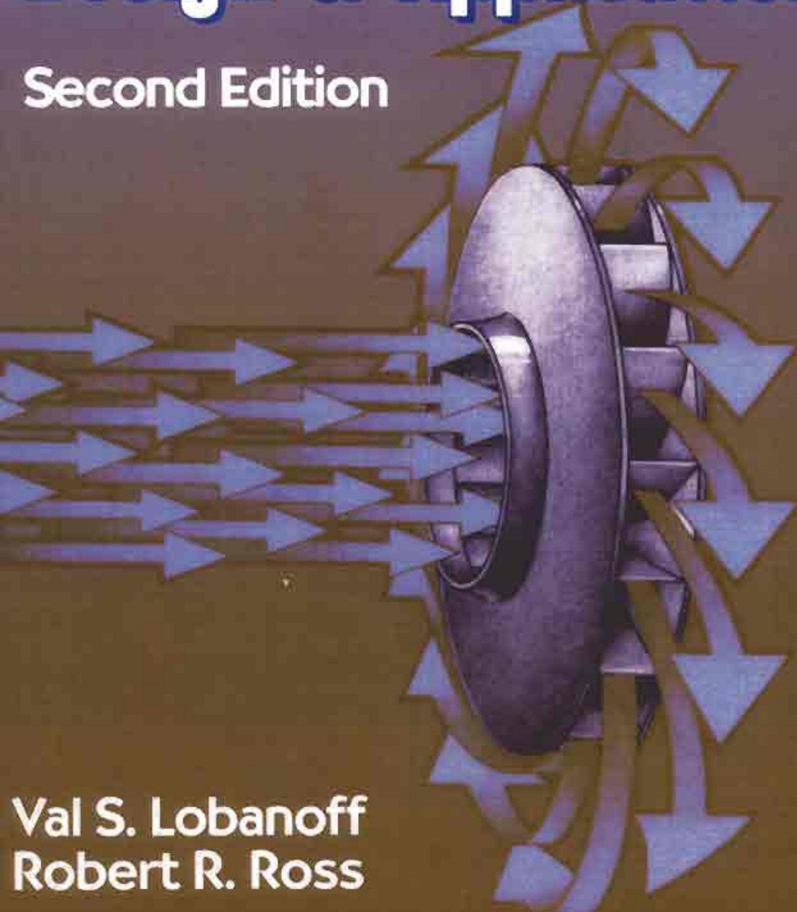
CENTRIFUGAL PUMPS
Design & Application

Second Edition



Val S. Lobanoff
Robert R. Ross

A practical reference stressing hydraulic design, performance prediction, analysis, and evaluation

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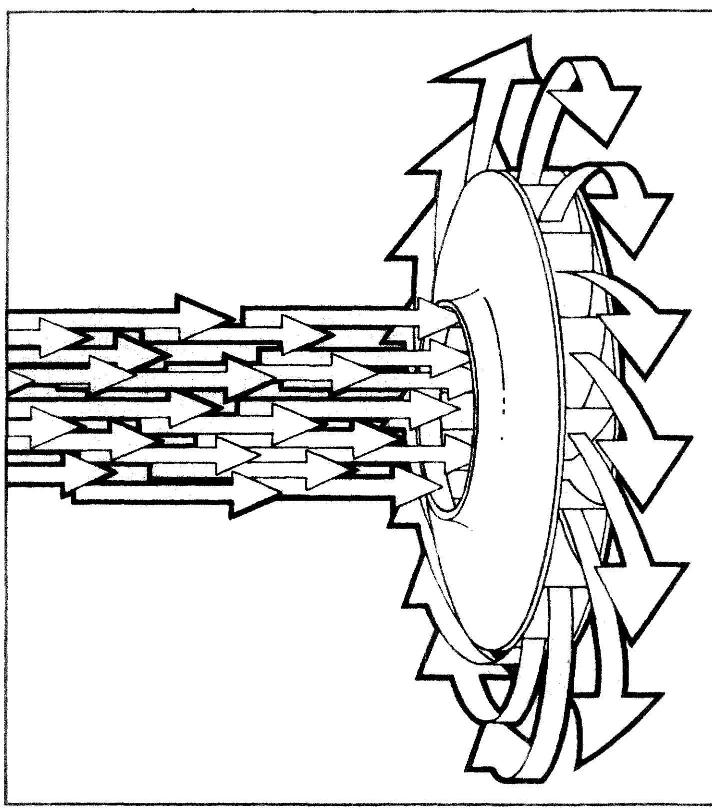

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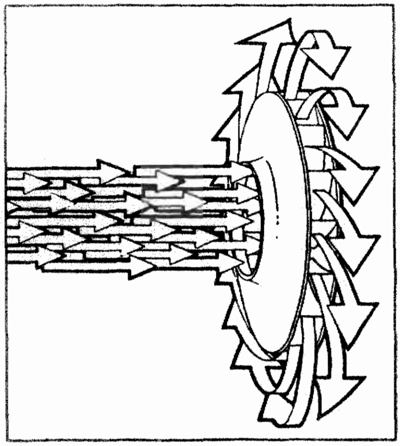

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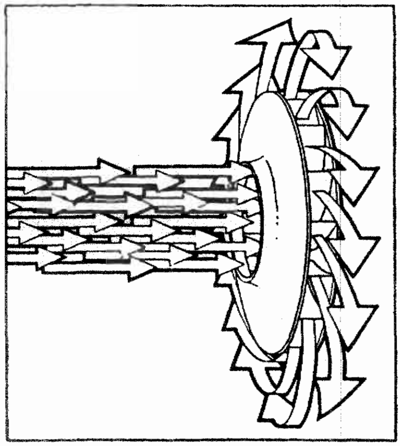

Preface

When Val and I decided to collaborate and write the first edition, our goal was to produce an easy-to-read, easy-to-understand, practical textbook stressing hydraulic design, that could be of hands-on use to the pump designer, student, and rotating equipment engineer. Although feedback from readers indicates that we achieved our desired goal, we did recognize that we had omitted several important topics. We had said little about the design of chemical pumps and touched only lightly on the extensive range of composite materials and the manufacturing techniques used in nonmetallic pump applications. We had totally ignored the subject of mechanical seals, yet we fully recognized that a knowledge of seal fundamentals and theory of operation is essential to the pump designer and rotating equipment engineer.

Another major omission was the subject of vibration and noise in centrifugal pumps. With today's high energy pumps operating at ever increasing speeds, it is essential that we understand the sources of pump noise and causes of vibration that result from installation, application, cavitation, pulsation, or acoustic resonance.

Although we had touched lightly on rotor dynamics, we felt this subject deserved to be expanded, particularly in the areas of bearing stiffness and damping, seal effects, and the evaluation of critical speed calculations. Finally, we had said nothing about the knowledge necessary to extend pump life during installation and operation, which requires a deep understanding of bearings, lubrication, mechanical seal reliability, and the external alignment of pump and driver.

This second edition was therefore written to incorporate these subjects, and in this regard, I have been fortunate in soliciting a number of friends and colleagues, each expert in his chosen field to assist me. With the help of Heinz Bloch, Gordon Buck, Fred Buse, Erik Fiske, Malcolm Murray, Jim Netzel, and Fred Szenasi, I have expanded and improved the second edition in a manner I know would have made Val proud. Readers of the first edition will find this book of even greater practical value, and new readers will gain an in-depth practical knowledge from the extensive experience of the authors and contributors.

Robert R. Ross

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Part 1

Elements of Pump Design

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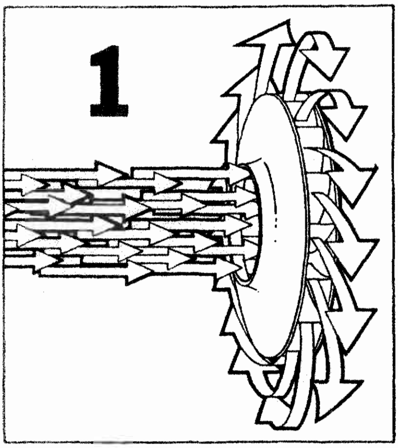


Introduction

System Analysis for Pump Selection

Before a pump can be selected or a prototype designed, the application must be clearly defined. Whether a simple recirculation line or a complex pipeline is needed, the common requirement of all applications is to move liquid from one point to another. As pump requirements must match system characteristics, analysis of the overall system is necessary to establish pump conditions. This is the responsibility of the user and includes review of system configuration, changes in elevation, pressure supply to the pump, and pressure required at the terminal. Relevant information from this analysis is passed on to the pump manufacturer in the form of a pump data sheet and specification. From the information given, the following will ultimately determine pump selection.

- Capacity range of liquid to be moved
- Differential head required
- NPSHA
- Shape of head capacity curve
- Pump speed
- Liquid characteristics
- Construction

Differential Head Required

The head to be generated by the pump is determined from the system head curve. This is a graphical plot of the total static head and friction

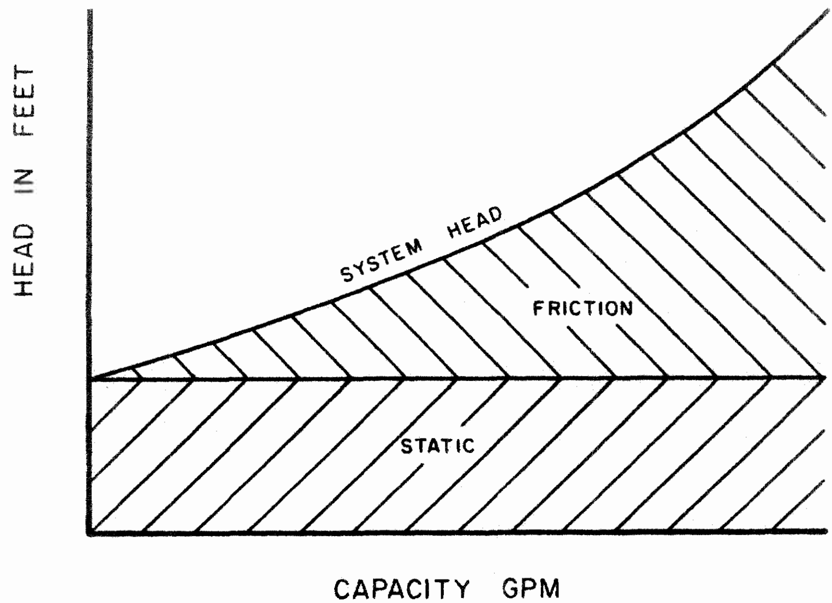


Figure 1-1. System head curve.

losses for various flow rates. For any desired flow rate, the head to be generated by the pump or pumps, can be read directly (Figures 1-1 and 1-2).

NPSHA

Net positive suction head available (NPSHA) is of extreme importance for reliable pump operation. It is the total suction head in feet of liquid absolute, determined at the suction nozzle and referred to datum, less the vapor pressure of the liquid in feet absolute. This subject is discussed in detail in Chapter 9.

Shape of Head Capacity Curve

The desired shape of the head capacity (H-Q) curve is determined during analysis of the system. Most specifications call for a continuously rising curve (Figure 1-3) with the percentage rise from the best efficiency point (BEP) determined by system limits and mode of operation. Unsta-

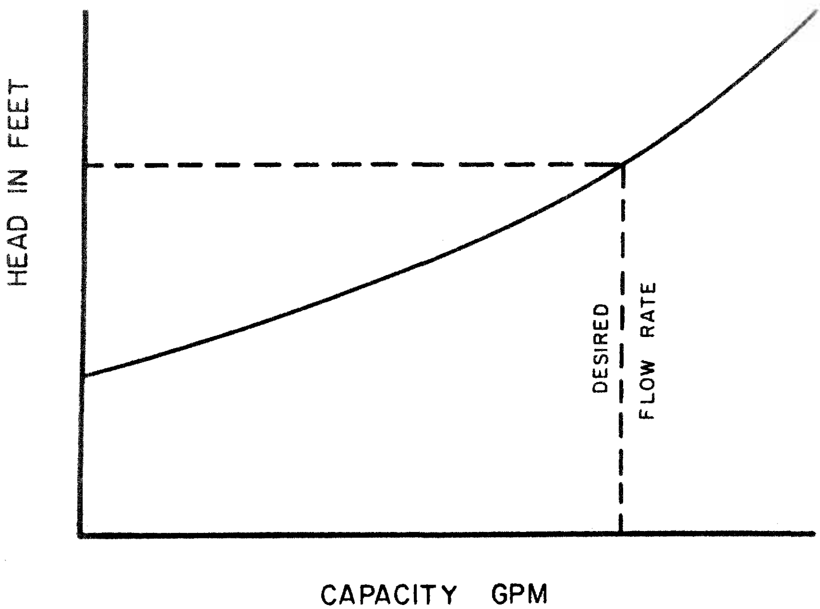


Figure 1-2. The system head curve establishes pump conditions.

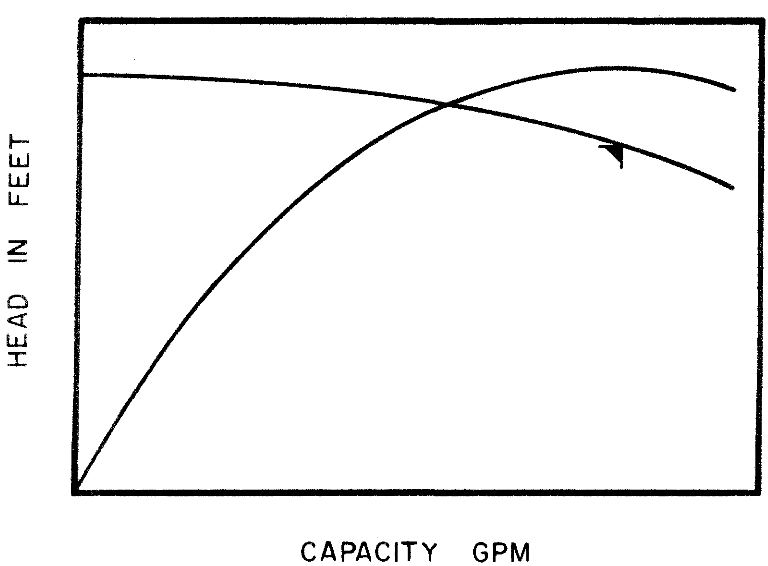


Figure 1-3. Continuously rising head capacity curve.

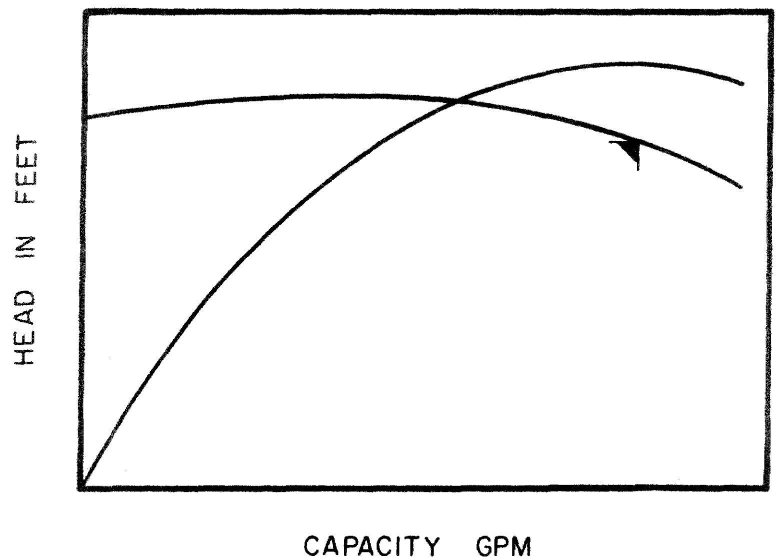


Figure 1-4. Unstable or hooked head capacity curve.

ble or hooked curves (Figure 1-4) where the maximum developed head is at some flow greater than zero are undesirable in applications where multiple pumps operate in parallel. In such applications, zero flow head may be less than system head, making it impossible to bring a second pump on line. It is also possible for pumps to deliver unequal flow with the discharge pressure from one pump determining the flow rate from another. These legitimate reasons have resulted in many specifications forbidding the use of unstable curves for any application. This is most unfortunate as in many instances such curves are perfectly suitable. More importantly, pumps with unstable curves will develop more head and be more efficient than their continuously rising counterparts. It should be noted that this tendency of instability is normally confined to the lower range of specific speeds. As specific speed increases, the H-Q curve becomes more stable. Specific speed is defined in Chapter 2, and design parameters to correct instability in the low specific speed range will be discussed in Chapter 3.

Pump Speed

Pump speed may be suggested by the user to match electric frequency or available driver speed. The pump manufacturer, however, has the ultimate responsibility and must confirm that the desired speed is compatible with NPSHA and satisfies optimum efficiency selection.

Liquid Characteristics

To have reasonable life expectancy, pump materials must be compatible with the liquid. Having intimate knowledge of the liquid to be pumped, the user will often specify materials to the pump manufacturer. When the pump manufacturer is required to specify materials, it is essential that the user supply all relevant information. Since liquids range from clear to those that contain gases, vapors, and solid material, essential information includes temperature, specific gravity, pH level, solid content, amount of entrained air and/or dissolved gas, and whether the liquid is corrosive. In determining final material selection the pump manufacturer must also consider operating stresses and effects of corrosion, erosion, and abrasion.

Viscosity

As liquid flows through a pump, the hydrodynamic losses are influenced by viscosity and any increase results in a reduction in head generated and efficiency, with an increase in power absorbed (Figure 1-5). Centrifugal pumps are routinely applied in services having viscosities below 3,000 SSU and have been used in applications with product viscosities up to 15,000 SSU. It is important to realize that the size of the internal flow passages has a significant effect on the losses, thus the smaller the pump is, the greater are the effects of viscosity. As the physical size of a pump increases, the maximum viscosity it can handle increases. A pump with a 3-in. discharge nozzle can handle 500 SSU, while a pump with a 6-in. discharge nozzle can handle 1,700 SSU. Centrifugal pumps can handle much higher viscosities, but beyond these limits, there is an increasing penalty loss. When viscosity is too high for a particular size pump, it will be necessary to go to a larger pump. A reasonable operating range of viscosity versus pump size is shown in Figure 1-6. Methods to predict pump performance with viscous liquids are clearly defined in the *Hydraulic Institute Standards*.

Specific Gravity

When pumping a nonviscous liquid, pumps will generate the same head uninfluenced by the specific gravity of the liquid. Pressure will change with specific gravity and can be calculated from:

$$\text{Differential pressure (psi)} = \frac{\text{Differential head (ft)} \times \text{sp gr}}{2.31}$$

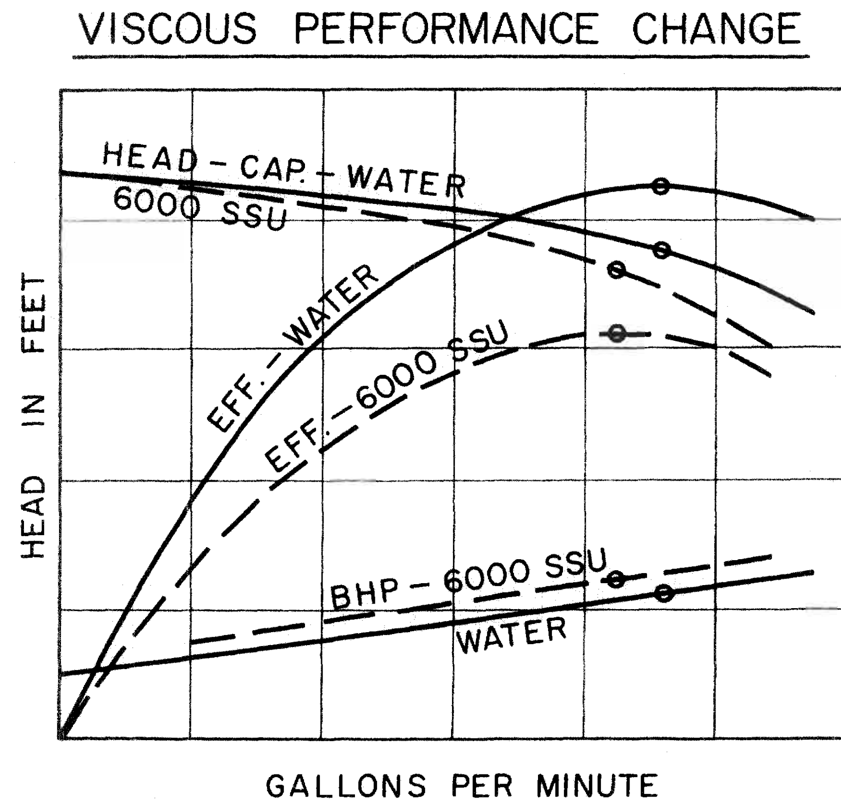


Figure 1-5. Viscous performance change.

Thus, pumps with a change in product density generating the same head will show a change in pressure, and horsepower absorbed by the pump will vary directly with the change in specific gravity (Figures 1-7 and 1-8). A pump being purchased to handle a hydrocarbon of 0.5 specific gravity will normally have a motor rating with some margin over end of curve horsepower. During factory testing on water with 1.0 specific gravity, the absorbed horsepower will be two times that of field operation, thus preventing use of the contract motor during the test. In such instances the pump manufacturers standard test motor is used.

Construction

Pump construction is quite often spelled out on the pump data sheet. General terms like horizontal, vertical, radial split, and axial split are

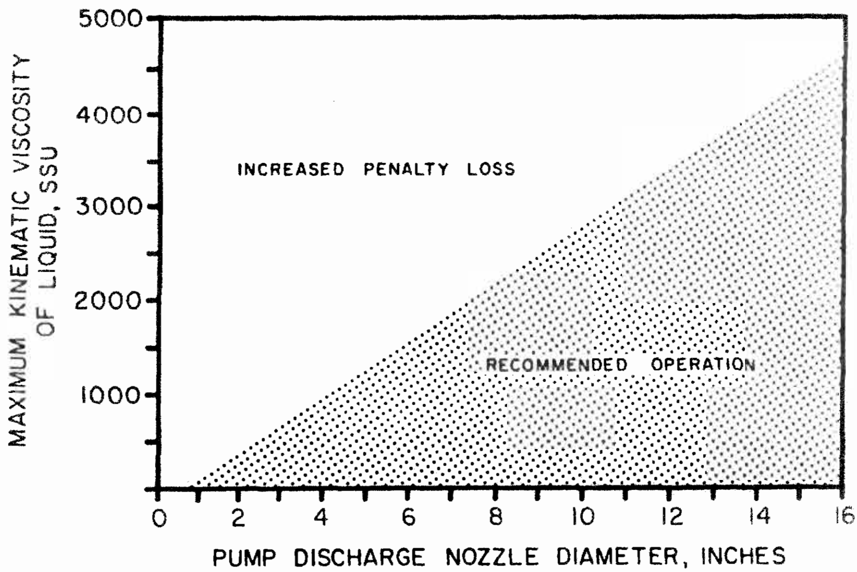
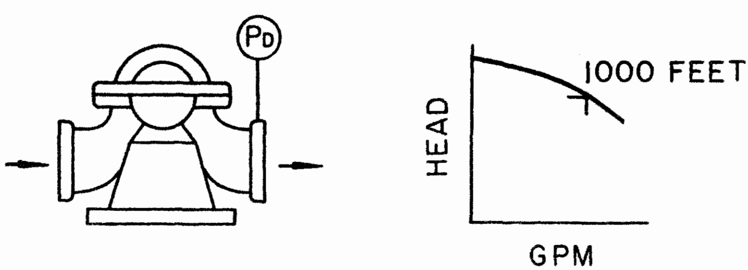


Figure 1-6. Maximum liquid viscosity for centrifugal pumps (from C.E. Petersen, Marmac, "Engineering and System Design Considerations for Pump Systems and Viscous Service," presented at Pacific Energy Association, October 15, 1982).



LIQUID	SPECIFIC GRAVITY	P_D
ETHANE	.5	216 PSI
CRUDE OIL	.85	368 PSI
WATER	1.0	433 PSI

Figure 1-7. Pressure vs. specific gravity.

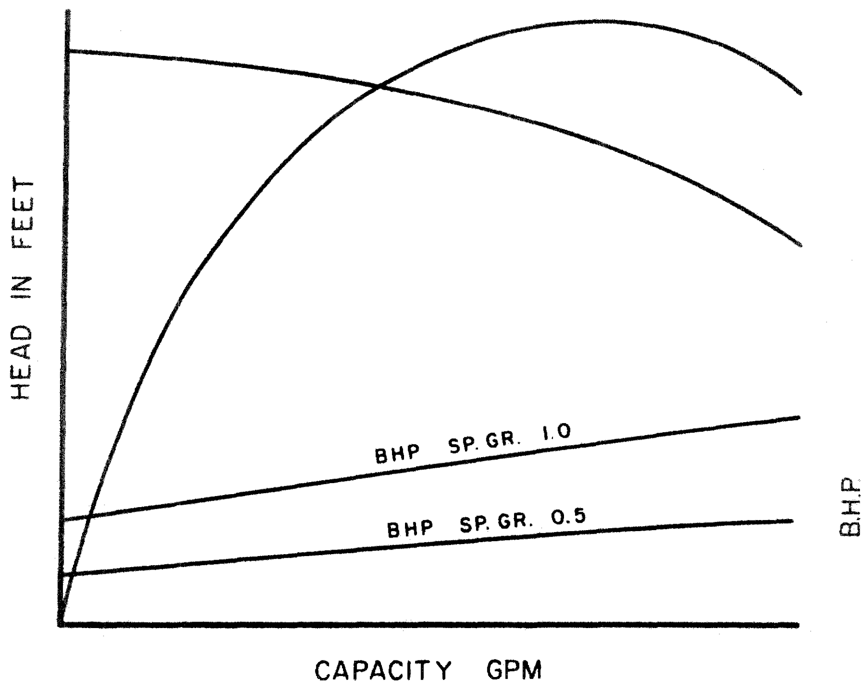
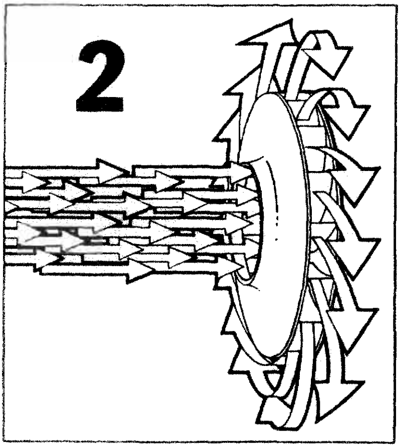


Figure 1-8. Horsepower change with specific gravity.

normally used. For most applications, construction is determined by reliability, ease of maintenance, available real estate, and operating parameters. Ultimately however, it is the pump manufacturer's responsibility to select appropriate construction.

Pump Selection

From the information supplied in the data sheet, a pump can normally be selected from the pump manufacturer's sales book. These are normally divided into sections, each representing a particular construction. Performance maps show the range of capacity and head available, while individual performance curves show efficiency and NPSHR. If the pump requirements fall within the performances shown in the sales book, the process of selection is relatively simple. When the required pumping conditions, however, are outside the existing range of performance, selection is no longer simple and becomes the responsibility of the pump designer.



Specific Speed and Modeling Laws

Specific speed and suction specific speed are very useful parameters for engineers involved in centrifugal pump design and/or application. For the pump designer an intimate knowledge of the function of specific speed is the only road to successful pump design. For the application or product engineer specific speed provides a useful means of evaluating various pump lines. For the user specific speed is a tool for use in comparing various pumps and selecting the most efficient and economical pumping equipment for his plant applications.

A theoretical knowledge of pump design and extensive experience in the application of pumps both indicate that the numerical values of specific speed are very critical. In fact, a detailed study of specific speed will lead to the necessary design parameters for all types of pumps.

Definition of Pump Specific Speed and Suction Specific Speed

Pump specific speed (N_S) as it is applied to centrifugal pumps is defined in U.S. units as:

$$N_s = \frac{RPM \times GPM^{.5}}{H^{.75}}$$

Specific speed is always calculated at the best efficiency point (BEP) with maximum impeller diameter and single stage only. As specific speed can be calculated in any consistent units, it is useful to convert the calculated number to some other system of units. See Table 2-1. The suction specific speed (N_{SS}) is calculated by the same formula as pump specific speed (N_S)

Table 2-1
Specific Speed Conversion

UNITS			
Capacity	Head/ Stage	U.S. to Metric Multiply By	Metric to U.S. Multiply By
Ft^3/Sec	Feet	.0472	21.19
M^3/Sec	Meters	.0194	51.65
M^3/Min	Meters	.15	6.67
M^3/Hr	Meters	1.1615	.8609

but uses NPSHR values in feet in place of head (H) in feet. To calculate pump specific speed (N_s) use full capacity (GPM) for either single- or double-suction pumps. To calculate suction specific speed (N_{ss}) use one half of capacity (GPM) for double-suction pumps.

$$N_{ss} = \frac{RPM \times GPM^{.5}}{NPSHR^{.75}}$$

It is well known that specific speed is a reference number that describes the hydraulic features of a pump, whether radial, semi-axial (Francis type), or propeller type. The term, although widely used, is usually considered (except by designers) only as a characteristic number without any associated concrete reference or picture. This is partly due to its definition as the speed (RPM) of a geometrically similar pump which will deliver one gallon per minute against one foot of head.

To connect the term specific speed with a definite picture, and give it more concrete meaning such as GPM for rate of flow or RPM for rate of speed, two well known and important laws of centrifugal pump design must be borne in mind—the affinity law and the model law (the model law will be discussed later).

The Affinity Law

This is used to refigure the performance of a pump from one speed to another. This law states that for similar conditions of flow (i.e. substantially same efficiency) the capacity will vary directly with the ratio of speed and/or impeller diameter and the head with the square of this ratio at the point of best efficiency. Other points to the left or right of the best efficiency point will correspond similarly. The flow cut-off point is usually determined by the pump suction conditions. From this definition, the rules in Table 2-2 can be used to refigure pump performance with impeller diameter or speed change.

Table 2-2
Formulas for Refiguring Pump Performance with Impeller Diameter or Speed Change

Diameter Change Only	Speed Change Only	Diameter and Speed Change
$Q_2 = Q_1 \left(\frac{D_2}{D_1}\right)$	$Q_2 = Q_1 \left(\frac{N_2}{N_1}\right)$	$Q_2 = Q_1 \left(\frac{D_2}{D_1} \times \frac{N_2}{N_1}\right)$
$H_2 = H_1 \left(\frac{D_2}{D_1}\right)^2$	$H_2 = H_1 \left(\frac{N_2}{N_1}\right)^2$	$H_2 = H_1 \left(\frac{D_2}{D_1} \times \frac{N_2}{N_1}\right)^2$
$bhp_2 = bhp_1 \left(\frac{D_2}{D_1}\right)^3$	$bhp_2 = bhp_1 \left(\frac{N_2}{N_1}\right)^3$	$bhp_2 = bhp_1 \left(\frac{D_2}{D_1} \times \frac{N_2}{N_1}\right)^3$

Q_1, H_1, bhp_1, D_1, and N_1 = Initial capacity, head, brake horsepower, diameter, and speed.
Q_2, H_2, bhp_2, D_2, and N_2 = New capacity, head, brake horsepower, diameter, and speed.

Example

A pump operating at 3,550 RPM has a performance as shown in solid lines in Figure 2-1. Calculate the new performance of the pump if the operating speed is increased to 4,000 RPM.

Step 1

From the performance curve, tabulate performance at 3,550 RPM (Table 2-3).

Step 2

Establish the correction factors for operation at 4,000 RPM.

$$4{,}000/3{,}550 = f = 1.13.$$
$$f^2 = 1.27.$$
$$f^3 = 1.43.$$

Step 3

Calculate new conditions at 4,000 RPM from:

$$Q_2 = Q_1 \times 1.13.$$
$$H_2 = H_1 \times 1.27.$$
$$bhp_2 = bhp_1 \times 1.43.$$

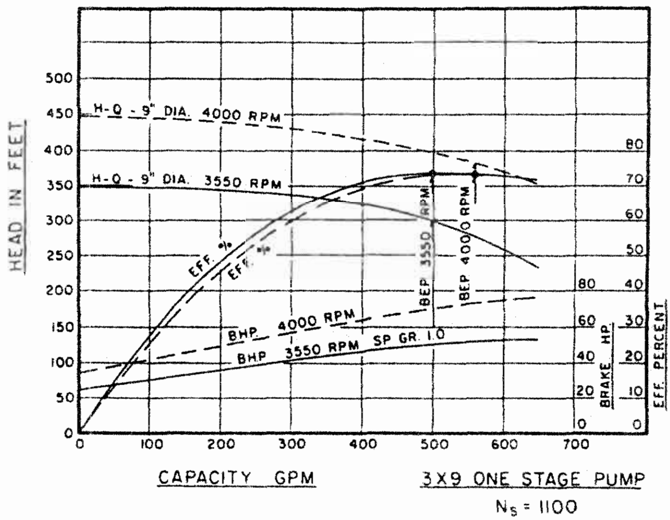


Figure 2-1. New pump factored from model pump—different speed.

Table 2-3
Tabulated Performance at 3,550 RPM

GPM	H(ft)	Eff. %	bhp
0	350	0	25
100	349	28	31
200	345	48	36
300	337	52	42
400	325	70	46
500	300	74	51
600	260	73	54
650	235	72	53

Results are tabulated in Table 2-4 and shown as a dotted line, in Figure 2-1. Note that the pump efficiency remains the same with the increase in speed.

Specific Speed Charts

We have prepared a nomograph (Figure 2-2) relating pump specific speed and suction specific speed to capacity, speed, and head. The nomograph is very simple to use: Locate capacity at the bottom of the graph, go vertically to the rotating speed, horizontally to TDH, and vertically to

Table 2-4
Tabulated Performance at 4,000 RPM

GPM	H(ft)	Eff. %	bhp
0	445	0	37
113	443	28	45
226	438	48	52
337	427	52	60
452	412	70	66
565	381	74	73
678	330	73	77
732	298	72	76

the pump specific speed. To obtain suction specific speed continue from the rotating speed to NPSHR and vertically to the suction specific speed. Pump specific speed is the same for either single-suction or double-suction designs.

For estimating the expected pump efficiencies at the best efficiency points, many textbooks have plotted charts showing efficiency as a function of specific speed (N_s) and capacity (GPM). We have prepared similar charts, but ours are based on test results of many different types of pumps and many years of experience.

Figure 2-3 shows efficiencies vs. specific speed as applied to end-suction process pumps (API-types). Figure 2-4 shows them as applied to single-stage double-suction pumps, and Figure 2-5 shows them as applied to double-volute-type horizontally split multi-stage pumps.

Figure 2-5 is based on competitive data. It is interesting to note that although the specific speed of multi-stage pumps stays within a rather narrow range, the pump efficiencies are very high, equal almost to those of the double-suction pumps. The data shown are based on pumps having six stages or less and operating at 3,560 RPM. For additional stages or higher speed, horsepower requirements may dictate an increase in shaft size. This has a negative effect on pump performance and the efficiency shown will be reduced.

As can be seen, efficiency increases very rapidly up to N_s 2,000, stays reasonably constant up to N_s 3,500, and after that begins to fall off slowly. The drop at high specific speeds is explained by the fact that hydraulic friction and shock losses for high specific speed (low head) pumps contribute a greater percentage of total head than for low specific speed (high head) pumps. The drop at low specific speeds is explained by the fact that pump mechanical losses do not vary much over the range of specific speeds and are therefore a greater percentage of total power consumption at the lower specific speeds.

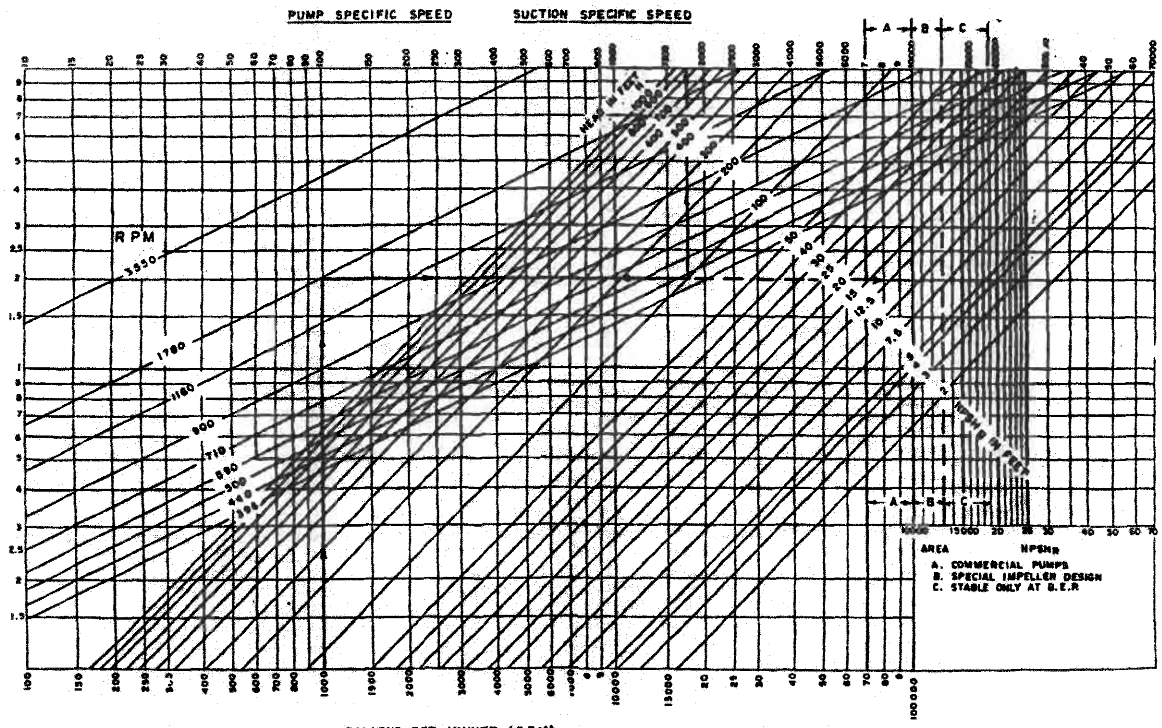


Figure 2-2. Specific speed and suction specific speed nomograph.

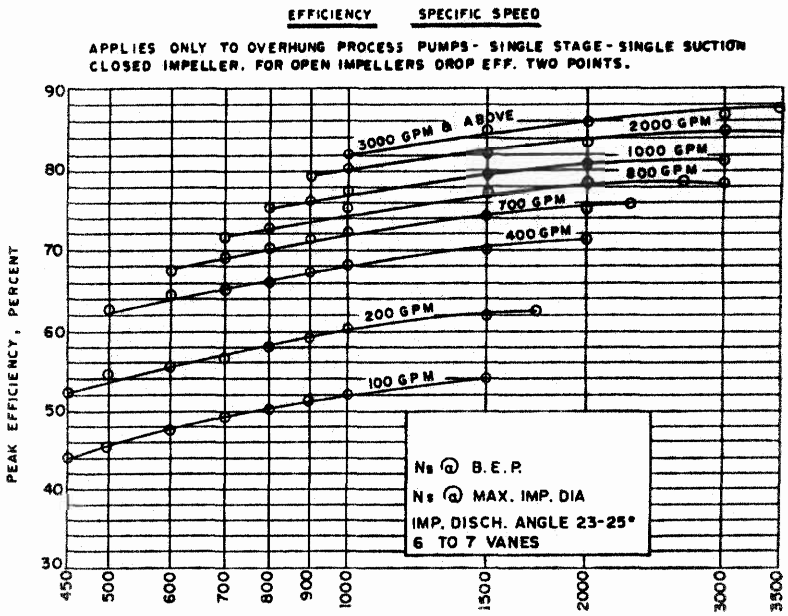


Figure 2-3. Efficiency for overhung process pumps.

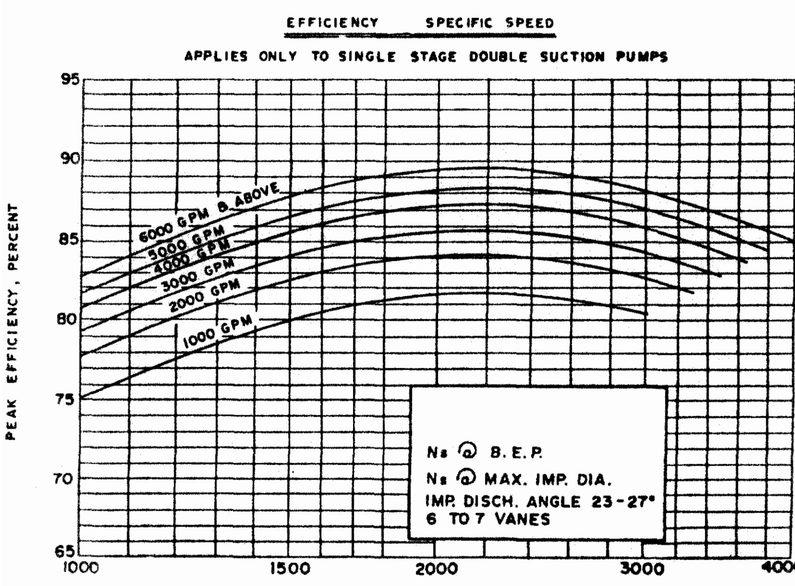


Figure 2-4. Efficiency for single-stage double-suction pumps.

Correction for Impeller Trim

The affinity laws described earlier require correction when performance is being figured on an impeller diameter change. Test results have shown that there is a discrepancy between the calculated impeller diame-

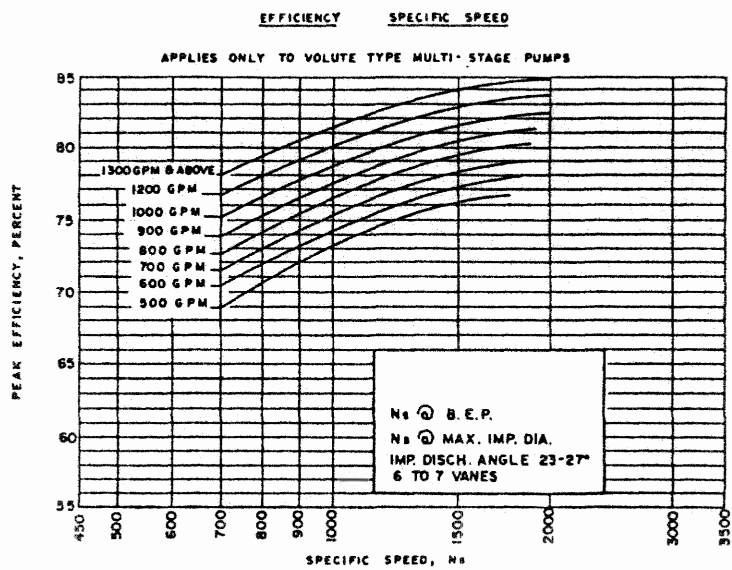


Figure 2-5. Efficiency for double-volute-type horizontally split multi-stage pumps.

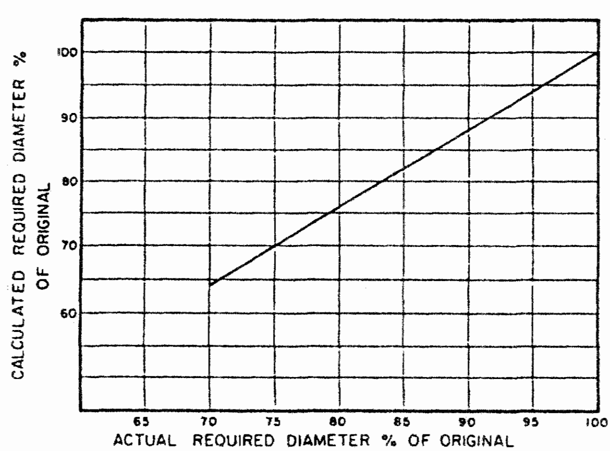


Figure 2-6. Impeller trim correction.

ter and the achieved performance. The larger the impeller cut, the larger the discrepancy as shown in Figure 2-6.

Example

What impeller trim is required on a 7-in. impeller to reduce head from 135 ft to 90 ft?

Step 1.

From affinity laws:

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

$$90 = 135\left(\frac{D_2}{7}\right)^2$$

$$D_2 = 5.72 \text{ in.}$$

Calculated percent of original diameter = 5.72/7 = .82

Step 2.

Establish correction factor:

From Figure 2-6 calculated diameter .82 = Actual required diameter .84.

Trim diameter = 7 × .84 = 5.88 in.

Impeller trims less than 80% of original diameter should be avoided since they result in a considerable drop in efficiency and might create unstable pump performance. Also, for pumps of high specific speed (2,500–4,000), impeller trim should be limited to 90% of original diameter. This is due to possible hydraulic problems associated with inadequate vane overlap.

Model Law

Another index related to specific speed is the pump modeling law. The "model law" is not very well known and usually applies to very large pumps used in hydroelectric applications. It states that two geometrically similar pumps working against the same head will have similar flow conditions (same velocities in every pump section) if they run at speeds inversely proportional to their size, and in that case their capacity will vary with the square of their size. This is easily understood if we realize that the peripheral velocities, which are the product of impeller diameter and RPM, will be the same for the two pumps if the diameter increase is inversely proportional to the RPM increase. The head, being proportional to the square of the peripheral velocity, will also be the same. If the velocities are the same, the capacities will be proportional to the areas, i.e. to the square of the linear dimensions.

As a corollary, the linear dimensions of similar pumps working against the same head will change in direct proportion to the ratio of the square root of their capacities, and the RPM in inverse proportion to the same ratio. This permits selection of a model pump for testing as an alternative to building a full-size prototype. The selected model must agree with the following relationship:

$$\frac{N_1}{N} = \left(\frac{D}{D_1}\right)\left(\frac{H_1}{H}\right)^{.5}$$

$$\frac{Q_1}{Q} = \left(\frac{D_1}{D}\right)^2\left(\frac{H_1}{H}\right)^{.5}$$

$$\frac{1-\eta_1}{1-\eta} = \left(\frac{D}{D_1}\right)^n$$

where D_1, N_1, H_1, and η_1, are model diameter, speed, head, and efficiency, and D, N, H and η, are prototype diameter, speed, head, and efficiency. n will vary between zero and 0.26, depending on relative surface roughness.

Other considerations in the selection of a model are:

1. Head of the model pump is normally the same as the prototype head. However, successful model testing has been conducted with model head as low as 80% prototype head.
2. Minimum diameter of the model impeller should be 12 in.
3. Model speed should be such that the specific speed remains the same as that of the prototype.
4. For meaningful evaluation prototype pump and model pump must be geometrically similar and flow through both kinematically similar.
5. Suction requirements of model and prototype should give the same value of sigma (see Chapter 9).

An example of model selection is described in detail in the *Hydraulic Institutes Standards, 14th Edition.*

Factoring Law

In addition to the affinity law and model law, there are some other principles of similarity that are very useful to the pump designer. The "factoring law" can be used to design a new pump that has the same specific speed and running speed as the existing model but which is larger or smaller in size. Factoring the new pump can be determined by using capacity or impeller diameter ratio. From BEP condition, a new pump head and flow are calculated from:

$$Q_l = Q_m \times f^3$$
$$H_l = H_m \times f^2$$

$$\frac{D_l}{D_m} = f$$

where Q_l = Capacity of new pump
H_l = Head of new pump
D_l = Impeller diameter of new pump
Q_m = Capacity of model pump
H_m = Head of model pump
D_m = Impeller diameter of model pump

Thus, the capacity will change with factor cubed (f^3). The head and all areas will change with factor squared (f^2), and all linear dimensions will change directly with factor (f).

The specific speed of the model pump should be the same as the new pump or within ±10% of the new pump's specific speed. The model pump should ideally be of the same type as the new pump.

Example

If we take the 3-in. × 9-in. pump shown in solid lines on Figure 2-1, with a performance at the BEP of 500 GPM; 300 ft; 74% efficiency; 55 maximum HP; 3,550 RPM; and 9-in. impeller diameter, and increase pump size to 700 GPM at BEP and running at 3,550 RPM, the new performance would be:

$$\frac{Q_l}{Q_m} = \frac{700}{500} = f^3 = 1.40$$

$$f^2 = 1.25$$

$$f = 1.12$$

Table 2-5
Tabulated Performance of Model Pump
(Model 9-In. Diameter Impeller)

Q_m	H_m	Eff.$_m$ %	bhp_m
0	350	0	25
100	349	28	31
200	345	48	36
300	337	52	42
400	325	70	46
500	300	74	57
600	260	73	54
650	235	72	54

Table 2-6
Tabulated Performance of New Pump
(New Pump 10⅛-In. Diameter Impeller)

Q_1	H_1	Eff.$_1$ %	bhp_1
0	437	0	50
140	436	28	55
280	431	49	62
420	421	53	84
560	406	71	81
700	375	75	88
840	325	74	93
910	293	72	94

Applying these factors to the model, the new pump performance at BEP will be: 700 GPM; 375 ft; 75% efficiency; 94 maximum HP; 3,550 RPM; and 10⅛-in. impeller diameter.

To obtain complete H-Q performance refer to Tables 2-5 and 2-6.

The new pump size will be a 4-in. × 6-in. × 10-in. with performance as shown in Figure 2-7.

All major pump manufacturers have in their files records of pump performance tests covering a wide range of specific speeds. Each test can be used as a model to predict new pump performance and to design same. In the majority of cases, the required model specific speed can be found having the same running speed as required by the new pump. There are cases, however, where the model pump running speed is different than required by pump in question.

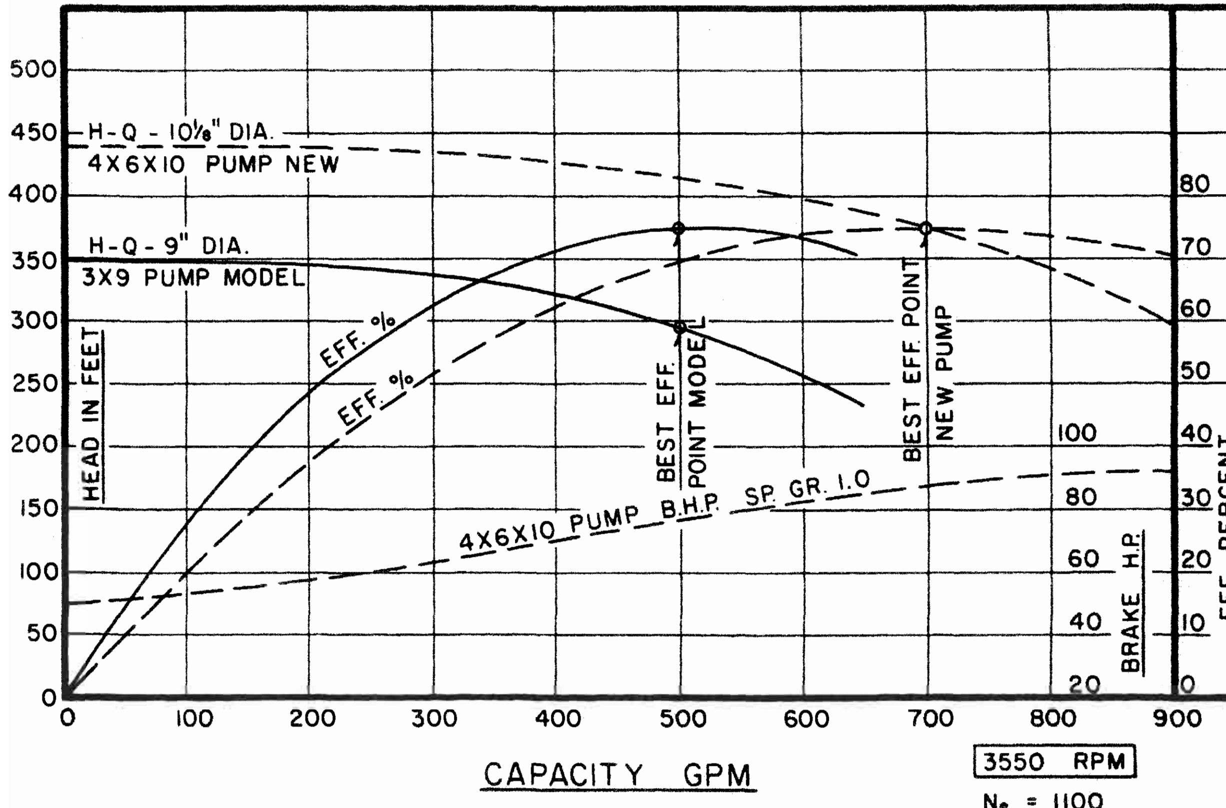


Figure 2-7. New pump factored from model pump—same speed.

If a new pump has the same specific speed (N_s) as the model but is to run at a different rotating speed, head, and flow, the new pump performance will be related to the model by:

$$Q_l = Q_m \times f^3 \times \frac{N_l}{N_m} \tag{2-1}$$

$$H_l = H_m \times f^2 \times \left(\frac{N_l}{N_m}\right)^2 \tag{2-2}$$

Example

A new pump is required for 750 GPM, 600-ft head, operating at 5,000 RPM. The calculated specific speed is 1,100. The model chosen is shown in Figure 2-1, peaks at 500 GPM, 300 ft, and operates at 3,550 RPM. Specific speed = 1,100.

From Equation 2-1:

$$f^3 = \frac{750 \times 3{,}550}{500 \times 5{,}000} = 1.07$$

$$f^2 = 1.05$$

$$f = 1.02$$

Performance at 5,000 RPM can be calculated by applying the following factors to the 3,550 RPM performance,
From Equation 2-1:

$$Q_l = Q_m \times 1.07 \times \left(\frac{5{,}000}{3{,}550}\right)$$

$$= Q_m \times 1.5$$

From Equation 2-2:

$$H_l = H_m \times 1.05 \times \left(\frac{5{,}000}{3{,}550}\right)^2$$

$$= H_m \times 2.07$$

Table 2-7
Tabulated Performance at 3,550 RPM

GPM	H(ft)	Eff. %
0	350	0
100	349	28
200	345	48
300	337	52
400	325	70
500	300	74
600	260	73
650	235	72

Table 2-8
Tabulated Performance at 5,000 RPM

GPM	H(ft)	Eff. %	bhp
0	700	0	91
150	698	28	94.4
300	690	48	109
450	674	52	142
600	650	70	140
750	600	74	153
900	520	73	161
975	470	72	160

For complete performance conversion, see Tables 2-7 and 2-8. The hydraulic performance of the new 4-in. × 6-in. × 9-in. pump operating at 5,000 RPM is shown in Figure 2-8.

Conclusion

There is no question that specific speed is the prime parameter for evaluating design of centrifugal pumps, evaluating pump selections, and predicting possible field problems. It is obvious, however, that no single parameter can relate to all aspects of final pump design. Different pump specifications covering a wide variety of applications force the designer to consider additional factors that may have an unfavorable effect on pump hydraulic performance. Predicted performance can be affected by any of the following:

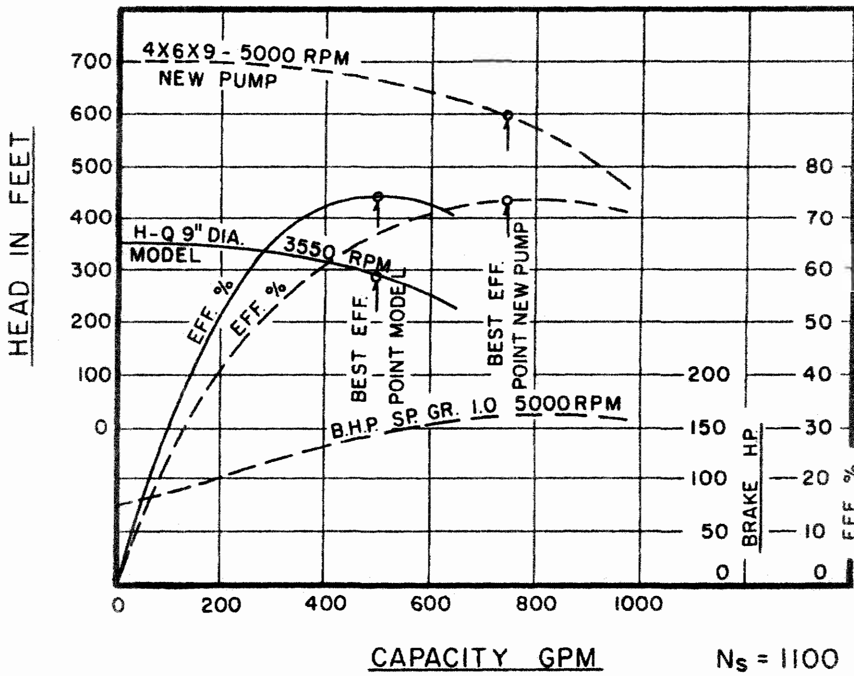
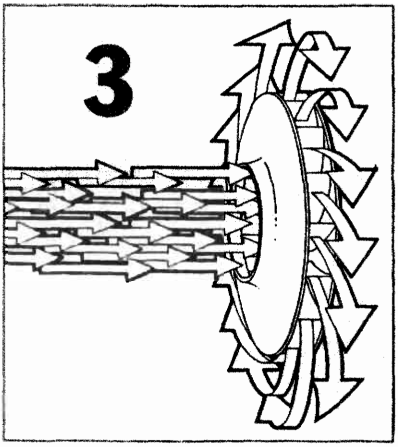


Figure 2-8. Performance change with speed change.

- *Mechanical Considerations*
  High horsepower/large shaft diameter
  High suction pressure
  Operating speed
  Operating temperature
  Running clearances
- *Pump Liquid*
  Slurries
  Abrasives
  High viscosity
  Dissolved gases
- *System Considerations*
  NPSHA
  Suction piping arrangement
  Discharge piping arrangement
  Shape of H-Q curve
  Runout conditions
  Vibration limits
  Noise limits

Pumps selected as models must have reliable hydraulic performance based on accurate instrumentation. Unreliable model tests used as a basis for new development will lead to total disaster. It is strongly suggested that the selected model pump be retested to verify its hydraulic performance.

In the following chapters of this book "specific speed" will be referred to with many additional charts, curves, and technical plots.



3 Impeller Design

In this chapter detailed information for designing a centrifugal pump impeller will be presented. This information will apply to a new design where a model in the same specific speed is not available. The design factors shown are based on a theoretical approach and many years of collecting thousands of performance tests in various specific speeds.

The following example, will demonstrate the procedure necessary to design and layout a new impeller.

Example

The requirements for a new pump are shown on curve Figure 3-1, which at Best Efficiency Point (BEP) is:

2,100 GPM—450 ft—3,600 RPM—Liquid—Water

Step 1: Calculate pump specific speed.

$$N_s = \frac{RPM \times (GPM)^{.5}}{(H)^{.75}}$$

$$= \frac{3{,}600 \times (2{,}100)^{.5}}{(450)^{.75}} = 1{,}688$$

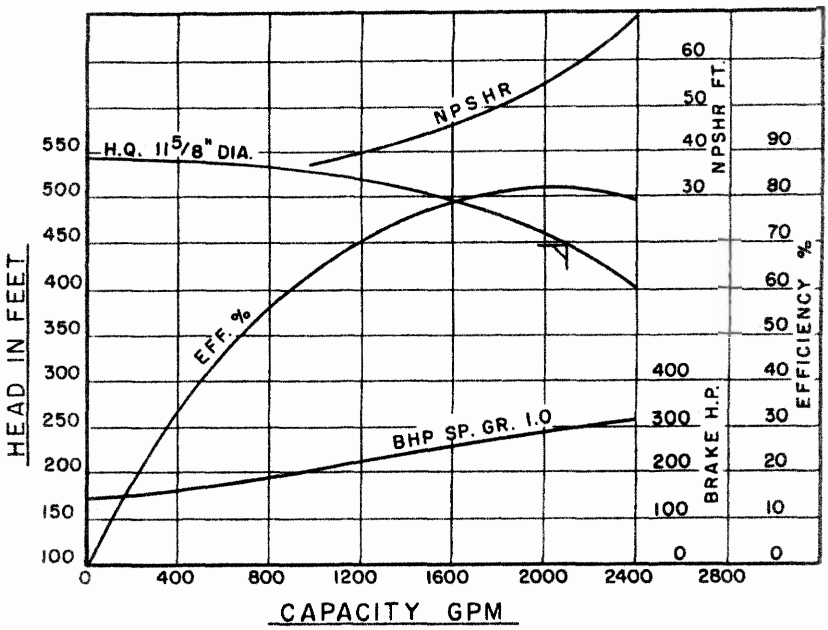


Figure 3-1. Required performance of new pump.

Step 2: Select vane number and discharge angle.

The desired head rise from BEP to zero GPM is 20% continuously rising (Figure 3-1). To produce this head rise, the impeller should be designed with six equally spaced vanes having a 25° discharge angle (Figure 3-2).

Step 3: Calculate impeller diameter.

From Figure 3-3:

Head constant $K_u = 1.075$

$$D_2 = \frac{1{,}840 \times K_u \times H^{.5}}{RPM}$$

$$= \frac{1{,}840 \times 1.075 \times (450)^{.5}}{3{,}600}$$

$$= 11.66 \text{ in. (say } 11^5/_8 \text{ in.)}$$

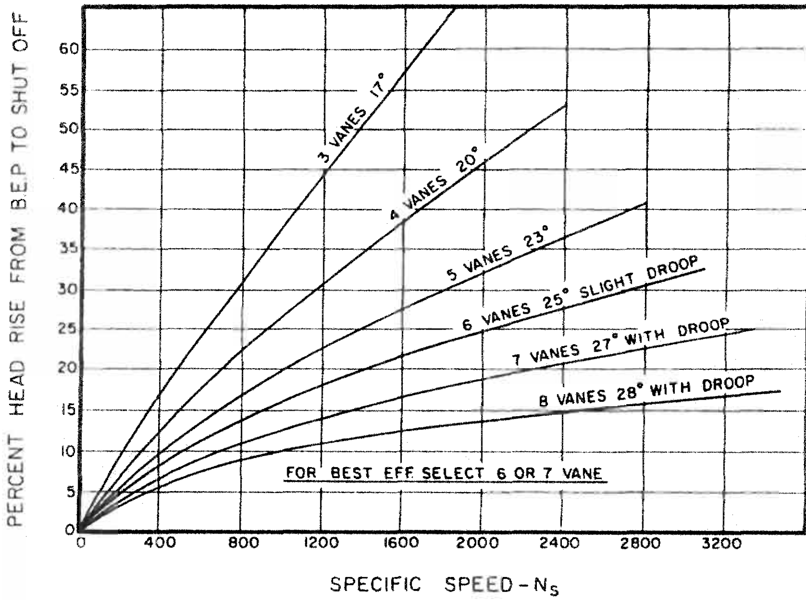


Figure 3-2. Percent head rise.

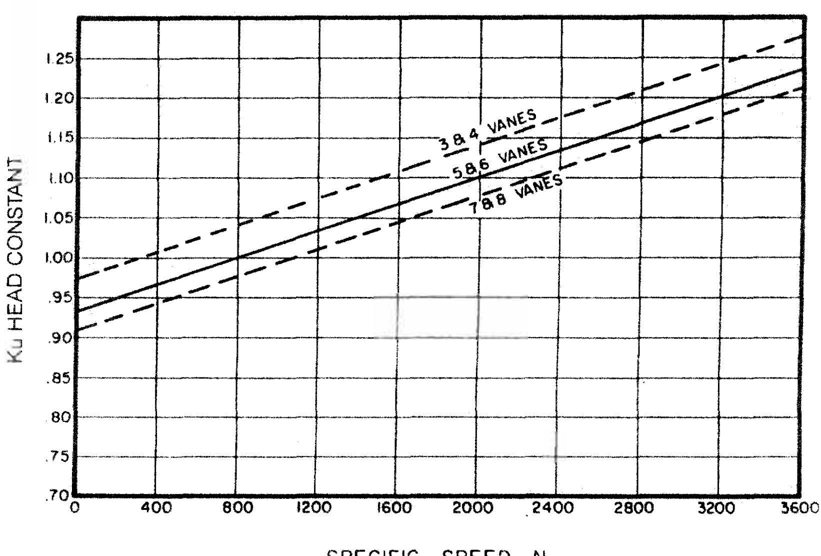


Figure 3-3. Head constant.

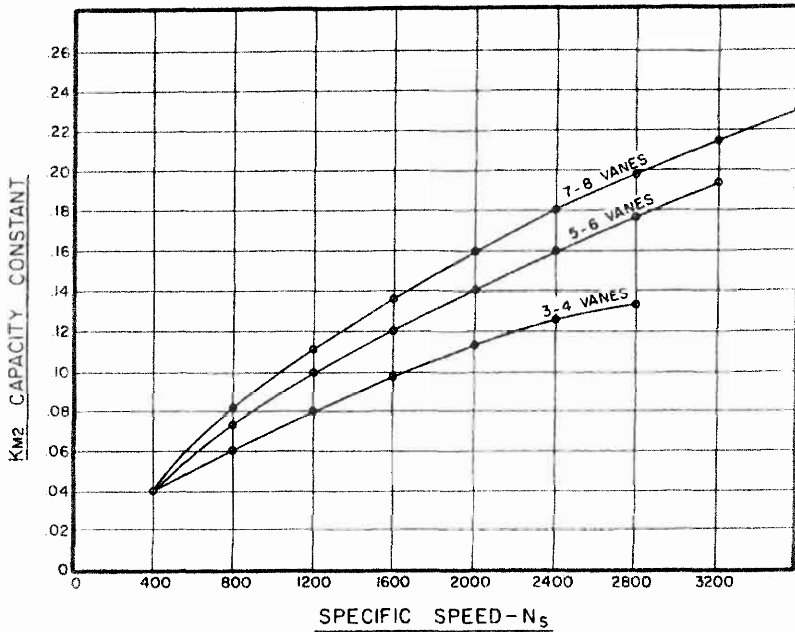


Figure 3-4. Capacity constant.

Step 4: Calculate impeller width b_2.

From Figure 3-4:

$$K_{m2} = .125$$

$$C_{m2} = K_{m2} \times (2gH)^{.5}$$

$$= .125 \times 170 = 21.3 \text{ ft/sec}$$

$$b_2 = \frac{GPM \times .321}{C_{m2} \times (D_2\pi - ZS_u)}$$

Estimated S_u = ½ in. (This will be confirmed during vane development and the calculation repeated if necessary.)

$$b_2 = \frac{2{,}100 \times .321}{21.3(11.66\pi - 6 \times .5)} = 1.09 \text{ in.}$$

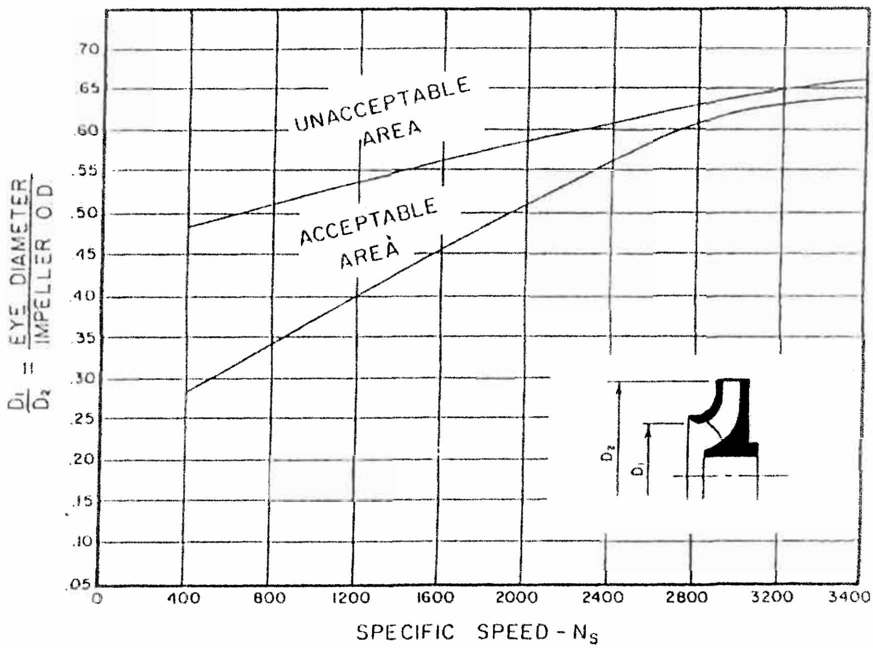


Figure 3-5. Impeller eye diameter/outside diameter ratio.

Step 5: Determine eye diameter.

From Figure 3-5:

$D_1/D_2 = .47$

$D_1 = 11.66 \times .47 = 5.5$ in.

Step 6: Determine shaft diameter under impeller eye.

Methods for calculating shaft diameter are discussed in detail under shaft design. For this exercise, the shaft diameter under the impeller eye is assumed to be 2 in.

Step 7: Estimate impeller eye area.

Eye area = Area at impeller eye − shaft area

= 23.76 − 3.1 = 20.66 sq in.

Step 8: Estimate NPSHR.

From Figure 3-6:

$$C_{m1} = \frac{2{,}100 \times .321}{20.66} = 32.63 \text{ ft/sec}$$

$$U_t = \frac{5.5 \times 3{,}600}{229} = 86.5 \text{ ft/sec}$$

From Figure 3-6:

NPSHR = 59 ft

$$N_{ss} = \frac{3{,}600 \times (2{,}100)^{.5}}{(59)^{.75}} = 7{,}749$$

Step 9: Determine volute parameters.

To finalize the impeller design, we must consider any mechanical limitations of the pump casing. Designing the impeller alone is not sufficient as we must see how it physically relates to the volute area, cutwater diameter, volute width, etc.

- *Volute area.* Figure 3-7 shows a number of curves for volute velocity constant K_3. These represent the statistical gathering efforts of a number of major pump companies. Figure 3-8 shows the average of these curves and is recommended for estimating volute area.

$$\text{Volute area } A_8 = \frac{0.04 \times \text{GPM}}{K_3 \times (H)^{.5}}$$

$$= \frac{.04 \times 2{,}100}{.365 \times (450)^{.5}}$$

$$= 10.85 \text{ sq in.}$$

The calculated volute area is the final area for a single volute pump. For double-volute pumps, this area should be divided by 2, and for diffuser type pumps it should be divided by the number of vanes in the diffuser casing.

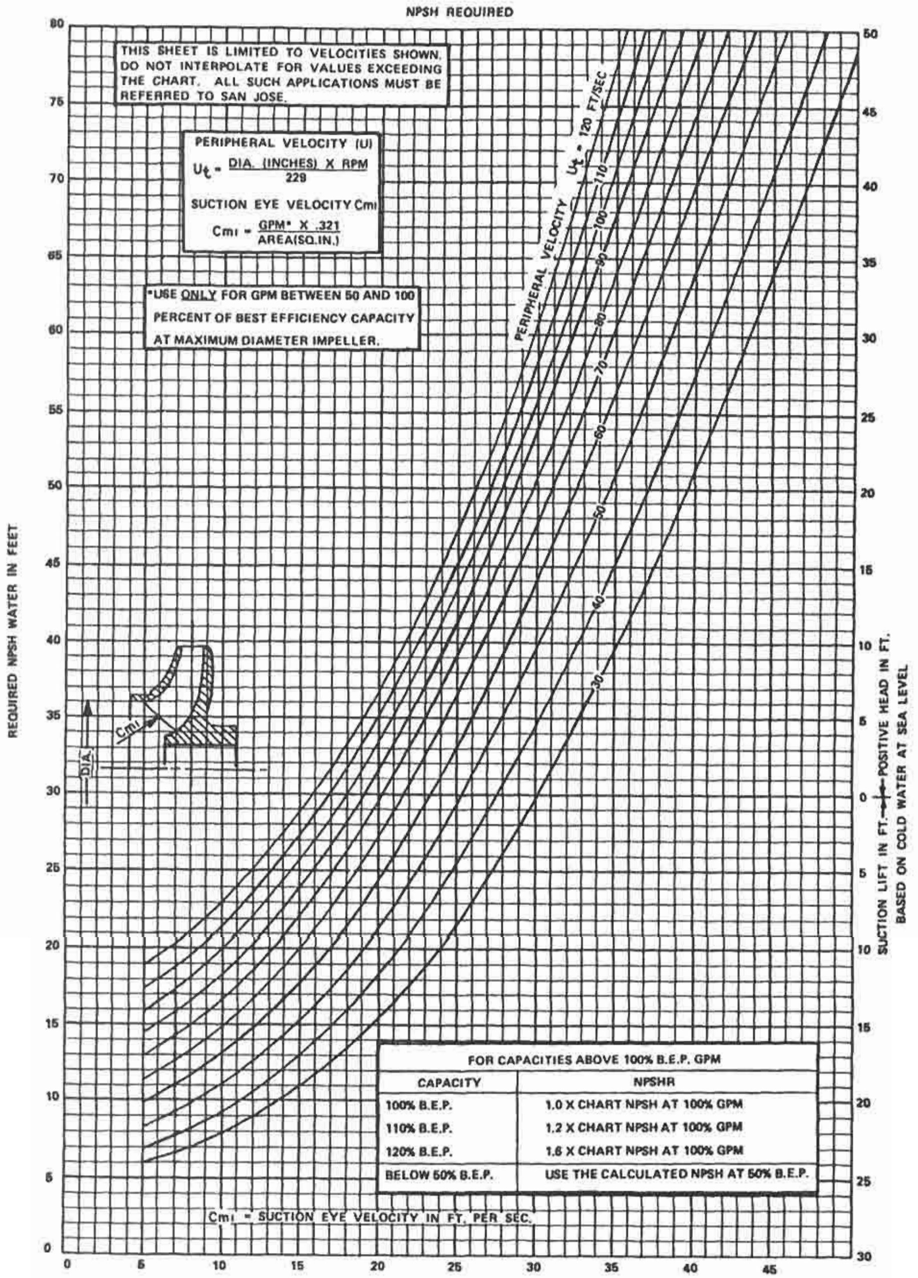


Figure 3-6. NPSHR prediction chart.

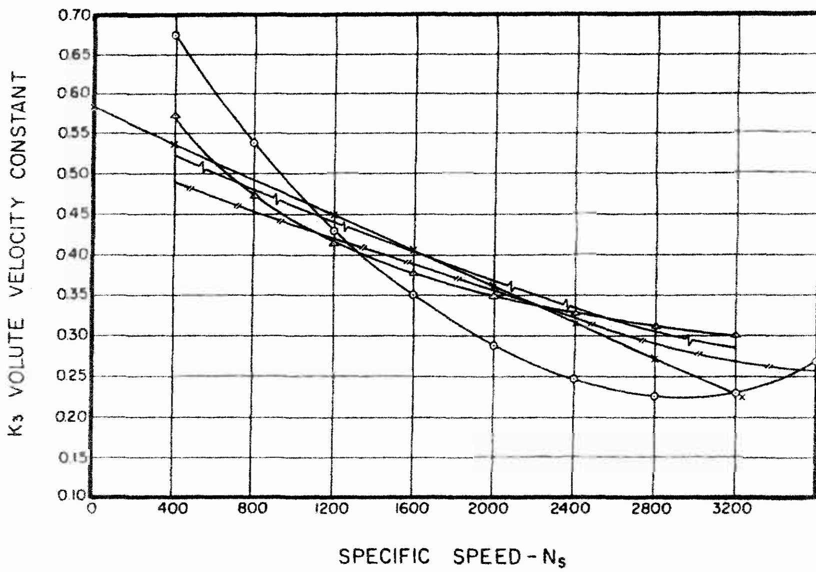


Figure 3-7. Volute velocity constant (data acquisition of different pump companies).

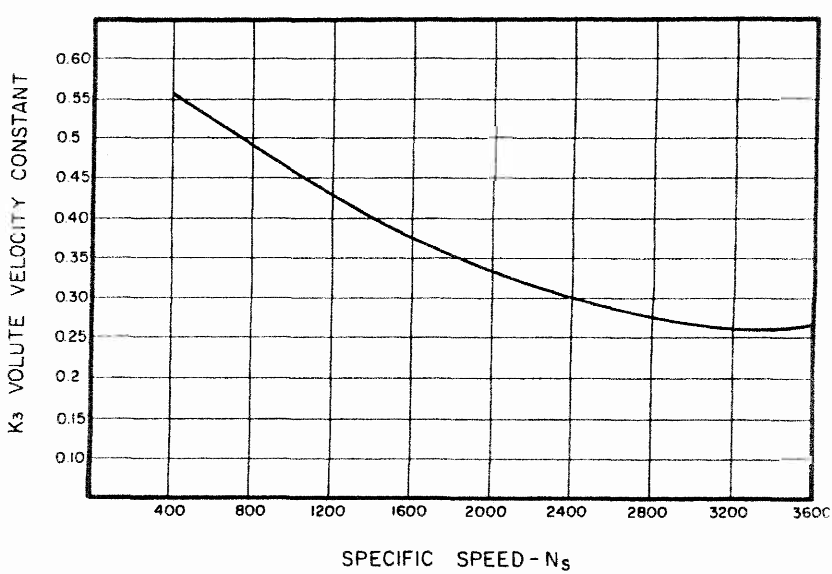


Figure 3-8. Volute velocity constant (author's recommendation).

- *Establish volute width.* In determing the width of the volute, the need to accommodate impellers of different diameter and b_2 must be considered. Distance from the impeller shroud to the stationary casing should be sufficient to allow for casting inaccuracies yet still maintain a satisfactory minimum end play. The values shown in Table 3-1 reflect those published by Stepanoff and are reasonable guidelines.

  Volute width $= 1.09 \times 1.75 = 1.9$ in. (say 2 in.)
- *Establish cutwater diameter.* A minimum gap must be maintained between the impeller diameter and volute lip to prevent noise, pulsation, and vibration, particularly at vane passing frequency. From Table 3-2, $D_3 = D_2 \times 1.07 = 11^5/_8 \times 1.07 = 12^7/_{16}$ in.

  Impeller design data shown in this chapter are applicable to:

- Specific speeds from 400 to 3,600.
- Open or closed impellers.
- Volute-type or diffuser-type casings.
- Single- or multi-stage units.
- Vertical or horizontal pumps.
- Single suction or double suction impellers.

In establishing discharge geometry for double suction impellers, use factors as given in this chapter. Do not divide the specific speed by

Table 3-1
Guidelines for Volute Width

Volute Width b_3	Specific Speed N_s
2.0 b_2	< 1,000
1.75 b_2	1,000–3,000
1.6 b_2	> 3,000

Table 3-2
Guidelines for Cutwater Diameter

Specific Speed N_s	Cutwater Diameter D_3
600–1000	$D_2 \times 1.05$
1000–1500	$D_2 \times 1.06$
1500–2500	$D_2 \times 1.07$
2500–4000	$D_2 \times 1.09$

$(2)^{.5}$, as final results will not be accurate. On the impeller suction side however, divide the inlet flow and eye area by 2 for each side of the impeller centerline.

Impeller Layout

Development of Impeller Profile (Plan View)

Lay out the impeller width b_2 at full diameter (Figure 3-9). Begin to develop the hub and shroud profiles by expanding this width by approximately 5° on each side of the vertical center line toward the suction eye. Complete these profiles by ending in such a way as to produce the required eye area. The area change from eye to discharge should be gradual. To minimize axial and radial thrust, make the impeller as symmetrical as possible about its vertical centerline.

Development of Impeller End View

Draw a circle to the full impeller diameter. Divide the circle into several even-degree segments. In this case 15° segments have been chosen (Points 1–10). The smaller the segments, the more accurate the vane layout will be.

Impeller Inlet Angles

Inlet angles are established from a layout of the velocity triangle, which shows the various component velocities of flow, entering the impeller (Figure 3-10). The vector connecting U_t and C_{M1} represents the angle of flow θ. Vane angle B_1 is drawn to intersect P_{S1} and should be greater than θ to allow for recirculated flow and nonuniform velocity. For reasonable design the prerotation angle should not exceed 30° and for optimum NPSH it is recommended that P_{S1} = 1.05 to 1.2 times C_{M1}.

Development of Impeller Vane

Begin the vane development by drawing a line equal to the discharge angle. On the end view estimate and locate the distance a_r. Transmit a_r to discharge angle line on the vane development establishing the distance a. Transmit a to the front shroud on the impeller profile (plan view). Measure R_2, then scribe R_2 on the end view, locating Point 2. Check to see if the estimated a_r was approximately correct, making a new estimate and repeating the process if necessary. Continue this same procedure to the minimum impeller cut diameter, in this case Point 4. Complete the vane development for the shroud contour by drawing several lines equal to the

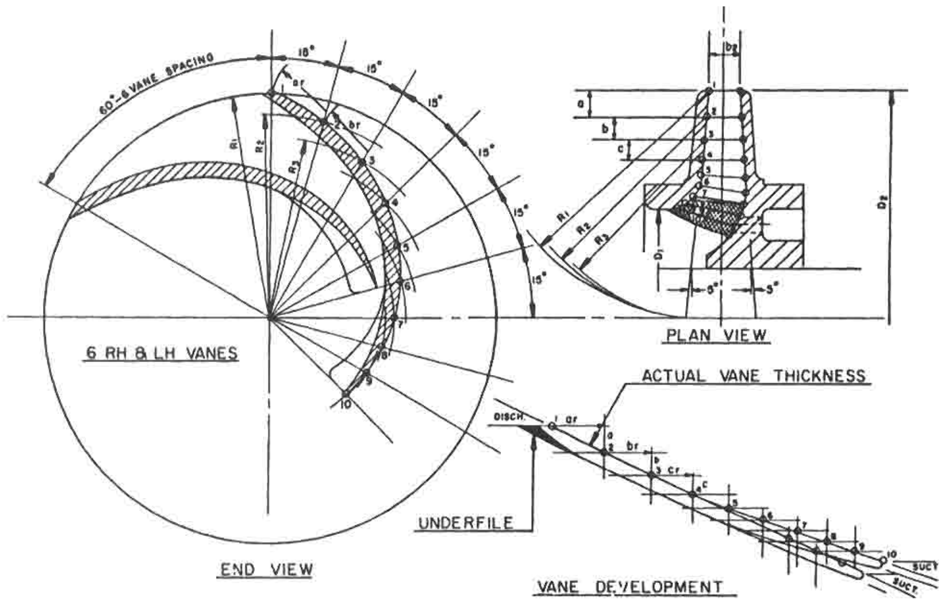


Figure 3-9. Impeller layout.

suction entrance angles to establish match between discharge angle and suction angles. Once a match is established with a smooth curve, lay out the remaining vane points until the eye diameter is reached, in this case at Point 10.

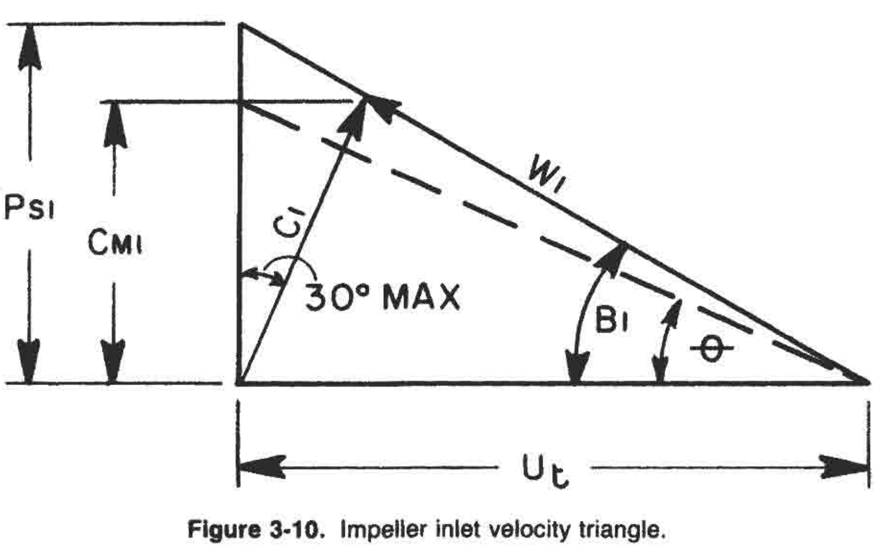


Figure 3-10. Impeller inlet velocity triangle.

The vane development and layout for the back shroud is done in the same manner, taking into account the required hub vane angle. This procedure should result in Point 10 reaching the estimated hub vane diameter. If this diameter is missed by more than 1/4-in., the hub vane angle should be changed and the layout repeated.

Complete the vane layout by adding vane thickness (Figure 3-11) and indicating a slight underfile at the vane OD and a thinning at the suction. This underfile will produce higher head and improved efficiency. The vane spacing location of the second vane on the end view will be determined by the number of vanes.

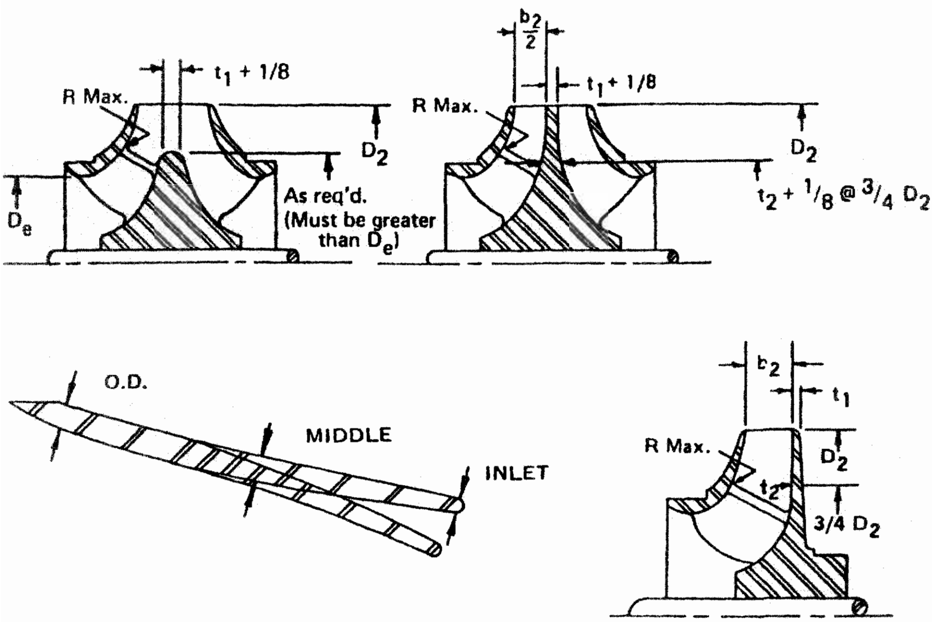
At this point it is necessary to check the ratio of the vane area A_v to the suction eye area A_e. Referring to Figure 3-12, this ratio should be 0.4 to 0.6. The area A_v is the area which is shaded on the plan view. If the area ratio falls outside the limits shown, it will be necessary to change the plan view profile and/or vane thickness.

Pumps in the higher specific speed range (N_s 1,000–5,000) have suction velocity triangles that dictate different suction vane angularity at the impeller hub and impeller eye. This difference in angularity restricts vane removal from the core box, requiring a segmented core and more costly patterns and casting. Hydraulically, it is very necessary, and we must learn to live with it. On low specific speeds however, (N_s 500–1000) a straight vane is acceptable and will not jeopardize hydraulic performance.

Design Suggestions

The following points should be kept in mind when designing an impeller.

- Standardize the relationship between the number of vanes and discharge vane angle. The relationship shown in Figure 3-13 is suggested. Number and angularity of vanes greatly affect H-Q pump performance. Standardization as shown will lead to more accurate performance prediction.
- Figure 3-14 shows the effect of the number of impeller vanes in the same casing, on H-Q performance. Please note that with less vanes, the lower is the BEP head and efficiency and steeper is the shape of H-Q curve.
- It is possible to maintain same BEP, Q, and efficiency by increasing b_2 where number of vanes and angularity are reduced (Figure 3-15). For best efficiency, the velocity ratio between volute and impeller peripheral must be maintained. Similarly for the same impeller diameter, vane number and discharge angle BEP will change with change in b_2 (Figure 3-16).
- Avoid using even number of vanes in double volute pumps.



IMP. DIA	MIN. VANE THICKNESS O.D.	MIDDLE	INLET	MIN. SHROUD THICKNESS t_1	t_2
> 6-11	7/32	5/16	1/8	1/8	3/16
> 11-15	1/4	3/8	3/16	5/32	5/16
> 15-19	9/32	7/16	3/16	3/16	3/8
> 19-23	5/16	1/2	1/4	7/32	7/16
> 23-31	11/32	9/16	1/4	1/4	9/16
> 31-35	3/8	5/8	5/16	5/16	5/8
> 35-39	7/16	11/16	5/16	3/8	11/16
> 39-50	1/2	3/4	3/8	7/16	3/4

Figure 3-11. Recommended minimum impeller vane and shroud thickness for castability (standard cast materials).

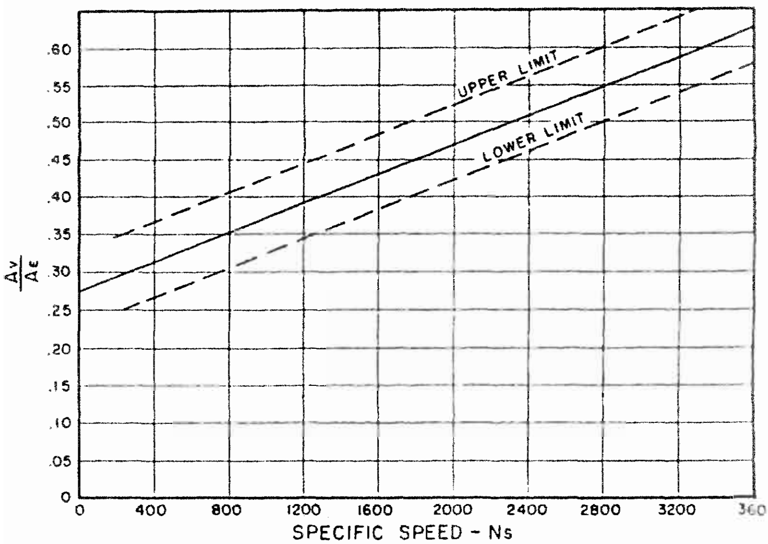


Figure 3-12. Area between vanes/eye area ratio.

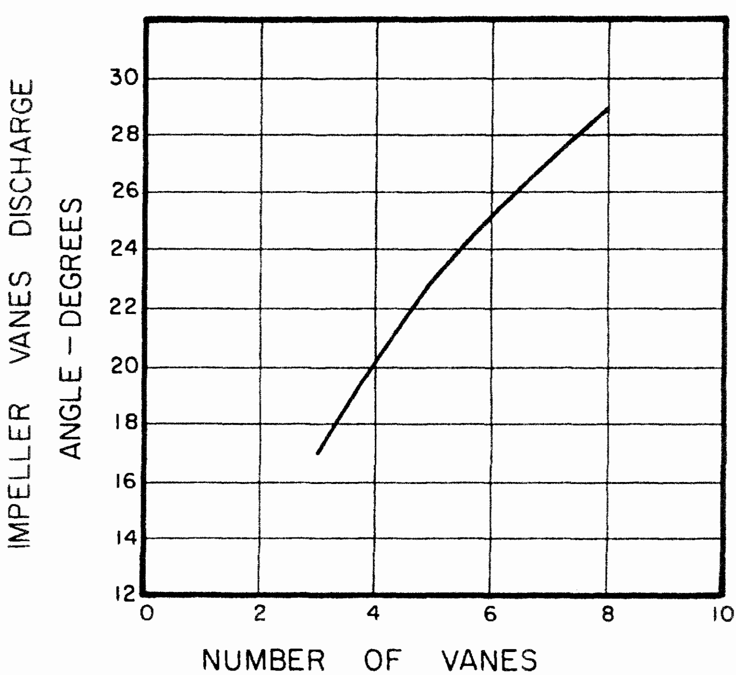


Figure 3-13. Suggested standard for vane number.

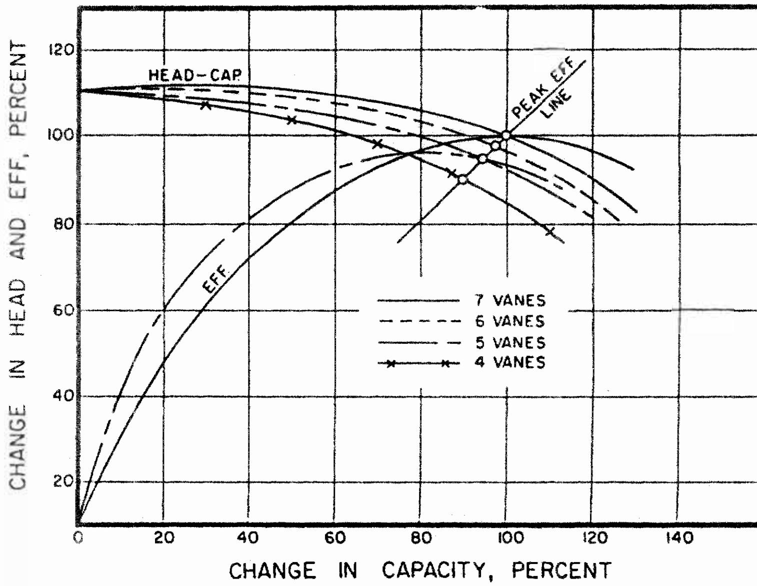


Figure 3-14. Influence of vane number on pump performance.

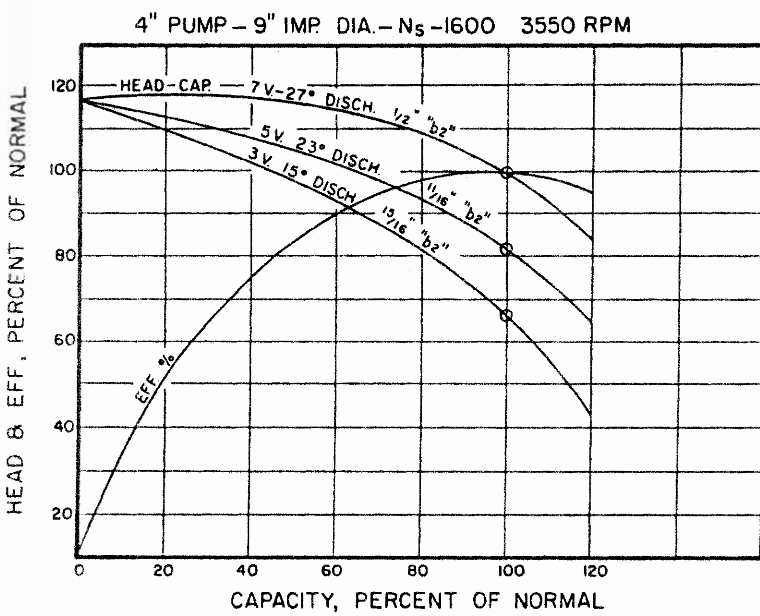


Figure 3-15. Constant BEP for different vane numbers and discharge angles by change in b_2.

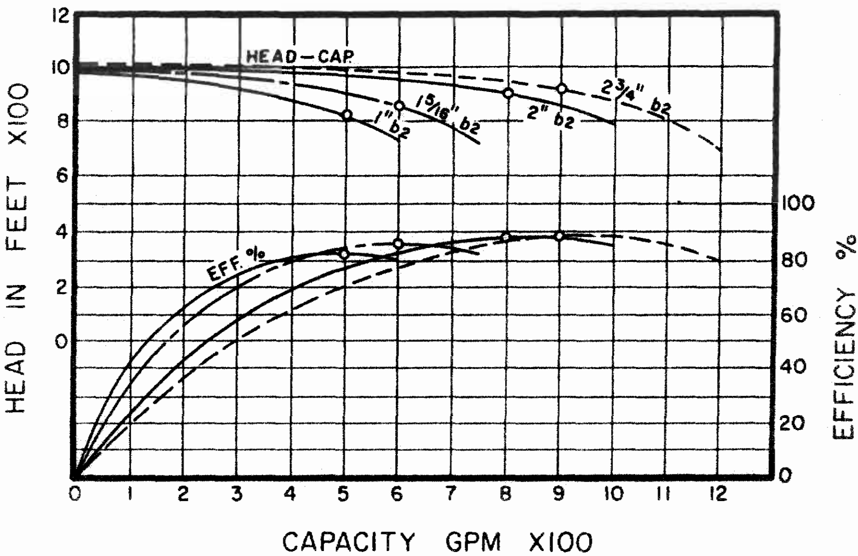


Figure 3-16. Influence of b_2 on pump performance.

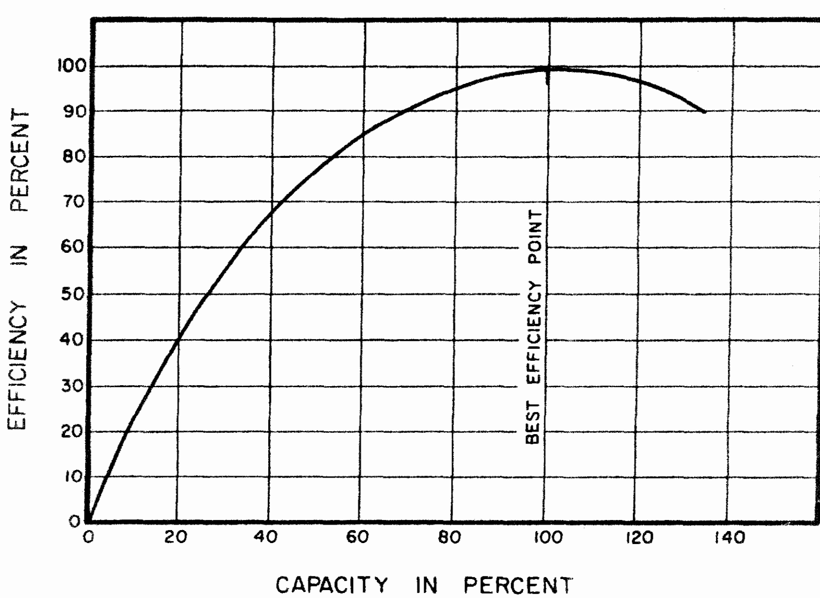
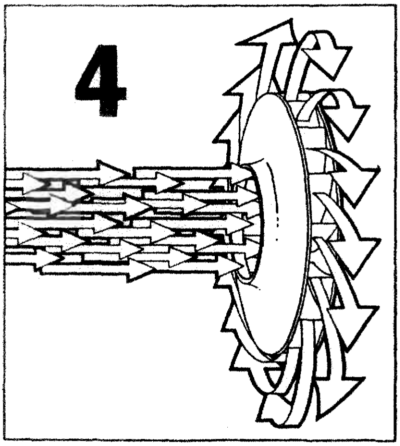


Figure 3-17. Predicting efficiency on both sides of BEP.

- To predict efficiency on both sides of BEP, use Figure 3-17.
- When designing a new pump, always have a performance curve and general sectional drawing.
- Keep neat and accurate design records.
- Try to use a symmetrical impeller to avoid excessive axial thrust.
- Consider mechanical requirements applicable to the new design.

Notation

K_u	Speed constant $= U_2/(2gH)^{.5}$
U_2	Impeller peripheral velocity (ft/sec) $= D_2 \times RPM/229$
g	Gravitational constant (32.2 ft/sec^2)
H	Impeller head (ft)
D_2	Impeller outside diameter (in.) $= 1840\ K_uH^{.5}/RPM$
D_1	Impeller eye diameter (in.)
K_{m2}	Capacity constant $= C_{m2}/(2gH)^{.5}$
C_{m2}	Radial velocity at impeller discharge (ft/sec) $= Q\ .321/A_2$
D_3	Volute cutwater diameter (in.)
A_2	Impeller discharge area (sq in.) $= (D_2\pi - ZS_u)b_2$
Z	Number of impeller vanes
S_u	Vane thickness at D_2 (in.)
b_2	Inside impeller width at D_2 (in.) $= Q\ .321/C_{m2}(D_2\pi - Z\ S_u)$
Q	GPM at BEP
K_3	Volute velocity constant $= C_3/(2gH)^{.5}$
C_3	Volute velocity (ft/sec.)
A_8	Volute throat area (sq. in.) $= 0.04\ Q/K_3(H)^{.5}$
C_{m1}	Average meridianal velocity at blade inlet (ft/sec) $= .321\ Q/Ae$
Ae	Impeller eye area at blade entry sq in.
w_1	Relative velocity of flow (ft/sec)
θ	Angle of flow approaching vane
C_1	Absolute velocity of flow (ft/sec)
P_{s1}	C_{m1}/R_1
R_1	Factor in determining B_1
B_1	Blade angle at outer radius of impeller eye
U_t	Peripheral velocity of impeller blade (ft/sec) $= D_1 \times RPM/229$
Av	Area between vanes at inlet sq in.



General Pump Design

It is not a difficult task to design a centrifugal pump; however, designing the right pump for a specific application related to a specific industry and service requires an extensive knowledge of hydraulics. Also required is experience with industrial specifications, end users, and contractors' special requirements and many years of practical experience in engineering and marketing.

The variables that exist for pump requirements are so numerous that the design of the right pump in the right service is a complex project. There is no such product as the "universal pump."

As an example, let us take a pump that is required to produce 500 GPM and 200-ft head, rotating at 2 or 4 pole speed. In any or all industries this hydraulic requirement exists; however, the mechanical specifications are entirely different for each and every industry. For instance, the type of pumps used in the pulp and paper industry are entirely different from pumps used in the petroleum industry, petrochemical industry, or chemical industry. Thus, the pump required to deliver 500 GPM and 200-ft head, will be different for each of the following applications:

- Slurry
- Boilerfeed
- Pipeline
- Nuclear
- Municipal
- Agricultural
- Marine
- Cargo

Mechanical variables include:

- Open or closed or semi-open impellers.
- Single-stage or multi-stage.
- Vertical or horizontal.
- With water jacket or without.
- Overhung or two-bearings design.
- Close-coupled or coupled units.
- Stiff shaft or flexible shaft design.
- Single volute, double volute, quad volute.
- Short elbow-type or turbine-type diffusers.
- Mechanical seals or packing.
- Stuffing boxes with bleed-offs or with clean flush injection.
- Ball, sleeve, or Kingsbury-type bearings.
- Oil rings, forced feed, oil mist, submerged or grease packed lubrication.

It can be seen from the variables listed that it is a complex job for a pump designer to design the right pump for the right environment.

After complete hydraulic and mechanical specifications are established, the designer should be ready for pump layout documents.

General pump design can be classified in the following categories:

1. Design a new pump to satisfy basic engineering requirements such as shape of H-Q curve, NPSHA, efficiency, etc.
2. Design a new pump to satisfy special applications such as boilerfeed, nuclear coolant, pipeline, large circulator, condensate, secondary recovery, etc.
3. Design a new line of pumps, such as API pumps, ANSI pumps, double-suction pumps, pulp and paper pumps, building trade pumps, boilerfeed pumps, etc.

Performance Chart

For pumps in any category, an overall performance chart should be prepared (if not available) as a first step in the design study. This chart will establish the flow and head for each pump, establish the number and size of pumps required to satisfy the range chart, and avoid overlap or gaps between pump sizes. Even if only one pump is required, the range chart should be confronted to be sure that the new pump fits into the overall planning.

Many old pump lines have poorly planned range charts, resulting in similar pump overlaps and uncovered gaps between pump sizes. Such a

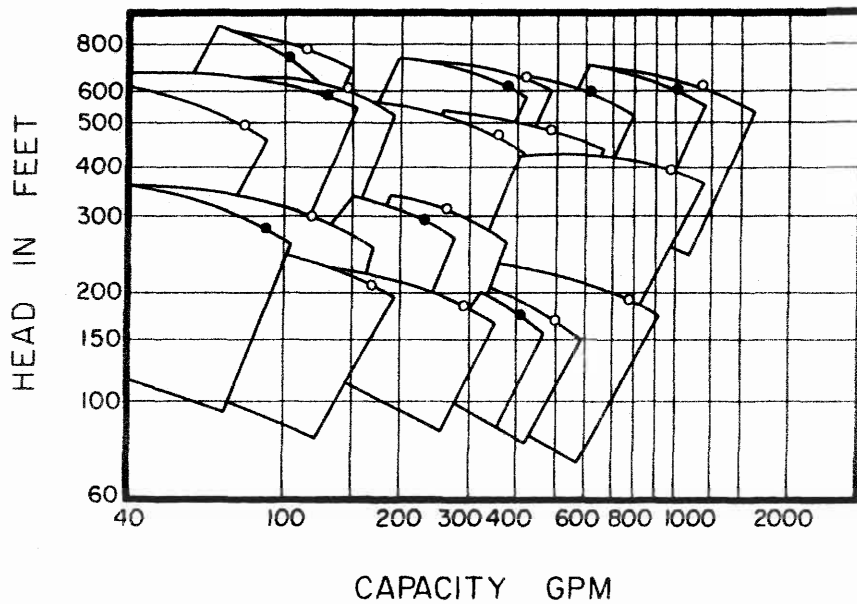


Figure 4-1. Poorly planned performance chart.

chart is shown in Figure 4-1. Black dots show pumps that could be eliminated, permitting a substantial reduction in inventory without hurting overall hydraulic performance. This type of chart is not recommended. A suggested properly layed out performance range chart is shown in Figure 4-2. Steps to develop this chart are now described.

Step 1

Establish BEP of lowest capacity and lowest head pump required. In this example, this is 86 GPM, 150-ft head, which is achieved by a 1-in. pump, and a 7-in. diameter impeller.

Step 2

Extend BEP capacity coverage at a constant 150-ft head by multiples of 1.75. Thus,

Second pump BEP = 86 GPM × 1.75 = 150 GPM.
Third pump BEP = 150 GPM × 1.75 = 260 GPM, etc.

In this manner, the base line BEP is established.

Step 3

Establish next size pump by multiplying each base line BEP by
- GPM × 1.75
- Head × 1.45

Example: Smallest pump on chart has BEP of 86 GPM and 150-ft head. Thus, BEP for next size pump = 86 × 1.75 = 150 GPM and 150 × 1.45 = 220 ft. Repeat this step for each base line BEP. As can be seen from the chart, the head of 220 ft now requires a 9-in. impeller.

Step 4

For all additional pump BEP's multiply preceding pump flow by 1.75 and head by 1.45.

The constants 1.75 and 1.45 are recommended for a well-planned performance chart following a number of constant specific speed lines. There are no gaps between pumps, each performance block can be covered by normal impeller trim, and there are no overlaps. The chart is also helpful to the designer. In this example, only six small pumps have to be

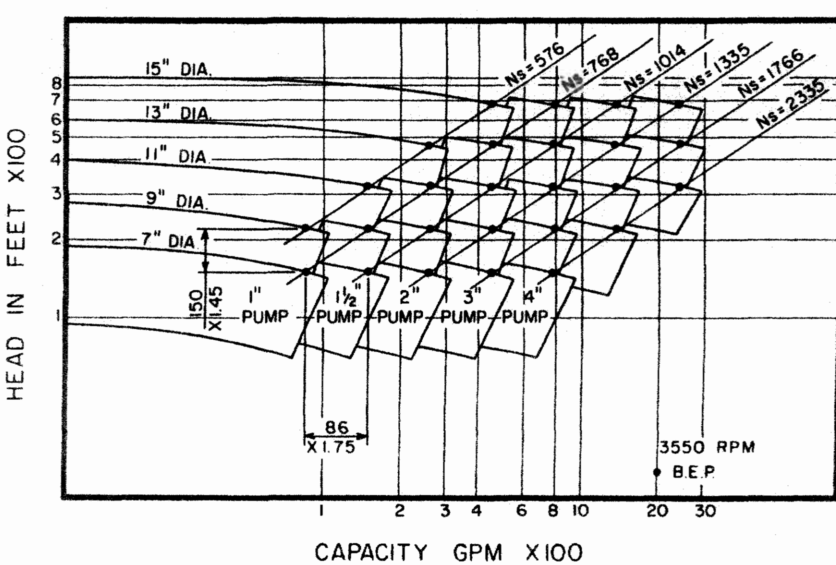


Figure 4-2. Recommended performance chart.

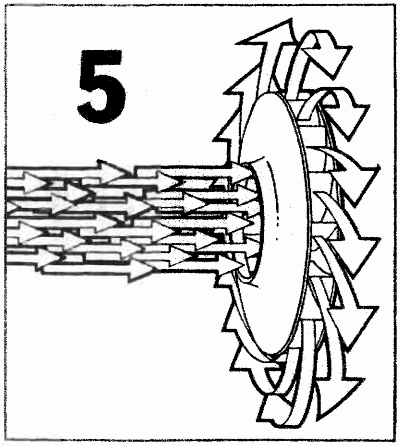
designed and test checked. The other sizes that follow specific speed lines can be factored up and their performance accurately predicted.
designed and test checked. The other sizes that follow specific speed lines can be factored up as described in Chapter 2 and their performance accurately predicted.

After the hydraulic performance range chart is complete, the designer should check the mechanical requirements as outlined by the applicable industrial specification.

If a complete line is being designed, the following mechanical features should be checked.

- Shaft sizes
- Bearing arrangements
- Stuffing box design
- Bolting for maximum pressures
- Suction pressures
- Pump axial and radial balance
- Bed plates, motor supports, etc.
- Gasketing
- Lubrication

The standardization and the use of existing parts should be considered at this time; however hydraulic performance should never be sacrificed for mechanical or cost reasons. If sacrifice becomes necessary, adjust pump hydraulics accordingly. Mechanical design features will be covered in other chapters.



Volute Design

The object of the volute is to convert the kinetic energy imparted to the liquid by the impeller into pressure. The pump casing has no part in the (dynamic) generation of the total head and therefore deals only with minimizing losses.

The absolute velocity of the liquid at the impeller discharge is an important parameter in pump casing design. This velocity is, of course, different from the average liquid velocity in the volute sections, which is the primary casing design parameter. The relationship between these two velocities is given indirectly in Figure 5-1. This relationship is given in a slightly different form in Figure 3-7.

Volutes, like all pump elements, are designed based on average velocities. The average velocity is, of course, that velocity obtained by dividing the flow by the total area normal to that flow. Designs are usually based only on the desired BEP flow, and the performance over the rest of the head-capacity is merely estimated. The results of many tests in which the pressure distribution within the volute casing was measured indicate that:

1. The best volutes are the constant-velocity design.
2. Kinetic energy is converted into pressure only in the diffusion chamber immediately after the volute throat.
3. The most efficient pumps use diffusion chambers with a total divergence angle between 7 and 13 degrees.
4. Even the best discharge nozzle design does not complete the conversion of kinetic energy. This was indicated on the Grand Coulee model pump where the highest pressure was read seven pipe diameters from the discharge flange.

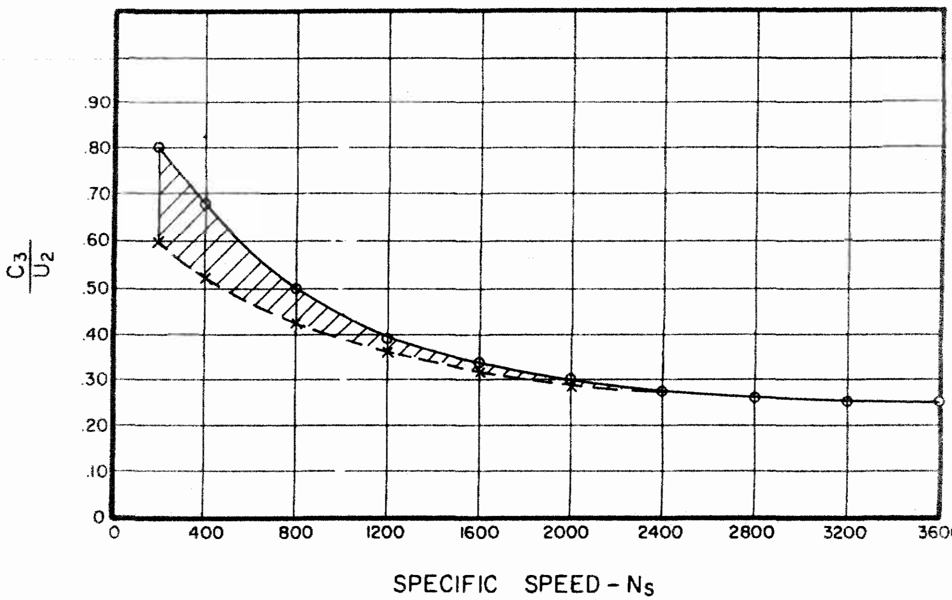


Figure 5-1. Volute velocity/impeller peripheral velocity ratio.

The hydraulic characteristics of a volute casing are a function of the following design elements:

- *Impeller diameter*
- *Cutwater diameter*—This is directly related to specific speed as shown in Table 3-2.
- *Volute lip angle*—This is selected to suit the absolute flow angle at the impeller discharge. However, considerable deviation is acceptable in low to medium specific speed pumps.
- *Volute areas*—These are so arranged that areas increase gradually from the volute tongue or cutwater, toward the volute nozzle, thus accommodating the discharge along the impeller periphery.
- *Volute width* (b_3)—This is made 1.6 to 2.0 times the impeller width (b_2) (Table 3-1).
- *Discharge nozzle diameter*
- *Throat area*—This area is the most important factor in determining the pump capacity at BEP and is determined using Figure 3-8.

Types of Volute Designs

There are several different volute designs being manufactured today.

Single-Volute Casing Designs

Single-volute pumps have been in existence from Day One. Pumps designed using single-volute casings of constant velocity design are more efficient than those using more complicated volute designs. They are also less difficult to cast and more economical to produce because of the open areas around impeller periphery. Theoretically they can be used on large as well as small pumps of all specific speeds. Stepanoff gives a complete description of single-volute casing design.

In all volute pumps the pressure distribution around the periphery of the impeller is uniform only at the BEP. This pressure equilibrium is destroyed when the pump is operating on either side of the BEP, resulting in a radial load on the impeller. This load deflects the pump shaft and can result in excessive wear at the wearing rings, seals, packing, or bearings. In extreme cases, shaft breakage due to fatigue failure can result. The magnitude of this radial load is given by:

$$P = KHD_2B_2 \text{ sp gr}/2.31$$

Values of the experimental constant K are given in Figure 5-2. For a specific single-volute pump it reaches its maximum at shutoff and will vary between 0.09 and 0.38 depending upon specific speed. The effect of the force will be most pronounced on a single-stage pump with a wide b_2 or a large-sized pump.

It is safe to say that with existing design techniques, single-volute designs are used mainly on low capacity, low specific speed pumps or pumps for special applications such as variable slurries or solids handling.

Double-Volute Casing Designs

A double-volute casing design is actually two single-volute designs combined in an opposed arrangement. The total throat area of the two volutes is identical to that which would be used on a comparable single-volute design.

Double-volute casings were introduced to eliminate the radial thrust problems that are inherent in single-volute designs. Test measurements, however, indicate that while the radial forces in a double volute are greatly reduced, they are not completely eliminated. This is because al-

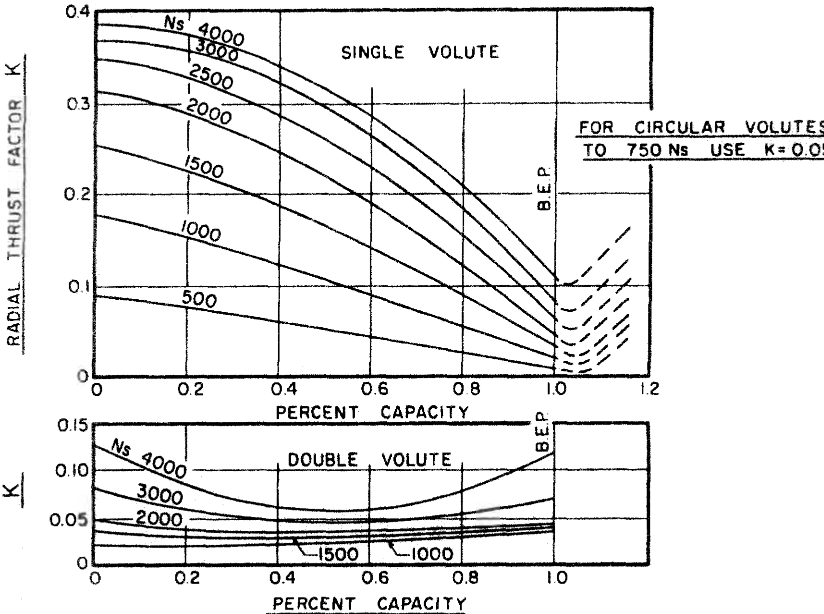


Figure 5-2. Radial thrust factor.

though the volute proper is symmetrical about its centerline, the two passages carrying the liquid to the discharge flange often are not. For this reason, the pressure forces around the impeller periphery do not precisely cancel, and a radial force does exist even in double-volute pumps.

Values of the constant K have been established experimentally by actually measuring the pressure distributions in a variety of double-volute pumps. The data presented in Figure 5-2 apply to conventional single-stage double-volute pumps and indicate substantial reductions in the magnitude of K. Tests on multistage pumps with completely symmetrical double-volute casings indicate that the radial thrust is nearly zero over the full operating range of the pump.

The hydraulic performance of double-volute pumps is nearly as good as that of single-volute pumps. Tests indicate that a double-volute pump will be approximately one to one and one-half points less efficient at BEP, but will be approximately two points more efficient on either side of BEP than a comparable single-volute pump. Thus the double-volute casing produces a higher efficiency over the full range of the head-capacity curve than a single volute.

Double-volute pump casings should not be used in low-flow (below 400 GPM) single-stage pumps. The small liquid passages behind the long dividing rib make this type of casing very difficult to manufacture and almost impossible to clean. In large pumps double-volute casings should be generally used and single-volute designs should not be considered.

The Double-Volute Dividing Rib (Splitter)

The dividing rib or splitter in double-volute pumps causes considerable problems in the production of case castings. This is particularly true on small-capacity pumps where flow areas are small and a large unsupported core is required on the outside of the dividing rib (splitter).

The original double-volute designs maintained a constant area in the flow passage behind the splitter. This concept proved to be impractical due to casting difficulties. In addition the consistently small flow areas caused high friction losses in one of the volutes, which in turn produced an uneven pressure distribution around the impeller. Most modern designs have an expanding area in this flow passage ending in equal areas on both sides of the volute rib.

The effects of volute rib length on radial thrust are shown in Figure 5-3. Note that the minimum radial thrust was achieved during Test 2 for which the dividing rib did not extend all the way to the discharge flange. Also note that even a short dividing rib (Test 4) produced substantially less radial thrust than would have been obtained with a single volute.

Triple-Volute Casings

Some pumps use three volutes symmetrically spaced around the impeller periphery. Total area of the three volutes is equal to that of a comparable single volute. The triple volute casing is difficult to cast, almost impossible to clean, and expensive to produce. We do not see any justification for using this design in commercial pumps.

Quad-Volute Casings

Approximately 15 years ago a 4-vane (quad) volute was introduced. Later this design was applied to large primary nuclear coolant pumps (100,000 GPM, 10–15,000 HP). The discharge liquid passage of these pumps is similar to that of a multi-stage crossover leading to a side discharge. There is no hydraulic advantage to this design.

The only advantage of this design is its reduced material cost. The overall dimensions of quad-volute casing are considerably smaller than those of a comparable double-volute pump.

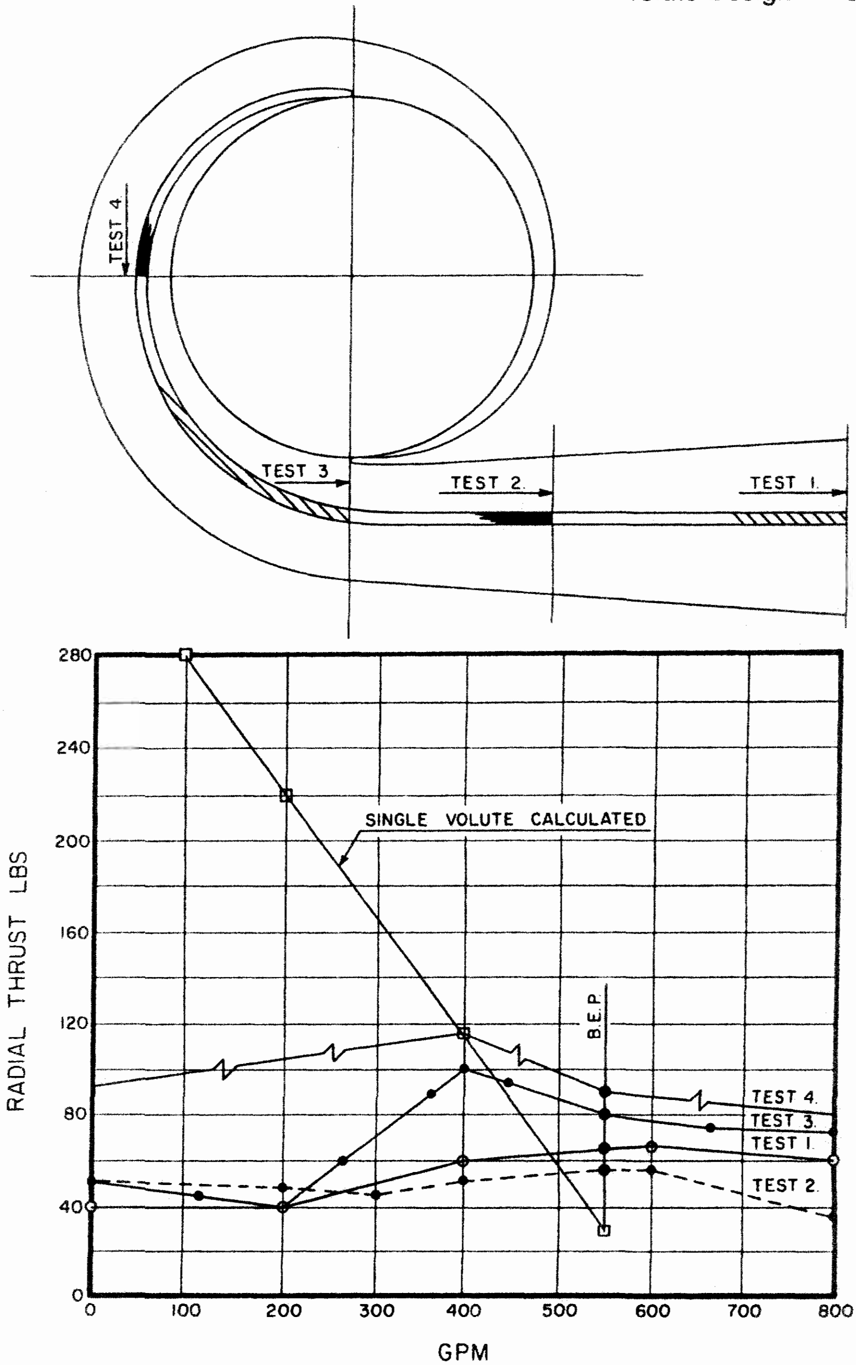


Figure 5-3. Influence of volute rib length on radial thrust.

Circular-Volute Casings

Several pump manufacturers have conducted tests to evaluate the hydraulic performance of pumps with circular volutes. A study of the results of these tests reveals that circular volutes improve the hydraulic performance of small high head or low specific speed units and impair the performance of high specific speed pumps. Specifically, pump efficiency is improved below specific speeds of 600. For specific speeds above 600 the efficiency of circular-volute designs will be 95% of that possible with conventional volute designs. This can be explained by remembering that in a conventional volute a uniform pressure and velocity distribution exists around the impeller periphery only at the BEP. At off-peak capacities, velocities and pressures are not uniform. For circular volutes the opposite is true. Uniform velocity and pressure exist only at zero flow. This uniformity is progressively destroyed as the capacity is increased. Therefore, at BEP the casing losses are generally greater than those of the conventional volute. For low specific speed pumps, however, there is some gain in efficiency due to a circular volute since the benefits of the improved surface finish in the machined volute outweigh the problems created by the nonuniform pressure distribution. A comparison of the efficiency of circular and conventional volutes is shown in Figure 5-4. The use of a circular volute design should be considered in the following instances:

- For small, high head, low specific speed (N_s 500–600) pumps.
- For a pump casing that must accommodate several impeller sizes.
- For a pump in which foundry limitations have dictated an overly wide impeller b_2.
- For a pump that must use a fabricated casing.
- For a pump that requires that the volute passage be machined in the case casting.

General Design Considerations

It was pointed out previously that the casing itself represents only losses and does not add anything to the total energy developed by the pump. In designing pump casings it is therefore important to utilize all available means of minimizing casing losses. However, commercial considerations dictate some deviations from this approach, and experience

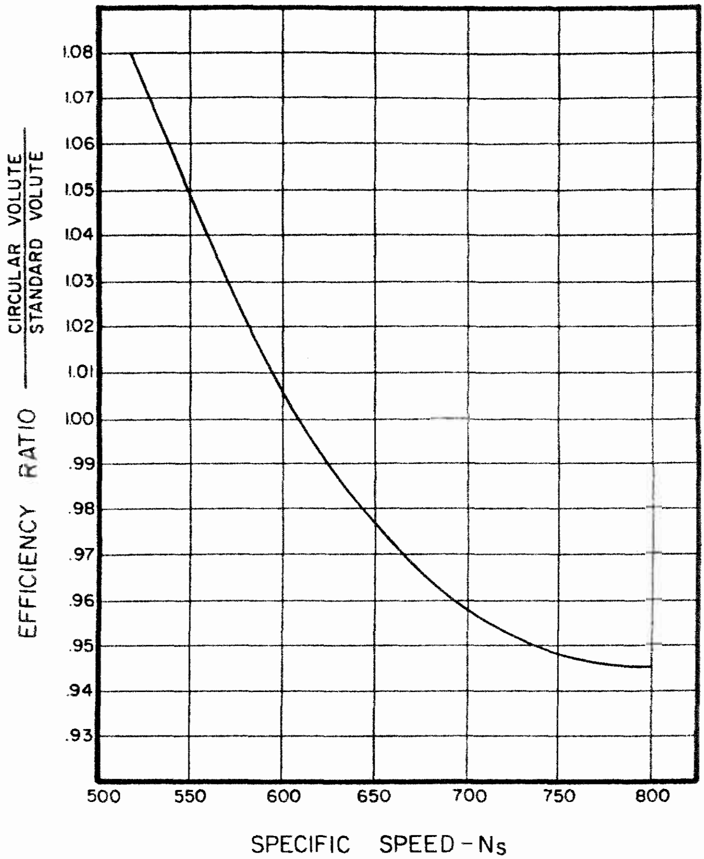


Figure 5-4. Efficiency comparison of circular and conventional volutes.

has shown that these do not have a significant effect on casing losses. The following design rules have shown themselves to be applicable to all casing designs:

1. Constant angles on the volute sidewalls should be used rather than different angles at each volute section. Experience has shown that these two approaches give as good results and the use of constant wall angles reduces pattern costs and saves manufacturing time.
2. The volute space on both sides of the impeller shrouds should be symmetrical.

3. All volute areas should be designed to provide a smooth change of areas.
4. Circular volutes should be considered for pumps below a specific speed of 600. Circular volutes should not be considered for multi-stage pumps.
5. The total divergence angle of the diffusion chamber should be between 7 and 13 degrees. The final kinetic energy conversion is obtained in the discharge nozzle in a single-stage pump and in both the discharge nozzle and crossover in a multi-stage pump.
6. In designing a volute, be liberal with the space surrounding the impeller. In multi-stage pumps in particular, enough space should be provided between the volute walls and the impeller shroud to allow one-half inch each way for end float and casting variations. A volute that is tight in this area will create axial thrust and manufacturing problems.

The Use of Universal Volute Sections for Standard Volute Designs

It has been noted that when the volute sections of different pumps are factored to the same throat area; their contours are almost identical. Any differences that do exist can be traced to mechanical considerations or the designer's whim, rather than any important principle of hydraulic design. Similarly, factoring the impeller width and the radial gap between the impeller and the cutwater reveals that the values of these parameters also lie in a very narrow random range.

In other words, the entire discharge portion of the pump casing when viewed in cross section and factored to a common throat area has only minor variations throughout the entire specific speed spectrum. This fact enables us to eliminate the usual trial-and-error method of designing volute sections while still consistently producing casings to a high standard of hydraulic design. To facilitate this process we have prepared a set of "universal" volute drawings on which the typical volute sections described above have been laid out for a 10 sq in. throat area. Once the designer has chosen his throat area, he can quickly produce the required volute sections by factoring the sections shown for the "universal" volute. Sections for a single-volute pump are shown in Figures 5-5 and 5-6, and sections for a double volute pump are shown in Figure 5-7.

The Design of Rectangular Double Volutes

For low capacity (500–600 GPM) or low to medium specific speed ($N_s < 1,100$) pumps, a rectangular volute design should be considered.

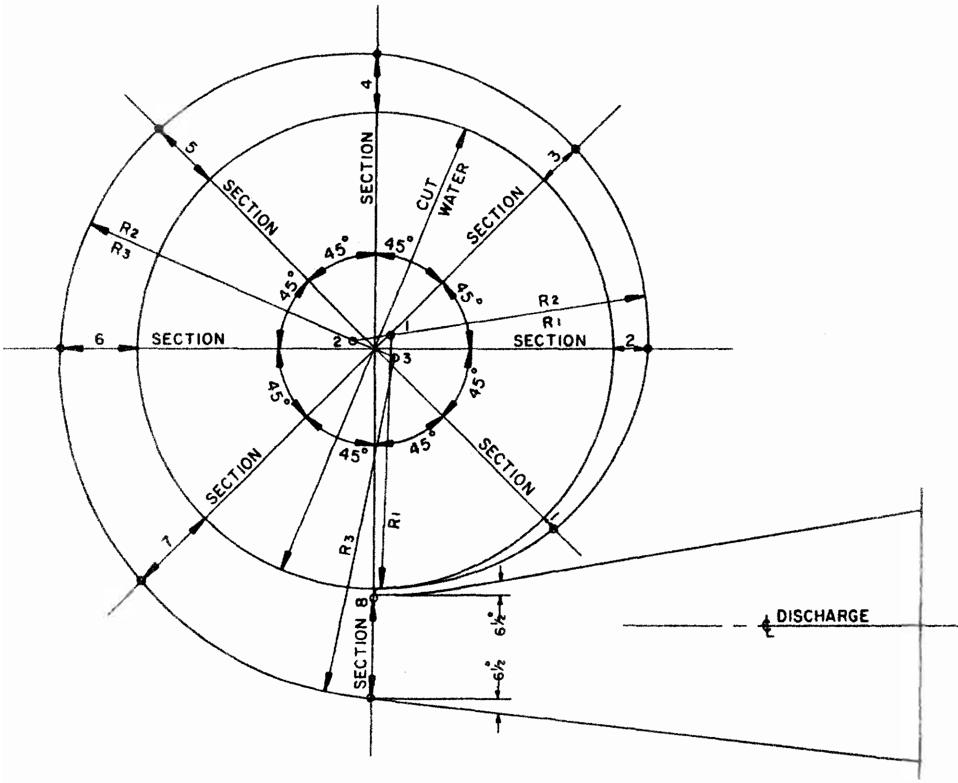


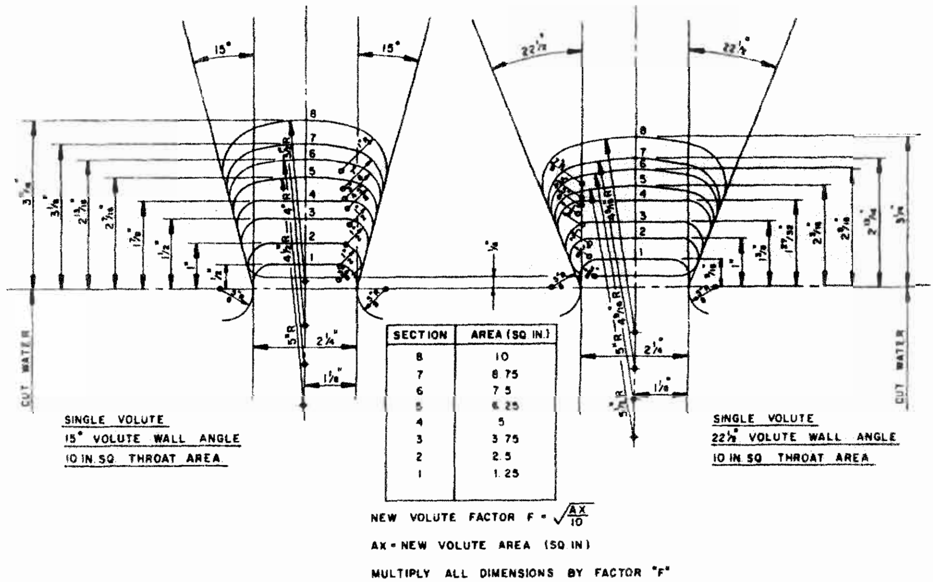
Figure 5-5. Typical single-volute layout.

The universal volute sections for such a design are shown in Figure 5-8. A rectangular volute casing requires the same throat area as a standard volute casing and should be laid out according to the principle of constant velocity.

Rectangular volutes are widely used in small single-stage and multistage pumps. The benefits of the rectangular volute are strictly economical. The simple volute section yields a considerable cost savings due to reduced pattern costs and production time. Over the range of specific speeds where it is used the hydraulic losses are negligible.

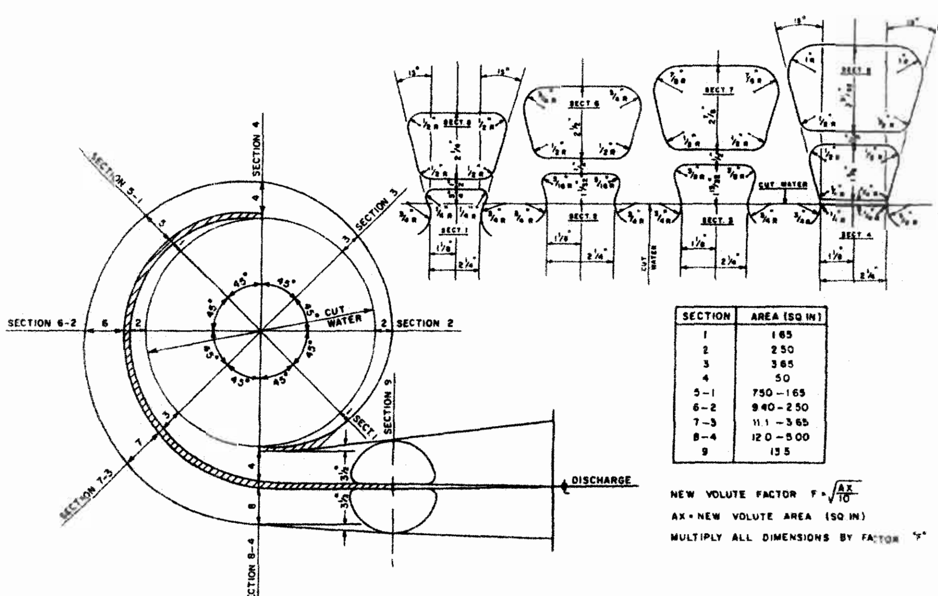
The Design of Circular Volutes

The details of a typical circular volute casing design are shown in Figure 5-9. The ratio between the impeller diameter, D_2, and the volute di-



SECTION	AREA (SQ. IN.)
8	10
7	8.75
6	7.5
5	6.25
4	5
3	3.75
2	2.5
1	1.25

Figure 5-6. Universal volute sections for single-volute pump.



SECTION	AREA (SQ. IN.)
1	1.65
2	2.50
3	3.65
4	5.0
5–1	7.50 – 1.65
6–2	9.40 – 2.50
7–3	11.1 – 3.65
8–4	12.0 – 5.00
9	13.5

Figure 5-7. Universal volute sections and typical layout for trapezoidal double-volute pump.

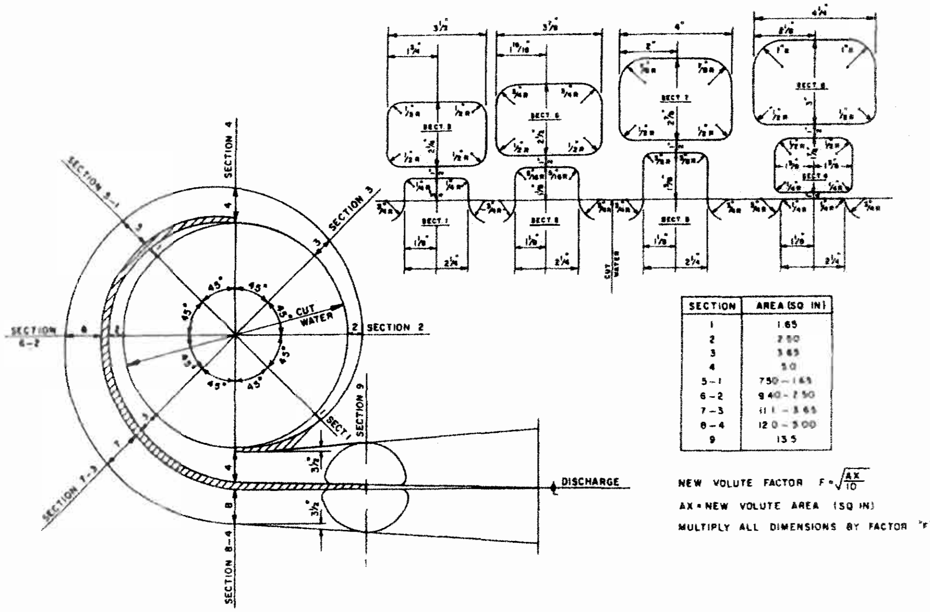


Figure 5-8. Universal volute sections and typical layout for rectangular double volute pump.

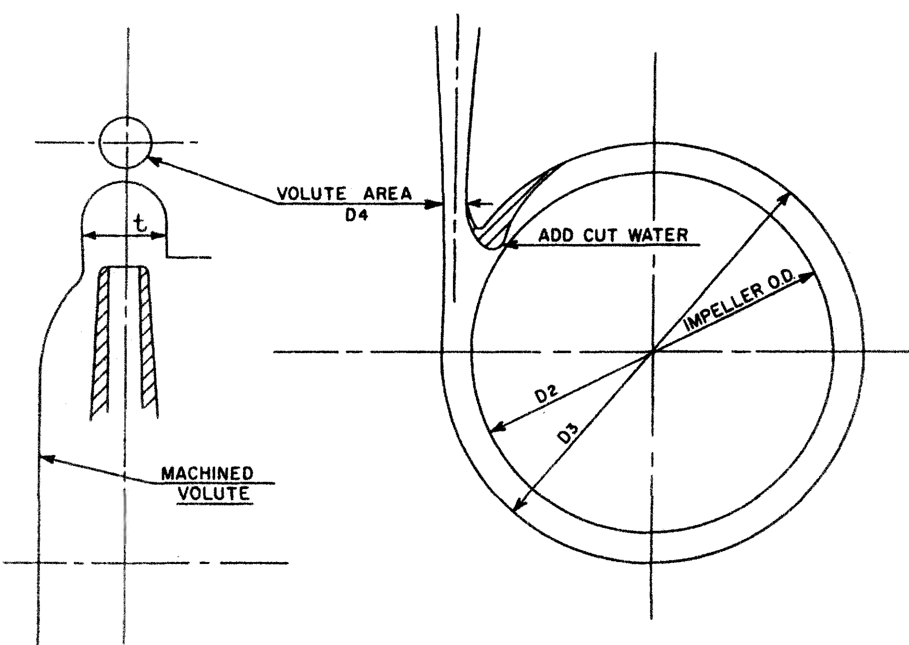
ameter, D_3, is quite important and should not be less than 1.15 or more than 1.2. The volute width, t, should be chosen to accommodate the widest (maximum flow) impeller that will be used in the casing. The capacity at BEP can be controlled by the choice of the volute diameter, D_4. Generally, the best results are obtained by selecting the volute width and diameter for each flow requirement. To minimize liquid recirculation in the volute, a cutwater tongue should be added. Tests have shown the addition of a cutwater tongue can reduce radial loads by up to 20%.

General Considerations in Casing Design

There are several considerations in the casing design process that apply to all volute types. These are as follows:

- The most important variable in casing design is the *throat area.* This area together with the impeller geometry at the periphery establishes the pump capacity at the best efficiency point. The throat area should be sized to accommodate the capacity at which the utmost efficiency is required, using Figure 3-8. Where several impellers in the same casing are to be considered, the throat area should be sized for the standard impeller and increased by 10% to maintain the efficiency of the high capacity impeller.

- Since significant energy conversion takes place in the *diffusion chamber,* the design of this element should be done with extreme care.
- The *volute* should be designed to maintain constant velocity in the volute sections.
- The overall *shape of the volute sections* should be as shown in Figures 5-5, 5-6, 5-7, and 5-8. The use of these figures will save the designer time and introduce consistency into the design process.
- The *volute spiral* from the cutwater to the throat should be a streamlined curve defined by no more than three radii.



$\frac{D3}{D2}$ = 1.15 TO 1.12

D4 = DRILL TO SUIT REQUIRED G.P.M.

t = TO SUIT IMPELLER WIDTH

Figure 5-9. Typical layout for circular volute pump.

Manufacturing Considerations

Casings, particularly of the double-volute design, are very difficult to cast. In small- and medium-sized pumps the volute areas are small and the liquid passages are long, requiring long unsupported cores. In volute-type multi-stage pumps the problem is more pronounced since there are several complicated cores in a single casing.

Casing Surface Finish

To minimize friction losses in the casing, the liquid passages should be as clean as possible. Since cleaning pump casings is both difficult and time consuming, an extreme effort to produce smooth liquid passages should be made at the foundry. The use of special sand for cores, ceramic cores, or any other means of producing a smooth casting should be standard foundry practice for producing casings.

Particularly with multi-stage pumps, however, even the best foundry efforts should be supplemented by some hand polishing at points of high liquid velocities such as the volute surfaces surrounding the impeller and the area around the volute throat. Both of these areas are generally accessible for hand polishing. In addition, both cutwater tongues should be sharpened and made equidistant from the horizontal centerline of each stage. The same distance should be maintained for each stage in a multi-stage pump.

Casing Shrinkage

Dimensional irregularities in pump casings due to shrinkage variations or core shifts are quite common. Shrinkage variations can even occur in castings made of the same material and using the same pattern. The acceptance or rejection of these defects should be based upon engineering judgments. However, knowing that shrinkage and core shifts are quite common, the designer should allow sufficient space for rotating element end float. The allowance for total end float should be a minimum of one-half inch.

Conclusion

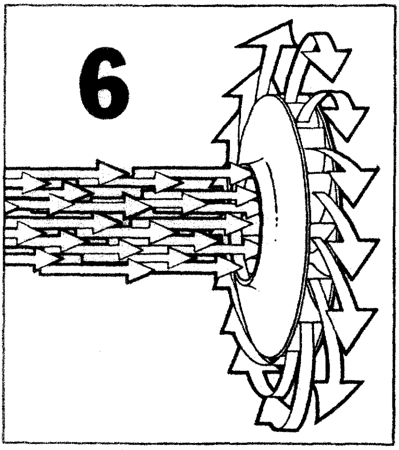
Although it is often claimed that casings are very efficient, this is misleading, since the hydraulic and friction losses that occur in the casing can only reduce the total pump output and never add to it. It is the designer's responsibility to do his utmost to minimize these losses.

Notation

P	Radial force (lbs)
H	Impeller head (ft)
D_2	Impeller diameter (in.)
B_2	Total impeller width including shrouds at D_2 (in.)
K	Experimental constant
sp gr	Specific gravity

Reference

Stepanoff, A. J., *Centrifugal and Axial Flow Pumps,* 2nd edition, John Wiley and Sons, Inc., New York, 1957, pp. 111–123.



6 Design of Multi-Stage Casing

Multi-stage casing discussed in this chapter will be related to one type of pump, namely, a horizontally split, opposed impellers, double-volute design, similar to one shown in Figure 6-1. This type of pump offers the following features:

- Pump casings, when properly bolted, are suitable for high working pressures. Many pumps in service are operating at 3,000 to 4,000 psi discharge pressure.
- The pumps are axially balanced allowing the use of standard ball bearings for many services. Only when shaft diameter is too big or rotating speed too high, will Kingsbury-type bearings be required.
- When NPSHA is low, double-suction first-stage impellers are available as a part of standard design.
- A completely assembled rotating element can be checked for shaft and ring run-out over the full length of the shaft.
- The most modern designs have split impeller hub case rings, horizontally split, allowing the assembled rotating element to be dynamically balanced.
- The pumps are easily designed for high pressure and speeds above 7,000 rpm.
- Removal of the top half of the pump casing exposes the complete rotating element for inspection or repair.

The impeller design for the multi-stage pump is the same as that for a one-stage unit, as described in Chapter 3. The double-volute design is also the same as that for a one-stage pump as described in Chapter 5.

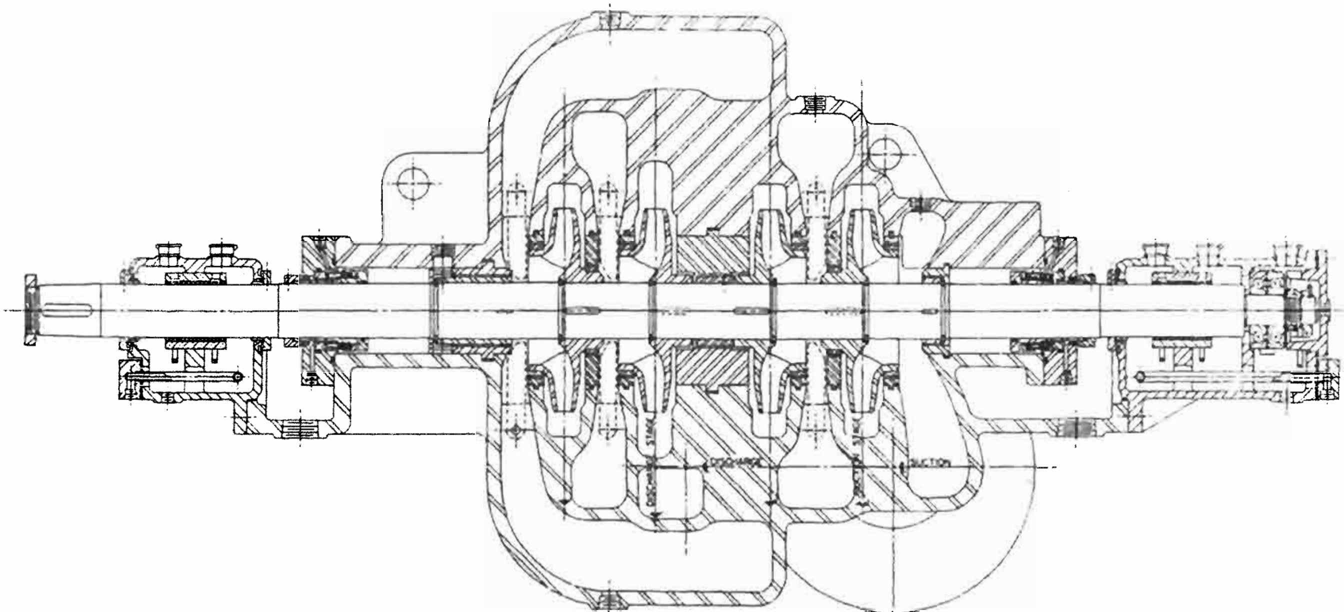


Figure 6-1. Horizontally split opposed impellers double-volute multi-stage pump (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

However, in a multi-stage casing, the liquid from one stage to the next stage must be transferred by means of a crossover passage. The term "crossover" refers to the channel leading from the volute throat of one stage to the suction of the next. Crossovers leading from one stage to the next are normally referred to as "short" crossovers and are similar to return channels in diffuser pumps. These are normally designed in right hand or left hand configurations, depending upon the stage arrangement. Crossovers that lead from one end of the pump to the other or from the center of the pump to the end are normally referred to as "long" crossovers.

The stage arrangements used by various pump manufacturers are shown schematically in Figure 6-2. Arrangement 1 minimizes the number of separate patterns required and results in a minimum capital investment and low manufacturing costs. However, with this arrangement a balancing drum is required to reduce axial thrust. Arrangement 2 is used on barrel pumps with horizontally split inner volute casings. Arrangement 3 is the most popular arrangement for horizontally split multi-stage pumps and is used by many manufacturers. Finally, with Arrangement 4 the series stages have double volutes while the two center stages have staggered volutes. This design achieves a balanced radial load and an efficient final discharge while requiring only one "long" crossover, thereby reducing pattern costs and casing weight.

General Considerations in Crossover Design

The principal functions of a crossover are as follows:

- To convert the velocity head at the volute throat into pressure as soon as possible, thereby minimizing the overall pressure losses in the crossover.
- To turn the flow 180° from the exit of one stage into the suction of the next.
- To deliver a uniformly distributed flow to the eye of the succeeding impeller.
- To accomplish all these functions with minimum losses at minimum cost.

Velocity cannot be efficiently converted into pressure if diffusion and turning are attempted simultaneously, since turning will produce higher velocities at the outer walls adversely affecting the diffusion process. Furthermore, a crossover channel that runs diagonally from the volute

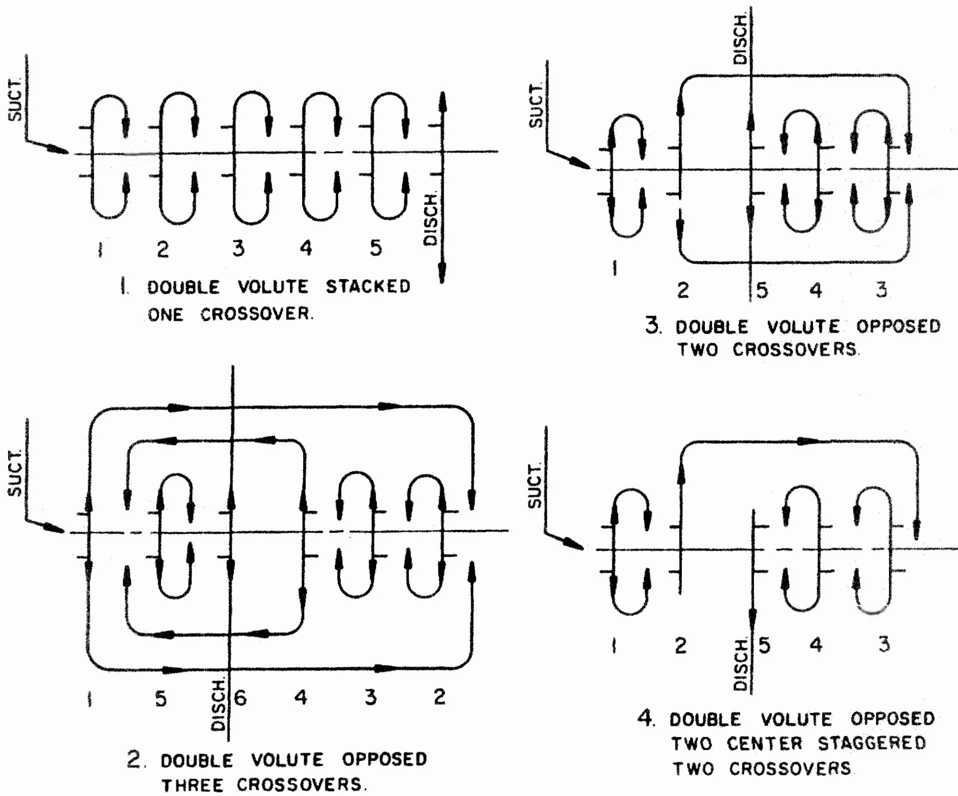


Figure 6-2. Multi-stage pump stage arrangements.

throat to the suction of the next stage imparts a spiral motion to the flow resulting in prerotation and hydraulic losses. For these reasons, the multi-stage pumps of 25 or more years ago were designed with high looping crossovers. To achieve radial balance these crossovers were in both the top and bottom casing halves. This design, referred to as the "pretzel" casing, was very costly, difficult to cast, and limited to a maximum of eight stages.

These problems prompted a study to evaluate the performance of various crossover shapes. A 4-in. pump delivering 1,200 GPM at 3,550 RPM was selected as a model and the three crossover configurations shown in Figure 6-3 were tested. For these tests the pump hydraulic passages were highly polished (60–80 micro-inches), ring clearances were minimized and component crossover parts were carefully matched using a template. Configuration 1 was designed with a total divergence angle of

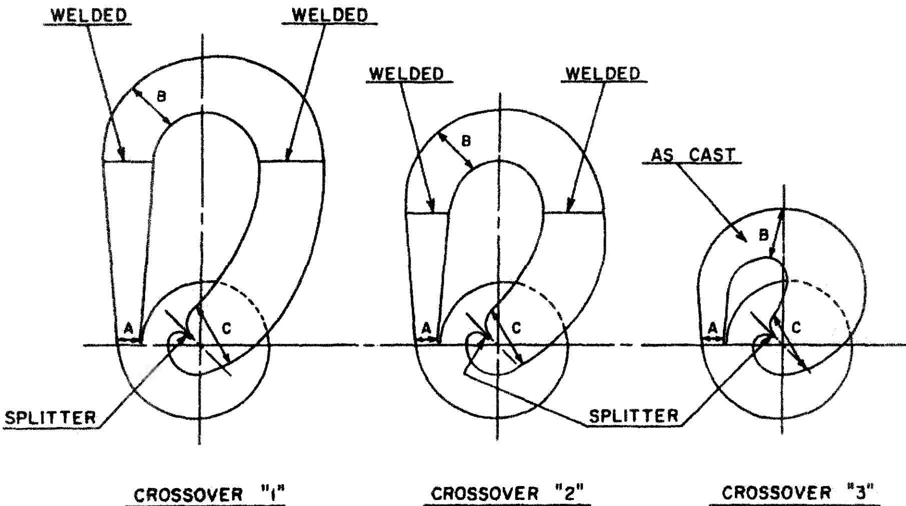


Figure 6-3. Configurations evaluated during crossover performance study.

7° in the passage between the volute throat and the entrance to the "U" bend. From this point the area was held constant to the impeller eye. To prevent prerotation a splitter was added to the suction channel. Crossovers 2 and 3 were designed maintaining the same areas at sections A, B, and C with the same divergence angle but progressively reducing the radial extent of the crossover. The "U" bend on Crossovers 1 and 2 were cast separately from the casing and highly polished before welding. Crossover 3 was cast as a single piece, and the "U" bend polished only in the accessible areas.

The results of testing all three crossover configurations are shown on Figure 6-4. The tests indicated that Crossover 1 yielded a peak efficiency four points higher than Crossover 3. Subsequent testing of commercial units, however, indicated the difference to be only two points. The difference in improvement was attributed to the poor quality of the commercial castings and the use of normal ring clearances. The two-point efficiency loss associated with Crossover 3 was deemed commercially acceptable and was incorporated in multi-stage pumps of up to fourteen stages by all the West Coast manufacturers. These pumps were suitable for higher pressures, easily adaptable to any number of stages, odd or even, and readily castable even in double-volute configurations.

Specific Crossover Designs

A successful multi-stage pump development should produce a product that has excellent hydraulic performance, low manufacturing cost, and requires a minimum initial capital investment. These three items become the basic design requirements during the layout of horizontally split multi-stage pumps. Hydraulically, the pump design should achieve the best possible efficiency, as well as the highest head per stage, thereby minimizing the number of stages required. The best available technology should therefore be utilized to produce the most efficient volutes and impellers. Although crossover design has only a secondary effect on pump efficiency, it too should use every available "trick" to achieve the best possible results.

Figure 6-5 shows short and long configurations of the two basic types of crossovers normally used on multi-stage pumps. Both have been tested by the West Coast pump companies. Results of these tests indicate that the radial diffusion type is approximately one point more efficient than the diagonal diffusion type.

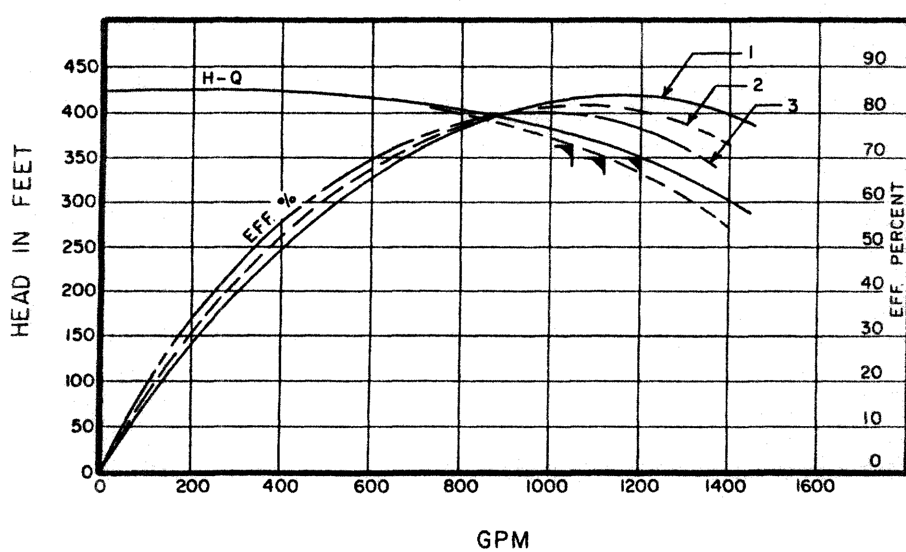


Figure 6-4. Results of crossover performance study.

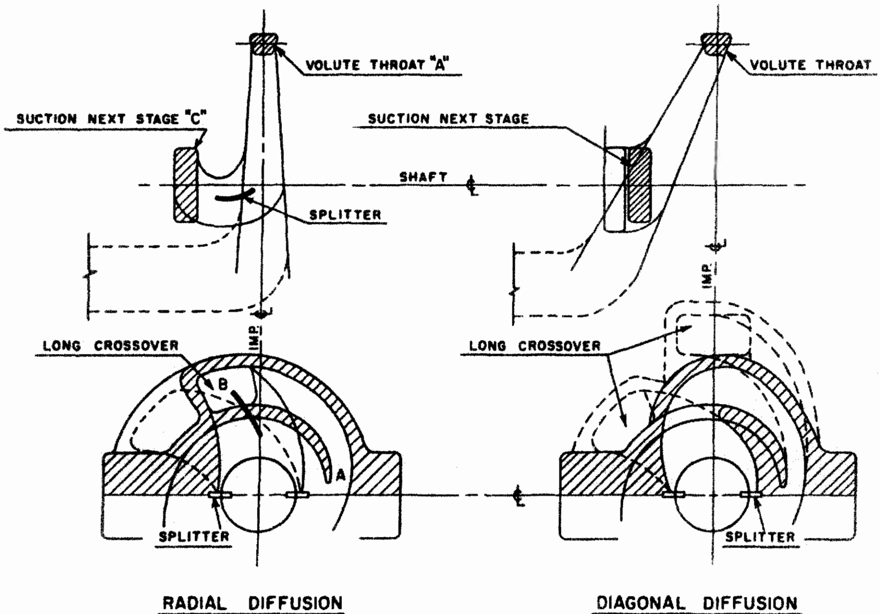


Figure 6-5. Radial and diagonal diffusion crossovers.

Crossovers with Radial Diffusion Sections

The radial diffusion type crossover shown in Figure 6-5 has a diffusion section that follows the volute periphery along the impeller centerline, diffusing with a total divergence angle of 7° up to the point where area at "B" is four times the volute throat area. This point should be reached before the "U" bend to the suction channel. The suction channel should be sized to accommodate the largest capacity impeller that will be used in the pump. The area of the suction return channel should be consistent immediately after the "U" bend. Some designers prefer to decelerate slightly at the impeller eye; however, recent tests indicate that better efficiency is obtained if the liquid is accelerated as it approaches the impeller eye.

Tests on this type of crossover indicate a total head loss equal to 86% of the inlet velocity head. The addition of a welded splitter in the "U" bend will reduce this loss to 65%. However, the cost of adding this splitter is generally prohibitive, and it is not generally used.

From a theoretical viewpoint, crossover channels should have a circular cross section to minimize friction losses. However, all the designs on the market today have rectangular shapes for practical reasons.

To attain the best hydraulic performance, anti-rotation splitters must be added to the suction channel at the impeller eye. The best overall results are obtained by placing two splitters at the casing parting split as shown in Figure 6-5.

The long crossover is identical to the short or series configuration up to the area "B," where the long channel that traverses the pump begins. This long channel should be designed with a "window" at the top for cleaning and a properly shaped plate matched to the crossover opening before welding. The configuration of the long crossover is also shown in Figure 6-5.

Crossovers with Diagonal Diffusion Sections

The diagonal diffusion type crossover shown in Figure 6-5 leads the liquid from the volute throat to the suction of the next impeller while traveling diagonally around the periphery of the volute. This design has one long radius turn as compared to the "U" bend used in the radial diffusion type. Other than these differences, both types of crossover have the same diffusion, area progression etc. Figure 6-5 also shows the configuration in which the long crossover channel climbs over the short crossover.

Even though this crossover has only a single long-radius turn, it is not as efficient as the radial type. This can be attributed to the diagonally located channel, which imparts a spiral motion to the fluid leaving the volute throat, resulting in hydraulic losses larger than those in the "U" bend.

Mechanical Suggestions

In previous chapters, we have described design procedures for centrifugal pump impellers and volutes applicable to one-stage or multi-stage units and in this chapter crossovers for multi-stage pumps only. However, hydraulic considerations alone for multi-stage pumps are not sufficient to complete a final unit. Mechanical details must be considered. This refers to patterns, foundry methods, mechanical bolting, and quality controls.

Patterns

Multi-stage casings have quite complex shapes of liquid passages, including crossovers, crossunders, double volutes at each stage, etc. For this reason, pattern equipment must be of high quality sectional design to allow for variations in number of stages. Normal practice is to make the first pattern a four stage of hard wood with each stage being a separate section. For additional stages, the sections are duplicated in plastic material. For each stage combination, plastic sections should be assembled on their own mounting boards. This arrangement will allow several pumps of different stages to be produced at the same time.

Foundries

The core assembly for multi-stage pumps is very complex as shown in Figure 6-6. It shows a 12-stage 4-in. pump with a single-suction first-stage impeller.

In order for the rotating element to fit into the pump casing, each volute core must be assembled perpendicular to the shaft centerline. To assure perpendicularity, a special gauge should be made for this purpose.

It is also vitally important to cast casing with wet area surfaces as smooth as possible. For this reason, the casing cores should be made from "green sand" or ceramic materials.

The major hydraulic loss in multi-stage pumps is friction loss. To minimize this, the as-cast-surface roughness of the internal passages should be a minimum of 125 micro inches. The smoother the wet areas, the less the cost of hand polishing or grinding will be.

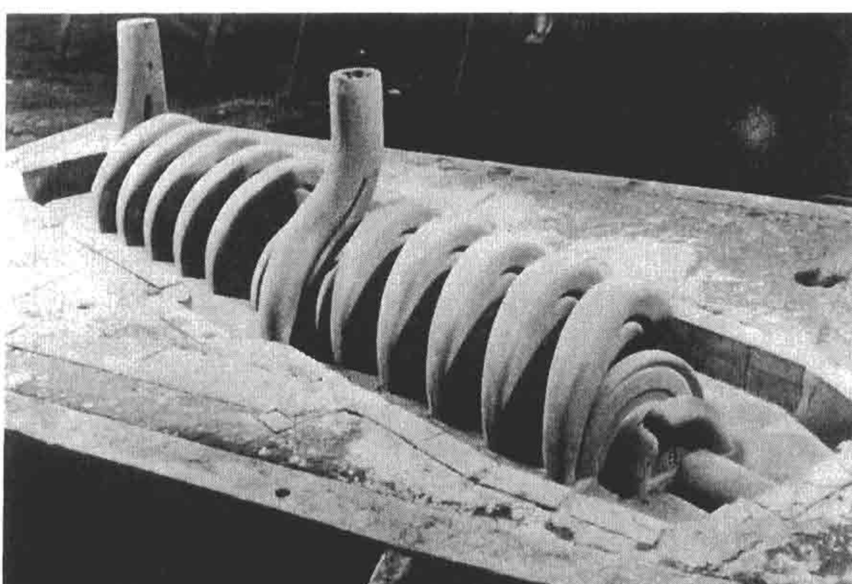


Figure 6-6. Core assembly.

Casing Mismatch

On horizontally split pump casings mismatching between the upper and lower casing halves is quite common. This mismatch is normally corrected by hand filing using a template (Figure 6-7). The same template is used by the machine shop to define bolt locations and also by the foundry. It is quite important that the fluid passages in both halves of the casing match in both the horizontal and vertical planes since any mismatching will adversely affect pump performance.

Check for Volute Interference

After the pump casing is completely finished to satisfy quality specification, the assembled rotating element should be installed in the casing (as shown in Figure 6-8) to check for interference against volute walls. Also check the element for total end float, which should be no less than one-half inch total or one-quarter inch on each end.

General Design Suggestions

- It is important to have the volute symmetrical about shaft centerline.
- A 1/16-in. deep relief should be machined around each stud at the split. At each stud at assembly the metal will rise. The relief will allow the gasket to lay flat, reduce gasket area, and increase gasket unit pressure.
- It is very important to install splitters at each impeller suction entrance. These can be installed recessed at the casing split or cast on casing hub rings.
- Horizontally split centrifugal pumps are designed for relatively high pressures; 3,000 psi hydrotest is very much standard by many manufacturers. Pressure up to 6,000 psi hydro can be obtained by special design of bolting and case.

It is essential (to prevent stage-to-stage bypassing) to have all bolts located as close as possible to the open area. The high pressure differential in opposed impeller design multi-stage pumps is between two center stages and at the high pressure end. Particular attention to bolting should be paid in these two areas. It is sometimes advantageous to provide elevated bosses to bring bolting closer to open area.

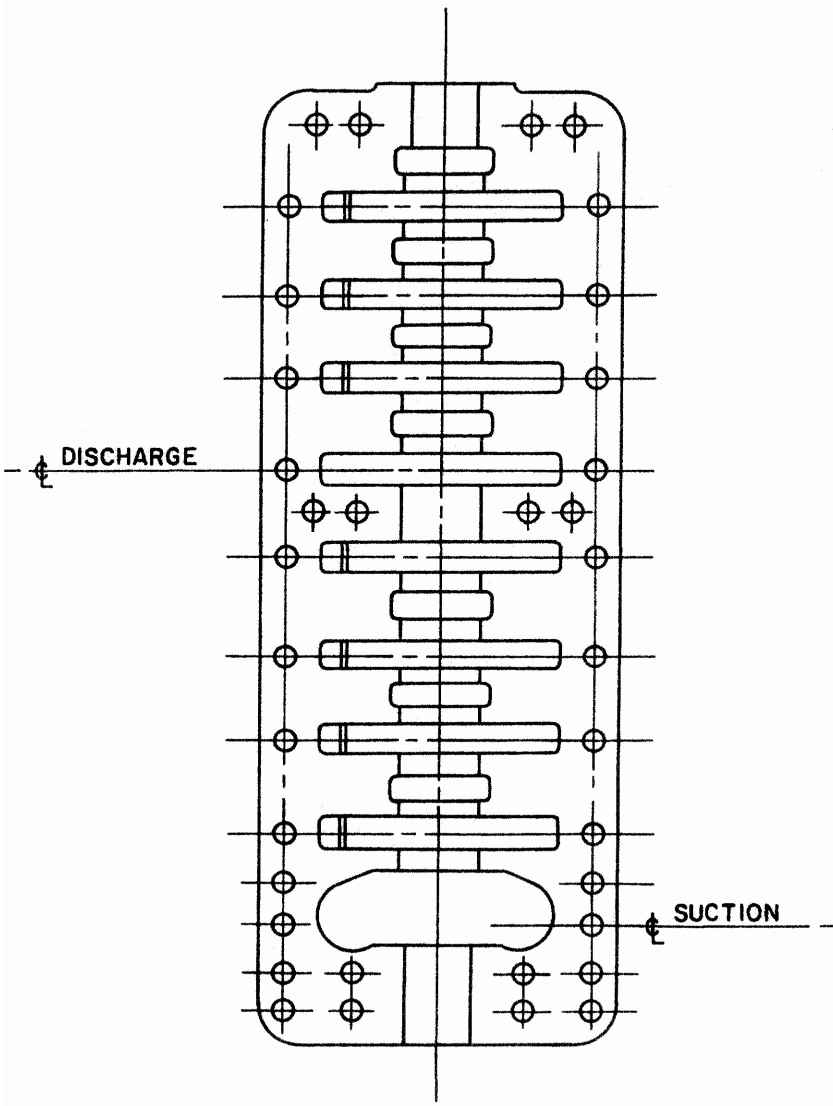


Figure 6-7. Split case template.

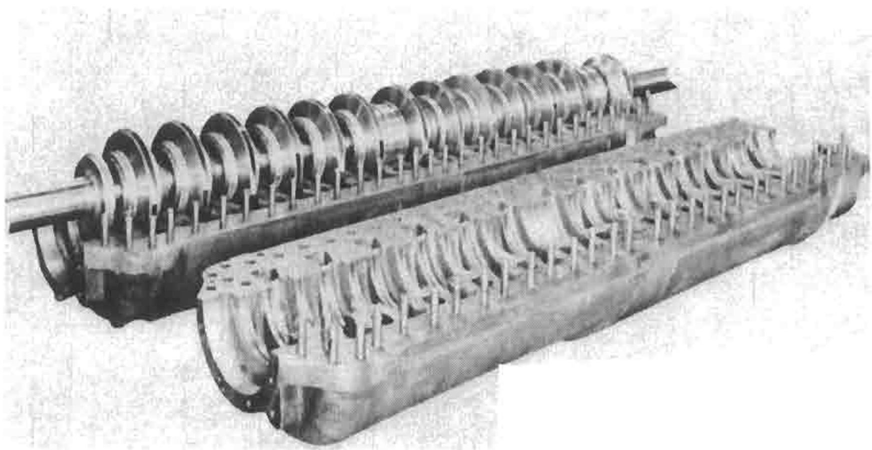


Figure 6-8. Multi-stage rotating element.

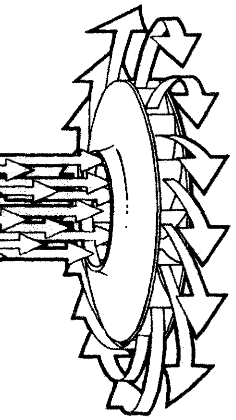
As a guide to selecting bolt size use:

$$\text{Number of bolts} = \frac{P \times A}{S \times A_R}$$

Notation

- P Hydrotest pressure (psi)
- A Total area at split minus total area of bolt hole relief diameter (sq in.)
- S Allowable bolt stress (psi)
- A_R Root area of bolt thread (sq in.)

7



Double-Suction Pumps and Side-Suction Design

The double-suction single-stage pump is, perhaps, the most widely used pump throughout the industrial world. Applications range from light duty building trade pumps to heavy duty pipeline injection pumps (Figure 7-1).

The double-suction is a very simple machine whose initial cost is relatively low. Above 700–1,000 GPM, efficiency is high and required NPSH is low. All modern double-suction pumps are designed with double-volute casing to maintain hydraulic radial balance over the full range of the head-capacity curve. Having a double-suction impeller, the pumps are theoretically in axial balance. Double-suction pumps often have to operate under suction lift, run wide open in a system without a discharge valve, or satisfy a variable capacity requirement. These pumps may be quite large, pump high capacities, and handle pumpage with gas or entrained air. In spite of all this, they are expected to operate without noise or cavitation. Suction passage design should therefore be based on the best available technical "know-how," and liberties should not be taken during the design process.

Double-Suction Pump Design

Pump Casing

The double-suction, double-volute casing is designed in identical manner to the single-suction pump, as described in Chapter 5. The specific speed, N_s, of a double-suction pump is identical to the single-suction unit. *Do not divide* calculated pump specific speed by the square root of

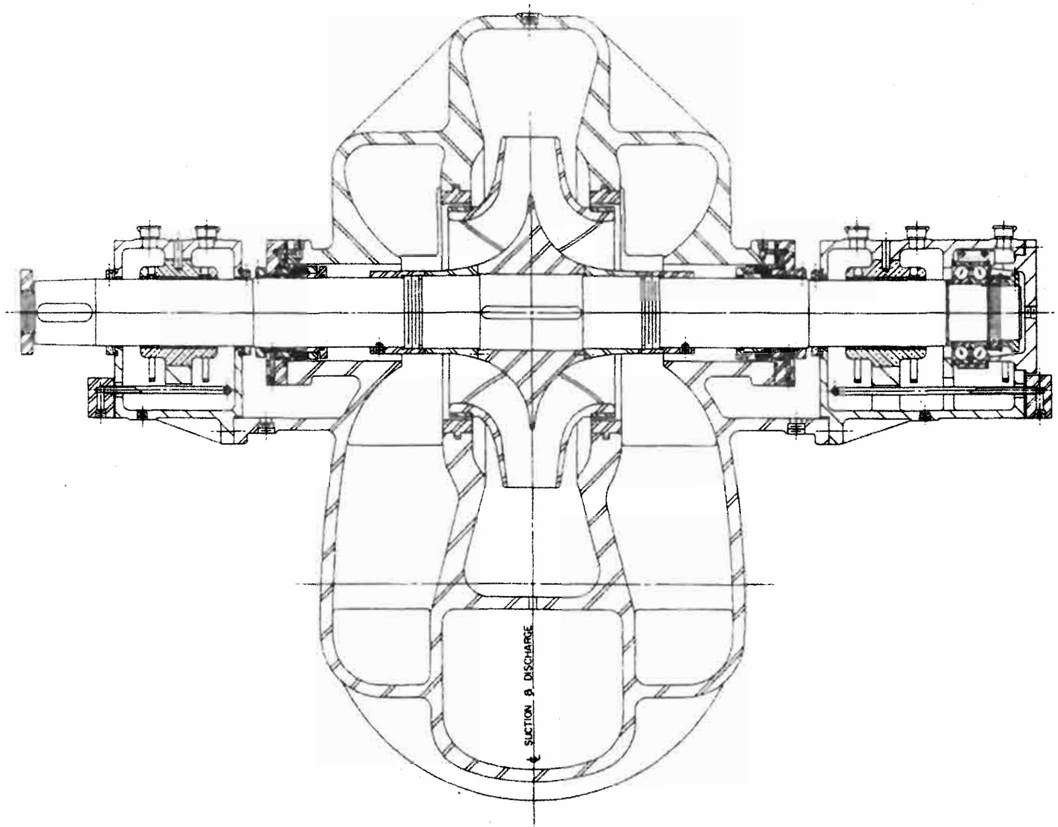


Figure 7-1. Single-stage, double-suction, pipeline pump (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

two. Experience shows that this procedure will give misleading design factors and unfavorable test results.

Design of the suction approach to the pump impeller for double-suction pumps will differ from the single-suction design. This will be covered in detail in the following paragraphs.

Double-Suction Impeller

The method for calculating impeller diameter, impeller width, number of vanes, and vane angularity is identical to the procedure for the single-suction impeller described in Chapter 3. The method for impeller layout will also follow Chapter 3, with a double-suction impeller being considered two single-suction impellers back to back. With double entry the eye area is greater and the inlet velocity lower, thus reducing NPSHR.

Side Suction and Suction Nozzle Layout

The importance of hydraulic excellence in the design of liquid passage areas from suction nozzle to the impeller eye or eyes is quite often minimized or unfavorably adjusted for economic reasons. Experience shows that this approach leads to many field NPSH problems. The current trend in industry is one of reducing NPSHA; therefore, it is essential for optimum NPSHR that the design of the suction approach to the impeller eye be carefully controlled.

We know from experience that in the design of the side suction inlet a certain amount of prerotation of the incoming liquid is desirable. To obtain this condition, the baffle (or splitter) is provided. This splitter is rotated 30° to 45° from suction centerline in the direction of pump rotation. The splitter will locate the radial section of zero flow, and the areas will progressively increase in both directions away from it.

The following drawings and information must be available to design a side suction.

1. Volute layout.
2. Impeller layout.
3. Shaft or sleeve diameter at the impeller.
4. Suction nozzle size.

Layout of the laterally displaced side suction should be done in two parts:

1. Sketch an approximate end view and profile (Figures 7-2 and 7-3) using the following guidelines:

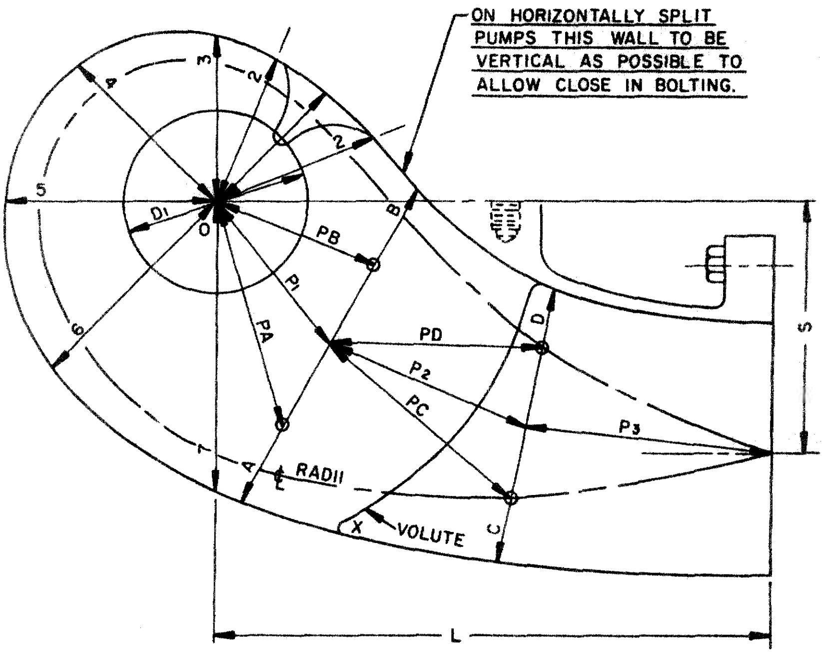


Figure 7-2. Double-suction layout—end view.

Table 7-1
Linear Dimensions of Suction Sections

Section	Approx. Dimension
1	$D_1 \times 0.84$
2	$D_1 \times 0.90$
3	$D_1 \times 0.95$
4	$D_1 \times 1.06$
5	$D_1 \times 1.17$
6	$D_1 \times 1.30$
7	$D_1 \times 1.65$
A-B	$D_1 \times 1.8$ to 2

Table 7-2
Areas of Suction Sections

Section	Area
6	.5 (Area at A-B)
5	.375 (Area at A-B)
4	.25 (Area at A-B)
3	.125 (Area at A-B)

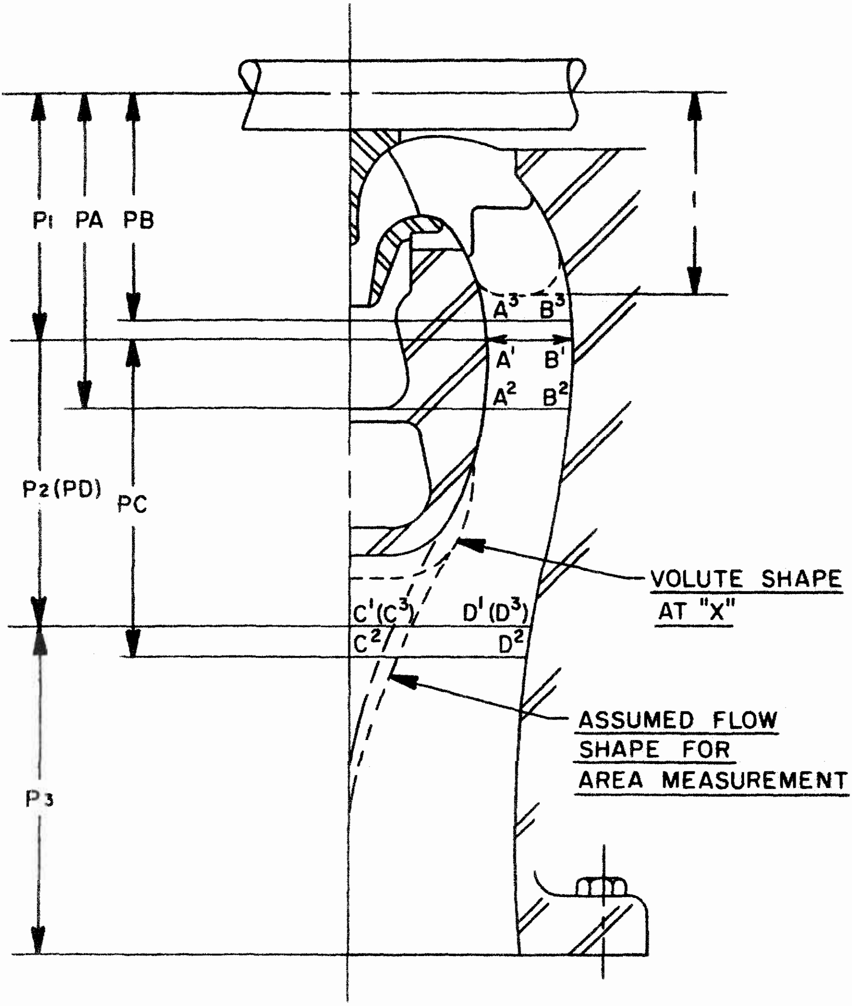


Figure 7-3. Double-suction layout—profile.

a. For linear dimensions of Sections 1 through 7, and A-B, use Table 7-1.
b. Area progression from nozzle to impeller should follow Figure 7-4. The range suggested permits impellers of different suction specific speeds to be used with a common suction.
c. Areas of Sections 3 through 6 measured normal to the flow are shown in Table 7-2.

2. Make final layout after checking all areas and dimensional location of suction nozzle.

Suction Layout (End View)

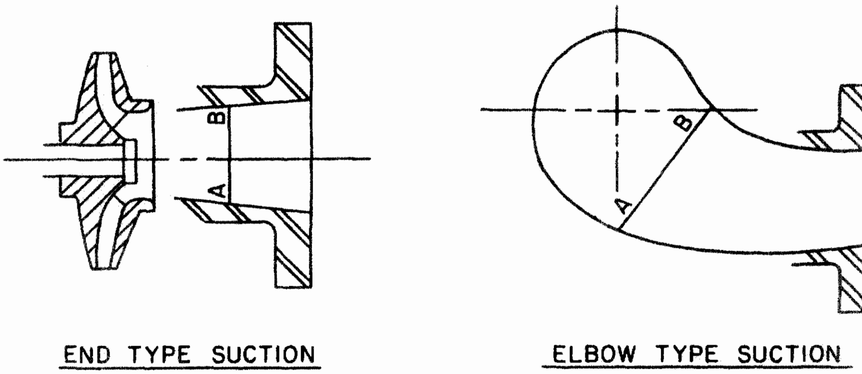
See Figure 7-2 for a diagram of the layout.

1. Draw a circle D_1 at point O.
2. Locate Sections 1 through 7 from Table 7-1.
3. Locate Section A-B perpendicular to flow, approximately D_1 dimension from O. For length of A-B use Table 7-1.
4. Locate nozzle dimensions L and S and mark off nozzle diameter. Dimension L should be only long enough for gradual area progression and clearance behind the flange bolting. This clearance becomes more critical on horizontally split pumps, where nozzle bolting may interfere with the parting flange.
5. Connect all points freehand.
6. Locate Section C-D somewhere between A-B and nozzle.
7. Layout volute metal line.
8. Lay in chords P_1, P_2, and P_3 to mid point of A-B, C-D, and nozzle respectively.

Suction Layout (Profile)

See Figure 7-3 for a diagram of the profile.

1. Layout impeller shape, including shaft and sleeve.
2. Layout volute shape and metal line at intersection of chord P_2.
3. Layout volute shape in dotted lines at location X to observe maximum blockage.
4. Curve metal line around volute sections into the impeller eye, maintaining minimum metal thickness.
5. Mark off chord lengths P_1, P_2, and P_3.
6. Locate nozzle diameter.
7. Approximate width of $A' - B'$ and $C' - D'$ to satisfy area progression from Figure 7-4.
8. Connect outer wall points freehand with ample curvature into impeller eye.
9. Lay in Sections 1 through 7. In this example only Section 1 is shown.
10. Develop Sections A-B and C-D as surfaces of revolution (Figure 7-5). Sections can be divided into any number of increments (e.g., PA and PB). Transfer these chords from the end view to the profile, measure dimensions $P' - A'$ $P' - B'$ and transfer to section layout.



LOCATION	AREA
IMPELLER EYE	100
SECTION A–B	120 TO 140
SUCTION FLANGE	132 TO 169

Figure 7-4. Suction area progression.

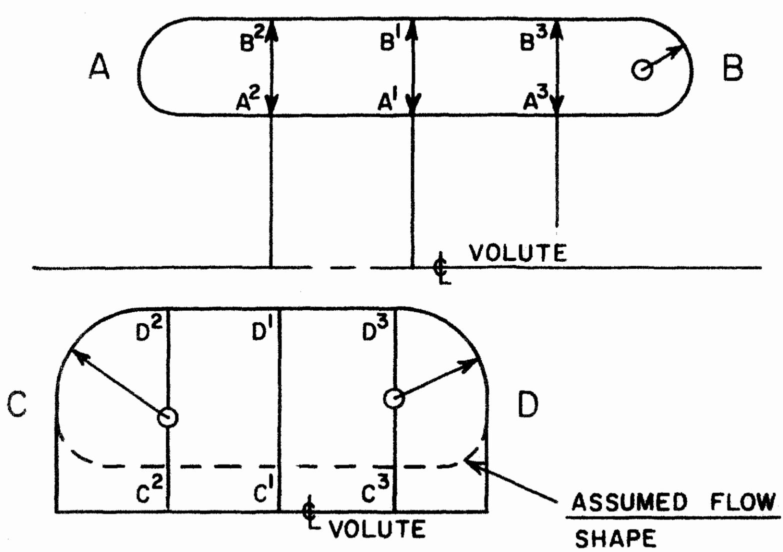


Figure 7-5. Double-suction layout—sections.

11. Repeat Step 10 for Sections C-D or any other sections deemed necessary.
12. Close sections with appropriate radii. These should be liberal for castability and should follow a smooth transition in the end view.
13. Check areas and if necessary adjust end view and/or profile view to satisfy area progression (Figure 7-4).

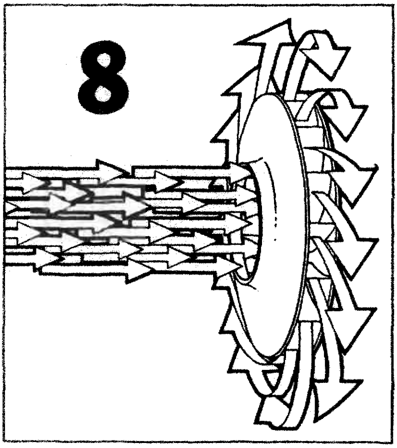
Example

Double Suction Pump:

- Eye diameter D_1 = 5.5 in.
- Shaft diameter = 3 in.
- Total eye area = 32 in.^{2}
- Area at A-B = 39 in.^{2}
- Suction nozzle = 8 in. Area = 50 in.^{2}
- Area section #6 = 19.5 in.^{2}
- Area section #5 = 14.6 in.^{2}
- Area section #4 = 9.75 in.^{2}
- Area section #3 = 4.9 in.^{2}

The double-suction casing in combination with a double-suction impeller has an inherent NPSH advantage over a single-suction combination. Reduction of the required NPSH is normally 40% to 50%. This NPSH advantage of the double suction model makes it possible to be adopted (in addition to the standard horizontally split, single-stage pump) to many different types of centrifugal pumps such as:

- Double-suction single-stage vertically split casing pumps, suitable for high temperatures and medium pressures.
- As a first stage in a horizontally split multi-stage double-volute-type pump.
- As a first stage in the barrel-type multi-stage units, suitable for very high pressures and temperatures up to 800°F.
- To the overhung API process single-stage pumps.
- In vertical pumps (can type) as a single-stage or as a first-stage arrangement, to reduce "can" length.
- Vertical in-line booster pumps.



NPSH

The expressions NPSHR and NPSHA are accepted abbreviations for net positive suction head required and net positive suction head available. Probably more has been written on NPSH than any other subject involving pumps. With so much literature available, one might assume that NPSH and its relationship to cavitation is well understood. Nothing could be further from the truth. To this day, NPSH is still misunderstood, misused, and misapplied, resulting in either costly over-design of new systems or unreliable operation of existing pump installations. Avoiding these problems requires accurate prediction of NPSHR, supply of sufficient NPSHA, and attention to suction piping approach. NPSHR can be considered the suction pressure required by the pump for safe, reliable operation.

Establishing NPSHA

Establishing NPSHA, which is the head available characteristic of the system that provides flow of liquid to the pump, is the responsibility of the system designer. As NPSHR increases with pump capacity, normal practice is to establish NPSHA at the operating condition, then add a reasonable margin to accommodate any anticipated increase in pumping capacity. All too often, future operating problems begin here. It is not unusual for the ultimate user to add some anticipated increase in capacity, then for insurance the contractor designing the system adds even more. When the pump designer finally gets the data sheet, he designs the impeller inlet and suction nozzle geometry for operating capacity, plus user margin, plus contractor margin. If these margins are not carefully controlled, the result can be an over-sized pump that operates well to the left of BEP (Figure 8-1). Such over-designed pumps are vulnerable to surging, recirculation, cavitation, noise, and vibration. This is particularly true with high-suction specific speed pumps above 11,000 where the inlet geometry has already been extended for minimum NPSH.

For minimum energy consumption and trouble-free operation, pumps should ideally be operated between 80% and 100% BEP. As this is not always possible or practical, pumps will often operate at lower flows. It is therefore important that the minimum flow for continuous trouble free operation be carefully considered by the pump designer. Minimum flow is influenced by physical pump size, margin between NPSHA and NPSHR, impeller inlet geometry, suction nozzle geometry, mode of operation, and last but not least, the liquid being pumped. With so many variables it is not unusual to find recommended minimum flows ranging from 10% to 60% BEP.

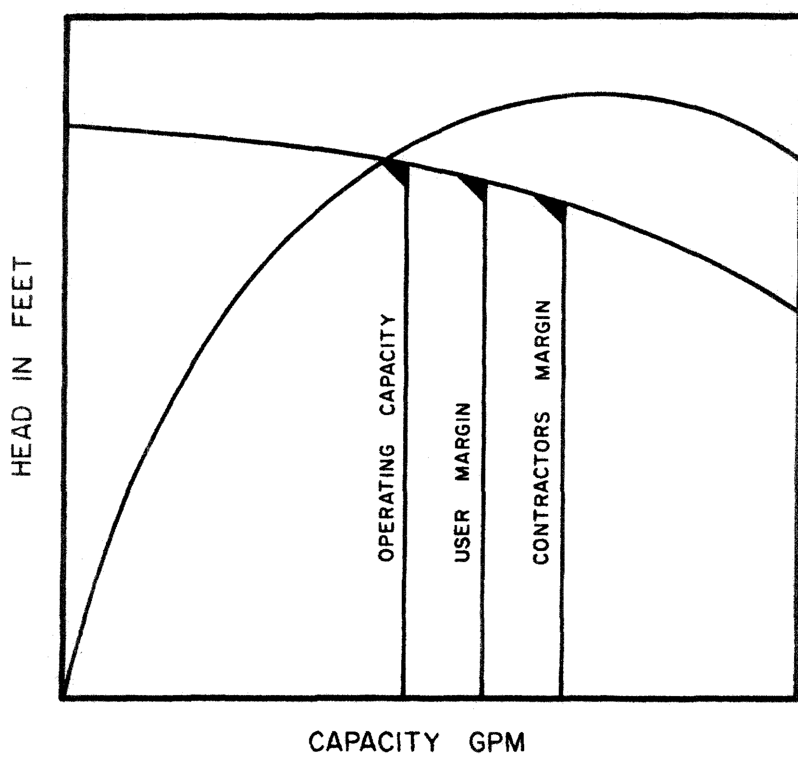


Figure 8-1. Margins of safety result in oversized pump.

Predicting NPSHR

The other side of the coin necessary for reliable operation is of course accurate prediction of NPSHR by the pump designer.

In considering NPSHR, it is necessary to understand that a centrifugal pump is designed as a hydraulic machine to move liquids. Any amount of entrained air or gas present will cause a deterioration in pump performance. Various tests substantiate the claim that a volume of only one percent air or gas will cause a loss of head and efficiency. As liquid travels from suction nozzle to impeller eye, it will experience pressure losses caused by friction, acceleration, and shock at blade entry. If the summation of these losses permits vaporization of the liquid, vapor bubbles will form in the impeller eye, travel through the impeller, and upon reaching a high pressure region, collapse. This collapse or implosion of the vapor bubbles is classic cavitation, which can lead to impairment of performance and impeller damage (Figures 8-2 and 8-3). Thus, predicting

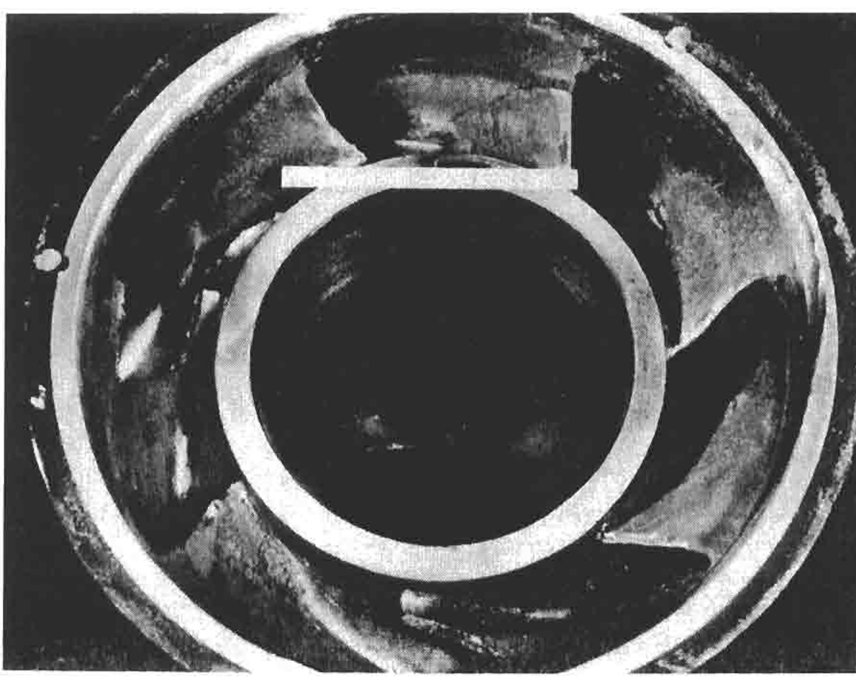


Figure 8-2. Cavitation damage—looking into impeller eye.

Figure 8-3. Cavitation damage—at leading edge of one vane.

NPSHR is in fact predicting the losses in the critical area between suction nozzle and the leading edge of the first-stage impeller blades (Figure 8-4).

Moderate Speed Pumps

One method used successfully for many years by pump designers will predict NPSHR with reasonable accuracy when the pump liquid is water. The inlet velocity of the liquid entering the impeller eye, C_{M1}, and the peripheral velocity of the impeller blade, U_t, are calculated and the ratio used to predict NPSHR (Figure 3-6). The chart is valid only for flows between 50% and 100% BEP at maximum impeller diameter. For capacities above 100% BEP, apply the factors indicated on the chart. Development of this chart is a result of acquisition of many years of pump test data. Values read from the chart will approximate the NPSHR established during pump testing using 3% head loss as criteria. When velocities exceed those shown, the chart should not be extrapolated.

The most commonly used method in determining the cavitation characteristics of a centrifugal pump is to cause a breakdown in the normal head capacity curve. This is done by holding the speed and suction pressure

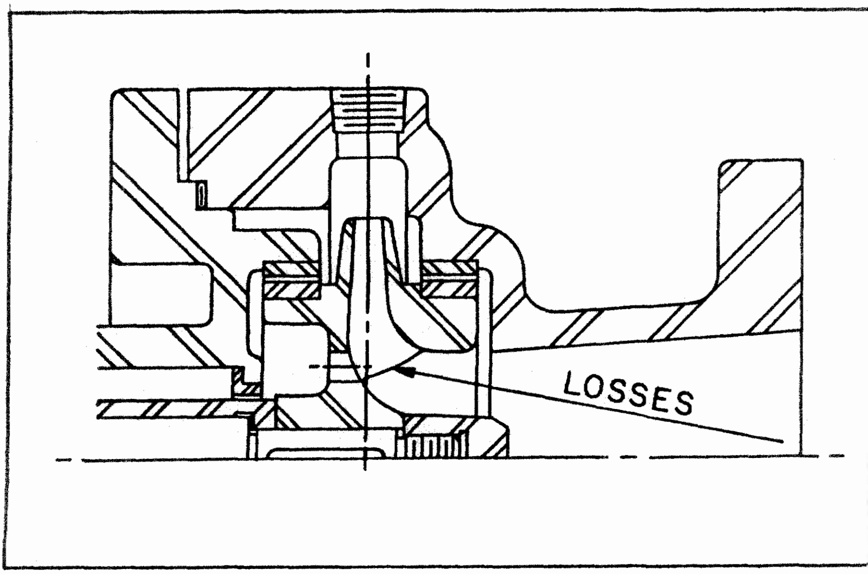


Figure 8-4. Pressure loss between suction nozzle and leading edge of impeller vane.

constant and varying the capacity, or by holding the speed constant and reducing the suction pressure at various capacities. Either of these methods will produce a breakdown in the head characteristic as shown in Figure 8-5, indicating a condition under which the performance of the pump may be impaired.

Accurately determining the inception of cavitation requires extreme control of the test and involves sophisticated instrumentation. The Hydraulic Institute has permitted a drop in head of 3% to be accepted as

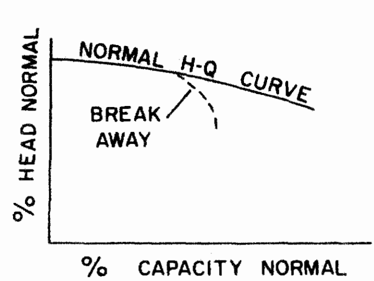


Figure 8-5. Loss in head during cavitation test.

evidence that cavitation is present under test conditions. They do caution, however, that where it is important to establish normal operation with an appreciable margin over the minimum required NPSH, values as low as 1% should be used. They further caution, that the pump should be operated above the break-away sigma if noise and vibration are to be avoided.

Influence of Suction Specific Speed (N_{ss})

When evaluating NPSHR one should always refer to the suction specific speed (N_{ss}) and not to sigma, whose value depends on the head developed. Thus the same impeller when tested before and after a diameter trim will show different values of sigma yet will behave identically under cavitation. In comparing cavitation performance and predicting NPSHR, we prefer to use the suction specific speed parameter. This can be applied in the same manner as sigma yet has more significance as it relates only to the inlet conditions and is essentially independent of discharge geometries and pump specific speed.

Figure 8-6 shows suction velocity triangles for N_{SS} from 7,000 to 16,000. It graphically shows that as suction specific speed increases, normal vane entrance angle becomes flatter, C_{M1} relative to shaft becomes smaller, and peripheral velocity at impeller eye becomes greater. The ratio of C_{M1} to U_t for higher suction specific speed, is very small and chances for cavitation are greatly increased. This cavitation would appear for the following reason: To move liquid from one point to another, a change of pressure gradient must take place. But as C_{M1} (velocity) is reduced, velocity head, $V^2/2g$, is also reduced. When it becomes lower than the head required to overcome impeller entrance losses, the liquid will backflow creating cavitation and pump damage.

Figure 8-7 represents a test of a four-inch pump with eight different suction specific speed impellers. Best efficiency point of all impellers is the same. Impeller profile is also the same, but impeller eye geometry is different for each suction specific speed. Some tests on different types of pumps might show different results, but the trend should be the same. This test shows the stable cavitation-free window for seven suction specific speeds and could be used as a guide for pump selection.

High Speed Pumps

The prediction methods and NPSH testing just described under moderate speed pumps are valid and well substantiated for low- to moderate-speed pumps. It is our opinion, however, that neither method will ensure

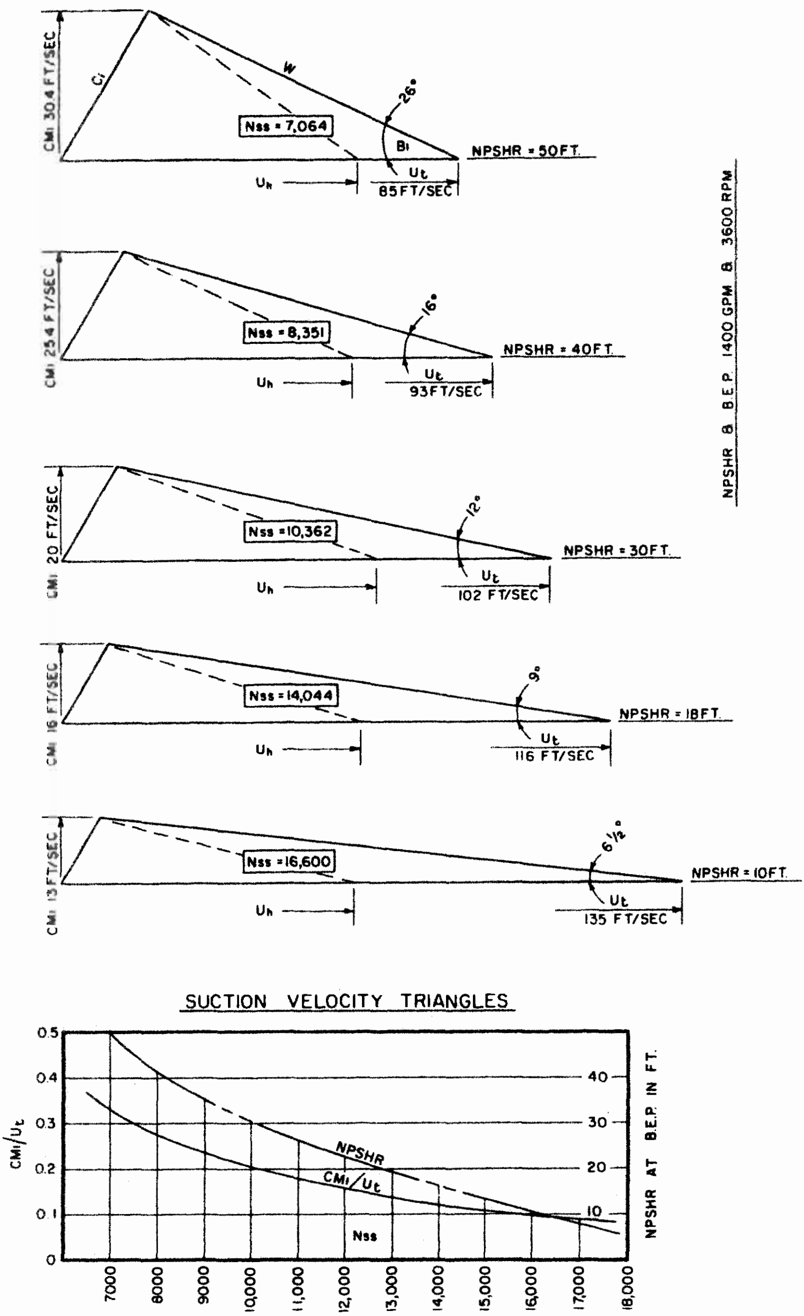


Figure 8-6. Change in velocity triangle with suction specific speed.

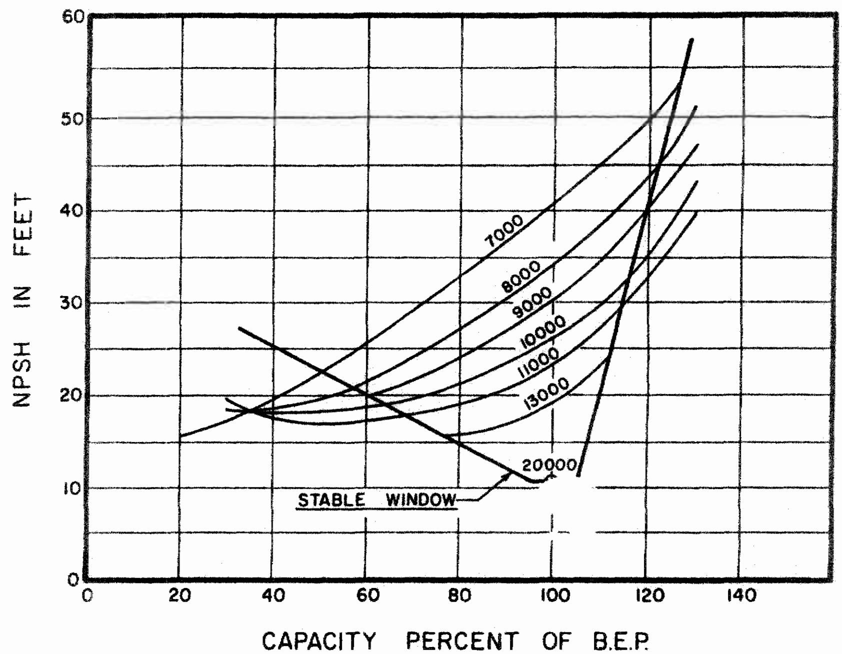


Figure 8-7. Stable operating window vs. suction specific speed.

damage-free operation on high speed pumps. The term "high speed" does not refer to RPM, but to blade peripheral velocities above 160 ft/sec. Thus a pump operating at high RPM with a small eye diameter may be less critical than a slower pump with a larger eye.

A weakness of the 3% head loss suppression test is that even when conducted at full speed it fails to provide any evidence of the extent of cavitation damage when the pump operates in this 3% zone for extended periods. This is particularly true of high-speed pumps operating on water where cavitation damage can occur at NPSHA values above 3% head loss and in some instances, even with 0% head loss. Even the inception of cavitation detected by acoustic noise testing does not establish the value of NPSHA for damage-free operation.

To prevent damage and permit safe operation of high-speed pumps, it is suggested that pumps be supplied with NPSH predicted on a theoretical basis and that this characteristic performance be termed the "damage free" or "cavitation free" NPSHR curve. This method of prediction could be useful in complying with specifications that require a guarantee of 40,000 hours damage-free operation due to cavitation.

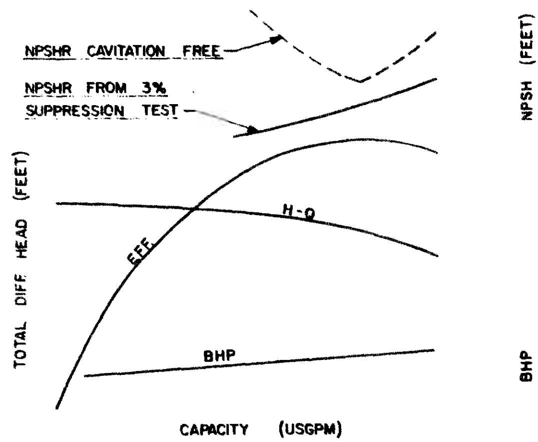


Figure 8-8. Performance curve showing NPSHR cavitation free and NPSHR 3% head loss.

"Cavitation-free NPSHR" cannot be demonstrated by a suppression test, as no head loss will be evident. Therefore, to satisfy the normal requirement for testing on the pump manufacturer's test stand, it is suggested that two NPSHR curves be offered with the pump quotation. These would be "cavitation-free NPSHR" and the conventional 3% head loss NPSHR (Figure 8-8).

Cavitation-Free NPSHR

As described earlier, cavitation is the formation of vapor-filled cavities in the pumped liquid resulting from a sufficient reduction of the liquid pressure to vaporize a proportion of the liquid. To prevent this vaporization and the damage associated with it, the pump designer must first consider the head losses in the most critical area, which is between the inlet nozzle of the pump and the leading edges of the first-stage impeller blades. This head loss is a result of the following factors:

1. Head loss due to friction.
2. Head drop due to fluid acceleration, which is the energy required to accelerate the flow from the suction nozzle to the impeller eye.
3. Head shock loss due to blade entry, which is the localized drop at the blade leading edge and is a function of the angle of attack and blade entry shape.

Both the friction losses and the acceleration losses are proportional to the square of the liquid velocity as it approaches the impeller eye with a constant of proportionality designated by K_1. The losses due to blade entry are proportional to the square of the velocity of flow relative to the blade leading edge with a constant of proportionality designated by K_2. To prevent cavitation, these losses must be compensated by supplying adequate NPSH to the pump. This "cavitation-free" NPSHR can then be expressed as:

$$NPSHR = K_1 C_{M1}^2/2g + K_2 w^2/2g \quad (8\text{-}1)$$

where the first term, $K_1 C_{M1}^2/2g$, represents the friction and acceleration losses, and the second term, $K_2 w^2/2g$, represents the blade entry losses. From this, it can be seen that, in small pumps of low speed, the first term is predominant, while for large and/or high speed pumps, the second term is the controlling factor and the first term is of secondary importance. This explains why it is often possible to reduce NPSHR on moderate speed pumps by changing to a larger eye impeller. As cavitation is most likely to occur in the region where the relative velocity, w, is highest, the calculation is based only on the maximum diameter of the blade tip, D_t, at the impeller entry.

The incidence angle, α, that influences K_2 is the difference between the inlet blade angle, B_1, and the flow angle, θ (Figure 8-9). B_1 is determined from C_{M1} multiplied by factor R_1, which allows for the effects of recirculated flow, Q_L, and nonuniform velocity distribution. As the leakage, Q_L, does not remain constant due to internal erosion, and as many engineers differ in their selection of R_1, it is seldom if ever that α equals zero. Leakage Q_L through impeller wear ring clearances and balance

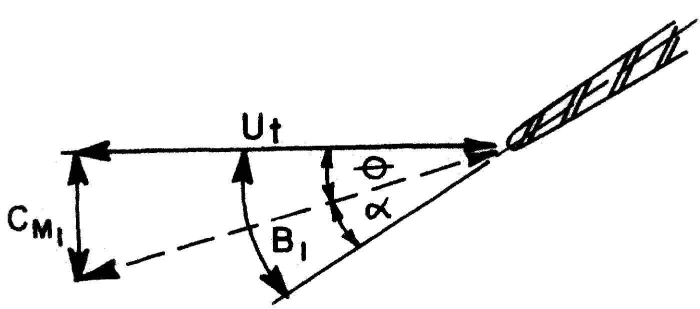


Figure 8-9. Blade and flow angle at impeller inlet.

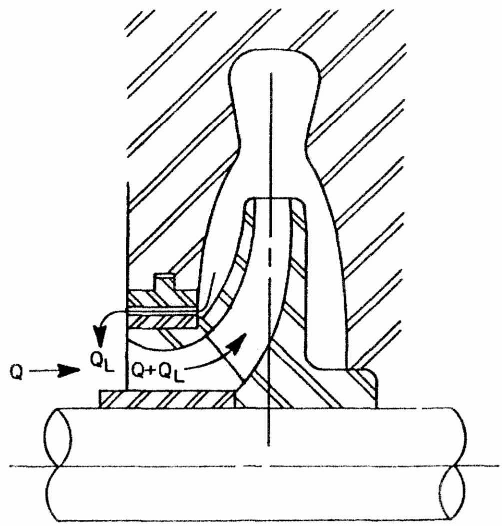


Figure 8-10. Leakage across wear ring back to impeller eye.

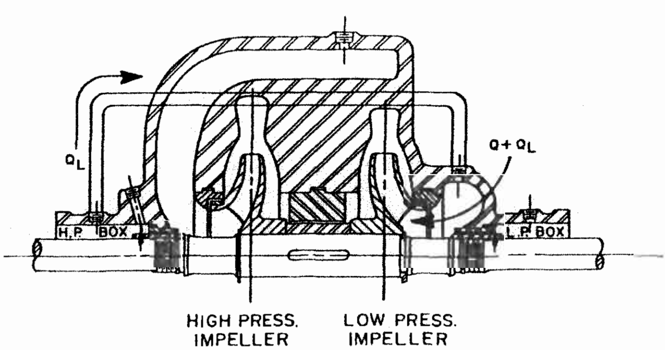


Figure 8-11. Leakage across high pressure bushing back to impeller eye.

lines in a multi-stage pump (Figures 8-10 and 8-11) should be added to the flow Q entering the impeller eye. This leakage will vary with the head developed and therefore has more influence on α during low flow operation. Pumps of low specific speed where ring leakage can be a significant percentage of pump flow will show increased NPSHR with increased ring clearance. One example is shown in Figure 8-12.

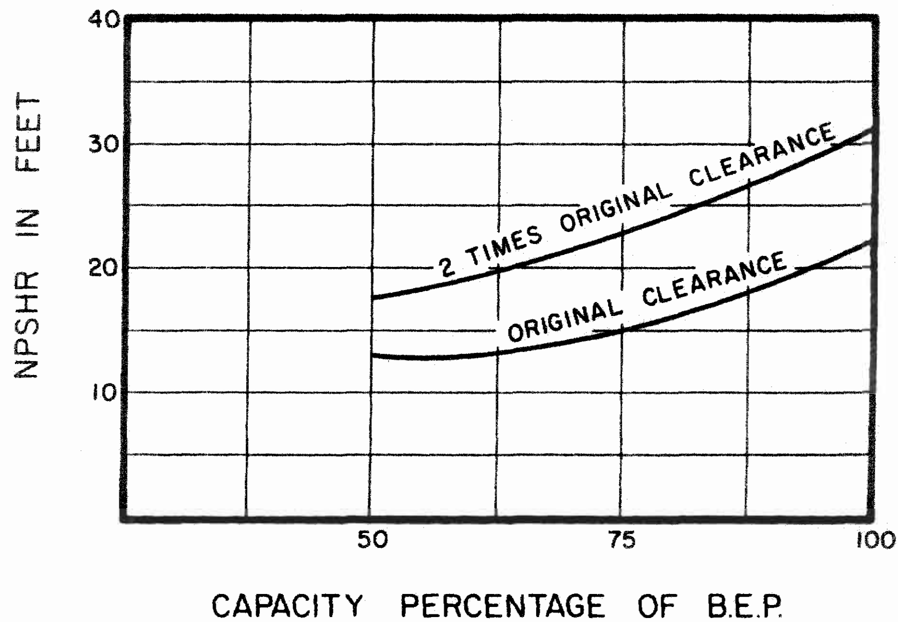


Figure 8-12. Influence of wear ring clearance on NPSHR.

Influence of Suction Nozzle

K_1 is largely influenced by the pump suction nozzle approaching the impeller eye. To confirm values of K_1, a pump was modified by installing area reducing steel plates in the suction approach as shown in Figures 8-13, 8-14, 8-15, and 8-16. The plates were first formed in wood conforming to existing suction nozzle core box shape, then manufactured in steel and welded in place. NPSHR tests were conducted before and after the modification in accordance with the standards of the Hydraulic Institute, using 3% head loss as the criterion. The results, shown in Figure 8-17, illustrate the influence of nozzle geometry approaching the impeller and lead to the observations now described.

The test pump was of the double-inlet split-case type. This complex suction passage from nozzle to impeller eye, makes it difficult to determine analytically the expected losses. It is documented that a design of this type, which changes the direction of flow at least four times and moreover splits the flow into two separate channels, can have appreciable influence on the energy loss—sometimes as high as 2% to 3% of the effective head of one impeller. The improvements experienced in the test are difficult to confirm analytically. However, it is obvious that the reduced energy loss could not result from velocity change alone.

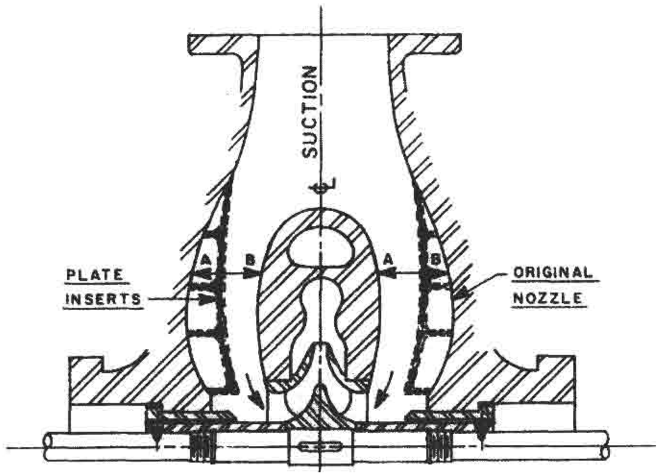


Figure 8-13. Plate inserts in double-suction nozzle.

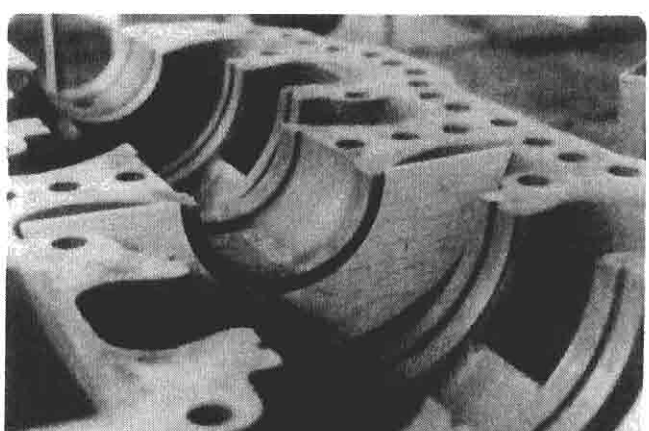


Figure 8-14. Wooden templates for plate inserts.

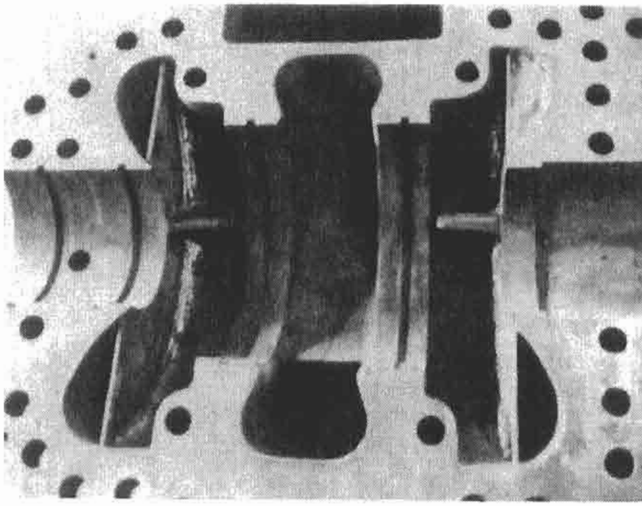


Figure 8-15. Plate inserts installed in upper half case.

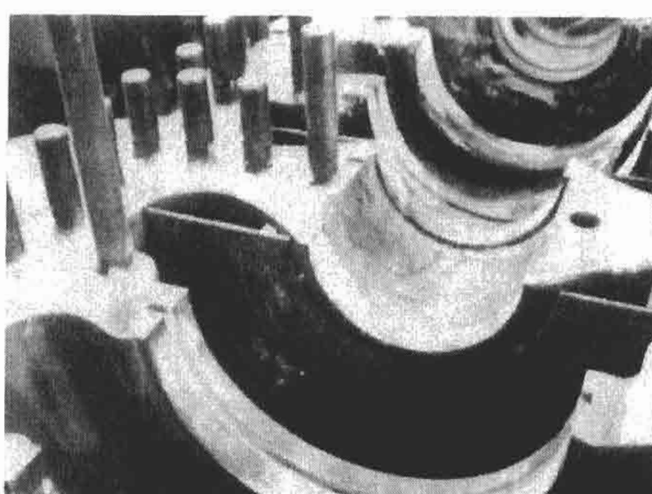


Figure 8-16. Plate inserts installed in lower half case.

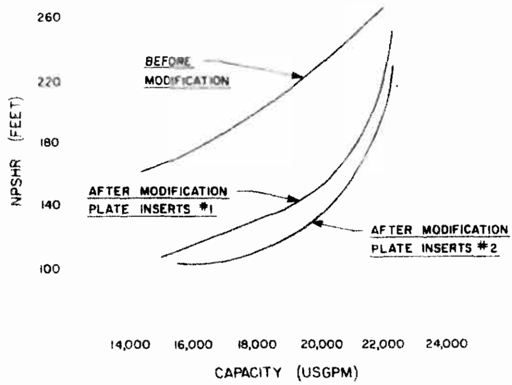


Figure 8-17. Influence of plate inserts on NPSHR.

It can be assumed that this improvement is influenced by reduced separation, improved flow stability and streamlining, a progressive increase in velocity, and the resulting reduced turbulence. It must also be accepted that any disturbance in the approach to the impeller can cause unequal distribution of flow rates into the two impeller eyes at different locations. These diversions from the correct angle of attack at the leading edge of the blades produce a corresponding head loss.

The test confirms that the cavitation characteristics of a good impeller design can be impaired by poor suction nozzle design. This is particularly true with double-entry impeller pumps where the complex nozzle geometry can adversely affect K_1. A well designed suction nozzle has a gradual decrease in area from nozzle to impeller, allowing a progressive increase in velocity. Area distribution guidelines are shown in Figure 7-4. The range suggested permits impellers of different suction specific speeds to be used with a common suction.

Using the actual suction nozzle area ratio at Section A-B gives a reasonable means of estimating K_1 (Figure 8-18). For the same incidence angle α, K_2 has a higher value at capacities above design. For estimating K_2 use Figure 8-19.

Influence of Liquid

The boiling of the liquid in the process of cavitation is a thermal process and is dependent on the liquid properties, pressure, temperature, latent heat of vaporization, and specific heat. To make this boiling possible, the latent heat of vaporization must be derived from the liquid flow.

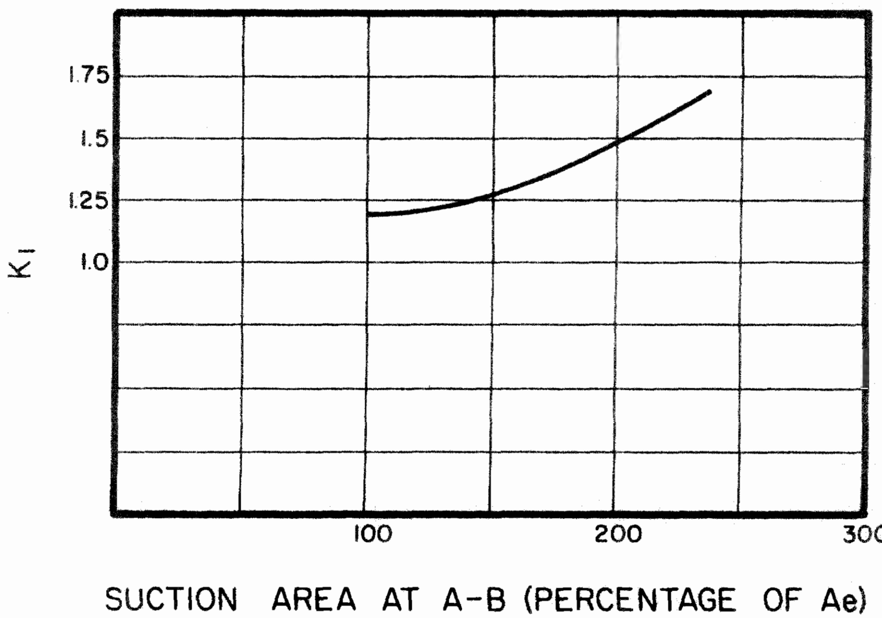


Figure 8-18. Estimating K_1.

The extent of cavitation damage depends on the proportion of vapor released, the rapidity of liberation, and the vapor specific volume. Taking this into consideration the cavitation face calculation can be corrected by applying a gas-to-liquid ratio factor C_b (Figure 8-20). On this basis, cold water must be considered the most damaging of the commonly pumped liquids. Similarly, this difference in behavior applies to water at different temperatures. A review of the properties of water and its vapor at several temperatures shows the specific volume of vapor decreases rapidly as pressure and temperature increase. This difference in behavior under cavitating conditions makes cold water more damaging than hot.

The problems associated with cold water are substantiated by operating experience in the field, where pumps handling certain hydrocarbon fluids or water at temperatures significantly higher than room temperature will operate satisfactorily with a lower NPSHA than would be required for cold water.

$$\text{NPSHR} = [K_1 C_{M1}^2/2g + K_2 w^2/2g]C_b$$

$$= [(K_1 + K_2)C_{M1}^2/2g + K_2 U^2/2g]C_b \qquad (8\text{-}2)$$

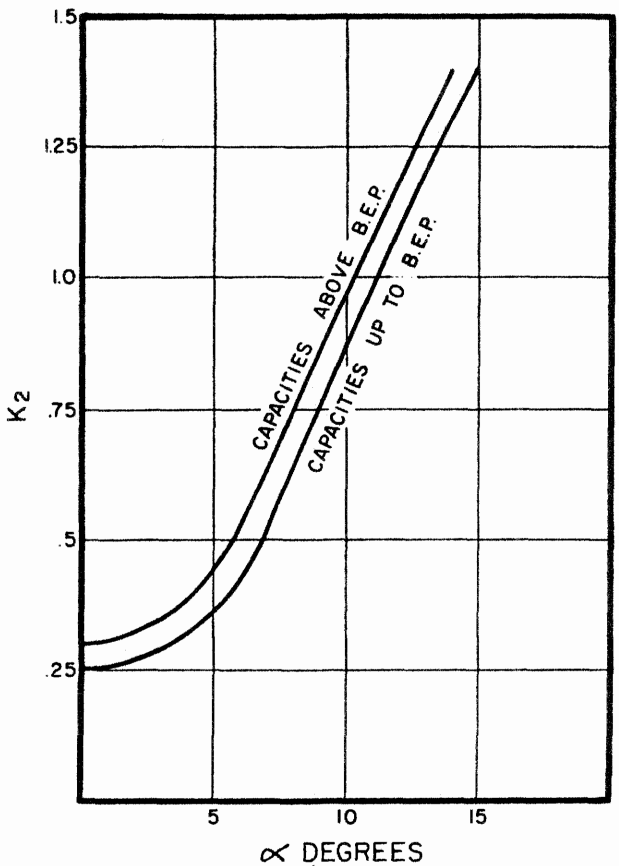


Figure 8-19. Estimating K_2.

Example

Ignoring internal leakage back to the impeller, calculate NPSHR cavitation free for the pump now described.

- Capacity—1,800 GPM
- Product—Water
- Temperature—70°F
- Speed—8,100 RPM
- Impeller eye area—17.2 sq in.
- Eye diameter—5 in.
- Inlet blade angle—15°
- Suction area at A-B—24 sq in.

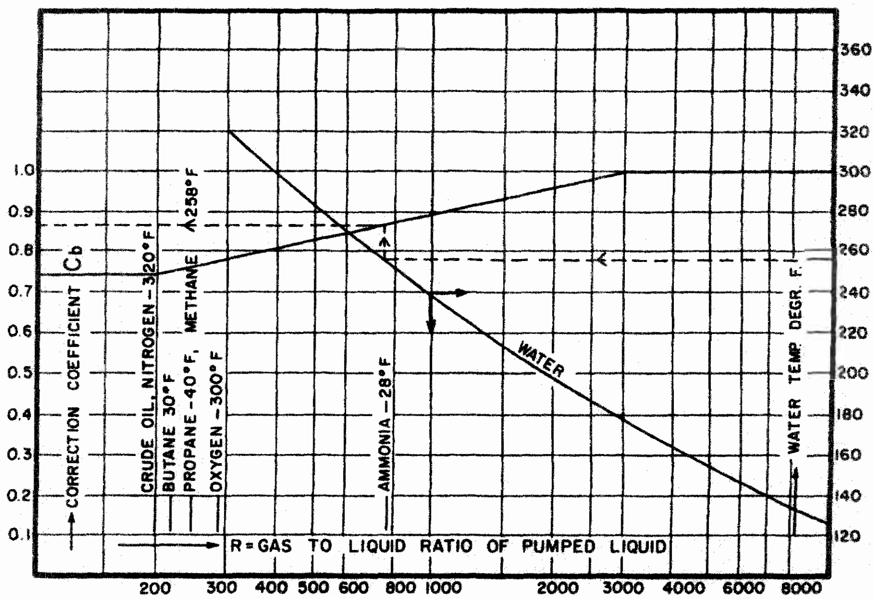


Figure 8-20. Gas-to-liquid ratio vs. NPSH correction factor, C_b (from D. J. Vlaming, "A Method of Estimating the Net Positive Suction Head Required by Centrifugal Pumps," ASME 81-WA/FE-32).

Step 1: Determine K_1.

Ratio at A-B = 24/17.2 = 1.4

From Figure 8-18:

$K_1 = 1.25$

Step 2: Calculate θ.

From Figure 8-9:

$$
\begin{aligned}
\tan\theta &= C_{M1}/U_t \\
&= .321\ Q/Ae \div D_tN/229 \\
&= .321 \times 1{,}800/17.2 \div 5 \times 8{,}100/229 \\
&= 33.6/176.8 = 0.19
\end{aligned}
$$

$$\theta = 10.75°$$

Step 3: Determine K_2.

From Figure 8-9:

$$\begin{aligned}\alpha &= B_1 - \theta \\ &= 15 - 10.75 \\ &= 4.25^\circ\end{aligned}$$

From Figure 8-19, as 1,800 GPM is less than BEP:

$$K_2 = .32$$

Step 4: Calculate NPSHR from Equation 8-2.

$$\text{NPSHR} = [(1.25 + .32)33.6^2/64.4 + .32 \times 176.8^2/64.4]C_b$$

From Figure 8-20,

$$\begin{aligned}C_b &= 1.0 \\ \text{NPSHR} &= 27.5 + 155.3 \\ &= 183 \text{ ft}\end{aligned}$$

Figures 8-21, 8-22, and 8-23 show examples of calculating NPSHA.

Suction Piping

As described earlier, NPSHR is influenced by suction nozzle design. Similarly, poor suction piping can adversely affect NPSHR and pump performance. Double-suction first-stage impellers are particularly vulnerable to a nonuniform approach of the liquid. If elbows are located close to the pump, they should be oriented to provide equal distribution of flow into both eyes of the impeller. An elbow parallel to the pump shaft directly before the pump is conducive to spiral flow and unbalanced flow distribution into the two eyes. Unequal flow distribution can result in excessive vibration, high axial thrust loads, noise, and cavitation.

Effect of Viscosity

Although the influence of viscosity is predictable on other hydraulic characteristics, particularly head, capacity, and efficiency, little general information is available to indicate the effect on NPSHR. From experience we know that up to 2,000 SSU we are safe in using water NPSHR

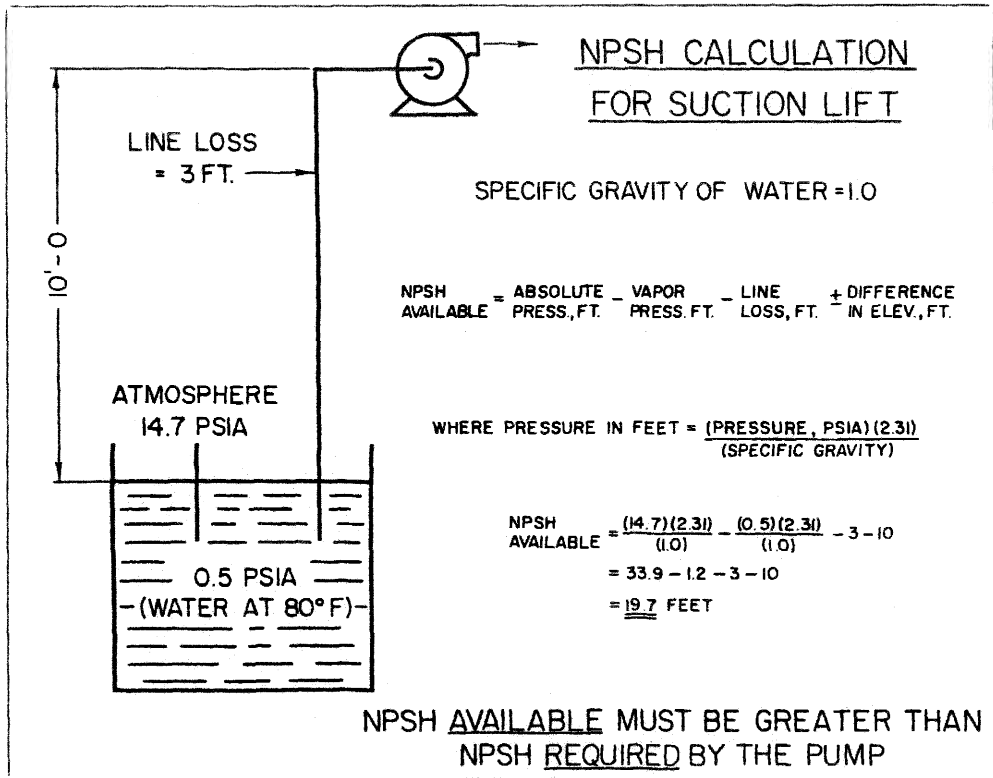
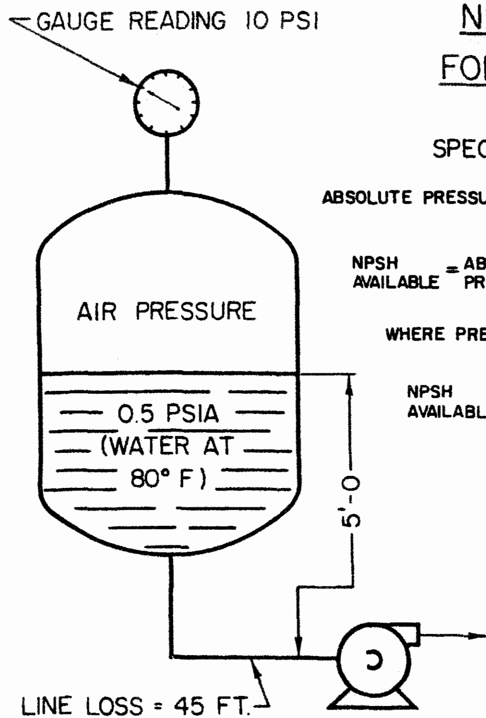


Figure 8-21. Calculating NPSHA for suction lift.



NPSH CALCULATION FOR PRESSURED DRUM

SPECIFIC GRAVITY OF WATER = 1.0

ABSOLUTE PRESSURE = GAUGE PRESSURE + ATMOSPHERIC PRESSURE
= GAUGE PRESSURE + 14.7

$$\text{NPSH AVAILABLE} = \text{ABSOLUTE PRESS., FT.} - \text{VAPOR PRESS., FT.} - \text{LINE LOSS., FT.} \pm \text{DIFFERENCE IN ELEV., FT.}$$

$$\text{WHERE PRESSURE IN FEET} = \frac{(\text{PRESSURE, PSIA})(2.31)}{(\text{SPECIFIC GRAVITY})}$$

$$\text{NPSH AVAILABLE} = \frac{(10+14.7)(2.31)}{(1.0)} - \frac{(0.5)(2.31)}{(1.0)} - 45 + 5$$

$$= 57.1 - 1.2 - 45 + 5$$

$$= \underline{\underline{15.9}} \text{ FEET}$$

NPSH AVAILABLE MUST BE GREATER THAN NPSH REQUIRED BY THE PUMP

Figure 8-22. Calculating NPSHA for pressure drum.

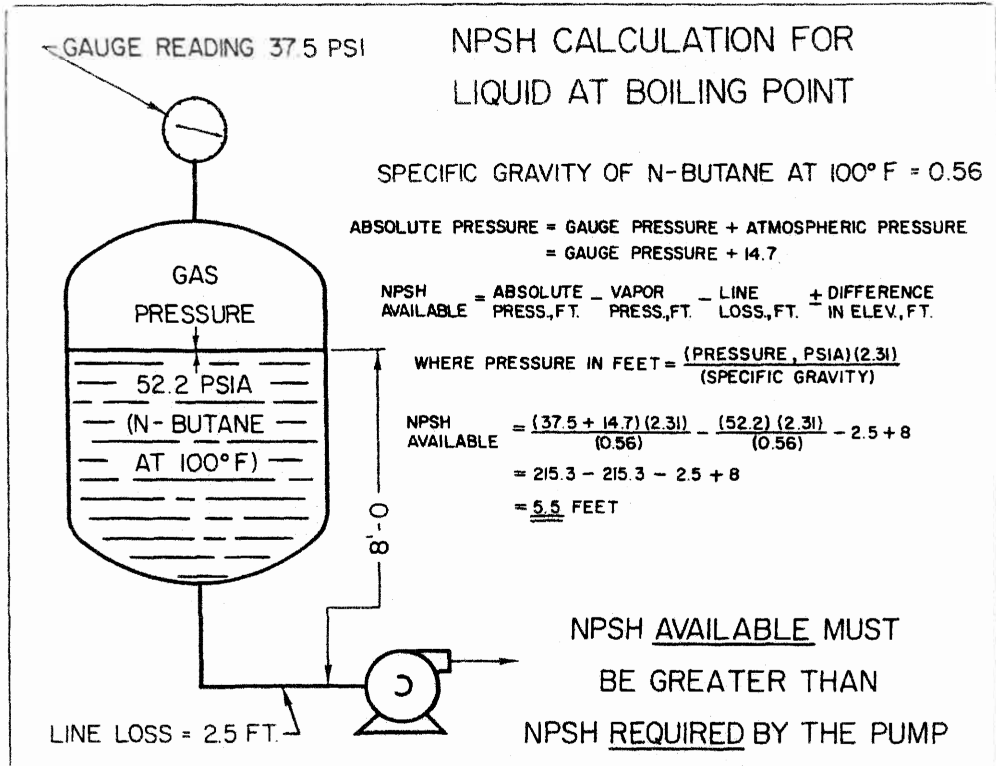


Figure 8-23. Calculating NPSHA for liquid at boiling point.

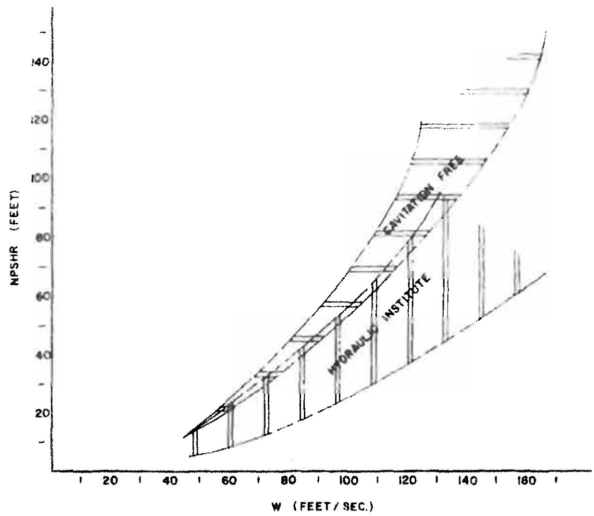


Figure 8-24. NPSHR cavitation free compared with NPSHR Hydraulic Institute.

values. Above 2,000 SSU, it is necessary to use the pump designer's judgment or experimentally determine the cavitation characteristics.

Figures 8-24 through 8-26 compare our cavitation-free NPSHR with NPSHR as permitted by the Hydraulic Institute. The Hydraulic Institute values shown were established by test and cavitation-free values calculated. Of 98 pump tests reviewed, 60 were selected for compilation of data. Criterion for selection was a witnessed suppression test conducted in accordance with the standards of the Hydraulic Institute. To reduce scatter, test data were confined to capacities limiting α to a maximum of $6\frac{1}{2}°$. The pumps used were limited to a maximum impeller blade peripheral velocity of 120 ft/sec. At higher velocities, the difference between cavitation-free and Hydraulic Institute will be more evident. Suction specific speed values lower than those shown in Figure 8-26 can be expected. General details of the pumps are:

- Number of pumps—50
- Number of tests—60
- Pump size (discharge nozzle)—1 in. to 30 in.
- Impeller diameter—7 in. to 37 in.
- Number of stages—One
- Impeller entry—Single suction: 26 pumps. Double suction: 24 pumps.
- Specific speed range—500 to 4,750

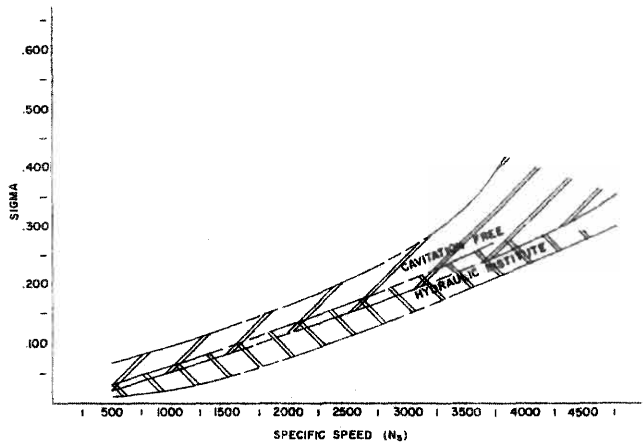


Figure 8-25. Sigma cavitation free compared with Sigma Hydraulic Institute.

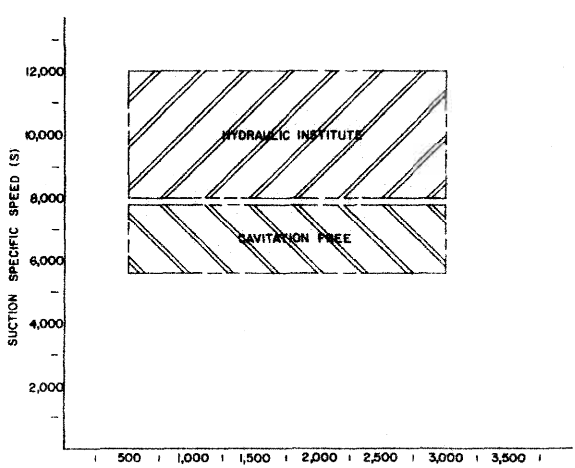


Figure 8-26. Suction specific speed cavitation free compared with suction specific speed Hydraulic Institute.

Notation

K_1	Friction and acceleration loss coefficient
K_2	Blade entry loss coefficient
NPSHR	Net positive suction head required (based on test or by calculation)
NPSHA	Net positive suction head available on site
C_{M1}	Average meridional velocity at blade inlet (ft/sec) = .321 Q/Ae
w	Relative velocity of flow (ft/sec)
U_t	Peripheral velocity of impeller blade (ft/sec) = $D_tN/229$
g	Gravitational acceleration (ft/sec^2)
θ	Angle of flow approaching blade
α	Angle of incidence
σ	Sigma $= \dfrac{NPSH}{H}$
H	Head generated per impeller (ft)
B_1	Blade angle at outer radius of impeller eye
C_b	Gas-to-liquid ratio factor
Q	Pump capacity (GPM)
Q_L	Recirculated leakage entering impeller (GPM)
R_1	Factor in determining B_1
D_t	Diameter at blade tip (in.)
N	Speed (RPM)
N_{SS}	Suction specific speed $= \dfrac{N\,(Q)^{0.5}}{NPSH^{0.75}}$ (for double suction impellers, use $(Q/2)^{0.5}$)
BEP	Best efficiency point on performance curve
N_s	Specific speed $= \dfrac{N\,(Q)^{0.5}}{H^{0.75}}$
Ae	Impeller eye area at blade entry (sq in.)
U_h	Peripheral velocity of impeller blade at hub (ft/sec)
C_1	Absolute velocity of flow (ft/sec)

References

Hydraulic Institute Standards Of Centrifugal Rotary And Reciprocating Pumps, 14th edition.

Lobanoff, V., "What Is This NPSH, *Oil ,Gas Journal*, February 24, 1958.

Ross, R. R., "Theoretical Prediction of Net Positive Suction Head Required (NPSHR) for Cavitation Free Operation of Centrifugal Pumps," United Centrifugal Pumps.

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