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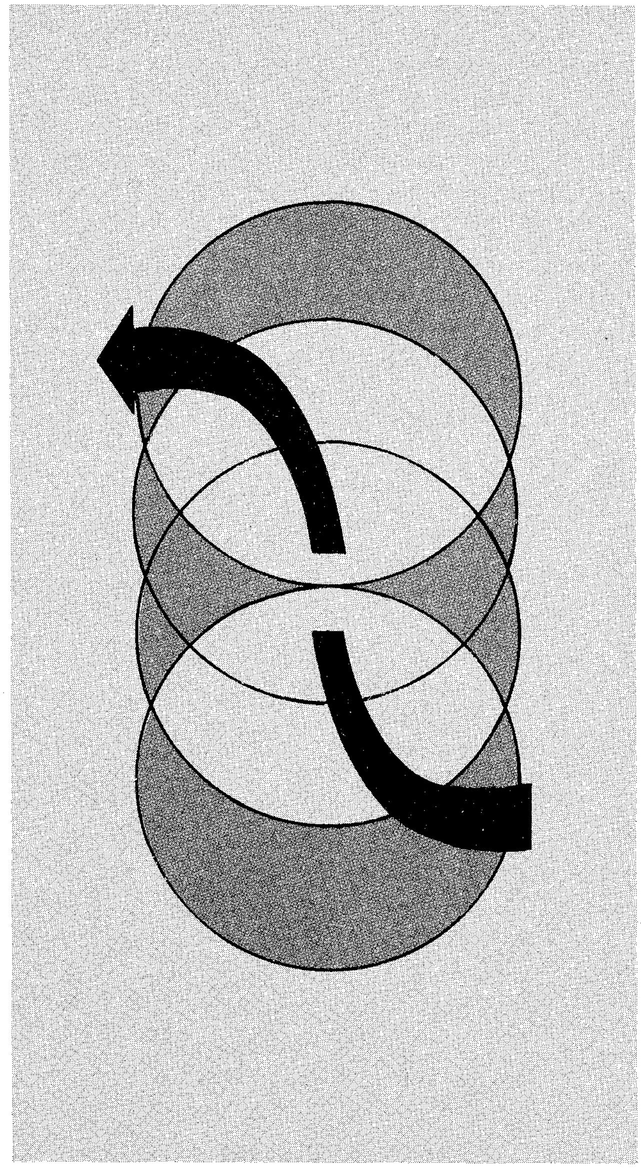

Selecting Large Pumping Units

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This monograph provides guidelines for designers or estimators who plan or lay out large-capacity pumping plants. Empirical charts and curves and analytical methods are presented to facilitate selection of the best type of pump and to estimate pump performance characteristics, submergence, dimensions, and mass. An example is given starting with basic head and capacity requirements, and working through selection of pump type and estimation of speed, specific speed, submergence, dimensions, pump mass, and power requirements. Included are illustrations of typical pumping plants.

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

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Preface

The objective of this monograph is to provide guidelines for selecting the type of pump to best meet large-capacity pumping requirements and for estimating the performance characteristics, required submergence, dimensions, and mass of the pump. The guidelines and data presented are based on both Bureau of Reclamation experience and basic theory and from recommendations in literature cited in the bibliography. The results should be sufficiently accurate for initial plant layout and cost estimation.

This monograph was prepared by William H. Duncan, Jr., mechanical engineer, and Carlos G. Bates, Head, Hydraulic Machinery Section, Mechanical Branch, Division of Design, Engineering and Research Center, Denver. Richard N. Walters made a substantial contribution to the technical presentation.

Letter Symbols and Quantities

Symbol	Quantity	Metric unit	U.S. customary unit
$d_{(i)}$	Spiral case diameter at (*i*)	mm	ft
D_1	Discharge diameter of impeller	mm	ft
D_3	Inlet diameter of impeller	mm	ft
f	Frequency	Hz	Hz
g	Gravitation constant (acceleration)	m/s^2	ft/s^2
h	Pump best efficiency head (design head)	m	ft
H	Head produced by pump	m	ft
H_a	Atmospheric pressure (head)	m	ft
H_s	Suction head	m	ft
H_L	Head loss (suction side)	m	ft
H_v	Vapor pressure head of water	m	ft
K_u	Speed constant		
K_3	Experimental design constant		
n	Rotational speed	r/min	r/min
n'	Trial rotational speed	r/min	r/min
n_s	Pump specific speed	$\frac{(\text{r/min})\sqrt{\text{m}^3/\text{s}}}{\text{m}^{0.75}}$	$\frac{(\text{r/min})\sqrt{\text{gal/min}}}{(\text{ft})^{0.75}}$
n_s'	Trial pump specific speed	$\frac{(\text{r/min})\sqrt{\text{m}^3/\text{s}}}{\text{m}^{0.75}}$	$\frac{(\text{r/min})\sqrt{\text{gal/min}}}{(\text{ft})^{0.75}}$
NPSH	Net positive suction head	m	ft
Q	Capacity (discharge)	m^3/s	ft^3/s or gal/min
S	Suction specific speed	$\frac{(\text{r/min})\sqrt{\text{m}^3/\text{s}}}{\text{m}^{0.75}}$	$\frac{(\text{r/min})\sqrt{\text{gal/min}}}{(\text{ft})^{0.75}}$
V	Velocity of water	m/s	ft/s
P	Power	watt	horsepower
α	Speed ratio factor		
σ	Cavitation coefficient (Thoma-sigma)		
η	Pump design efficiency	percent	percent

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Introduction

This monograph covers pumping unit capacities ranging from 3 to 280 m^3/s (100 to 10 000 ft^3/s). Units smaller than 3 m^3/s usually can be found in manufacturers' catalogs wherein design and estimating data are readily available. Presently, the largest pumps the Bureau operates are rated 62.3 m^3/s at 38 m (2200 ft^3/s at 125 ft) of total head, and it is not foreseen that USBR will require larger units. However, pump-turbines have been built for larger capacities; the USBR is considering pump-turbines of 710 m^3/s (25 000 ft^3/s) capacity.

In selecting the number and size of units to perform required duties, consideration must be given to reliability, flexibility, and cost.

Whereas it may be a wise decision to select only one unit for a powerplant supplying power to an interconnected transmission system, it could be a very poor selection to have only one unit if a water supply was entirely dependent on uninterrupted pumping capability. Thus, more units would be expected in a pumping system than in a power system. The time scheduled for maintenance and the effects of an unscheduled outage of the largest unit should be considered.

Standard designs and identical hydraulic units are desirable from an engineering and maintenance standpoint. However, the units should be selected to match variations in head or capacity without causing excessive loss in efficiency and unusual wear problems. The water to be pumped should be analyzed and pump materials selected accordingly to resist corrosion. Priming equipment usually is avoided by setting the impeller inlet edge below minimum water surface elevation and/or providing adequate positive suction head for water to fill the pump case.

Pumps are classified by distinguishing features such as:

- Impeller characteristics (axial flow, mixed flow, radial flow, open, semiopen or enclosed, single suction or double suction, etc.),
- Pump casing design (spiral, single volute, double volute or diffuser, turbine, circular, etc.),
- Orientation of pump shaft axis (vertical, inclined or horizontal),
- Intake design (wet pit, dry pit), and
- Number of stages.

Figure 1 is a general guide for selecting the type of pump best suited to meet various head and capacity requirements. However, in selecting the type pump best suited to a particular situation, economics of plant construction, efficiency of the units, and operation and maintenance costs should be considered.

Pump and motor dimensions and costs can be minimized by using high rotational speeds. However, in providing optimum performance at high rotational speeds, a pump will require deep submergence, possibly leading to increased plant construction costs. Likewise, capital expenditures to increase unit efficiency by using a diffusion casing, enlarging flow passages, or other means should be compared with the savings in power costs during the life of the project.

To select a pump and prepare preliminary designs, operational requirements must be analyzed and estimates made of rotational speed, submergence requirements, pump dimensions, pump mass, efficiency, and power requirements.

Capacity

A plant serving a distribution canal or pipeline obviously requires more regulating capability than a plant pumping from one reservoir to another or to a feeder canal. The former may require a number of units or even two or more sizes of units to meet demand. For small plants, using catalog-size pumps, a common selection is:

- Two units at one-third plant capacity,
- One unit at one-sixth, and
- Two units at one-twelfth plant capacity.

This combination provides flow increments of one-twelfth plant capacity while only one-third capacity is lost when the largest unit is out of service. For large plants with specially designed units, variable-pitch pumps in axial and mixed-flow designs may be economical. Such units can deliver from 50 to 100 percent of maximum capacity at either a variable or constant head and operate with good efficiency. Thus, a plant containing four fixed-pitch units and two variable-pitch units can deliver any capacity from one-twelfth to maximum capacity with good efficiency. The cost of variable-pitch pumps is about 30 percent higher than fixed-pitch pumps. Offsetting this cost, a fixed-pitch pump requires a larger—consequently more expensive—motor to accommodate overcapacity in the pump design and variation in head.

Other methods for obtaining flexibility in discharge rate are: multispeed, variable speed, throttling, and bypassing. These methods have not proved economical for high capacity irrigation pumping.

Head

This monograph discusses *best efficiency* heads ranging from 3 to 300 m (10 to 1000 ft). As illustrated in figure 1, the approximate head range of a single stage for three types of pumps (classified by impeller design) is:

Head range		Pump flow type	Customary name
meters	feet		
3-9	10-30	Axial	These are referred to as *propeller pumps* from the design in the impeller and the lifting action of the blades on the liquid.
9-18	30-60	Mixed	Francis-style double-curvature vanes usually are used in the impeller design of these pumps commonly called *Francis pumps*.
18-300	60-1000	Radial	Since centrifugal forces define the principal action of these pumps, they often are referred to as *centrifugal pumps*.

Note that all three types of pumps considered are often classified in the category of centrifugal pumps because of the rotary action of the impellers.

Mixed-flow, variable-pitch pumps have been used for heads up to 76 m (250 ft). Single-stage pumps are desirable for reasons of lower cost and simplicity. However, multistaging is applicable to improve efficiency, to obtain a steeper head-discharge curve, or to reduce *required* net positive suction head. Single-stage pump-turbines are being built for 610 m (2000 ft) of head or more. At the A. D. Edmonston Pumping Plant on the California Aqueduct [1][1], the pumps are four-stage, for a total design head of 600 m (1970 ft), resulting in an optimum pump specific speed considering efficiency and submergence requirements.

The head range that a pump must operate within is an important consideration. For a canal relift plant, the head may be nearly constant when individual discharge lines are used. The head will vary more if several pumps are manifolded to a single discharge line. The San Luis Pumping/Generating Plant, California [2], is notable as it has a head range from 30 to 100 m (98 to 330 ft) for filling a large reservoir. Two speeds are used to satisfy the head range. At Snake Creek Pumping Plant, North Dakota, the requirement is to pump from 0 to 23 m (0 to 75 ft) of static head. Interchangeable bowls and impellers fulfill the specified performance.

Variation in head requires a deeper submergence, especially when the variation from the design head is to lower heads. When a unit is required to pump from a storage reservoir to a canal, the head will vary; but the minimum head will occur at maximum suction head (full reservoir)—which is helpful.

[1] Numbers in brackets refer to items in the bibliography.

USBR practice is to add a small percentage to pump design capacity to allow for wear prior to scheduled overhauls (fig. 2).

Pump Specific Speed

The specific speed n_s *of a pump is:*

$$n_s = \frac{n\sqrt{Q}}{h^{0.75}}$$

where:

n = rotational speed, r/min,
h = best efficiency head developed, m (ft), and
Q = best efficiency discharge, m^3/s (gal/min).

Pump specific speed is defined as the rotational speed at which a given pump or geometrically and hydraulically similar pump discharges 1 m^3/s of discharge under 1 m of head (1 gal/min at 1 ft of head) while operating at the best (peak) efficiency point. The pump specific speed characterizes the type and shape of the impeller and is used to predict other important pump characteristics, dimensions, and mass. To obtain an approximate value of n_s in units of r/min, m^3/s, and m, multiply U.S. customary n_s (r/min, gal/min, and ft) times 0.019 36.

For double suction pumps, it is USBR practice to use one-half the capacity ($Q/2$) of the pump to calculate n_s and S. Thus, an identical specific speed versus head graph applies to both single and double-suction pumps.

The range of pump specific speed can be categorized:

- High specific speeds greater than 155 (8000) usually indicate an axial-flow-type impeller.
- Low specific speeds of 87 (4500) or less indicate the radial-flow-type impeller.
- Medium values of specific speed generally indicate a mixed-flow-type impeller.

There are no definite limits to define the operating regimes for the three types of impellers. However, experience has shown that an axial-flow-type impeller cannot be used efficiently for high heads and is seldom used for heads greater than 9 m (30 ft) per stage. The radial-flow type is the most efficient at high heads and has been used for heads up to 610 m (2000 ft) per stage. Figure 1 shows the usual operating regimes for the different types of pumps. Figure 3 presents curves of expected pump efficiency versus specific speed for various pump capacities.

Net Positive Suction Head

NPSH (net positive suction head) is defined as the total suction head above vapor pressure at the *highest point* of the impeller inlet edge. The NPSH *available* for the pump, at a given site, is calculated from the equation:

$$\text{NPSH} = H_a + H_s - H_v - H_L$$

where:

H_a = atmospheric pressure head,
H_s = suction head,
H_v = water vapor pressure head, and
H_L = suction side head losses.

Figure 4 illustrates NPSH. A particular pump design requires a certain minimum NPSH head (required) to prevent cavitation. The *available* NPSH at the plant site must be equal to or greater than the *required* NPSH. Operation with less than the *required* NPSH will cause the head and efficiency to drop, and destructive cavitation will occur on the impeller blades.

Figure 5 illustrates the upper limit of pump specific speed versus design head for various conditions of H_s. The figure is based on data from pumps and pump-turbines which are operational. A higher speed and a wide range in head generally necessitate a higher value of *available* NPSH.

Suction Specific Speed

Suction specific speed S is defined as:

$$S = \frac{n\sqrt{Q}}{(\text{NPSH})^{0.75}} = n_s \left(\frac{h}{\text{NPSH}}\right)^{0.75}$$

where:

h = best efficiency head developed,
n = rotational speed,
Q = best efficiency discharge, and
NPSH = net positive suction head at the site, or absolute suction head less vapor pressure head.

The parameter S is used for pumps to describe the suction characteristics of an impeller. An S value within 153 to 155 (7900 to 8000) has been found to produce the best performance. If an impeller is designed for a higher suction specific speed to reduce the *required* NPSH, the blade entrance-angle must be flattened. This results in lower efficiency and a larger impeller-eye diameter for a given capacity [3]. Except for very special cases, higher suction specific speeds should not be considered.

Thoma Cavitation Coefficient

The parameter used for defining the operating condition, with respect to cavitation, is commonly known as the Thoma cavitation coefficient and is represented by the Greek letter sigma σ. It is the ratio of NPSH to total pump head or:

$$\sigma = \frac{\text{NPSH}}{H}$$

For large pumping units, USBR defines critical sigma as the sigma value at which cavitation causes a 1-percent loss in head. Critical sigma is determined by model tests for each impeller design. With the critical sigma values known, the required pump submergence can be determined with respect to suction water level for any environment. Critical sigma and *required* NPSH increase rapidly with discharge above the best efficiency point. Bureau experience has shown that if a pump is operated at a head considerably below *best* efficiency head (high discharge), no amount of *available* NPSH will prevent cavitation.

Figure 6 shows the recommended sigma value versus pump specific speed. Figure 7 illustrates typical variations of critical sigma versus discharge for different pump specific speeds. Note that the curves on figures 5 and 6 are based on a constant suction specific speed S of approximately 154 (7950). A similar curve to figure 5 in the *Hydraulic Institute Standards* [4] exhibits a varying suction specific speed. Figure 5 generally indicates a lower pump specific speed limit per given design head and suction head up to a specific speed of approximately 70 (3700), and higher limits above 70 than recommended in the *Hydraulic Institute Standards.*

Affinity Laws and Hydraulic Similarity

For a given centrifugal pump of constant impeller discharge diameter D, the performance variables of capacity Q, head H, power P, and rotational speed n, at points of equal efficiency η, vary according to the relations:

- Capacity is directly proportional to speed,
- Head is proportional to the square of the speed, and
- Power is proportional to the cube of the speed.

For mixed-flow and radial-flow pumps, when speed is held constant and the impeller discharge diameter is varied slightly, the relations between points of equal efficiency can be expressed as:

- Capacity is directly proportional to diameter,
- Head is proportional to the square of the diameter, and
- Power is proportional to the cube of the diameter.

If both n and D are varied, both relations apply simultaneously.

The affinity laws which follow from these relations can be applied to calculate changes in pump performance due to varied rotational speed or impeller discharge diameter of a given pump.

For constant diameter:	For constant speed:
$\frac{Q_a}{Q_b} = \frac{n_a}{n_b}$	$\frac{Q_a}{Q_b} = \frac{D_a}{D_b}$
$\frac{H_a}{H_b} = \left(\frac{n_a}{n_b}\right)^2$	$\frac{H_a}{H_b} = \left(\frac{D_a}{D_b}\right)^2$
$\frac{P_a}{P_b} = \left(\frac{n_a}{n_b}\right)^3$	$\frac{P_a}{P_b} = \left(\frac{D_a}{D_b}\right)^3$

where:
a and b denote the same pump run at different speeds.

where:
a and b denote the same pump with slightly different impeller diameters.

For geometrically and hydraulically similar machines (equal specific speeds) the performance data obtained from one unit can be used to estimate the performance of another unit (i.e., model test data used to estimate prototype performance) using the following laws of pump scaling [5]:

$$\frac{Q_a}{Q_b} = \left(\frac{n_a}{n_b}\right)\left(\frac{D_a}{D_b}\right)^3$$

$$\frac{H_a}{H_b} = \left(\frac{n_a}{n_b}\right)^2\left(\frac{D_a}{D_b}\right)^2$$

$$\frac{P_a}{P_b} = \left(\frac{n_a}{n_b}\right)^3\left(\frac{D_a}{D_b}\right)^5$$

and a modified Moody equation to determine efficiency:

$$\frac{1 - \eta_a}{1 - \eta_b} = \left(\frac{D_b}{D_a}\right)^{0.14}$$

where:

a and b denote two geometrically and hydraulically similar pumps.

Unit Characteristics

Sample Design Problem

There is a requirement for three equal-size canal relift pumping units discharging into a common line where the unit discharge centerline is at approximately 1000 m. The pumps are to be submerged to avoid the need for vacuum priming equipment.

- Minimum capacity at maximum head (per unit) .... 14.2 m³/s
- Static lift .... 61 m
- Average water temperature .... 35 °C

Figure 1 shows that consideration should be given to the radial-flow, vertical-shaft, spiral-case-type pump.

- Canal water surface may vary from normal .... ±0.9 m
- Total suction side head loss per unit (including trashrack loss) . 0.3 m
- Discharge side head loss per unit through the manifold .... 1.5 m

With three units in operation at full capacity, the head loss in the discharge line (including velocity head loss at the discharge structure) is 5.5 m. The hydraulic units are to be designed for *best efficiency* with two units operating. The pumps will be driven by synchronous electric motors.

Head Range

Begin by calculating the required total head range. In this case, the best efficiency (design) head is to occur with two units operating and a static lift of 61.0 m.

Design head (100 percent head) = 61 + 0.3 + 1.5 + 5.5 (2/3)² = 65.2 m.

Note that, for a number of units using a common discharge line—during operation at less than full capacity—the discharge line head loss is roughly equal to:

$$\left[\frac{\text{(operating capacity)}}{\text{full capacity}}\right]^{2*} \times \text{(discharge line loss at full capacity)}$$

The minimum head will occur with one unit operating and a static lift of 61.0 - 0.9 = 60.1 m.

Minimum head = 60.1 + 0.3 + 1.5 + 5.5 (1/3)² = 62.5 m = 96 percent of design head.

The maximum head will occur with three units operating and a static lift of 61.0 + 0.9 = 61.9 m.

Maximum head = 61.9 + 0.3 + 1.5 + 5.5 = 69.2 m = 106 percent of design head.

Trial Pump Specific Speed

It is assumed the pumps are to be submerged 1 m (H_s = 1 m). Using figure 5, (pump specific speed versus design head at various suction heads) the upper limit of pump specific speed can be estimated and used as the trial pump specific speed n_s'. However, it is noted that figure 5 is based on sea level with a water temperature of 29 °C, and does not include suction side head losses H_L. Therefore, prior to using figure 5, the desired suction head H_s—at plantsite elevation and plantsite water temperature—should be corrected to sea level and a water temperature of 29 °C.

For better accuracy the trial pump specific speed can be calculated considering that suction specific speed S should approximate 154 and using the equations:

$$\text{NPSH} = H_a + H_s - H_v - H_L$$

$$S = n_s'\left(\frac{h}{\text{NPSH}}\right)^{0.75}, \text{ or } n_s' = \frac{S}{\left(\frac{h}{\text{NPSH}}\right)^{0.75}}$$

From figure 6, at the plantsite elevation of 1000 m and a water temperature of 35 °C:

H_a = 9.18 m atmospheric pressure, and H_v = 0.57 m vapor pressure

*When the discharge line head loss is large; use an exponent of 1.85 for the pipe friction portion of the loss, and 2.0 for any fitting loss and velocity head loss.

Suction side losses H_L were given as 0.3 m and suction head H_s has been assumed to be 1.0 m.

Thus, the *available* NPSH for the pumping plant with 1-m suction head H_s is estimated to be:

$$\text{NPSH} = 9.18 + 1.0 - 0.57 - 0.3 = 9.31 \text{ m}$$

and

$$n'_s = \frac{154}{\left(\frac{65.2}{9.31}\right)^{0.75}} = 35.8$$

Capacity Requirements

For estimating the capacity to be specified at design head *h,* reference is made to the performance curves of existing units of similar pump specific speeds. The curve on figure 8 shows a typical plant having a specific speed of 39. At 106 percent design head (maximum head for the example), capacity should be about 95 percent of that at best efficiency (design) head *h*. Therefore, to deliver 14.2 m³/s in the example plant (at maximum head), the capacity at design head (design capacity) should be increased to:

$$Q = \frac{14.2}{0.95} = 14.9 \text{ m}^3/\text{s}$$

The drop in efficiency and capacity caused by wear (between overhaul periods) also must be considered. Figure 2 presents percent loss in efficiency versus $\sqrt{h}/\sqrt{Q}$ for different water environments. Generally, canals are subject to contamination from windblown sand and silt equivalent to condition "B" (fig. 2);

thus:

$$\sqrt{\frac{h}{Q}} = \sqrt{\frac{65.2}{14.9}} = 2.1$$

From figure 2, a 2-percent loss in capacity can be predicted between 3-year overhaul periods. Therefore, the design capacity must be increased:

hence:

$$Q = (14.9)\ 1.02 = 15.2 \text{ m}^3/\text{s}$$

Rotational Speed and Pump Specific Speed

Calculate the trial rotational speed n' from the trial pump specific speed n'_s;

where:

$$n' = \frac{n'_s\, h^{0.75}}{Q} = \frac{35.8\ (65.2)^{0.75}}{15.2} = 210 \text{ r/min}$$

However, since the pumps are to be driven by synchronous electric motors, the pump rotational speed should equal a synchronous speed. Consideration should be given to the fact that for extremely large motors a multiple of four poles is preferred [6]; however, standard motors are available in most multiples of two poles.

Determine speed as follows:

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

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

Using a multiple of two poles, the closest 60 Hz synchronous speed to 210 r/min is:

$$n = \frac{60\ (120)}{34} = 212 \text{ r/min}$$

Therefore, the pump specific speed for the given condition is:

$$n_s = \frac{n\sqrt{Q}}{h^{0.75}} = \frac{212\ \sqrt{15.2}}{(65.2)^{0.75}} = 36.0$$

Pump Submergence

The submergence of the units can be estimated using $n_s = 36.0$. From figure 6, at best efficiency head, the recommended minimum sigma σ is:

$$\sigma = \frac{1212\,(n_s)^{1.33}}{10^6} = \frac{1212\,(36.0)^{1.33}}{10^6} = 0.142$$

From the performance curves on figure 8, at minimum head (96 percent design head), capacity will be about 105 percent of design capacity. Figure 7 (typical variation of critical sigma versus discharge) shows that at 105 percent capacity the expected critical sigma will be approximately 110 percent of critical sigma at *best efficiency* point. Therefore, at minimum head, critical sigma should be approximately:

$$\sigma = 0.142\,(1.10) = 0.156$$

The same procedure for maximum head (106 percent design head) results in a critical sigma value of 0.132;

whereas:

$$\sigma = \frac{\text{NPSH}}{H},\ \text{or NPSH} = \sigma H$$

At maximum head, NPSH = 0.132 (69.2) = 9.13 m
At design head, NPSH = 0.142 (65.2) = 9.26 m
At minimum head, NPSH = 0.156 (62.5) = 9.75 m

For a plant at an elevation of 1000 m and average water temperature of 35 °C, figure 6 shows:

- Atmospheric pressure head, H_a .... 9.18 m
- Water vapor pressure head, H_v .... 0.57 m
- Total suction side losses, H_L .... 0.3 m

Therefore, the highest point of the inlet edge of the impeller should be at least (fig. 4):

9.75 - 9.18 + 0.57 + 0.3 = 1.44 m below the inlet canal water surface elevation which produces minimum total head,

9.26 - 9.18 + 0.57 + 0.3 = 0.95 m below the inlet canal water surface elevation which produces design head, and

9.13 - 9.18 + 0.57 + 0.3 = 0.82 m below the inlet canal water surface elevation which produces maximum total head.

Additional submergence may be considered to provide a factor of safety against cavitation and loss of efficiency.

Pump Dimensions

After estimating the rotational speed and pump specific speed with the given design head, the curves and equations of figure 13 are used to estimate the impeller inlet diameter D_3 and the impeller discharge diameter D_1.

For estimating D_1, determine the speed constant K_u [7] from either the curve (fig. 13) or the polynomial approximation:

$$K_u = 0.82 + \frac{6.4\,n_s}{10^3} - \frac{3.3\,(n_s)^2}{10^6}$$

D_1 is calculated from the equation:

$$D_1 = \frac{84\,600\,K_u\sqrt{h}}{n}$$

With the values from the example, n = 212 r/min, h = 65.2 m, and n_s = 36,

calculate:

$$K_u = 0.82 + \frac{6.4\,(36)}{10^3} - \frac{3.3\,(36)^2}{10^6} = 1.05$$

and

$$D_1 = \frac{84\,600\,(1.05)\,65.2}{212} = 3383\ \text{m}$$

D_3 is determined by using figure 13:

Select the speed ratio factor α for the given n_s from either the curve or from the equation:

$$\alpha = 810 \left(\frac{n_s}{1000}\right)^{0.707}$$

For the given design head, calculate D_3 from the equation:

$$D_3 = \frac{550\,\alpha\sqrt{h}}{n}$$

whereupon:

$$\alpha = 810 \left(\frac{n_s}{1000}\right)^{0.707} = 77.226$$

and

$$D_3 = \frac{550\ (77.226)\ \sqrt{65.2}}{212} = 1618\ \text{mm}$$

It is noted that, with the approximate impeller inlet diameter D_3 known, the suction tube dimensions also can be estimated by using USBR Report No. HM-2 [8].

Using the estimated impeller discharge diameter D_1 and pump specific speed n_s, figure 14 is used to estimate the following spiral case dimensions:

R_1	= 0.180	= (3383)	=	609 mm
R_2	= .155	= (3383)	=	524 mm
R_3	= .125	= (3383)	=	423 mm
R_4	= .090	= (3383)	=	304 mm
$A \approx J$	= .835	= (3383)	=	2825 mm
E	= 1.065	= (3383)	=	3603 mm
F	= .980	= (3383)	=	3315 mm
G	= .890	= (3383)	=	3011 mm

Spiral case dimensions also can be approximated analytically by using Stepanoff's [7] volute velocity equation:

$$V = K_3 \sqrt{2gh}$$

where:

V = velocity in the spiral case, and
K_3 = an experimental design constant:

$$K_3 = 1.15\ (n_s)^{-0.33}$$

whence K_3 may be calculated.

Assuming V is constant and Q increases in direct proportion to the angular distance from the cut-water ("The wall dividing the initial section and the discharge nozzle portion of the casing * * * " [9]) or can be otherwise predicted, the spiral case diameter d can be approximated at various locations (i). From the equation of continuity:

$$d_{(i)} = \sqrt{\frac{Q_{(i)}\ 10^6}{0.7854\ V}}$$

To find the radial length from the unit centerline to the outside of the spiral case at any location (i), add one-half the impeller discharge diameter D_1, plus the spiral case diameter $d_{(i)}$, plus 0.1 times the impeller discharge diameter to allow for the diffuser ring. Thence, at any location (i) around the case, the radial length from the unit centerline to the outside of the spiral case can be approximated by:

$$\text{Radial length} = \sqrt{\frac{Q_{(i)}\ 10^6}{0.7854\ V}} + 0.6\ D_1$$

In the example, dimension F at location 2 (fig. 14) is calculated,
where:

$$h = 65.2\ \text{m},\ Q = 15.2\ \text{m}^3/\text{s, and}\ n_s = 36.0$$

$$K_3 = (n_s)^{-0.33} = 1.15\ (36.0)^{-0.33} = 0.35$$

$$V = K_3 \sqrt{2gh} = 0.35 \sqrt{2\ (9.82)\ 65.2} = 12.5\ \text{m/s}$$

Assuming location (2) is nearly 270° from the cutwater:

$$Q_{(2)} = \frac{270}{360} (Q_{design}) = 0.75\ (15.2) = 11.4\ \text{m}^3/\text{s}$$

$$d_{(2)} = \sqrt{\frac{(11.4)\ 10^6}{0.7854\ (12.5)}} = 1078\ \text{mm}$$

Therefore, $R_{(2)}$ = 538 mm as compared to $R_{(2)}$ = 524 mm from experience curves, and

$$F = d_{(2)} + 0.6\ D_1 = 1078 + 0.6\ (3383) = 3108\ \text{mm}$$

In this case, F (as calculated) is less than the dimension F of 3315 mm previously predicted by the experience curve of figure 14. This method, though perhaps less reliable, has the advantage of being easily programmed on a hand calculator to quickly calculate estimates. From figure 14:

Dimension	Empirical curves millimeters	Analytical method millimeters
E	3603	3274
G	3011	2909
$A \approx J$	2825	2652

Impeller and Total Pump Masses

In considering a spiral-case pump, for a particular design head h and impeller discharge diameter D_1, figure 15 is used to estimate the impeller mass and the total pump mass. The curves are based on data from existing pump designs and the equations shown (fig. 15) are polynomial approximations of the curves. Note that the total pump mass is expressed by two separate curves. One curve is used when the design head is less than 30 m and the other when the design head is greater than 45 m. Intermediate design heads require interpolation.

For the example, where the design head h was greater than 45 m and the discharge diameter is 3383 mm, the total pump mass is about 94 metric tons. The impeller mass is approximately 16 t. A similar computation can be made to estimate the mass of a vertical-column pump with the experience curves shown on figure 16.

Pump Power Requirement

When calculating the pump power requirement P, for subsequent motor sizing, performance curves for a pump of similar specific speed should be used to predict the operating conditions that will demand maximum power. In the sample problem, from the shape of the curves (fig. 8, Flatiron), the power requirement is expected to be greatest at maximum capacity.

Pump power requirement P in kilowatts is:

$$P = \frac{9.8\,QH}{\eta}$$

The pump efficiency η, at design head and capacity (best efficiency), is estimated from figure 3. Using the design parameters previously calculated, the pump best efficiency will be approximately 91 percent.

At design conditions:

$$P = \frac{9.8\,(15.2)\,65.2}{0.91} = 10\,670 \text{ kW}$$

However, from figure 8, at maximum capacity (105 percent design capacity) the power requirement will be 102 percent of the requirement at design capacity. Therefore, the maximum pump power requirement is:

$$P = 10\,670 \text{ (102 percent)} = 10\,880 \text{ kW}$$

A driver with a net output approximately 10 percent over the maximum pump requirement usually would be required to allow for overcapacity which the pump manufacturer may provide to assure his guarantee is fulfilled, and to provide for operation under conditions other than those anticipated.

Comments on Pump Selections

After determining the principal dimensions and mass of the pump, a layout can be made. With the aid of electrical and structural engineers, a cost estimate can be prepared. Consideration should be given to alternatives of rotational speed and submergence, number of stages, and style of pump relative to construction cost, operation and maintenance expense, and replacement life.

BIBLIOGRAPHY

[1] Jansen, Robert B., "Edmonston Pumping Plant: Nation's Mightiest," Civ. Eng., vol. 42, No. 10, pp. 67-71, October 1972.

[2] Stroh, Raymond G., Seitz, Bradley G., Lloyd, L. W., "Rapid Reversal of San Luis Pumping-Generating Units," IEEE Trans., Power Appar Syst., vol. PAS-89, No. 6, pp. 1106-1111, July/August 1970.

[3] Bates, Carlos G., "Pumping Unit Studies Final Evaluation Report," Prepared for Office of Saline Water, 39 p. Bureau of Reclamation, Denver, December 1965.

[4] Hydraulic Institute, "Standards for Centrifugal, Rotary, and Reciprocating Pumps," 13th ed., New York, 1975.

[5] Stelzer, R. S., Walters, R. N., "Estimating Reversible Pump-Turbine Characteristics," Engineering Monograph No. 39, 46 p., Bureau of Reclamation, Denver, 1977.

[6] Walters, Richard N., Bates, Carlos G., "Selecting Hydraulic Reaction Turbines," Engineering Monograph No. 20, rev. ed., 55 p., Bureau of Reclamation, Denver, 1976.

[7] Stepanoff, Alexey J., "Centrifugal and Axial Flow Pumps," 2d ed., J.Wiley, New York, 1957.

[8] Bureau of Reclamation, "Proportions of Elbow-Type Suction Tubes for Large Pumps," Report No. HM-2, 10 p., Denver, June 1964.

[9] Karassik, I. J., Krutzsch, W. C., Fraser, W. H., Messina, J. P., "Pump Handbook," McGraw-Hill, New York, 1976.

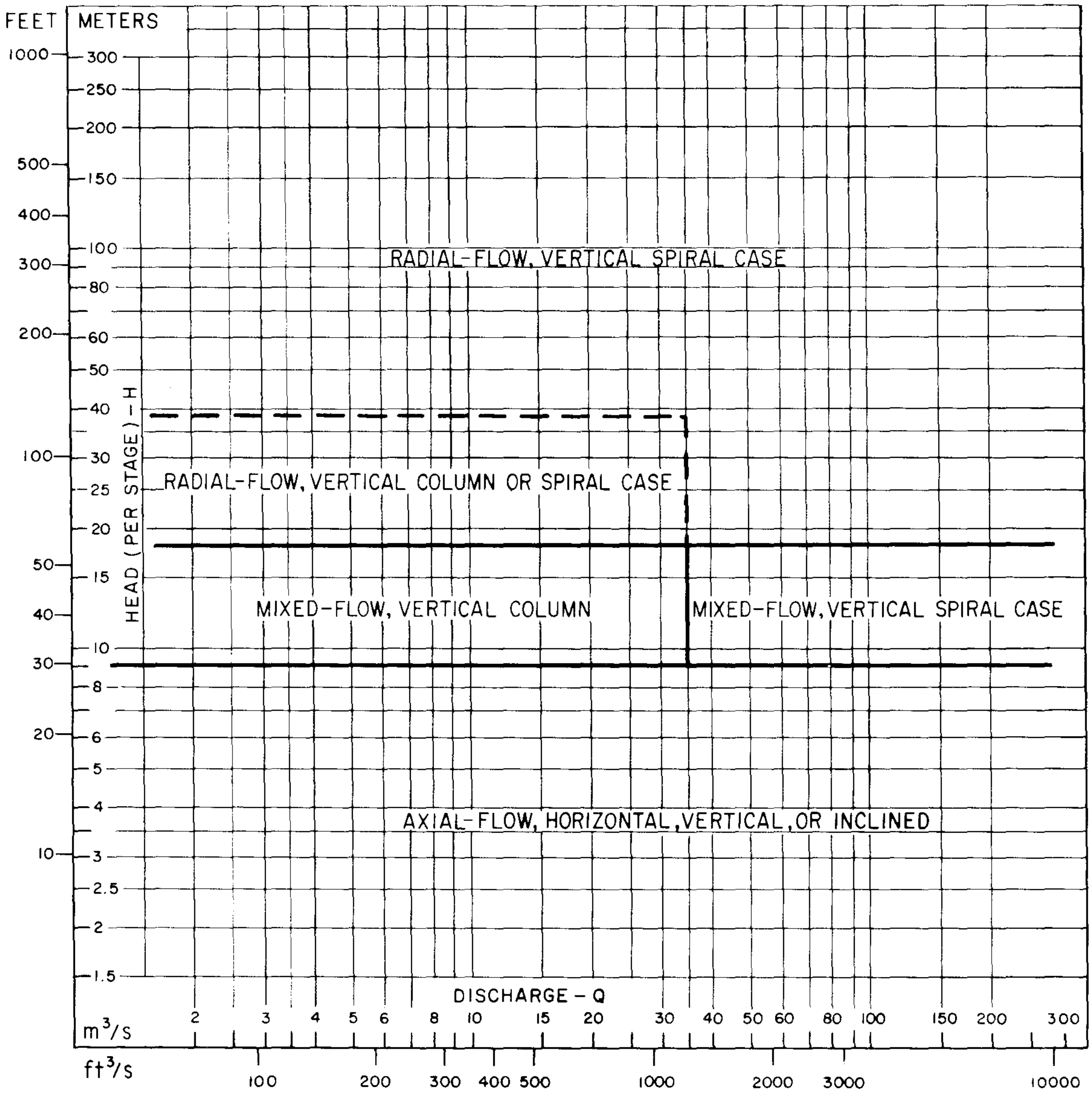


FIGURE 1.—*Large pump type selection guide. 106-D-380.*

A Contains no sand (0.0625–2 mm) or silt (0.004–0.0625 mm); but may contain clay (< 0.004 mm) with a mean concentration of less than 100 mg/L and organic material.

B Contains clay (< 0.004 mm) and silt (0.004 – 0.0625 mm) with a mean concentration of less than 500 mg/L, and for short periods, fine sand (0.0625 – 0.125 mm).

C Contains clay (< 0.004 mm) and silt (0.004 – 0.0625 mm), and sand (0.0625 – 2 mm) with a mean concentration of less than 2000 mg/L which can occur as fine sand (0.0625 – 0.125 mm) in small amount most of the year and coarse sand (0.125 – 2 mm) during flood periods.

D Contains clay (< 0.004 mm) and silt (0.004 – 0.0625 mm), and some fine sand (0.0625 – 0.125 mm) or frequently contains coarse sand (0.125 – 2 mm) and occasional gravel (2 – 8 mm) with a mean concentration of greater than 1000 mg/L.

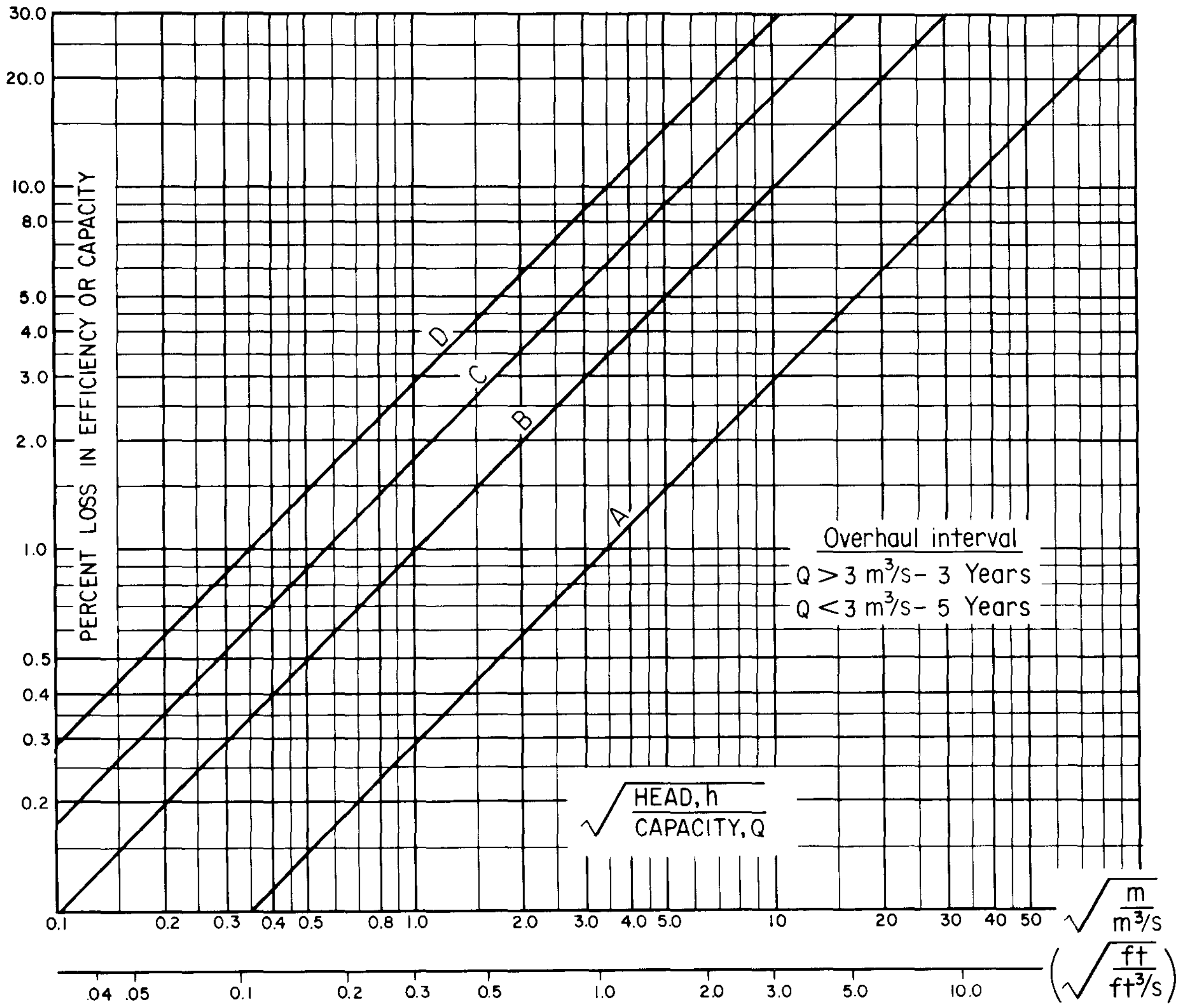


FIGURE 2.—*Loss in efficiency and capacity due to wear between overhauls. 106-D-381.*

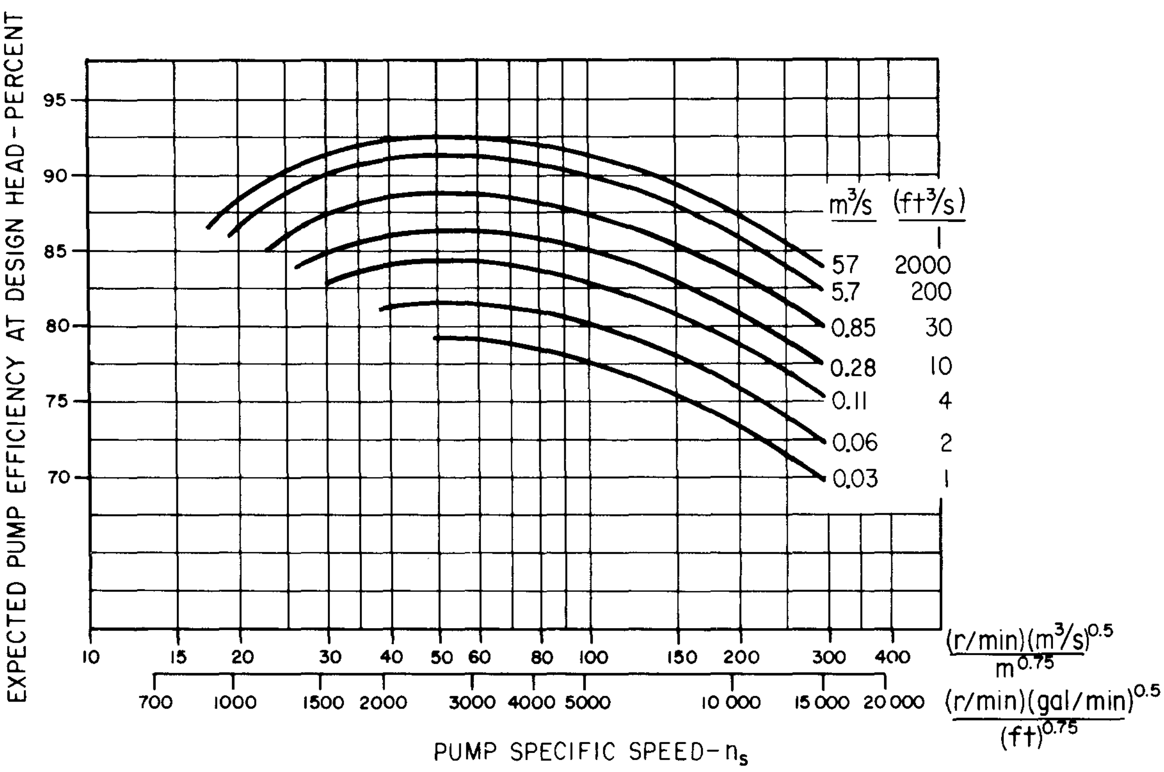


FIGURE 3.—*Expected pump efficiency versus specific speed. 106-D-382.*

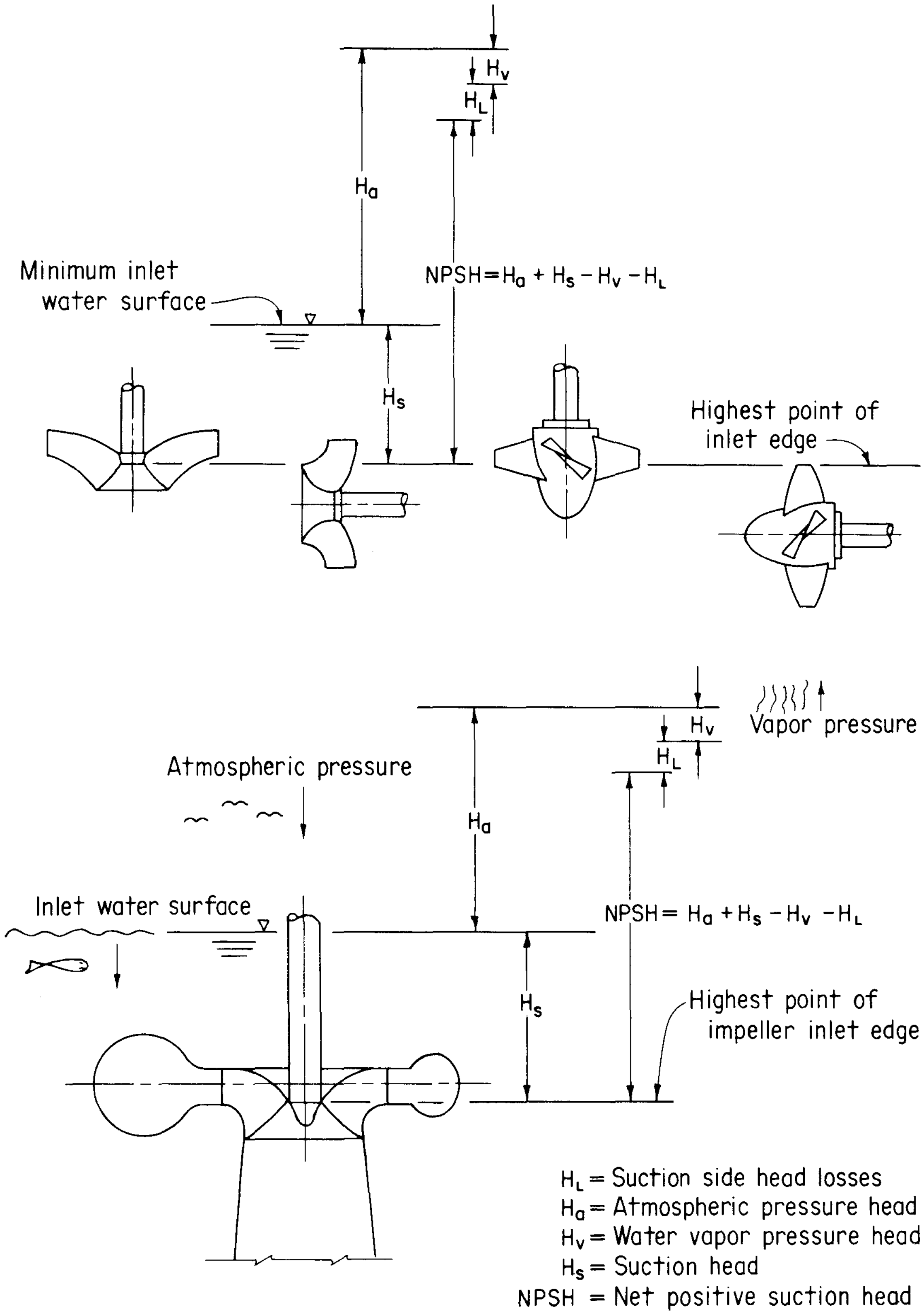


FIGURE 4.—*Net positive suction head. 106-D-383.*

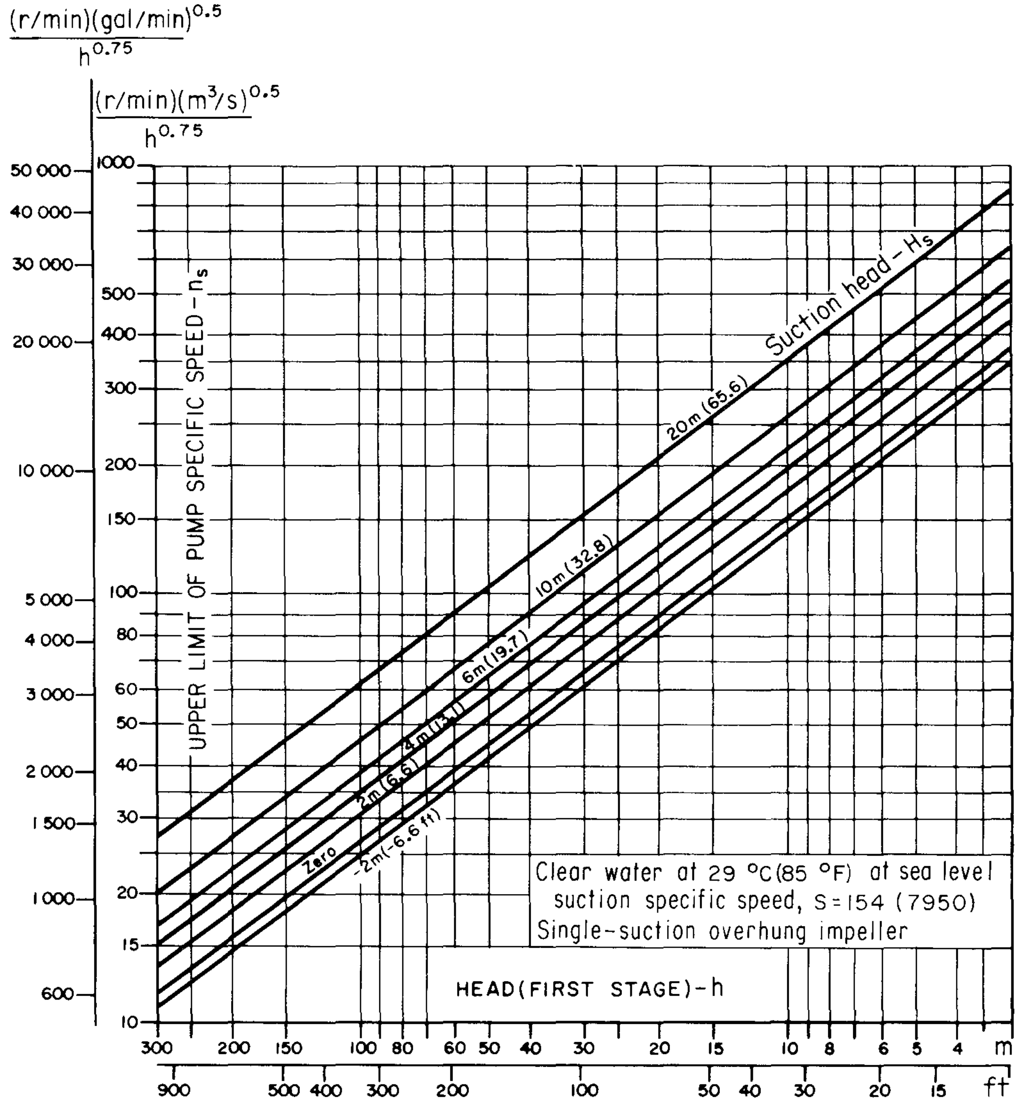
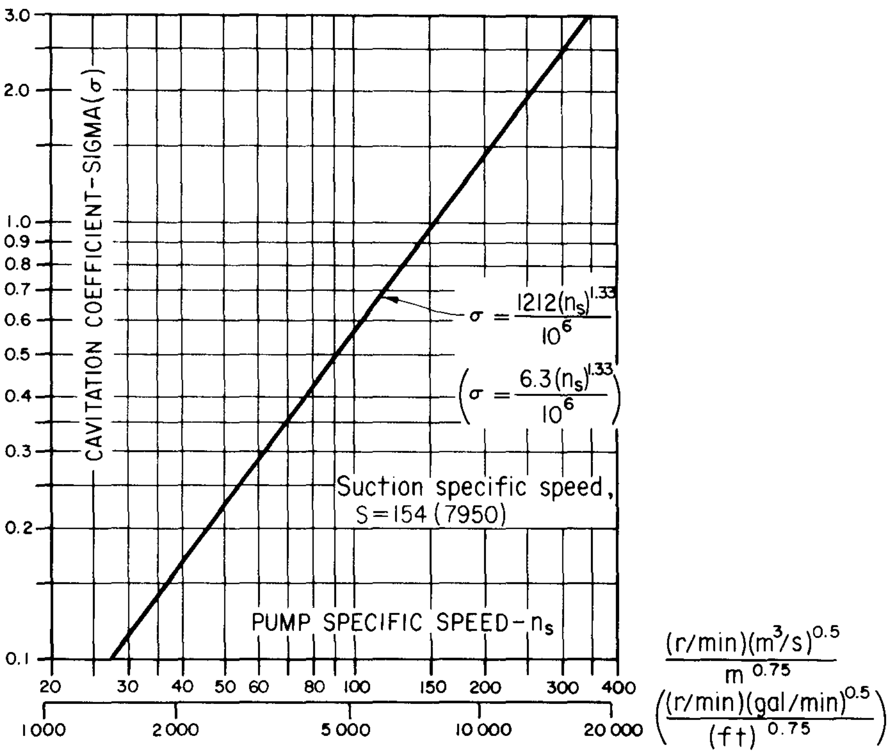


FIGURE 5.—*Pump specific speed versus design head. 106-D-384.*



ATMOSPHERIC PRESSURE				WATER PROPERTIES			
ALTITUDE		HEAD-H_a		TEMPERATURE		VAPOR PRESSURE-H_v	
METERS	FEET	METERS	FEET	°C	°F	METERS	FEET
0	0	10.351	33.959	5	41	0.089	0.292
500	1640	9.751	31.992	10	50	.125	.411
1000	3280	9.180	30.118	15	59	.174	.571
1500	4921	8.637	28.337	20	68	.239	.783
2000	6562	8.120	26.640	25	77	.324	1.062
2500	8202	7.628	25.026	30	86	.434	1.425
3000	9843	7.160	23.491	35	95	.577	1.892
3500	11483	6.716	22.034	40	104	.752	2.467
4000	13123	6.295	20.653	45	113	.977	3.206

FIGURE 6.—*Recommended minimum sigma at best efficiency point. 106-D-385.*

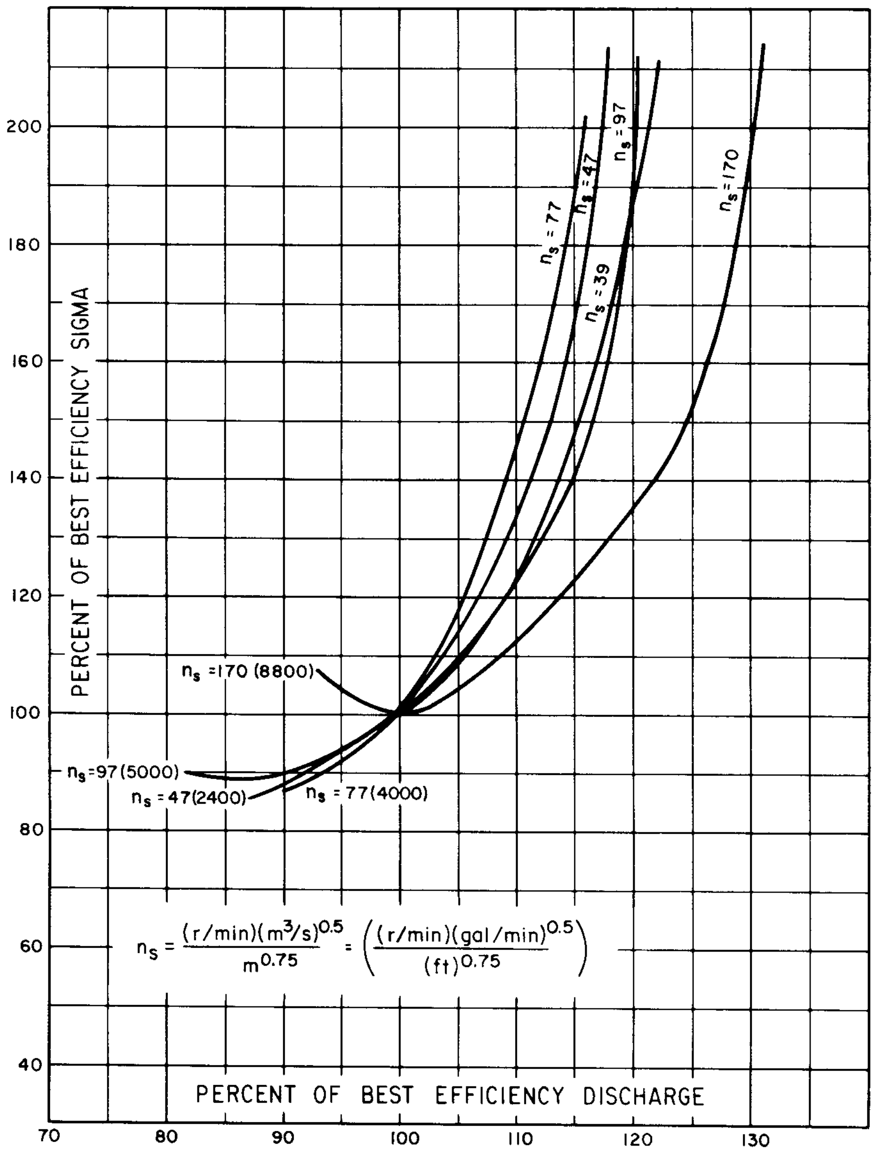
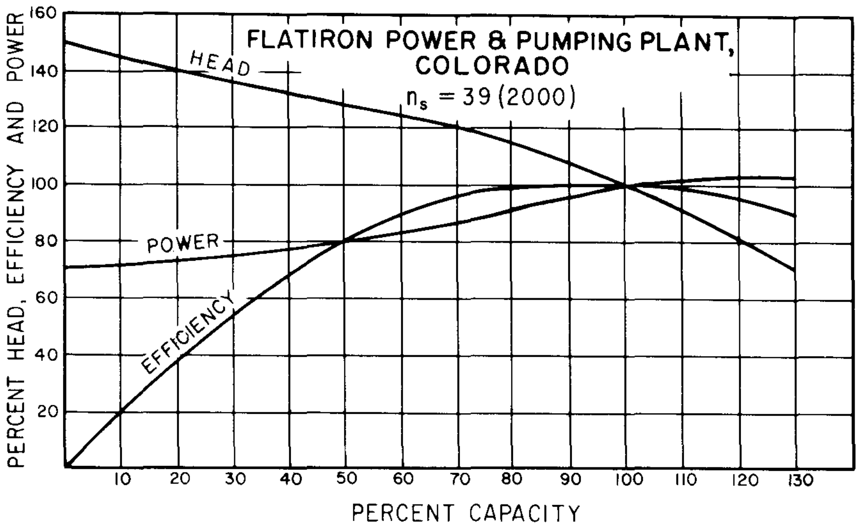


FIGURE 7.—*Typical variation of critical sigma versus discharge. 106-D-386.*



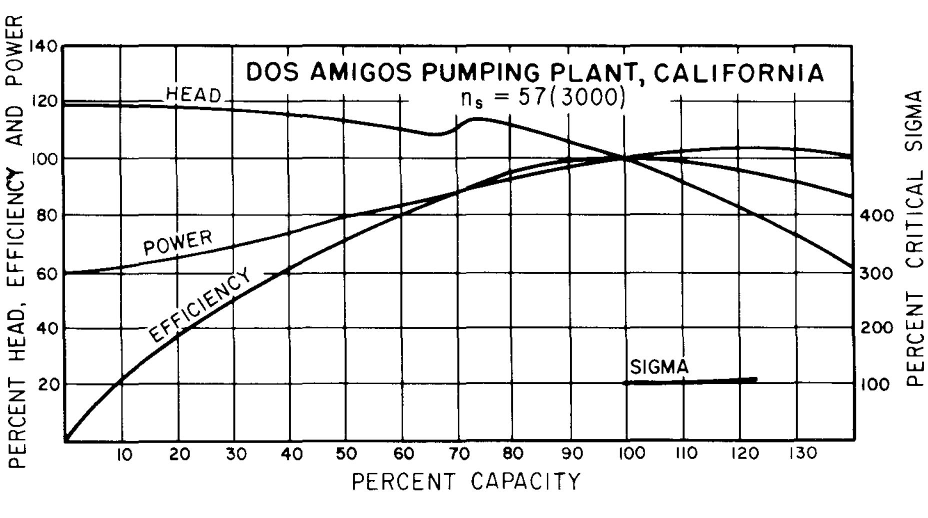
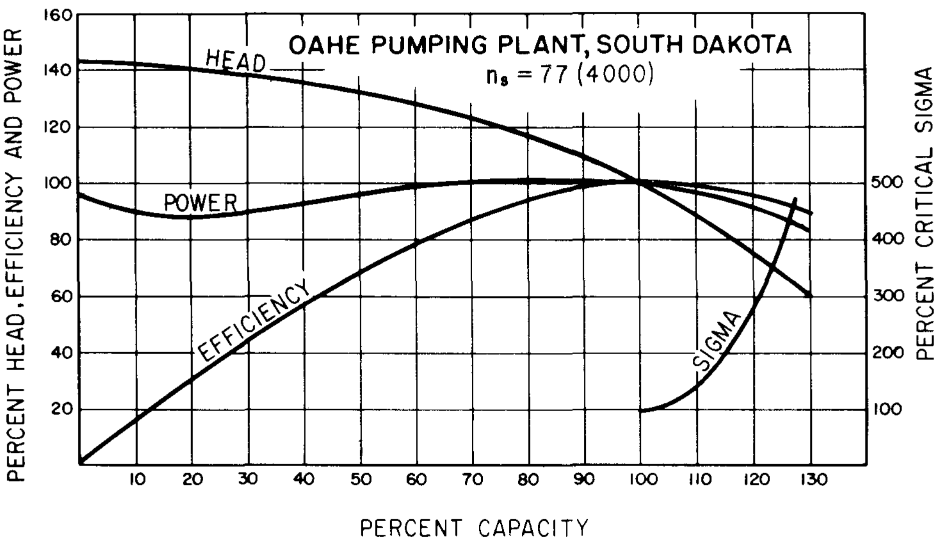


FIGURE 8.—*Flatiron and Dos Amigos performance curves. 106-D-387.*



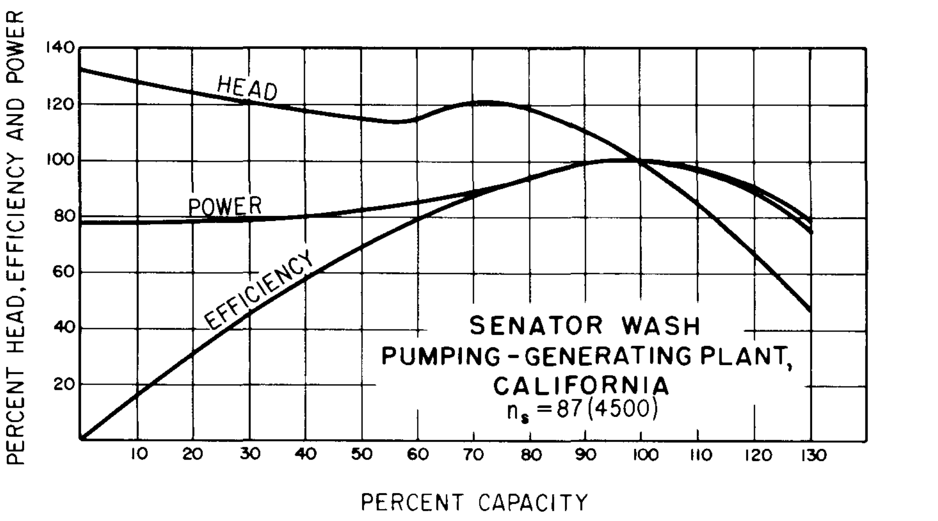
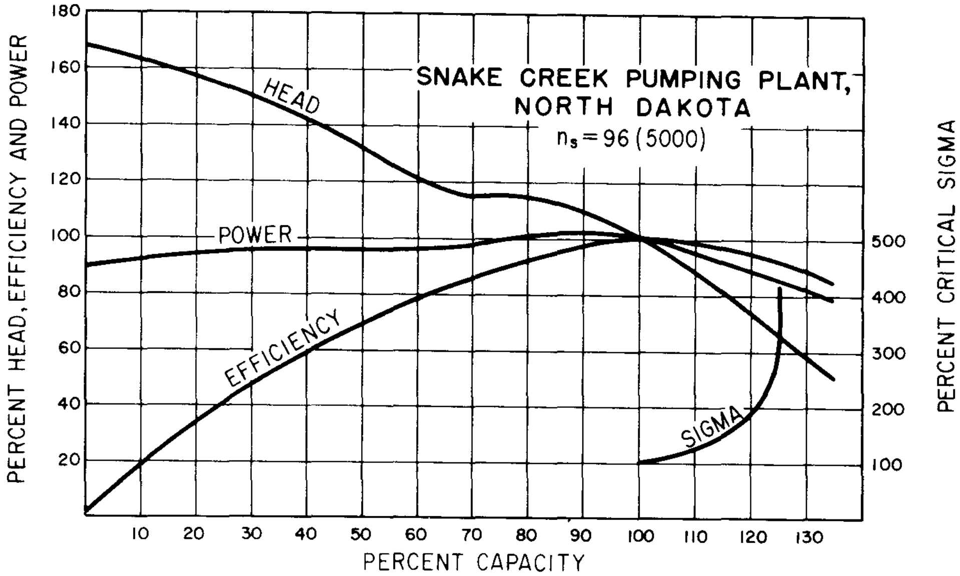


FIGURE 9.—*Oahe and Senator Wash performance curves. 106-D-388.*



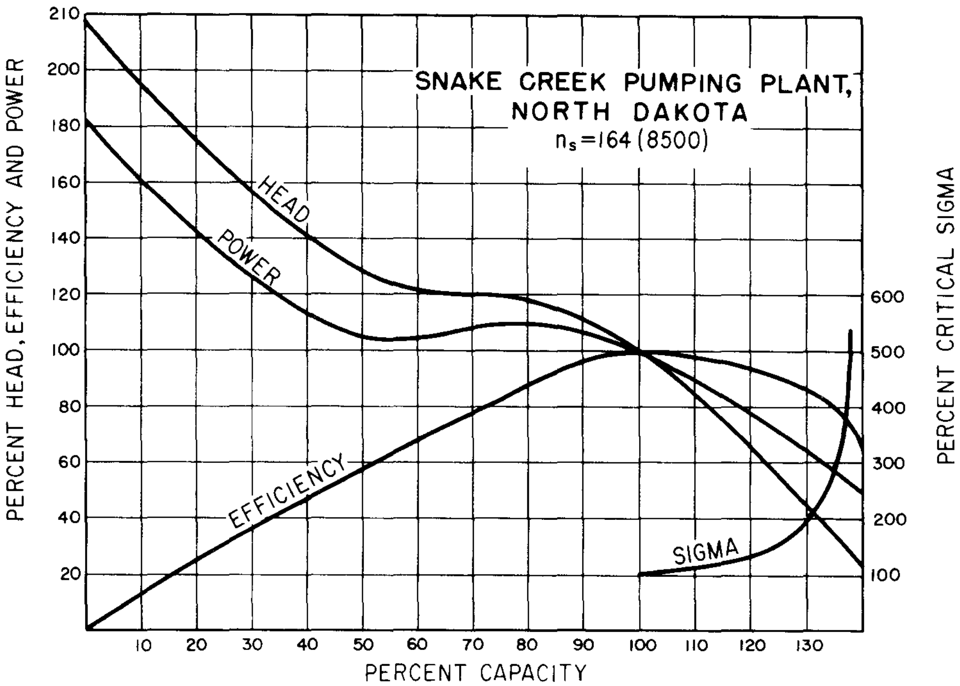


FIGURE 10.—*Snake Creek performance curves. 106-D-389.*

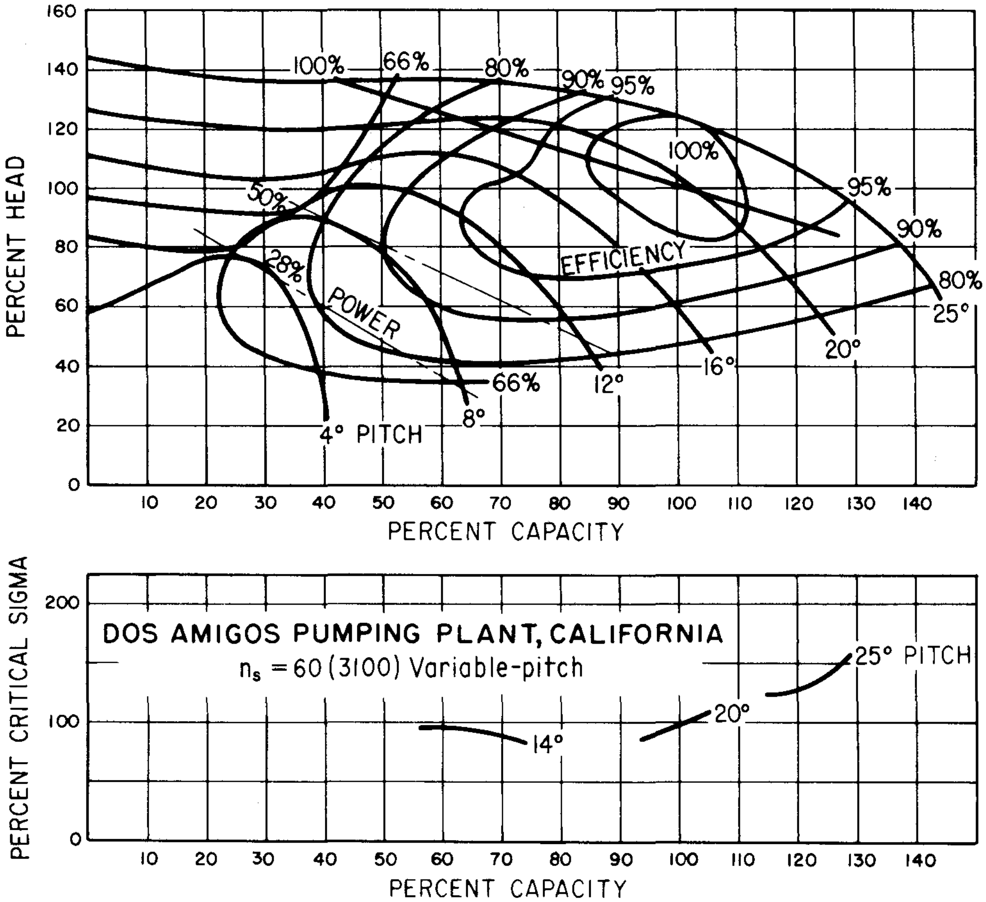


FIGURE 11.—*Dos Amigos performance curves. 106-D-390.*

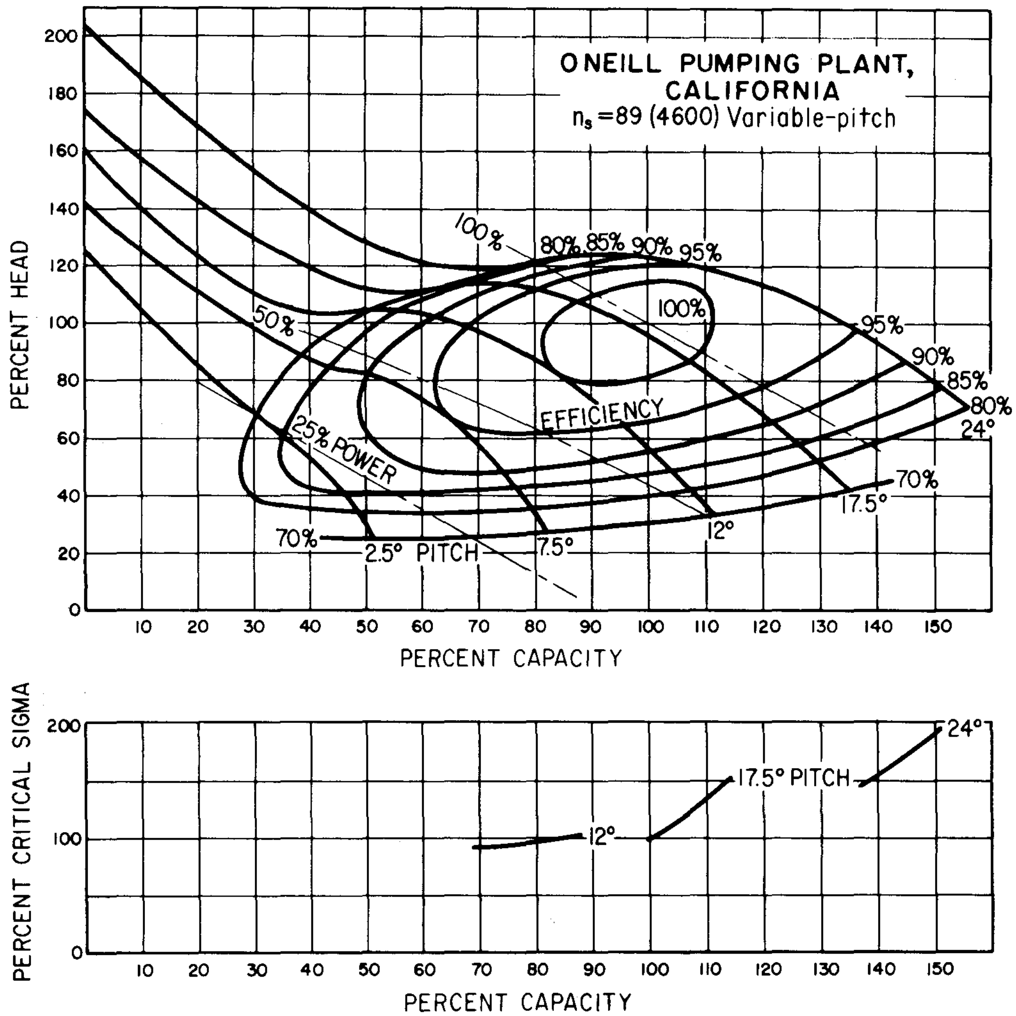


FIGURE 12.—*O'Neill performance curves. 106-D-391.*

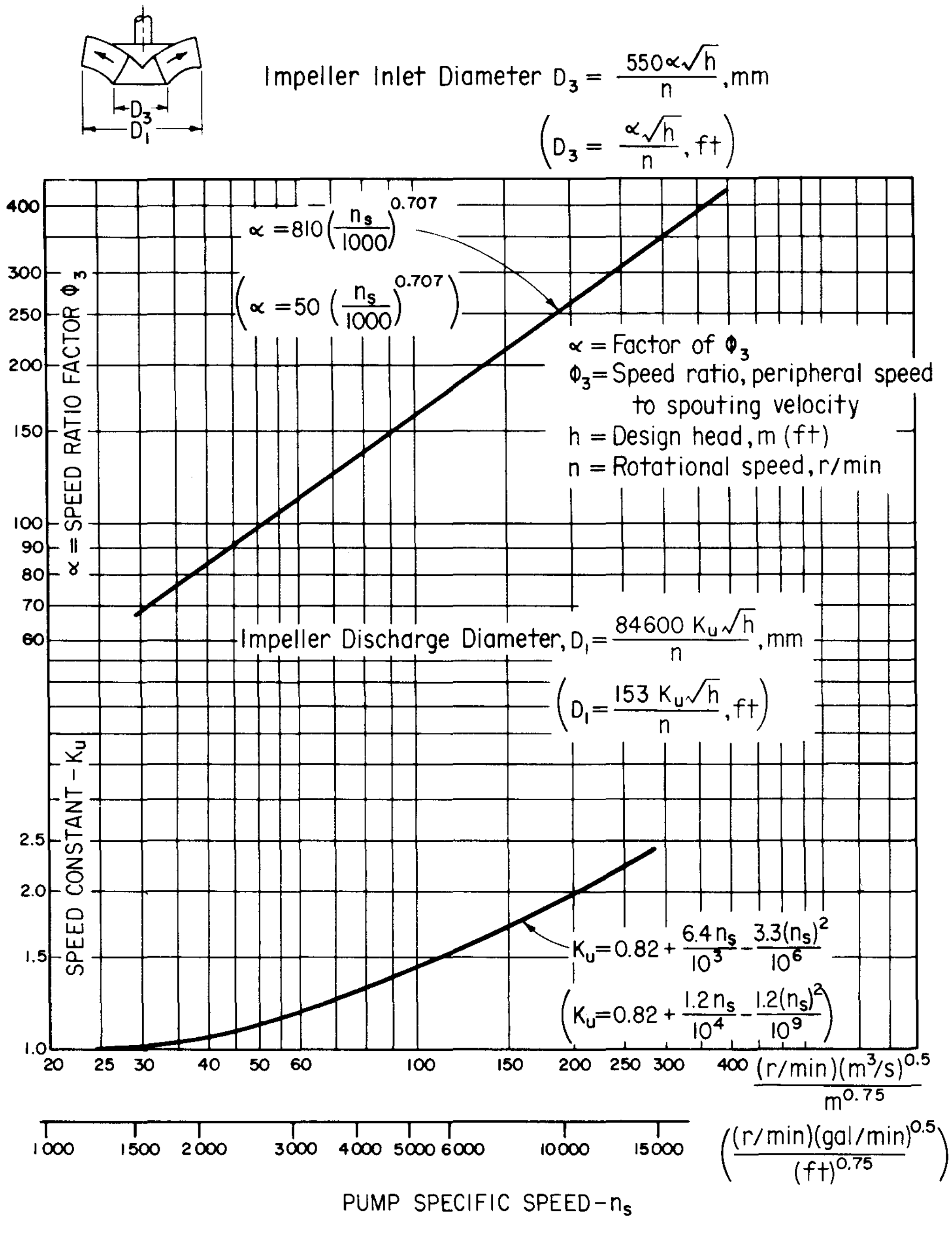


FIGURE 13.—*Inlet and discharge diameter approximations. 106-D-392.*

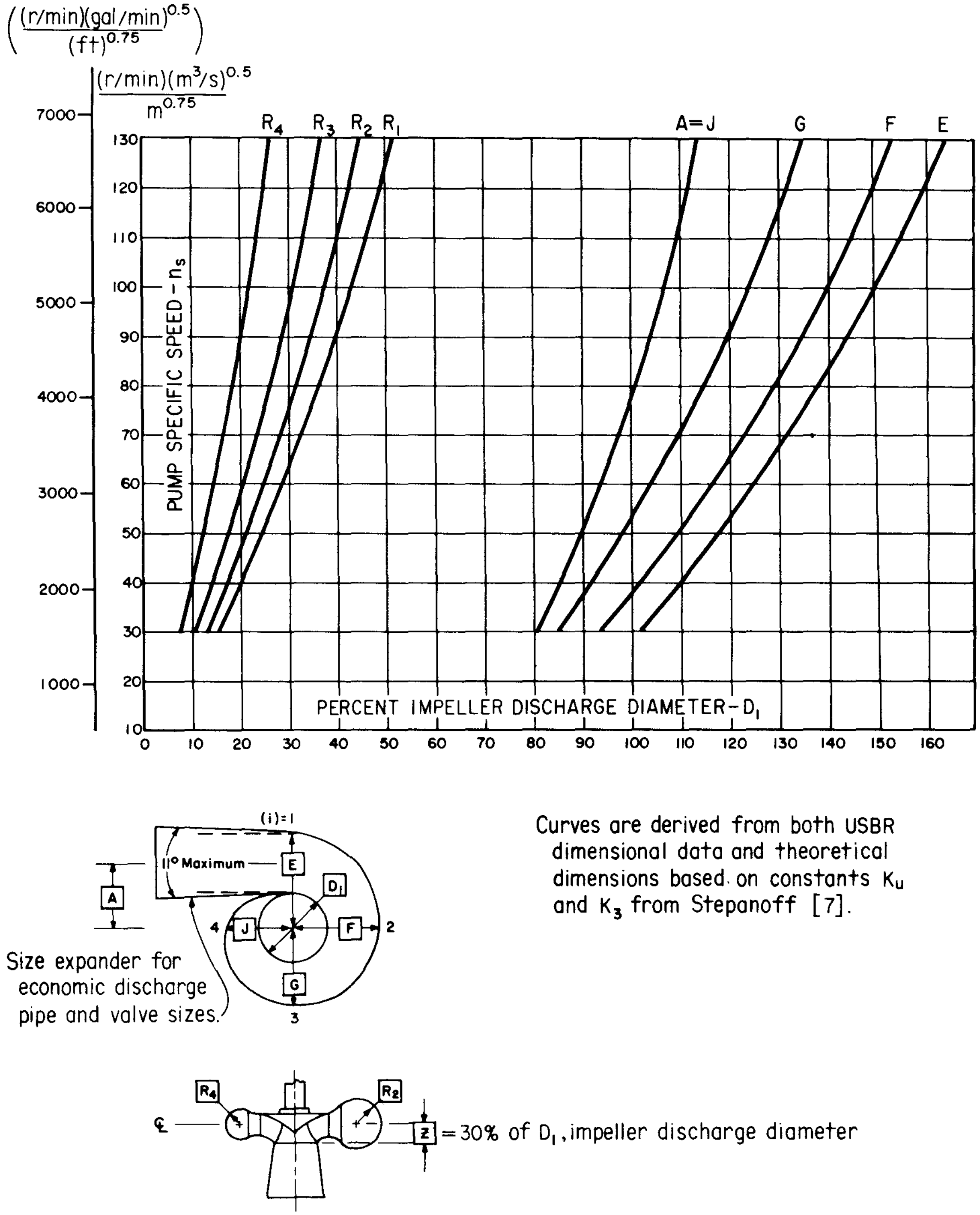


FIGURE 14.—*Approximate spiral case dimensions. 106-D-393.*

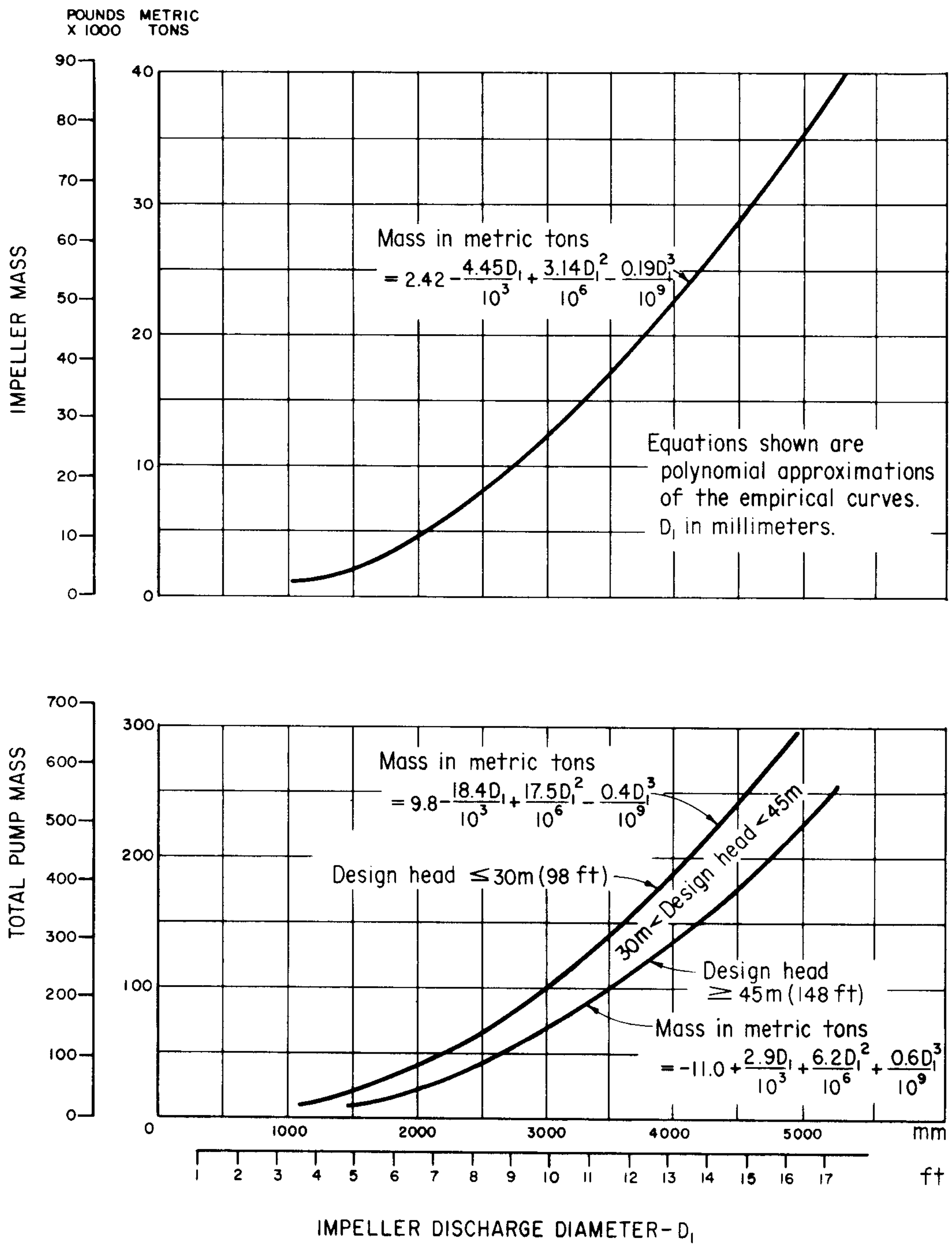


FIGURE 15.—*Impeller and total pump mass—centrifugal vertical spiral-case pump. 106-D-394.*

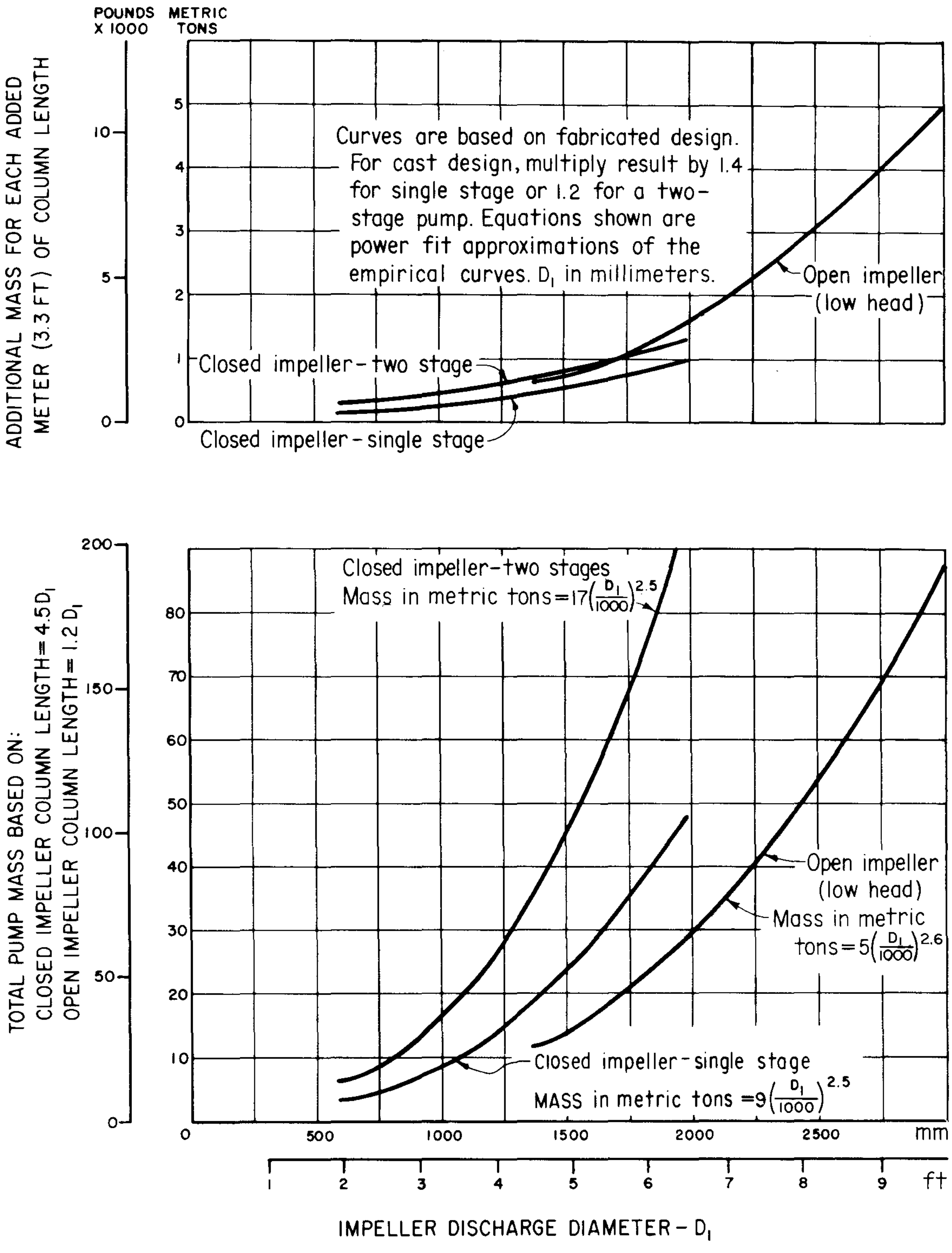


FIGURE 16.—*Vertical column pump mass. 106-D-395.*

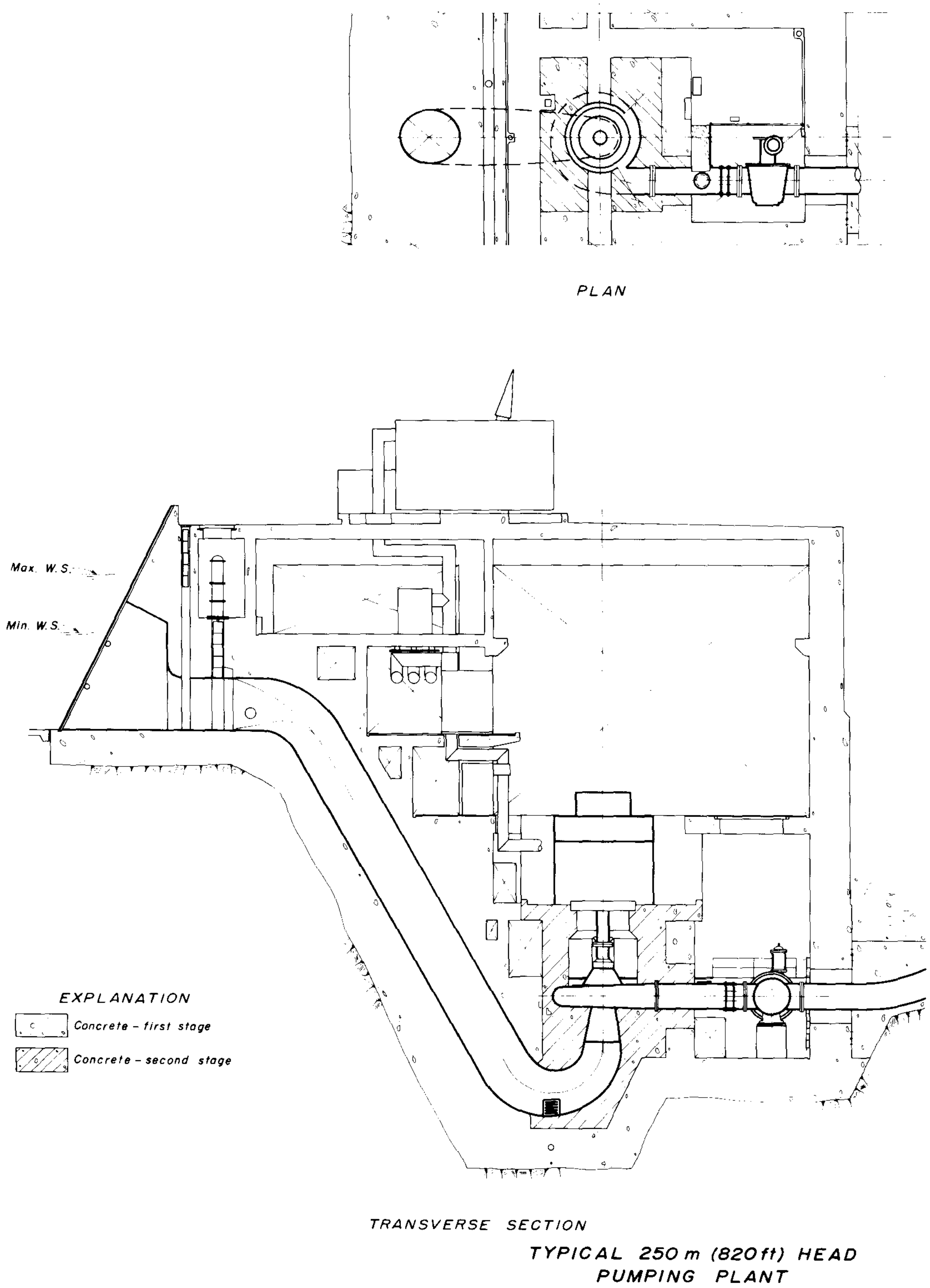


FIGURE 17.—*Typical 250-m (820-ft) head pumping plant. 106-D-396.*

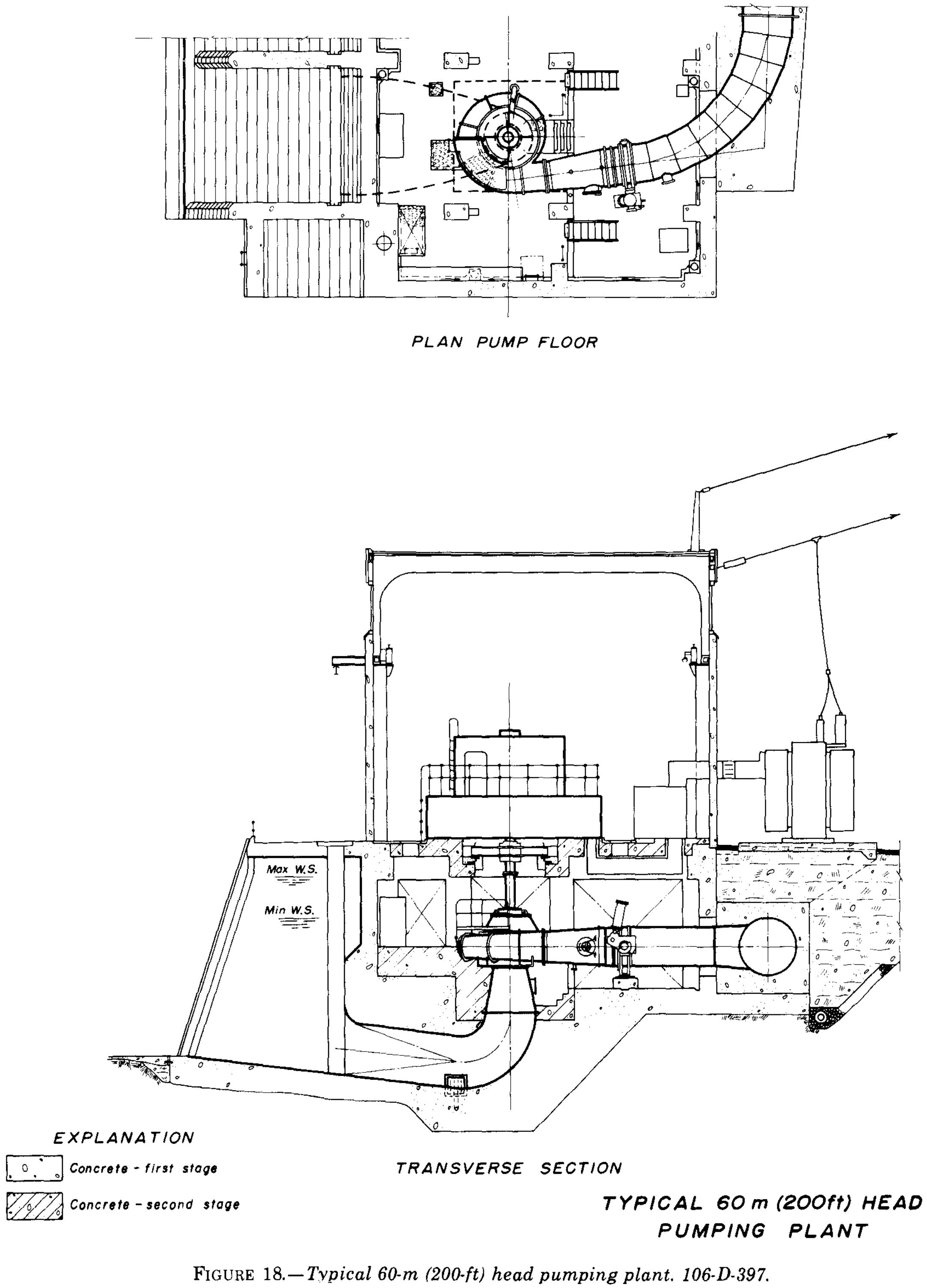


FIGURE 18.—*Typical 60-m (200-ft) head pumping plant. 106-D-397.*

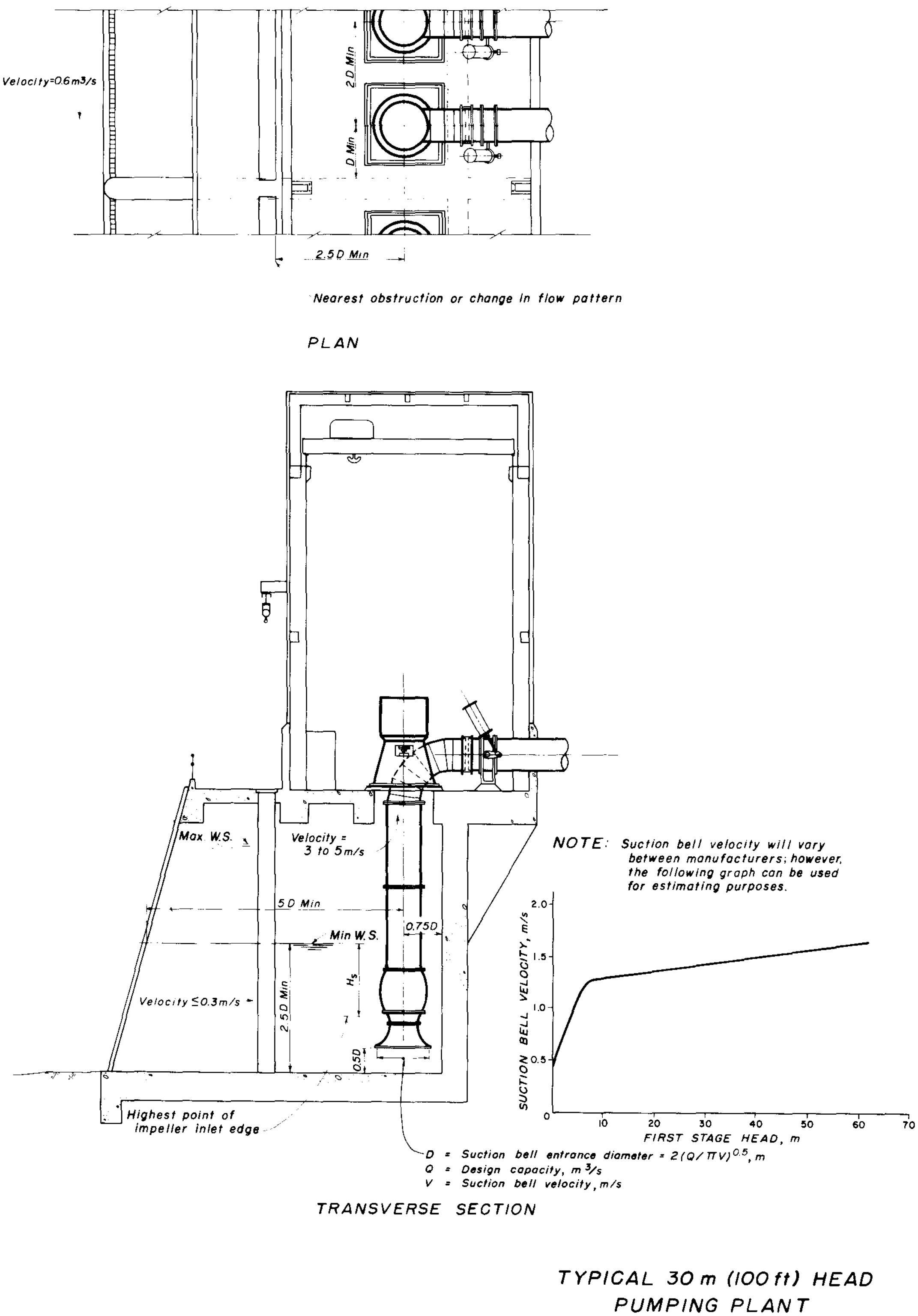


FIGURE 19.—*Typical 30-m (100-ft) head wet-pit pumping plant 106-D-398.*

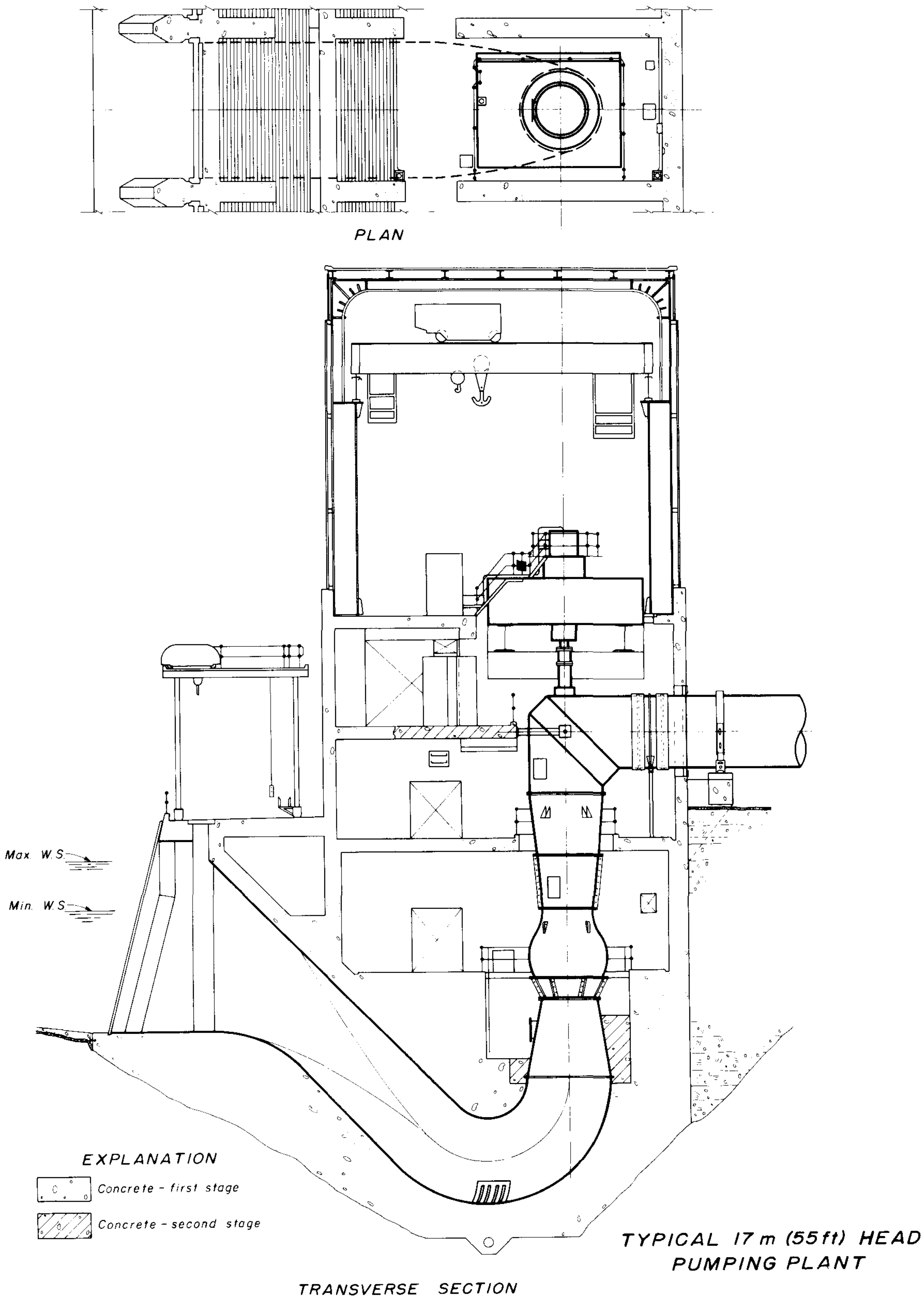


FIGURE 20.—*Typical 17-m (55-ft) head dry-pit pumping plant. 106-D-399.*

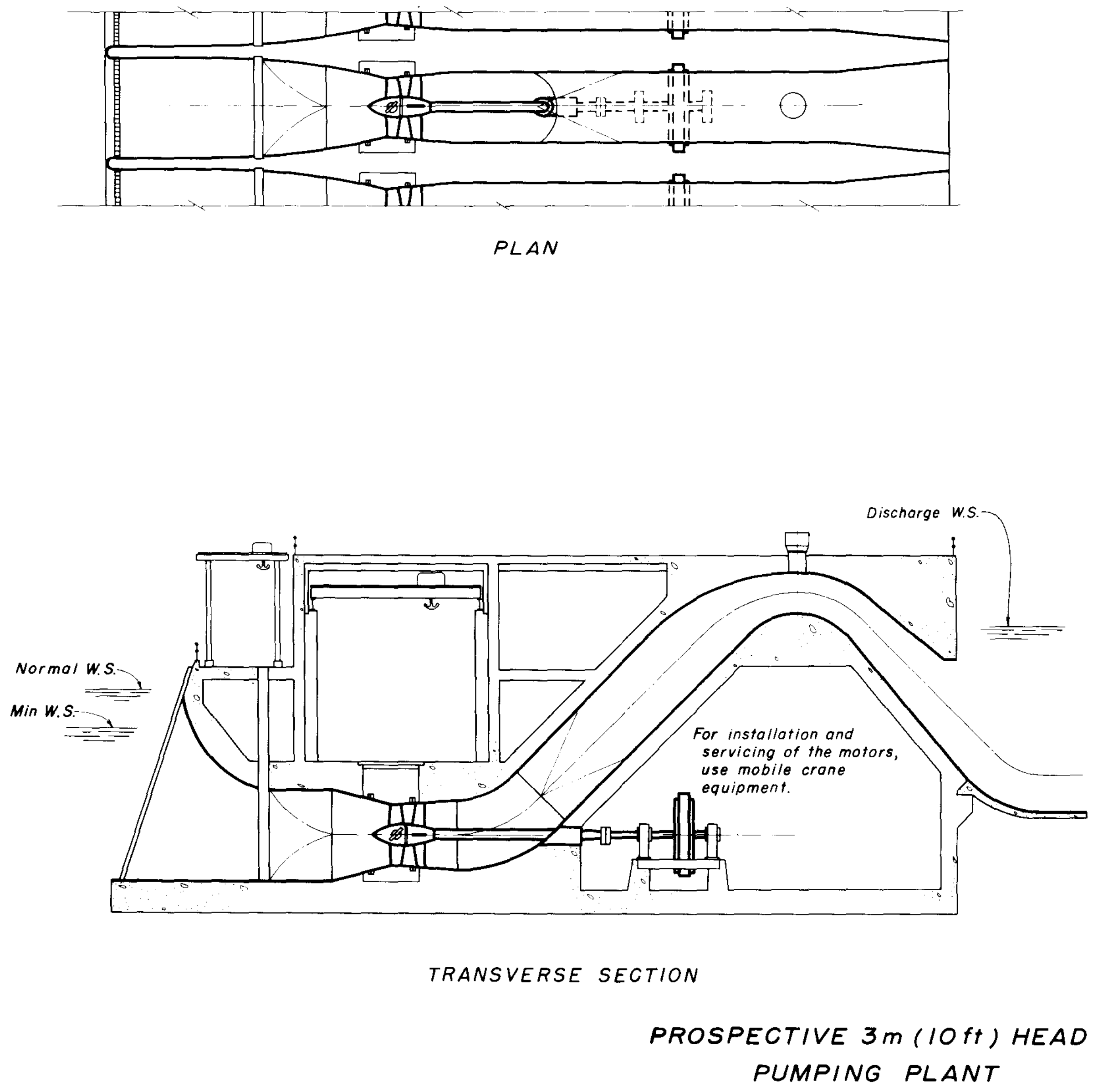


FIGURE 21.—*Prospective 3-m (10-ft) head pumping plant. 106-D-400.*

BASIC EQUATIONS

Using SI units	Using U.S. customary units
$d_{(i)} = \sqrt{\dfrac{Q_{(i)}\,10^6}{0.7854\,V}}$	
$D_1 = \dfrac{84\,600\,K_u\sqrt{h}}{n}$	$D_1 = \dfrac{153\,K_u\sqrt{h}}{n}$
$D_3 = \dfrac{550\,\alpha\sqrt{h}}{n}$	$D_3 = \dfrac{\alpha\sqrt{h}}{n}$
$K_3 = 1.15\,(n_s)^{-0.33}$	$K_3 = 0.44\left(\dfrac{n_s}{1000}\right)^{-0.33}$
$K_u = 0.82 + 6.4\,(n_s)\,10^{-3} - 3.3\,(n_s)^2\,10^{-6}$	$K_u = 0.82 + 1.2\,(n_s)\,10^{-4} - 1.2\,(n_s)^2\,10^{-9}$
$n_s = \dfrac{n\sqrt{Q}}{h^{0.75}}$	$n_s = \dfrac{n\sqrt{Q}}{h^{0.75}}$, Q in gal/min
$\text{NPSH} = H_a + H_s - H_v - H_L$	
Power input, kilowatts = $\dfrac{9.8\,Qh}{\eta}$ [1 kW = 101.971 (m·kg)/s]	Power input, horsepower = $\dfrac{Qh}{8.82\eta}$, Q ft³/s [1 hp = 550 (ft·lb)/s]
$S = \dfrac{n\sqrt{Q}}{(\text{NPSH})^{0.75}} = n_s\left(\dfrac{H}{\text{NPSH}}\right)^{0.75}$	$S = \dfrac{n\sqrt{Q}}{(\text{NPSH})^{0.75}} = n_s\left(\dfrac{H}{\text{NPSH}}\right)^{0.75}$, Q gal/min
Synchronous speed = $\dfrac{120\text{ (frequency)}}{\text{number of poles}}$	
Velocity in spiral case, $V = K_3\sqrt{2gh}$	$V = K_3\sqrt{2gh}$
$\alpha = 810\left(\dfrac{n_s}{1000}\right)^{0.707}$	$\alpha = 50\left(\dfrac{n_s}{1000}\right)^{0.707}$
$\sigma = \dfrac{\text{NPSH}}{H}$	
Min. recommended $\sigma = 1.2\,(n_s)^{1.33}\,10^{-3}$	Min. recommended $\sigma = 6.3\,(n_s)^{1.33}\,10^{-6}$

CONVERSION FACTORS

To convert from		To		Multiply by	
foot (International)	(ft)	meter	(m)	3.048	E-01*
cubic foot per second	(ft^3/s)	cubic meter per second	(m^3/s)	2.831 685	E-02
gallons per minute	(gal/min)	cubic meter per second	(m^3/s)	6.309 020	E-05
gallons per minute	(gal/min)	cubic feet per second	(ft^3/s)	2.23	E-03
horsepower, electric	(hp)	watt	(W)	7.460	E+02*
horsepower, metric	(hp)	watt	(W)	7.354 99	E+02
pound	(lb)	kilogram	(kg)	4.535 924	E-01

To convert from customary specific speed	To the index	Multiply by	
$\frac{(r/min)\sqrt{gal/min}}{(ft)^{0.75}}$	$\frac{(r/min)\sqrt{m^3/s}}{m^{0.75}}$	1.94	E-02
$\frac{(r/min)\sqrt{gal/min}}{(ft)^{0.75}}$	$\frac{(r/min)\sqrt{kW}}{m^{1.25}}$	6.06	E-02
$\frac{(r/min)\sqrt{gal/min}}{(ft)^{0.75}}$	$\frac{(r/min)\sqrt{hp_{metric}}}{m^{1.25}}$	7.07	E-02

*Indicates exact conversion. E-05 represents 10^{-5}.