.0 \text{ rpm}$$

For synchronous speed $N_p = 6000/N$

$N_p = 6000/272 = 22.1$ Use 24 poles

N = 6000/24 = 250 rpm ⟵ ANSWER

3. Using empirical equation for $N = F(P/H)$ solve for N and empirical equation $D = F(Q/N)$ solve for D using N from the solution of $N = F(P/H)$

Determine N from Equation:

$$N = F(P/H) = 3583.983\ (P/H)^{-0.4833} \quad \text{from Table 9}$$

$$N' = 3583.983\ \left(\frac{1200}{4.5}\right)^{-0.4833} = 240.9 \text{ rpm}$$

For synchronous speec $N_p = 6000/N$

$N_p = 6000/240.9 = 24.9$ Use 24 poles

$N = 6000/24 = 250$ rpm

Now using this N = 250 rpm determine turbine diameter D from

$$D = F(Q/N) = 4.1604\ \left(\frac{Q}{N}\right)^{0.3064}$$

$$= 4.1604\ \left(\frac{32.5}{250}\right)^{0.3064} = 2.23 \text{ m} \quad \longleftarrow \quad \text{ANSWER}$$

4. Using the more traditional approach, solve for $N_s = F(H)$, then find N from specific speed equation, then solve for $\phi = F(N_s)$, then use $D = F(\phi\frac{\sqrt{H}}{N})$ to solve for D.

Using Equation:

$$N_s = F(H) = 1553.445\ H^{-0.2918} \quad \text{from Table 9}$$

$$N_s = 1553.445\ (4.5)^{-0.2918} = 1001.6$$

$$N' = \frac{N_s\ H^{1.25}}{\sqrt{P}} = \frac{1001.6\ (4.5)^{1.25}}{\sqrt{1200}} = 189.5 \text{ rpm}$$

For synchronous speed $N_p = 6000/N$

$N_p = 6000/189.5 = 31.66$ Use 32 poles

$N = 6000/32 = 187.5$ rpm

Now find actual N_s

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{187.5\sqrt{1200}}{(4.5)^{1.25}} = 991.0$$

Using Equation:

$$\phi = F(N_s) = 0.166\ N_s^{0.3728} \quad \text{from Table 9}$$

$$\phi = 0.166\ (991.0)^{0.3728} = 2.173$$

Now solve for D using Equation

$$D = 84.47\ \phi\ \frac{H^{0.5}}{N} = \frac{84.47\ (2.173)(4.5)^{0.5}}{187.5}$$

$$D = 2.08 \text{ m} \longleftarrow \quad \text{ANSWER}$$

E. <u>Study of all Bulb Units Using Techniques and Regression Developed by Kpordze - Warnick</u>

1. Determine turbine diameter by Equation:

$$D = 0.1826\ (P/H)^{0.4462} \qquad \text{Eq. 25}$$

$$D = 0.1826\ \left(\frac{1200}{4.5}\right)^{0.4462} = 2.21 \text{ m} \longleftarrow \quad \text{ANSWER}$$

Then using this value of D determine turbine speed by Equation

$$N = F\left(\frac{\sqrt{H}}{D}\right) = 169.199\ \left(\frac{\sqrt{H}}{D}\right)^{0.926} \text{ from Eq. 30}$$

$$N' = 169.199\ \left(\frac{\sqrt{4.5}}{2.21}\right)^{0.926} = 162.8 \text{ rpm}$$

For synchronous speed N_p = 6000/N'

Therefore N_p = 6000/162.8 = 36.9 poles, Use 36 poles

$$N = 6000/36 = 166.7 \text{ rpm} \quad \longleftarrow \quad \text{ANSWER}$$

2. Using D from above (1) of 2.21 m = D and utilizing empirical equation

$$D = F\left(\frac{Q}{N}\right) = 4.181 \left(\frac{Q}{N}\right)^{0.3175} \quad \text{from Eq. 26}$$

or transposing to solve for N

$$N = \left(\frac{4.181}{D}\right)^{3.15} Q$$

$$N' = \left(\frac{4.181}{2.21}\right)^{3.15} (32.5) = 242.1 \text{ rpm}$$

For synchronous speed N_p = 6000/N.

N_p = 6000/242.1 = 24.8 poles   Use 24 poles

N = 6000/24 = 250 rpm ⟵ ANSWER

3. Using empirical Equation for N = F(P/H) solve for N and use empirical equation for D = F(Q/N) solve for D using the N from N = F(P/H) as selected to agree with a synchronous speed.

$$N = F\left(\frac{P}{H}\right) = 2152.856 \left(\frac{P}{H}\right)^{-0.4062} \quad \text{from Eq. 28}$$

$$N' = 2152.856 \left(\frac{1200}{4.5}\right)^{-0.4062} = 222.6$$

For synchronous speed N_p = 6000/N

N_p = 6000/222.6 = 26.9   Use 28 poles

N = 6000/28 = 214.3 rpm

Now using this N = 214.3 determine diameter D from Equation D = F(Q/N)

$$D = 4.181 \left(\frac{Q}{N}\right)^{0.3175} \quad \text{from Eq. 26}$$

$$D = 4.181 \left(\frac{32.5}{214.3}\right)^{0.3175} = 2.30 \text{ m} \quad \longleftarrow \quad \text{ANSWER}$$

4. Using the more traditional approach solve for $N_S = F(H)$, then find N from specific speed equation, then solve for $\phi = F(N_S)$, then use $D = F(\phi \frac{\sqrt{H}}{D})$ to solve for D.

Using Equation

$$N_s = F(H) = 1520.256\ H^{-0.2837} \text{ from Eq. 3}$$

$$N_s = 1520.256\ (4.5)^{-0.2837} = 992.2$$

$$N' = \frac{N_s\ H^{5/4}}{\sqrt{P}} = \frac{992.2\ (4.5)^{1.25}}{\sqrt{1200}} = 187.7 \text{ rpm}$$

For synchronous speed $N_p = 6000/N$

$$N_p = 6000/187.7 = 31.97 \quad \text{Use 32 poles}$$

$$N = 6000/32 = 187.5 \text{ rpm}$$

Now find actual N_S

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{187.5\sqrt{1200}}{(4.5)^{1.25}} = 991.0$$

Using Equation

$$\phi = F(N_s) = 0.0944\ N_s^{0.4605} \text{ from Eq. 19}$$

$$\phi = 0.0944\ (991.0)^{0.4605} = 2.26$$

Now solve for D using Equation

$$D = 84.47\ \phi \frac{H^{1/2}}{N} = 84.47 \frac{(2.26)(4.5)^{1/2}}{187.5} = 2.16\text{m}$$

$D = 2.16$ m $\longleftarrow$ ANSWER

F. Actual Manufactured Values of Diameter and Speed

D actual = 2.45 m

N = 157 rpm

APPENDIX 3

COMPLETE TABLE OF DATA

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
ARGENTINA								
RIO QUEQUEN	1982	-	4.15	5.5	170	1.00	425.0	N
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
OBERVOGAU	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
BISCHOFSHOFEN	1981	-	-	-	10000	-	136.4	VA
HAINBURG	1982	-	18.24	-	55800	-	100.0	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
PEOPLE'S REPUBLIC OF CHINA								
MA JI TANG	1984	ZI SHUI	6.56	310.0	18000	6.30	75.0	V

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
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
ALBAS	1965*	-	3.87	15.0	423	1.80	176.5	N
AGE	1981*	-	19.00	15.4	2608	1.50	428	N
BERGERAC	1980*	-	3.62	-	791	2.50	136	N
CAILLADE	1958*	-	3.50	5.3	154	1.12	257	N
CAPDENAC	1959*	-	6.00	15.0	751	1.80	260	N
MERCUS 1	1954*	-	3.50	9.5	283	1.65	182	N
MERCUS 2	1959*	-	3.90	9.9	318	1.40	254	N
MOTZ	1982*	-	9.40	10.0	790	1.25	395	N
ROCHEREAU	1982*	-	9.00	6.6	500	1.00	487	N
VERDUM	1957*	-	3.13	8.4	241	1.65	181	N
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

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
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	257.8	27100	5.60	100.0	V-C
STRASBOURG	1970	RHINE	14.50	219.2	29000	5.60	100.0	N
CASTET	1953	-	7.80	12.5	810	1.65	250.0	N
WADRINAU	1957	-	4.50	36.4	1480	3.05	107.0	N
SAINT-MALO	1959	-	4.80	227.0	9000	5.80	88.3	N
GERSTHRIM	1957	-	9.80	258.0	23000	5.60	107.0	N
BEAUCAIRE	1970	-	15.30	258.0	35000	6.25	93.8	N
GERVANS	1971	-	12.0	-	30000	6.52	93.8	N
AVIGNON	1973	-	10.50	350.0	30000	6.52	93.8	N
GAMBSHEIM	1974	-	13.20	-	24500	5.60	100.0	N
CHAUTAGNE	-	-	14.67	350.0	46600	6.40	107.0	N
BELLEY	-	-	14.70	350.0	46670	6.40	107.0	N
GERMANY								
PALZEM	1964	MOSELLE	3.40	50.0	1500	3.60	78.0	MA
GREVENMACHER	1962	MOSELLE	5.50	59.0	2600	3.20	120.0	EW
TRIER(TREVES)	1958	MOSELLE	5.10	95.0	4400	4.60	78.0	EW
DETZAM	1959	MOSELLE	7.00	95.0	5800	4.20	92.5	EW
WINTRICH	1963	MOSELLE	5.60	95.0	4900	4.60	83.0	EW
ZELTINGEN	1964	MOSELLE	4.00	95.0	3300	4.80	67.0	MA
ENKIRCH	1965	MOSELLE	5.10	95.0	4300	4.60	79.0	MA
NEEF(ST.ALDEGUND)	1964	MOSELLE	5.50	95.0	4000	4.60	76.0	EW
FRANKEL	1962	MOSELLE	4.10	95.0	3700	4.60	77.0	V
MUDEN	1962	MOSELLE	4.10	95.0	3600	4.60	77.0	V
LEHMEN	1966	MOSELLE	5.30	95.0	4600	4.60	85.0	V
BUCKENHOFEN	1960	ILLER	5.20	35.0	1500	2.45	166.7	EW

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
IVORY COAST								
SAN PEDRO	1982	SAN PEDRO	9.80	30.0	2600	2.05	272.7	V-C
JAPAN								
HITOKITA	1959	NATORI	12.00	12.5	1375	1.50	333.3	MI
KONAKAJIMA	1961	MABUCHI	9.20	29.0	2320	2.30	200.0	T
AKIRASHIMA	1964	TEDORI	13.70	40.0	4800	2.30	240.0	MI
OMATA	1960	WADA	13.00	30.0	3350	2.20	200.0	FF
JOGANJIGAWA(NO.1,2,3,4)	1964	JOGANJI	15.10	40.0	5340	2.47	240.0	FE
TAGUCHI	1966	HIROSE	12.40	58.2	6300	2.90	187.5	FE
KOIDE	1967	HIROSE	12.90	78.1	8800	3.40	150.0	FE
YANAGIHARA	1967	HIROSE	10.00	90.1	7850	4.00	125.0	T
HITOKITA	1959	NATORI	12.00	12.5	1375	1.50	333.0	MI
KOSHI	1959	SENDAI	8.00	22.0	1640	1.90	225.0	MI
SAIKAWA	1961	SAI	18.30	13.5	2216	1.43	450.0	FF
SHIMOAKA	1962	KITA	10.65	20.0	1840	1.84	240.0	FE
TAMAYODA 2	1964	ARA	16.80	30.0	4370	1.95	300.0	FE
MIZUKOSHI	1965	NISHIKI	12.12	12.0	1410	1.30	400.0	F/M
SEKINE	1967	HIROSE	9.50	99.0	8200	4.00	125.0	T
KUROTORI	1968	NARIHA	10.21	26.0	2310	2.10	225.0	FF
ISHII	1975	CHIKUGO	13.74	10.0	1176	1.27	450.0	FE
KUROKAWA 2	1975	SHIRO	22.70	11.13	2194	1.27	600.0	FE
IKEDA	1976	YOSHINO	10.73	62.0	5200	3.13	150.0	F/M
AKAO	1978	SHO	17.40	220.0	34000	5.10	128.6	FE
FUTAKAWA	1979	SHIZUNAI	12.00	73.0	7300	3.40	150.0	T
ARAMAKI	1966	-	9.50	108.0	8200	-	125.0	T
SAKUMA 2	1982	TENRYU	12.30	12.2	16800	4.49	125.0	FF
MONIWA	1961	-	16.3	-	1570	-	429.0	H
KAKIO	1962	-	11.9	-	860	-	500.0	H
OSAKABE	1962	-	10.35	-	540	-	514.0	H
KOREA								
NAM GANG	1972	-	8.70	93 .0	6500	3.00	199.5	J
PALDANG	1972	-	11.80	200 .0	21000	5.20	120.0	N
LUXEMBOURG								

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
FINSING	1961	-	10.60	35.0	3000	2.30	214.3	V
URSPRING	1963	LECH	8.10	52.0	3400	2.85	166.7	V
LECH 3	1963	LECH	9.20	47.5	4200	2.85	166.7	EW
SYLVENSTEIN	1960	ISAR	23.40	12.5	2500	1.46	452.0	V
IFFEZHEIM	1977	RHINE	11.70	267.5	27000	5.80	100.0	EW
LECHSTUFE 2	1968	LECH	15.20	52.3	7500	2.85	200.0	EW
LECHSTUFE 18	1973	LECH	12.80	47.5	6700	2.85	200.0	EW
LECHSTUF 23	1978	LECH	8.60	47.5	5000	2.85	187.5	EW
ISARWERK 3	1979	ISAR	4.50	32.5	1200	2.45	157.0	EW
LECHSTUFE 19	1980	LECH	8.70	47.5	4500	2.85	176.5	EW
LECHSTUFE 20	1984	LECH	9.40	47.5	4090	2.85	176.5	V
LECHSTUFE 22	-	LECH	9.77	47.5	-	2.85	176.5	V
GOTTFRIEDING	1977	ISAR	6.00	50.5	2710	2.92	135.0	V
REHLINGEN	1984	SAAR	7.6	30.0	2080	2.30	187.5	V
SCHODEN	1984	SAAR	5.70	30.0	1550	2.30	187.5	V
HUNGARY								
TISZA 2	1973	-	6.40	138.0	7200	4.30	107.0	GM
INDIA								
GANDAK	1966	-	6.10	112.0	5500	4.10	107.0	EW
KOSI	1984	-	7.70	-	5000	4.50	93.8	H
WESTERN YAMUNA	1982	-	-	-	-	-	-	
CANAL	1982	-	-	73.3	9080	3.15	187.5	FE
INDONESIA								
ANGKUP 1	1980*	-	9.0	5.70	425	0.90	659	N
HARUYAN	1980*	-	4.85	5.00	200	0.90	460	N
MEJAGUNG	1980*	-	14.87	5.10	640	0.90	802	N
WONODADI	1980*	-	3.60	8.30	235	1.25	280	N
IRAK								
MOSUL 2	-	TIGRIS	10.5	16.0	-	5.00	115.4	V
ITALY								
FIORINO NUOVO	1966	PIAVE	16.50	62.0	9000	3.00	187.5	RA
MELLEA 1	-	-	11.0	2.5	200	0.63	770	N
MELLEA 2	-	-	11.0	4.1	350	0.80	603	N

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
NORWAY								
GAMLEBRUFOSS	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
C.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
PHILLIPPINES								
MAGAT A	1984	-	3.50	13.80	381	1.50	239	N
MAGAT B	1984	-	3.50	13.80	381	1.50	239	N
MAGAT C	1984	-	2.80	11.70	253	1.50	214	N
MAGAT MATION 36	1985	-	9.96	10.28	837	1.25	400	N
TALAVERA	1983	-	14.80	-	645	-	-	N
PENARANDA	1983	-	7.80	-	323	-	-	N
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

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
ALCANADRE	1963*	-	2.49	18.30	379	-	136.0	N
SASTAGO	1969*	-	7.00	-	753	-	-	N
MENGIBAR	1974*	-	7.60	-	1700	-	-	N
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
LOVON	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	5.90	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

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
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	TW
	-	-						
UNITED KINGDOM								
AWE	1964	-	6.85	8.75	518	1.25	375	N
USA								
ROCK ISLAND	1978	COLUMBIA	12.10	481.00	54000	7.40	85.7	CL
VACEBURG	-	-	8.40	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	8390	4.85	94.7	VA
DAWSON	1982	-	5.5	96.3	4660	3.87	120.0	FE
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
USSR								
KISLAYAGUBSK	1961	-	2.50	19.10	400	3.30	92.0	N
KIEV	1966	DNIEPER	7.70	290.0	23000	6.00	85.7	KHAPKOV
KISLOGUBSKAYA	1965	-	1.28	-	400	3.30	72.0	N
KAMA	1968	-	21.0	130.0	21800	4.50	125.0	LMZ
PEREPAD	1972	-	11.20	230.0	20600	5.50	93.8	LMZ
SARATOV	1972	VOLGA	10.60	528.0	47300	7.50	75.0	LMZ
KANIEV	1972	-	8.40	240.0	18220	6.00	85.7	KHAPKOV
TCHEREPOVETZ	1967	-	15.00	175.0	21000	5.50	93.8	LMZ
YUGOSLAVIA								
IRON GATES 2	1984	DANUBE	7.40	425.0	28000	7.50	62.5	-
CAKOVEC	1979	DRAVA	18.55	250.0	42240	5.40	125.0	H & I

MANUFACTURERS:

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

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

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

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

DRAFT TUBE DIMENSIONS FOR BULB TURBINES

STATION	YEAR	DIA-METER	C	D	A_e	F	G	A_o	J	K	MANUF-ACTURER
URSTEIN	1969	4.95	73.14	13.25	73.14	-	-	73.14	19.3	13.8	V
ALTENWORTH	1976	6.70	105.68	18.85	105.68	-	-	122.72	32.0	29.0	V
ABWINDEN-AS.	1979	6.47	105.68	15.91	105.68	-	-	122.72	27.0	28.5	V
ABWINDEN-AS.	1979	6.45	105.68	15.40	105.68	46.00	-	122.72	-	-	VA
MELK	1982	7.10	124.69	17.10	105.68	49.00	-	122.72	30.0	29.5	V
GREIFFENSTEIN	1984	8.10	143.14	18.60	105.68	52.00	-	122.72	33.0	32.5	VA
KLEINMUENCHEN	1978	3.55	36.32	9.60	36.32	30.00	-	30.19	-	-	VA
MA JI TANG	1984	7.20	124.69	18.45	124.69	-	-	134.78	30.9	30.5	V
ANKKAPURHA	1983	6.20	95.03	15.35	95.03	-	5.94	36.32	-	-	TAM
VAJUKOSKI	1984	5.80	73.29	13.87	73.29	-	5.05	26.42	-	-	TAM
ARGENTAT	1957	3.20	40.72	13.30	-	-	-	50.27	-	-	V-C
ARGENTAT	1958	1.70	8.30	8.00	15.34	8.46	3.60	4.91	-	-	V-C
LA RANCE	1966	4.35	57.41	10.60	71.63	20.50	26.00	63.62	-	-	V-C
ABZAC	1958	1.23	-	2.23	-	-	1.97	3.30	-	-	V-C
MARCKOLSHEIM	1957	3.60	19.63	5.60	8.04	8.05	7.37	6.16	-	-	V-C
RABODANGES	1959	0.97	-	2.50	-	-	7.26	4.33	-	-	V-C
RHINAU	1960	3.60	19.63	5.70	25.00	-	10.00	11.20	-	-	V-C
GERSTHEIM	1967	5.15	66.48	14.75	88.36	19.70	23.20	78.54	-	-	V-C
GERSTHEIM	1968	3.60	19.63	5.60	16.00	-	11.10	12.18	-	-	V-C
STRASBOURG	1970	5.20	69.40	13.50	88.36	19.70	23.20	78.54	-	-	V-C
FANKEL	1962	3.82	69.40	12.50	69.40	-	-	78.54	17.00	21.5	V
MUDEN	1962	3.82	69.40	12.50	69.40	-	-	78.54	17.00	21.0	V
LEHMEN	1966	3.82	69.40	12.50	69.40	-	-	78.54	17.40	21.0	V
URSPRING	1963	3.30	32.37	9.30	32.37	-	-	32.37	12.13	16.0	V
SYLVENSTEIN	1960	-	-	-	-	-	-	6.16	-	8.5	V
LECHSTUFE20	1984	3.30	25.52	9.30	25.52	-	-	36.32	13.37	15.6	V
GOTTFRIEDING	1977	3.80	41.85	10.55	34.21	-	-	33.18	10.04	13.0	V
REHLINGEN	1984	2.60	19.63	7.67	19.63	-	-	19.63	12.07	10.5	V
SCHODEN	1984	2.60	19.63	7.67	19.63	-	-	19.63	12.07	10.5	V
SAN PEDRO	1982	1.73	9.08	3.95	9.08	-	9.9	12.35	-	-	V-C
GAMLEBROFOSS	1970	4.50	46.56	8.00	-	-	18.9	53.90	-	-	KMW
DOVIKFOSS	1975	7.10	103.87	14.20	-	-	23.7	98.00	-	-	KMW
SKOGSFORSEN	1959	2.40	14.19	7.50	-	-	11.00	18.00	-	-	KMW
HALLEFORS	1966	-	-	8.40	-	-	11.00	18.45	-	-	KMW
SPERLINGSHOLM	1967	-	-	7.30	-	-	10.20	12.90	-	-	KMW
PARKI	1970	5.50	69.40	11.30	-	-	22.00	79.21	-	-	KMW
VITTJARY	1974	6.60	-	13.10	-	-	-	-	-	-	KMW
BODUM	1975	6.60	-	13.10	-	-	26.20	100.10	-	-	KMW
LANDAFORS	1976	7.10	103.87	14.20	-	-	28.70	135.00	-	-	KMW

DRAFT TUBE DIMENSIONS FOR BULB TURBINES

STATION	YEAR	DIA-METER	C	D	A_e	F	G	A_o	J	K	MANUF-ACTURER
LJUSNEFORS	1976	7.10	103.87	14.20	-	-	-	-	-	-	KMW
ASELE	1981	6.80	113.10	-	-	-	27.40	110.24	-	-	KMW
SODERFORS	1979	7.10	113.10	12.70	-	-	28.90	114.49	-	-	KMW
JUVELN	1978	5.10	56.75	11.30	-	-	25.00	63.00	-	-	KMW
TORRON	1978	5.20	63.62	13.40	-	-	25.20	68.82	-	-	KMW
NAS 1	1979	6.60	113.10	13.00	-	-	27.50	99.54	-	-	KMW
AVESTA-	1982	5.00	113.10	15.50	-	-	27.40	111.30	-	-	KMW
MATFORS	-	6.45	103.87	12.50	-	-	26.60	100.47	-	-	KMW
LILLA EDET 4	1982	7.10	132.73	12.80	-	-	27.60	124.80	-	-	KMW
NAS2	1980	6.60	113.10	13.00	-	-	27.50	99.54	-	-	KMW
GRANBOFORSEN	1980	6.60	118.82	13.10	-	-	29.00	99.54	-	-	KMW
WINZNAU	1962	0.90	2.54	3.15	2.54	-	7.50	4.20	-	-	V-C
TASJO	1978	4.60	46.32	12.65	46.32	-	4.50	20.99	-	-	TAM
HOTING	1978	5.10	58.36	14.07	58.56	-	5.05	26.42	-	-	TAM
VIFORSEN	1982	5.30	58.36	13.38	58.56	-	5.05	26.42	-	-	TAM
IDAHO FALLS	1981	5.46	73.90	13.30	73.90	41.0	-	143.14	-	-	VA
PELTON REREG.	1982	5.82	76.98	14.30	76.98	40.7	-	72.39	-	-	VA

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T U B U L A R   T U R B I N E   D A T A

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)	HS	SIGMA	MANUFAC-TURER
FINLAND										
OKSAVA	1975	KALAJOKI	10.5	28.0	2610	2.40	250.0	3.65	0.61	TAM
KALLIOKOSKI	1976	PYHAJOKI	6.0	13.0	633	1.65	222.0	3.59	1.06	TAM
KALAJARVI	1976	SEINAJOK	13.5	15.0	1802	1.72	300.0	0.61	0.70	TAM
HERRFORS	1978	AHTAVANJOKI	4.0	12.0	410	1.72	167.0	2.46	1.91	TAM
FINNHOLM	1978	AHTAVANJOKI	6.0	12.0	635	1.72	222.0	3.26	1.12	TAM
PADINGINKOSKI	1979	KALAJOKI	4.0	30.0	1040	2.65	141.0	4.33	1.43	TAM
KATTILAKOSKI	1979	AHTAVANJOKI	10.5	27.0	2540	2.20	250.0	1.30	0.83	TAM
SOININKOSKI	1980	KOKEMAENJOKI	7.5	22.0	1433	2.20	200.0	3.60	0.85	TAM
HATTAR	1981	AHTAVANJOKI	6.1	20.0	1080	2.20	179.0	2.95	1.17	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	3.83	0.81	TAM
NOWAY										
BLAFALLI	-	MATREFJORDEN	27.0	36.7	8750	2.09	333.3	-5.96	0.61	V-C
FLATENFOSS	1981	NIDELV	10.0	60.0	5340	3.20	167.0	1.30	0.87	TAM
ROSTEFOSSEN	1969	-	9.5	-	1545	-	280.0	-	-	TAM
MAGO A	1984	ANDELVEN	7.2	12.0	770	1.72	214.0	4.46	0.76	TAM
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

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)	HS	SIGMA	MANUFAC-TURER
SWITZELAND										
LESSOC	1973	SARINE	20.7	16.1	2940	1.7	432.0	0.60	0.41	V-C
KALLNACH	1980	AAR	17.5	45.6	7050	2.5	250.0	-6.65	0.93	V-C
USA										
SAWMILL		ANDROSCOGGIN	5.3	16.6	760	2.0	-	-	-	ALLIS
SAWMILL		ANDROSCOGGIN	5.3	16.6	827	2.0	-	-	-	ALLIS
TRAICAO	-		7.0	-	257	-	-	-	-	ALLIS
TRUMAN	-		13.0	138.0	31500	6.5	-	-	-	ALLIS
LOWER PAINT	-		6.1	-	116	0.75	514.0	-	-	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
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
CONSOLIDATED PAPER CO.	1962	WISCONSIN	6.7	35.5	2090	2.794	150.0	-	-	ALLIS
ORILLIA WATER,L.&POWER	1964	SWIFT RAPIDS	14.3	21.0	2610	1.956	277.0	-	-	ALLIS
CITY OF NORWICH	1965	CONNECTICUT	4.7	36.0	1490	2.794	129.0	-	-	ALLIS
OZARK DAM	1965	ARKANSAS	10.7	290.0	25200	8.000	60.0	-0.40	0.97	ALLIS
WEBER FALLS	1967	OKLAHOMA	10.7	290.0	25200	8.000	60.0	-	-	ALLIS
CORNELL PROJECT	1972	WISCONSIN	11.0	107.0	10400	4.650	100.0	3.83	0.54	ALLIS
DOLBY PROJECT	1974	MAINE	14.6	33.0	4237	2.290	212.0	-	-	ALLIS
BAKER MILL	1978	MAINE	14.9	11.5	1500	1.500	306.0	1.35	0.59	ALLIS
GISBORNE DEV. PROJECT	1979	NOVA SCOTIA	19.0	22.0	3700	2.000	262.0	2.00	0.40	ALLIS
BROWN PAPER COMPANY	1979	MAINE	5.3	19.0	877	2.000	194.0	3.00	1.27	ALLIS
SALT RIVER PROJECT	1980	ARIZONA	10.6	17.0	1580	1.750	237.0	1.08	0.81	ALLIS
WOODWARD DAM	1980	CALIFORNIA	14.6	23.5	3000	2.000	213.0	1.00	0.61	ALLIS
GARVINS FALLS	1980	NEW HAMPSHIRE	9.1	42.0	3380	2.750	168.0	1.08	0.99	ALLIS
IMPERIAL IRRIGATION	1980	CALIFORNIA	6.9	34.0	2070	2.500	176.0	0.45	1.40	ALLIS

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)	HS	SIGMA	MANUFAC-TURER
WOONSOCKET FALLS	1981	RHODE ISLAND	5.9	23.0	1133	2.000	204.0	1.70	1.42	ALLIS
RILEY MILL	1981	MAINE	6.1	26.0	1390	2.250	177.0	-2.28	2.01	ALLIS
BLACKSTONE FALLS	1981	RHODE ISLAND	4.0	12.0	420	1.600	200.0	1.40	2.18	ALLIS
WELLS RIVER	1981	VERMONT	22.9	6.0	1150	1.000	605.0	-5.50	0.67	ALLIS
CITY OF STURGIS	1982	MICHIGAN	7.6	12.0	810	1.500	294.0	0.35	1.25	ALLIS
SHAWMUT	1982	MAINE	6.4	35.5	2000	2.750	160.0	1.68	1.31	ALLIS

MANUFACTURER:

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

CROSSFLOW TURBINES

NAME OF POWER STATION	DATE OF COMMIS-SIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M /S)	RATED CAPACITY PER UNIT (KW)	RUNNER DIAMETER (M)	TURBINE RUNNING SPEED (RPM)
AUSTRIA							
KRONLACHNER	1979	.	4.8	5.85	228	1.0	90.0
BELGIUM							
JOSEPH GAMBY	1970	.	4.25	3.7	124	0.8	97.0
CANADA							
GOUIN	1975	ST.MAURICE	12.5	3.0	306	0.8	180.0
RODDICKTON	1980	MARBLE	42.0	1.29	440	0.6	450.0
KINGCOME	1982	KINGCOME	147.0	0.072	84	0.4	1200.0
GRAET FALLS	.	.	16.76	235.8	35660	5.87	112.5
POINTE-DE BIOS	.	.	13.72	250.1	30950	6.23	97.3
FRANCE							
CERNAY	1981	.	8.0	6.00	377	1.0	177.0
PORTUGAL							
ALMONDA	1966	.	8.25	4.55	294	0.8	143.0
SWEDEN							
HANS-GARDAENAS	1981	.	5.8	4.33	205	0.8	123.0
BOSAGENS	1980	.	6.95	7.00	396	1.0	113.0
SWITZERLAND							
NIEDERGLATT	1965	.	9.33	4.8	353	0.20	152.0

CROSSFLOW TURBINES

NAME OF POWER STATION	DATE OF COMMIS-SIONING	NAME OF RIVER	RATED HEAD (M)	RATED FLOW (M /S)	RATED CAPACITY PER UNIT (KW)	RUNNER DIAMETER (M)	TURBINE RUNNING SPEED (RPM)
USA							
GOODYEAR LAKE 1	1980	.	9.8	8.5	654	1.0	131.5
GOODYEAR LAKE 2	1980	.	9.8	11.5	885	1.25	103.0
CORNEL 1	1981	FALL CREEK	35.0	2.5	712	0.8	325.0
CORNEL 2	1981	FALL CREEK	35.0	3.5	997	1.0	261.0
BRADFORD	1982	WAITS	21.64	6.0	1057	1.0	195.0
BRADFORD	1982	WAITS	21.64	3.0	528	0.8	244.0
GEORGETOWN	1983	CANAL	57.00	0.974	708	0.6	618.0
SPOTTED BEAR	1982	.	37.19	0.26	52	0.3	800.0
YUGOSLAVIA							
HE SOTESKA	1975	.	4.7	6.3	241	1.0	84.0

STANDARD TUBULAR TURBINE WATER PASSAGE DIMENSIONS

MANUFACTURER	DIAMETER	AE	L1	L	M	AO
NEYPICERYPIC	0.45	0.554	1.72	4.48	2.76	0.64
NEYPICERYPIC	0.63	1.039	2.10	5.93	3.83	1.254
NEYPIC	0.83	1.839	2.70	7.11	4.41	2.020
NEYPIC	1.00	2.630	2.90	8.20	5.30	3.170
NEYPIC	1.25	4.600	3.20	9.66	6.46	4.930
NEYPIC	1.50	5.515	3.70	11.23	7.53	7.08
NEYPIC	1.80	7.793	4.06	12.94	8.88	10.24
VOITH	0.50	1.91	2.63	8.53	5.90	-
VOITH	0.70	1.91	2.63	8.53	5.90	-
VOITH	0.90	1.91	2.63	8.53	5.90	-
VOITH	1.15	1.91	2.63	8.53	5.90	-
VOITH	1.40	1.91	2.63	8.53	5.90	-
VOITH	1.70	1.91	2.63	8.53	5.90	-
VOITH	2.00	1.91	2.63	8.53	5.90	-
VOITH	2.25	1.91	2.63	8.53	5.90	-
VOITH	2.50	1.91	2.63	8.53	5.90	-
VOITH	2.75	1.91	2.63	8.53	5.90	-
VOITH	3.00	1.91	2.63	8.53	5.90	-
ALLIS	0.75	1.61	2.50	-	-	3.00
ALLIS	1.00	1.47	2.30	-	-	3.00
ALLIS	1.25	1.41	2.20	-	-	3.00
ALLIS	1.50	1.37	2.20	-	-	3.00
ALLIS	1.75	1.35	2.20	-	-	3.00
ALLIS	2.00	1.33	2.00	-	-	3.00
ALLIS	2.25	1.31	2.00	-	-	3.00
ALLIS	2.50	1.29	2.00	-	-	3.00
ALLIS	2.75	1.27	2.00	-	-	3.00
ALLIS	3.00	1.17	2.00	-	-	3.00
TAMPELLA	1.40	6.45	1.50	-	8.25	9.00
TAMPELLA	1.65	9.18	1.80	-	9.75	12.96
TAMPELLA	1.90	12.30	2.05	-	11.25	16.81
TAMPELLA	2.15	15.18	2.30	-	12.70	21.16
TAMPELLA	2.40	19.24	2.60	-	14.20	27.04
TAMPELLA	2.65	16.80	2.50	-	11.20	25.00
TAMPELLA	2.90	20.01	2.80	-	12.20	30.25
TAMPELLA	3.20	24.00	3.10	-	13.50	36.00
TAMPELLA	0.90	3.20	2.40	-	5.30	4.00

STANDARD TUBULAR TURBINE WATER PASSAGE DIMENSIONS

MANUFACTURER	DIAM-METER	AE	L1	L	M	AC
TAMPELLA	1.15	5.00	3.05	-	6.80	6.25
TAMPELLA	1.40	7.50	3.70	-	8.25	9.00
TAMPELLA	1.65	10.44	4.35	-	9.75	12.96
TAMPELLA	1.90	13.74	5.05	-	11.25	16.81
TAMPELLA	2.15	17.48	5.70	-	12.70	21.16
TAMPELLA	2.40	21.84	6.35	-	11.20	27.04
TAMPELLA	2.65	26.97	3.80	-	11.20	25.00
TAMPELLA	2.90	32.13	4.20	-	12.20	30.25
TAMPELLA	3.20	38.64	4.60	-	13.50	36.00

APPENDIX 4

COMPUTER PROGRAMS

```
CMS FI IN DISK BULB4  DATA A (PERM;

* SAS PROGRAM FOR COMPUTING TURBINE CONSTANTS OF BULB TYPE UNITS;
* THE DATA OF THE BULB UNITS ARE IN A FILE NAMED BULB4;

DATA KOJO.NS;
     INFILE IN;
     LENGTH STATION $ 20;
     INPUT STATION &$  YEAR  HEAD FLOW POWER  DIAM SPEED MANUF &$  B
           C  D  E  F  G  H  J   K;

      PI    = 3.14159265;
      W     = (2.0*PI*SPEED)/(60.0);
     N11    = (SPEED*DIAM)/SQRT(HEAD);
     Q11    = FLOW/((DIAM**2)*SQRT(HEAD));
     P11    = POWER/((DIAM**2)*(HEAD**1.5));
     NS     = (SPEED*SQRT(POWER))/(HEAD**1.25);
     WS     = W*SQRT(FLOW)/((9.81*HEAD)**0.75);
     QCN    = FLOW/SPEED;
     POH    = POWER/HEAD;
     EFF    = POWER/(9.81*FLOW*HEAD);
     PHI    = (PI/(60.0*SQRT(2.0*9.81)))*N11;
     PHIFUN =  (PHI*SQRT(HEAD))/SPEED;
     IF NS =. THEN DELETE;
     LN11   = LOG10(N11);
     LQ11   = LOG10(Q11);
     LP11   = LOG10(P11);
     LNS    = LOG10(NS);
     LWS    = LOG10(WS);
     LQON   = LOG10(QON);
     LPOH   = LOG10(POH);
     LDIAM  = LOG10(DIAM);
     LHEAD  = LOG10(HEAD);
     LEFF   = LOG10(EFF);
     LPOW   = LOG10(POWER);
     LPHI   = LOG10(PHI);
     LFLOW  = LOG10(FLOW);
     LPHIFUN = LOG10(PHIFUN);
```

```
* THE NOTATIONS BELOW REFER TO TURBINE CIVIL WORKS DIMENTIONS;

     FPG    = (F+G);
     DPG    = (D + G);
     VEL    = (FLOW/E);
     DOE    = (D/E);
     LFPG   = LOG10(FPG);
     LDPG   = LOG10(DPG);
     LVEL   = LOG10(VEL);
     LB     = LOG10(B);
     LC     = LOG10(C);
     LD     = LOG10(D);
     LE     = LOG10(E);
     LF     = LOG10(F);
     LG     = LOG10(G);
     LH     = LOG10(H);
     LJ     = LOG10(J);
     LK     = LOG10(K);
     LDOE   = LOG10(DOE);
     KEEP STATION YEAR HEAD FLOW POWER DIAM  SPEED  MANUF B  C  D  E  F
          G  H  J  K  FPG DPG  VEL N11 Q11 P11 NS WS  QON POH  DOE PHI
          EFF PHIFUN LN11 LQ11 LP11 LNS  LWS  LQON  LPOH  LHEAD  LPOW
           LDIAM  LEFF LFPG LDPG  LVEL LB LC LD LE LF LG LH LJ LK
          LFLOW LDOE  LPHI LPHIFUN;
PROC PRINT DATA=KOJC.NS  PAGE;
     VAR STATION YEAR HEAD FLOW POWER DIAM  SPEED  MANUF B C D E F G H
        J  K  N11 Q11 P11 NS WS  QON POH  EFF FPG DPG  VEL  DOE  PHI
        PHIFUN LN11 LQ11 LP11 LNS LWS  LQON LPOH  LPOW LDIAM LHEAD LEFF
        LFPG LDPG  LVEL LB LC LD LE LF LG LH LJ LK LDOE LFLOW
        LFLOW LPHI LPHIFUN;
```

SAMPLE COMPUTER PROGRAM FOR COMPUTING REGRESSION RELATIONS

```
CMS FI KOJO DISK A A A;
DATA INSET;
     SET KOJO.NS;
     IF NS=. THEN DELETE;
     IF YEAR <= 1965 THEN GROUP =65;
     ELSE IF YEAR >1965 THEN GROUP =84;
PROC SORT; BY GROUP;
PROC GLM DATA=INSET; BY GROUP; MODEL LNS=LQ11;
 OUTPUT OUT=B.NEW01 (KEEP=GROUP NS LNS PLNS Q11 LQ11) P=PLNS;
 PROC PRINT; VAR NS  LNS PLNS Q11 LQ11; BY GROUP;
PROC GLM DATA =INSET; BY GROUP; MODEL LNS = LP11;
  OUTPUT OUT=B.NEW02 (KEEP=GROUP NS LNS PLNS P11 LP11) P=PLNS;
  PROC PRINT; VAR NS LNS PLNS P11 LP11 ; BY GROUP;
PROC GLM DATA=INSET; BY GROUP; MODEL LP11=LQ11;
  OUTPUT OUT=B.NEW03 (KEEP=GROUP P11 LP11 PLP11 Q11 LQ11) P=PLP11;
  PROC PRINT; VAR P11 LP11  PLP11 Q11 LQ11;  BY GROUP;
PROC GLM DATA=INSET; BY GROUP; MODEL LNS= LN11;
  OUTPUT OUT=B.NEW04  (KEEP=GROUP NS LNS PLNS N11 LN11) P=PLNS;
  PROC PRINT; VAR  NS LNS PLNS N11 LN11; BY GROUP;
PROC GLM DATA=INSET; BY GROUP; MODEL LPHI= LP11;
  OUTPUT OUT=B.NEW05  (KEEP=GROUP PHI LPHI PLPHI P11 LP11)  P=PLPHI;
  PROC PRINT; VAR PHI  LPHI PLPHI P11 LP11; BY GROUP;
PROC GLM DATA=INSET; BY GROUP; MODEL LPHI = LNS;
  OUTPUT OUT=B.NEW06  (KEEP=GROUP PHI LPHI PLPHI NS LNS)  P=PLPHI;
  PROC PRINT; VAR  PHI LPHI PLPHI NS  LNS;  BY GROUP;
PROC GLM DATA=INSET; BY GROUP; MODEL LDIAM = LPOH;
  OUTPUT OUT=B.NEW07  (KEEP=GROUP DIAM LDIAM PLDIAM POH LPOH) P=PLDIAM;
  PROC PRINT; VAR DIAM  LDIAM PLDIAM POH LPOH; BY GROUP;
PROC GLM DATA=INSET; BY GROUP; MODEL LDIAM = LPHIFUN;
  OUTPUT OUT=B.NEW08 (KEEP=GROUP DIAM LDIAM PLDIAM PHIFUN LPHIFUN)
      P=PLDIAM;
 PROC PRINT; VAR DIAM  LDIAM PLDIAM PHIFUN LPHIFUN; BY GROUP;
```

SAMPLE SAS GRAGH PROGRAM FOR PLOTTING GRAPHS OF REGRESSION RELATIONS

```
CMS FI B DISK A A A;
DATA INSET;
     SET TUBE.NEW01; SET TUBE.NEW02; SET TUBE.NEW03; SET TUBE.NEW04;
GOPTIONS  DEV=TEK4662;
  PROC GPLOT;
  PLOT LAE*LDIAM;
  SYMBOL1 I=RL  V=:  L=1;
  SYMBOL2 I=RL  V=PLUS L=2;
TITLE1;
FOOTNOTE .H=5 FIGURE 98.LOG OF ENTRANCE AREA  VERSUS LOG OF RUNNER DIAM
METER FOR STANDARD TUBE TURBINE;
  PROC GPLOT;
  PLOT LAO*LDIAM;
  SYMBOL1 I=RL  V=:  L=1;
  SYMBOL2 I=RL  V=PLUS L=2;
TITLE1;
FOOTNOTE .H=5 FIGURE 99. LOG OF EXIT AREA  VERSUS LOG OF RUNNER DIAMETER
FOR STANDARD TUBE TURBINE;
  PROC GPLOT;
  PLOT LL1*LDIAM;
  SYMBOL1 I=RL  V=:  L=1;
  SYMBOL2 I=RL  V=PLUS L=2;
TITLE1;
FOOTNOTE .H=5 FIGURE 100. LOG OF L1 VERSUS LOG OF RUNNER DIAMETERFOR ST
ANDARD TUBULAR TURBINE;
PROC GPLOT;
  PLOT LM*LDIAM;
  SYMBOL1 I=RL  V=:  L=1;
  SYMBOL2 I=RL  V=PLUS L=2;
TITLE1;
FOOTNOTE .H=5 FIGURE 101. LOG OF M VERSUS LOG OF RUNNER DIAMETER FOR STA
NDARD TUBULAR TURBINE;
```

APPENDIX 5

LIST OF TURBINE MANUFACTURERS

LIST OF TURBINE MANUFACTURERS

	Manufacturer Name	Address	Phone Contact	Contact Person	Type of Units
1.	Ateliers Bouvier	53 rue Pierre-Semard 3800 Grenoble (France)	(76) 96.63.36		P, F, K, T
2.	Allis Chalmers	P.O. Box 712 York. PA 17405 (USA)	(717)792-3511	Helmut Wirshal Selim Chacour	P, F, K, B, T
3.	Barber Hydraulic Turbine, Ltd.	Barber Point Box 340 Port Colborne. Ontario. L3K 5W1 Canada	(416)834-9303	M. R. Wilson	P, F
4.	Canyon Industries	6342 Mosquito Lake Road	(206)592-5552	Don New	P
5.	Dependable Turbines. Ltd	#7-3005 Murray St. Port Moody, B.C. V3H1X3 (Canada)	(604)461-3121	Robert Prior	P, F, K, Tu
6.	Escher Wyss, Ltd	CH-8023 Zirich, Switzerland (Swiss)	(01) 44.44.51	Dimtri Foca	P, F, K, T
		Sulzer Bros. Inc. 200 Park Ave. New York, NY 10017 (USA)	(212)949-0999		
7.	General Electric	Installation & Service Engineering Division-Small Hydro Operation One River Road Schenectady, N.Y. 12345	(518)385-7097 (480)974-4729	D.W. Lyke P.O. Box 6440 Salt Lake City, UT 84106	P, F, T
8.	Gilbert Gilkes & Gordon, Ltd	Kendal Cumbria LA9 7BZ England	(0589)20028	O.S. Shears	P, F, T, Tu
		Gilkes Pumps Inc. P.O. Box 628 Seabrook, TX 77586 (USA)	(713)474-3016	Alan S. Fife	P, F, T
9.	Hitachi, Ltd.	6-2 Otemachi, Chiyoda-ku Tokyo 100 (Japan)	(03)270-2111	M. Suzuki	P, F, K, T
10.	Hydro-Watt Systems	146 Siglono Road Coos Bay, OR 97420 (USA)		Mert. J. Junking	P, C
11.	Independent Power Developers, Inc.	Route 3, Box 174H Sandpoint, ID 83864 (USA)	(208)263-2166	William Delp Charles Green	P, C
12.	AB Karlstads Mekaniska Werkstad KMW or KaMeWa	Fack S-681 01 Kristinehamn (Sweden)	0550/15200	Hans G. Hansson Lars-Erik Lindestrom	P, F, K, T
13.	Kraerner Brug A/S	Kvaernerveien 10 Oslo 1, (Norway)	(472)676970	James Victory	P, F, K, T
			(212)752-7310	Kvaerner Moss, Inc. 31st Floor, 800 Third Ave. New York, N.Y. 10022	
14.	James Leffel & Co.	426 East St. Springfield, Ohio 45501 (USA)	(513)323-6431	Kim Brockl Kenneth W. Berchak	P, F, T
15.	Leroy Somer	Boulevard Marcellin-Leroy B.P.119-16004 Angouleme (France)	003345.62.41.11		
		NEEDS New England Energy Development Systems, Inc. 109 Main St. Amherst, MA 010002 (USA)	(413)256-8466	Michael Pill	T
16.	Little Spokane Hydroelectric	P.O. Box 82 Chattaroy, WA 99003 (USA)	(509)238-6810	Mike Johnson	P, T

LIST OF TURBINE MANUFACTURERS (continued)

Manufacturer Name	Address	Phone Contact	Contact Person	Type of Units
17. Mitsubishi Heavy Industries, Ltd.	5-1 Marunouchi 2-chome Chiyoda-ku Tokyo (Japan)	Tokyo 212-3111 (415)981-1910	Kenji Fukumasu Billy M. Tanaka	F, D
18. Neyrpic	Groupe Creusot-Loire B.P. 75 Centre de Tri 38041 Grenoble Cedex (France)	(76)96.48.30	Lucien Megnint	
	GE/Neypic 969 High Ridge Road Box 3834 Stanford, CT 06905 (USA)	(203)322-3887	Michael Guer	P, F, K, B, T
19. Obermeyer Hydraulic Turbins, Ltd	10 Front Street Collinsville, CT 06022 (USA)	(203)693-4292		P, F, B, T, C
20. Ossberger-Turbinenfabrik	D-8832 Weissenburg/Bay Pastfach 425 Bayern (West Germany)	0 91 41/40 91		
	F.W.E. Stapenhorst, Inc. 285 LaBrosse Ave. Pointe Claire, Quebec H9R 1A3 (Canada)	(514) 695-2044	F.W.E. Stapenhorst	
21. Small Hydroelectric Systems	5141 Wickersham Acme, WA 98220 (USA)	(206)595-2312	William Kitching	P
22. Tampella	Engineering Division SF-33100 Tampere 10 (Finland)	(931)-32 400	Georg von Graeveniyz	P, F, K, B, T
23. Toshiba	Power Apparatus Export 1-6 Uchisaiwai-cho Chyoda-ku, Tokyo 100 (Japan)		Hideki Yamada	
24. Vevey Engineering Works, Ltd	1800 Vevy (Switzerland)	(021) 51 0000 51	J. P. Kaufmann	P, F, K, B, T
25. J.M. Voith GmbH	P.O. Box 1940 D7920 Heidenheim (West Germany)	(07321)32.25.61	Peter Ulith Franz Wolfram	P, F, K, B, T

B = Bulb turbine
C = Cross-flow turbine
F = Francis turbine
K = Kaplan turbine

P = Pelton turbine
T = Tubular turbine
Tu = Turgo turbine