Ahmad Nourbakhsh
André Jaumotte
Charles Hirsch
Hamideh B. Parizi

Turbopumps & Pumping Systems

Springer

Turbopumps and Pumping Systems

Ahmad Nourbakhsh · André B. Jaumotte
Charles Hirsch · Hamideh B. Parizi

Turbopumps and Pumping Systems

With 157 Figures and 10 Tables

Springer

Prof. Dr. Ir. S. Ahmad Nourbakhsh
Faculty of Mechanical Engineering
University of Tehran
Tehran-Iran
Tel: +98-21-88005677
Fax: +98-21-88013029
anour@ut.ac.ir

Prof. Dr. Ir. Charles Hirsch, em.
Royal Flemish Academy
for Sciences and Arts
President, NUMECA International
Avenue Franklin Roosevelt, 5
B-1050 Brussels, Belgium
Tel: +32-2-642-2800
Fax: +32-2-647-9398
charles.hirsch@numeca.be

Baron Andre Jaumotte, Prof. Dr. Ir.,
Royal Academy of Sciences
and Arts in Belgium
Universite Libre de Bruxelles
Avenue Franklin Roosevelt
B-1050 Brussels, Belgium
Tel: +32-2-650.26.70
Fax: +32-2-650.27.10
sma@ulb.ac.be

Hamideh B. Parizi, PhD., P.Eng.
Vice President, Simulent Inc.
203 College Street, Suite 302
Toronto, ON., Canada, M5T 1P9
Tel : +1-416-979-5544
Fax : +1-416-979-5519
parizi@simulent.com

Library of Congress Control Number: 2007930456

ISBN: 978-3-540-25129-3 Springer Berlin Heidelberg New York

Springer is a part of Springer Science+Business Media
springer.com

Typesetting: Integra Software Services Pvt. Ltd., India

Cover design: Kirchner Design, Heidelberg

Printed on acid-free paper SPIN: 10963777 89/3180/Integra 5 4 3 2 1 0

Preface

Pumps are one of the mechanical devices that are widely used in many industries throughout the world. Turbopumps which are common types of pumps are used mainly for transporting water in agricultural units, industrial processes, cooling and heating systems, and power plants. But their application is not limited only for handling water. Turbopumps can be used for transporting oil products in oil and gas industries and refineries, as well as byproducts in pulp and paper industry, food and beverage in food industry, and contaminated water in sewage systems.

Because of the versatility of pump usage, the number of turbopumps in operation worldwide is very large and as a result the total energy consumption of these units can be considerable. To reduce the energy consumption by turbopumps, it is important to increase the efficiency of the pumps, as well as the system they are working in. The pump manufacturing companies and design engineers are constantly trying to improve the performances of pumps by designing more efficient pumps and by using the latest technologies available in terms of design software, new developed materials, and state of the art manufacturing tools. Today, computational fluid dynamics and sophisticated design software tools help designers to optimize the fluid flow inside the pumps and to design different pump elements with better hydraulic efficiencies.

However, the operation of turbopumps can be optimized a well by effectively using proper pumps and designing efficient piping systems. When pumps are selected correctly and installed properly in a system, they would work at their best efficiency points, thus, reducing energy consumption. Therefore, it is essential for all consulting engineers working in this field, pump users, and operating personnel to have enough knowledge about the principals of turbopump operation and most importantly have the expertise to select, install and operate turbopumps properly.

The authors have continuously collaborated as technical adviser and consultant with many industries in Germany, Belgium, U.S.A., France, England, Iran and Switzerland. Throughout these collaborations, they noticed there is a need for a reference book in this field that not only covers the basic information about pump design but most importantly addresses the needs of pump users who are involved in pump selection and operation. Therefore, the author's main objective was set to write a book to address this second group of engineers, i.e., pump users.

This book has been divided into two major Parts. In Part I, Turbopumps, the basic information about pumps classification, definitions, principal of operation, and construction elements presented. In Part II, Pumping Systems, the important parameters in pump operation, selection, pumping systems, and pump stations are presented.

The materials of this book were mainly collected and presented by authors during many years of their experiences in academic and industry communities. Materials from “Turbomachinary” course that has been taught by Professor A. Jaumotte at “L’Universite Libre de Bruxelles” and courses that were taught by Professor Ch. Hirsch at “Vrije Universiteit Brussel” are used continuously in the book. Also materials from courses taught and book written by Professor A. Nourbakhsh at the University of Tehran and book “Pump and Pumping” written by Professor A. Nourbakhsh and Dr. H. B. Parizi, while working at a major pump manufacturing company, were frequently used in different chapters of this book.

Contents

Part II Pumping Systems

About the Authors

Professor A. Jaumotte (Universite Libre de Bruxelles, ULB, Belgium) is a very known personality for all his efforts to develop turbomachinery science in Belgium. A. Jaumotte was president of the ULB and president of the Board of Directors of Von Karman Institute for Fluid Dynamics (Brussels) for several years. He was given the title of “Baron” by the King of Belgium for his exceptional contribution to his country. A. Jaumotte is a member of the Royal Academy of Sciences and Arts in Belgium.

Professor Ch. Hirsch (Vrije Universiteit Brussel, VUB, Belgium) is a very well known scientist in the field of computational fluid dynamics and turbomachinery around the world. The two books and more than 500 papers that he has published are valuable references in these fields. Ch. Hirsch is Founder and President of Numeca International in Belgium. Ch. Hirsch is a member of the Royal Flemish Academy of Sciences and Arts in Belgium.

Professor A. Nourbakhsh (University of Tehran, Iran) holds a Ph.D. from the Von Karman Institute and the Universite Libre de Bruxelles (ULB). He was awarded the Worthington prize (third place, golden medal) in Europe in the field of Pumps. He has published two books in the field of turbomachinery. A. Nourbakhsh is also a scientific collaborator at the Universite Libre de Bruxelles.

Dr. H. B. Parizi holds a PhD. in the field of Mechanical and Industrial Engineering form the University of Toronto (Canada). H. Parizi is presently the Vice President of Simulent Inc. where she is involved in developing various CFD software products. Dr. Parizi has many years of experiences in pump manufacturing companies and has published a book in the field of pump and pumping with A. Nourbakhsh.

Introduction

Turbopumps are members of turbomachines family, generally used for transporting liquids from a low level to a high level and/or through a system of pipelines. These machines are widely used in almost every industrial applications, from water supply, irrigation, oil and chemical processes to medical applications. Therefore, the majority of mechanical and process engineers should have at least the basic knowledge about these machines and their performances under different working conditions. Particularly, engineers who work in pump manufacturing companies must be quite familiar with different procedures involved in designing pumps and peripheral systems.

This book is divided into two major sections. In **Part I: Turbopumps**, the fundamental information about turbopumps, regardless of the piping systems that they are installed in, is presented. This part has been divided into five chapters. In Chap. 1, the basic description of pumps will be presented that includes a section about pump classification with respect to the flow rate and pressure as well as the turbopump application. The construction elements of a turbopump will also be introduced in this chapter.

In Chap. 2, the basic hydraulic parameters in pumps and pumping systems are defined and the relations between these parameters are determined. The basic laws and governing equations that are applied inside the pump and the liquid will be introduced. The one-dimensional governing equations for incompressible fluids passing through different sections of pumps and between the blades will be presented. Finally, the different sources of losses in a turbopump that reduce its efficiency will be briefly presented.

In Chap. 3, the similarity laws will be explained for turbopumps' families. These laws are very important in both pump design and pump selection procedures, and as will be seen, are widely used in obtaining the pump characteristic curves. Also, the definition of specific speed will be presented in this chapter.

In Chap. 4, cavitation will be introduced. Cavitation imposes an important limitation on pump application and has a major effect on its performance.

In Chap. 5, the axial and radial thrusts in a turbopump that are produced by flow movement inside the impeller and the methods to hydraulically balance these forces will be discussed.

The second part of the book, **Part II: Pumping Systems**, contains information about the pump performance in a system and under different conditions. This part is mainly useful for mechanical engineers who are involved in selection or design process of pump stations. In Chap. 6, the characteristic curves of the turbopumps are discussed in more detail. Topics such as the effect of rotational speed, impeller diameter, and liquid viscosity on the characteristic curves will be discussed in this chapter. Ways to modify the characteristic curves of the pump in order to use a specific pump to meet the required working conditions of a piping system will be shown.

In Chap. 7, those topics of fluid mechanics that are widely used in designing the piping systems will be reviewed. Of course for more detailed and in depth information, readers can refer to any fluid mechanic textbook. However, in this chapter the basic information, relations, diagrams, and charts that can be used to determine the friction losses in the pipes and fittings and to obtain the characteristic curve of a system will be presented.

Pumps' performances in different piping systems will be discussed in Chap. 8. Pumps' performances when operating in parallel and series, pumps working on different pipeline systems, the parameters that an engineer should consider to select a proper pump, and pipeline elements are then extensively discussed in this chapter. This chapter is one the most useful sections of this book for those engineers who are operating a pumping system or a pumping station.

In Chap. 9, the essential information about water hammer will be discussed. Water hammer is a common problem in longer pipeline systems and in this chapter the conditions under which it occurs in a piping system and the methods for preventing it and protecting the piping system, pump, and other elements in the system will be discussed.

In Chap. 10, the important steps in hydraulic design of a pump station will be presented. There are many parameters that have significant effects on the performance of a pump in a pump station. Among them are water hammer, formation of vortex in the pump inlet, and priming. Furthermore, intake and suction pipes for a pump must be designed in such a way that they can protect the pump from adverse conditions like solid objects inflow and cavitation. All these parameters and design conditions are addressed in this chapter.

Terms, Notations, and Units

Following is a table of the terms, abbreviations, and units mostly used throughout this book

Term	Abbreviation	SI Units	Unit used in practice
Flowrate	Q	m^3/s	m^3/h
Total head	H	m	m
Net positive suction head	NPSH	m	m
Input power	P	Watt	kW
Pump overall efficiency	η		
Rotational speed	n	rpm	rpm
Angular velocity	$\omega = 2\pi n$	1/s	1/min
Pressure*	p	N/m^2	bar
Density or volumetric mass	ρ	Kg/m^3	Kg/m^3
Specific weight	γ	kN/m^3	
Peripheral velocity	U	m/s	m/s
Absolute velocity	V	m/s	m/s
Relative velocity	W	m/s	m/s
Gravitational acceleration	g	m/s^2	m/s^2

* The atmospheric pressure p_a and the vapor pressure of the liquid, p_v are quoted as the absolute pressures. All other pressures are presented as gauge pressures

List of Figures

List of Tables

Part I
Turbopumps

Chapter 1
Introduction to Turbopumps

Pumps are devices that transfer energy from an external source to a liquid in order to move the liquid from one location to another. This process will increase the energy of the liquid after it leaves the pump.

The hydraulic characteristics of a specific piping system (simply referred to the *system*) as well as the physical and chemical characteristics of the liquid itself would determine what type of pump is most suitable for one specific application. These characteristics include liquid viscosity, specific weight, working temperature, corrosion property, possibly dissolved gases and suspended particles in the liquid. All of these elements along with the volume of the liquid that has to be pumped per unit of time and the required pressure have led to development of pumps with different structures, designs, and applications.

The mechanisms of energy transfer between pumps and the liquids are so different that it is impossible to utilize a unique theory to describe the process. For this reason, pumps are divided into different categories and each category is then defined and analyzed. In the following sections the classification of pumps in general and the description of turbopumps in particular are presented. To show the reader the difference between each type, a picture of each pump has been shown to illustrate the major differences.

1.1 Pump Classification

The classification of pumps is done based on different criteria. This classification could be based on the application, internal structure, the mechanism of energy transfer between pump and liquid, or based on the type of pumped liquid. The most common way of classification is, however, based on the mechanism of energy transfer. In this respect, pumps are divided into two major categories:

1. The "dynamic pumps" in which the energy transfer from the pump is continuous.
2. The "displacement pumps" in which the energy transfer between the pumps and the liquid is periodic or non-continuous.

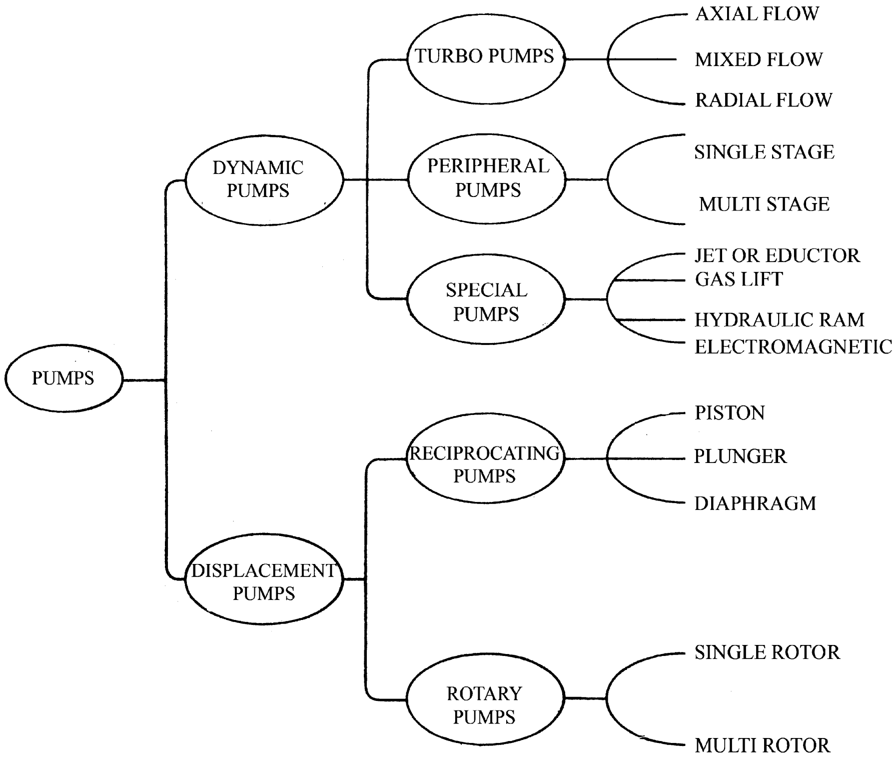


Fig. 1.1 Pump classification chart [1]

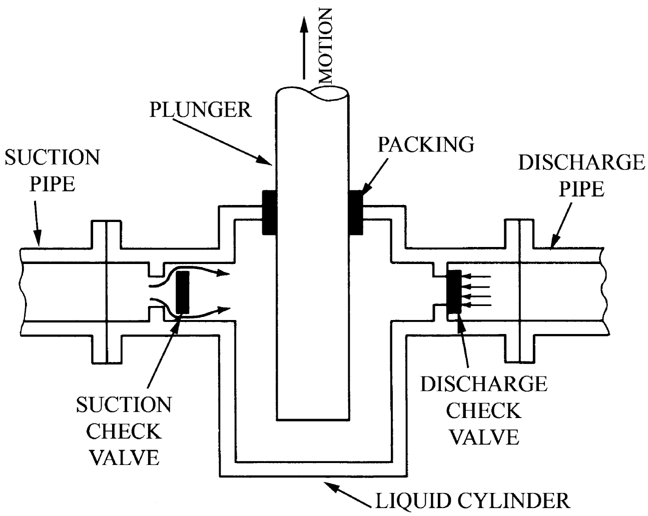


Fig. 1.2 Plunger reciprocating displacement pump [1]

In Fig. 1.1 the classification chart of pumps based on this definition is shown. As one can see, turbopumps belong to the first group, i.e. dynamic pumps. Since turbopumps are the most common pumps that are used in industry, this book is dedicated to this type of pumps. Some examples of the most common pumps are shown in Figs. 1.2–1.9.

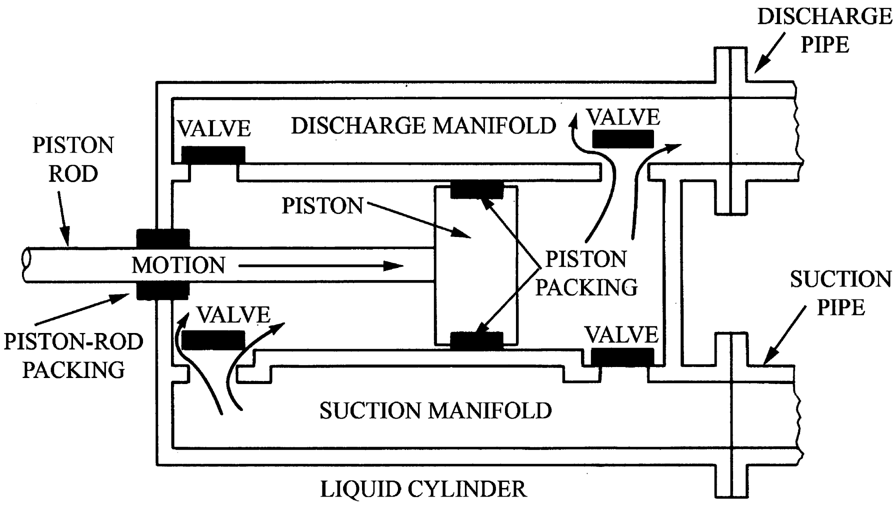


Fig. 1.3 Piston reciprocating displacement pump [1]

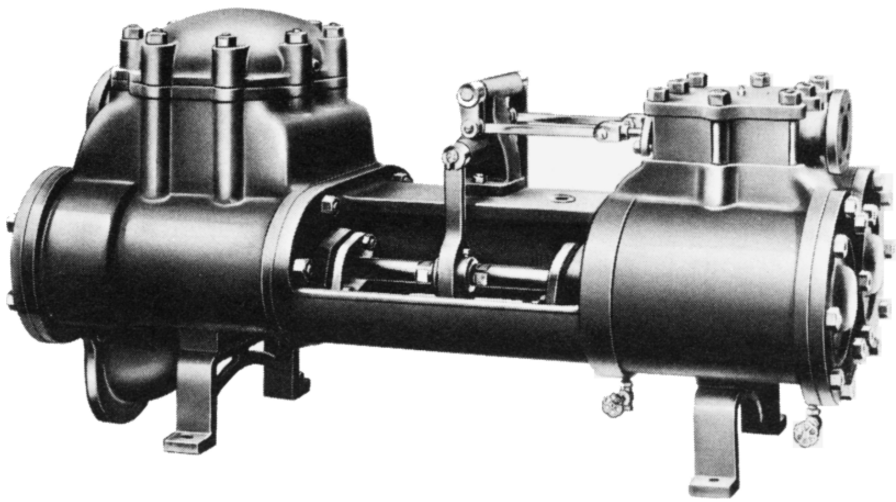


Fig. 1.4 Reciprocating steam pump (Worthington Pump Corporation) [1]

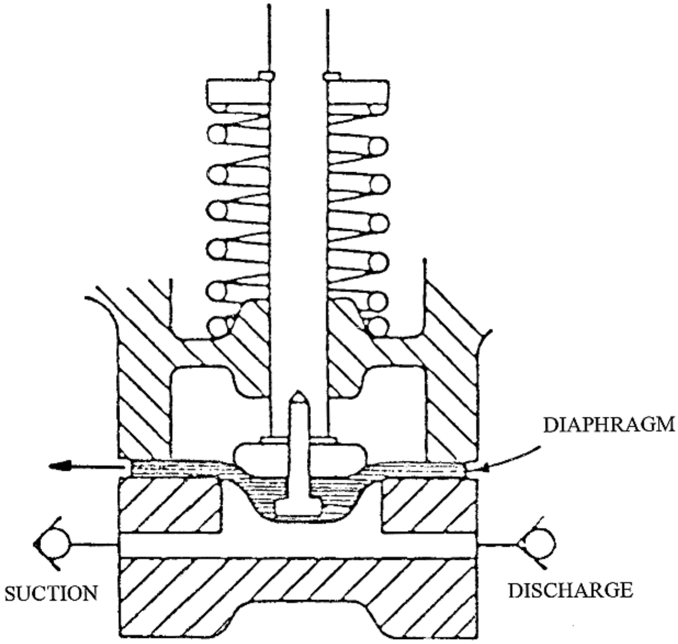


Fig. 1.5 Diaphragm pump [2]

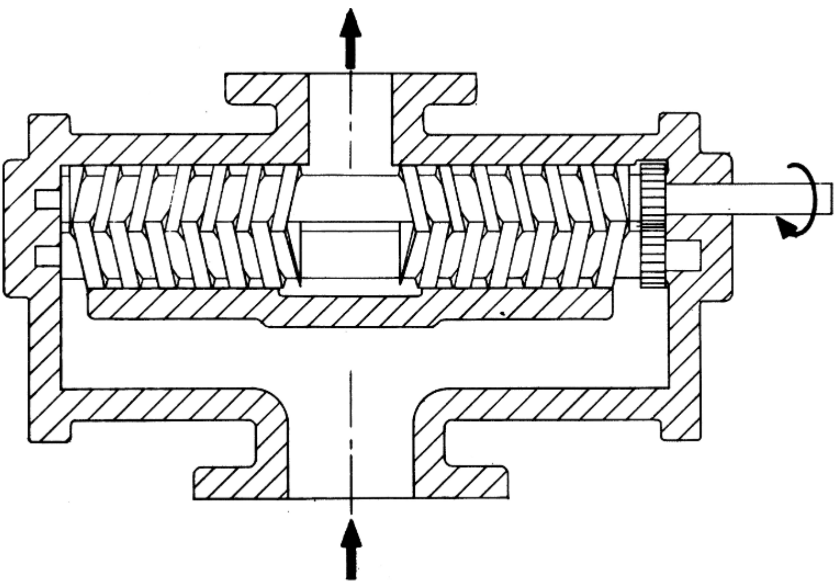
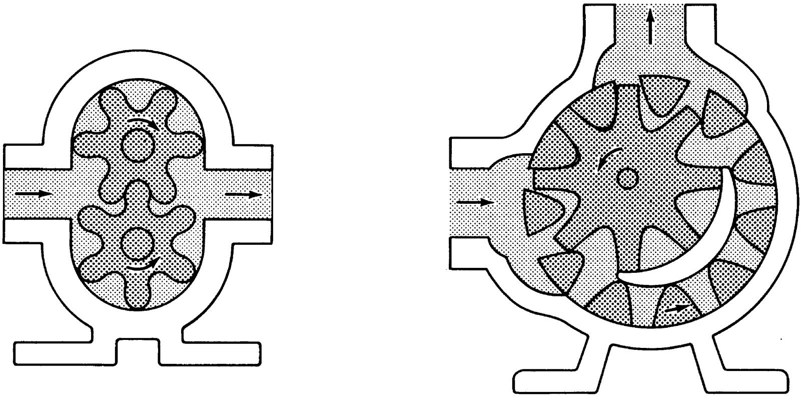


Fig. 1.6 Multiple-screw double-end arrangement (rotary pump) [1]



(a) (b)

Fig. 1.7 Rotary gear pumps. (**a**) External; (**b**) Internal [1]

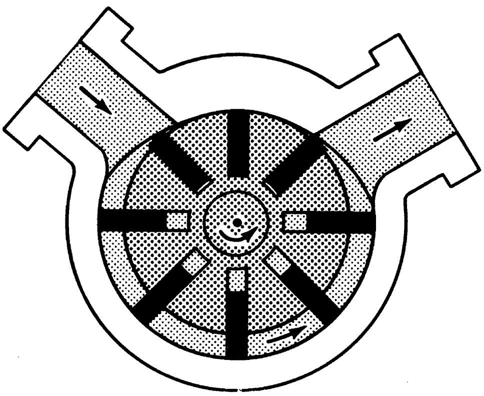


Fig. 1.8 Internal (vane-in-body) pump [2]

1.2 Turbopumps

Turbopumps or impeller pumps are machines in which the main moving part is a rotor with a number of blades mounted on a rotating shaft. The moment of momentum of the liquid passing continuously through the impeller is increased by action of blades. All impeller pumps are members of turbomachines family.

Due to the simple constructional elements, low ratio of volume to the power input, and numerous applications in the industry, these pumps are the most used among other types of pumps.

1.2.1 Turbopump Types

There are basically many types of pumps in the market. Most important types of pumps are single stage, axial flow, multi-stage, horizontal, vertical, submersible, and

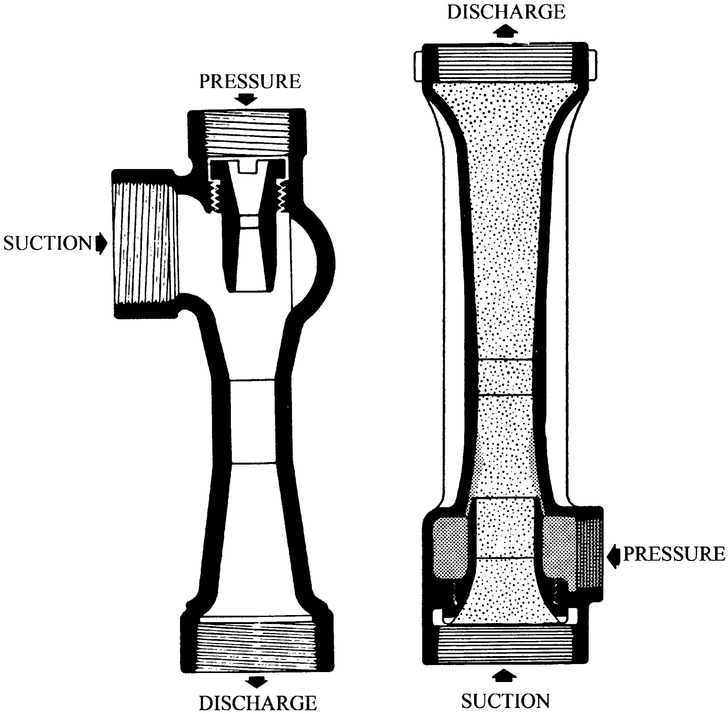


Fig. 1.9 Special pumps (jet pump) (Schutte and Koerting Co.) [1]

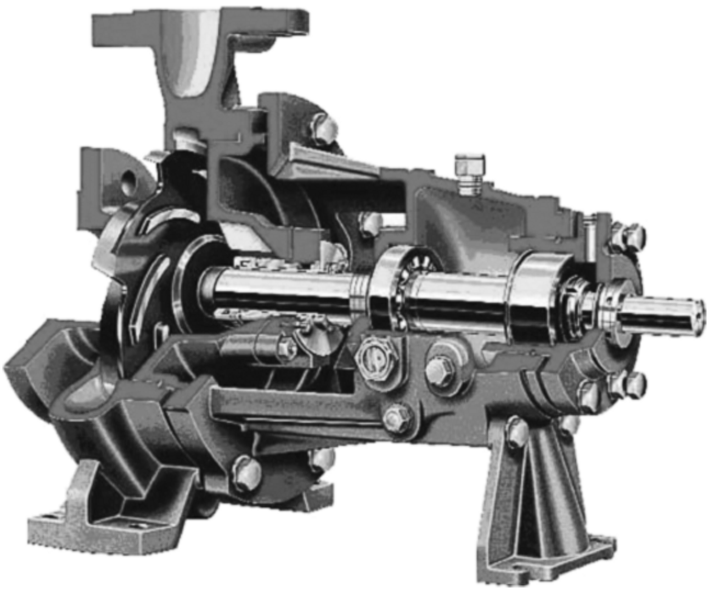


Fig. 1.10 Single stage centrifugal pumps (www.flowserve.com)

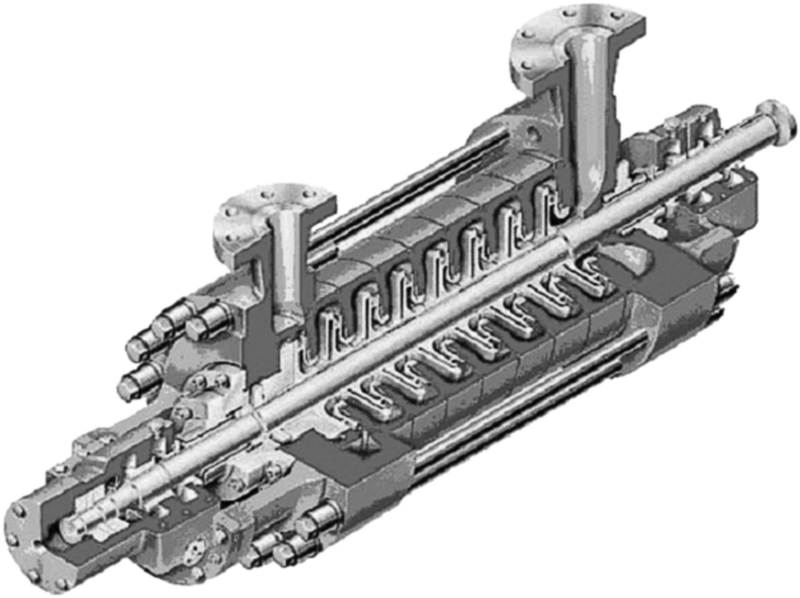


Fig. 1.11 Multistage centrifugal pumps (www.flowserve.com)

double suction. In multi-stage pumps, several rotors are mounted on a rotating shaft to increase the pressure. The number of blades on a rotor can be two or even one for handling the liquid with solids. In Figs. 1.10–1.17 different types of turbopumps are shown.

1.3 Turbopump Classification

The most common way for classification of impeller pumps is based on the path of liquid movement inside the rotor. If the liquid moves through the rotor on streamlines or stream surfaces approximately parallel to the axis of rotation, the machine is classified as an axial (propeller) flow pump (Fig. 1.18a). In radial centrifugal flow pumps, on the other hand, the directions of the fluid entering and leaving the impeller are perpendicular (Fig. 1.18b).

In the mixed flow impellers, flow direction changes from axial at the entrance to an angle greater than 90° at the exit, Fig. 1.18c.

Centrifugal, mixed, and axial flow pumps are used for producing high, medium, and low manometric heads, respectively. For very high heads and handling low flow rates, the positive displacement pumps are more suitable than the centrifugal pumps. In Fig. 1.19 a pump selection chart is shown. This chart can be used to find the suitable pump type for different working conditions.

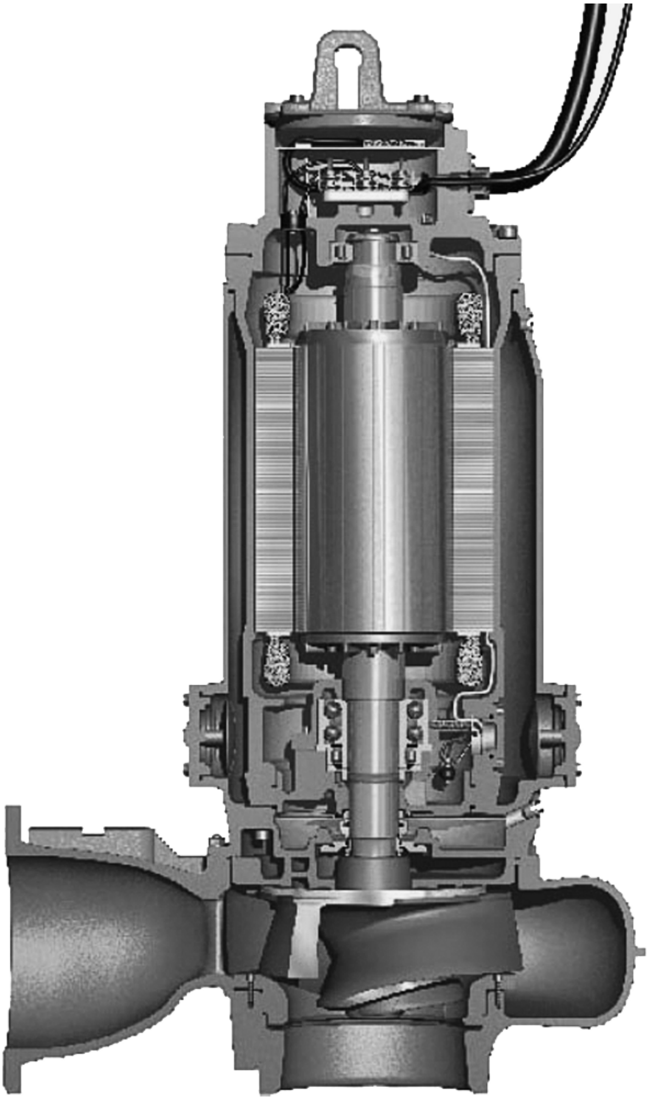
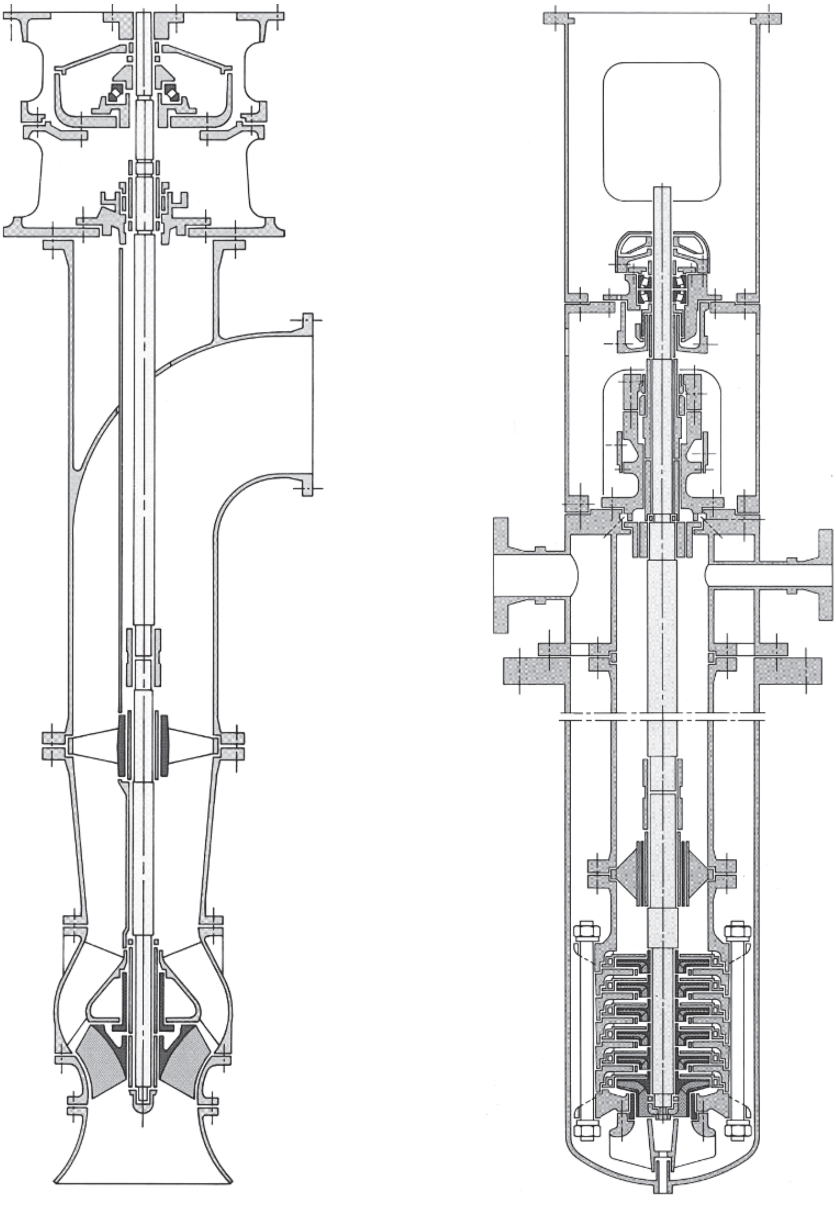


Fig. 1.12 Vertical submersible sludge pump (www.flygt.com)



(a) (b)

Fig. 1.13 (**a**) Vertical mixed flow single-stage pump; (**b**) Vertical multi-stage centrifugal pump [3]

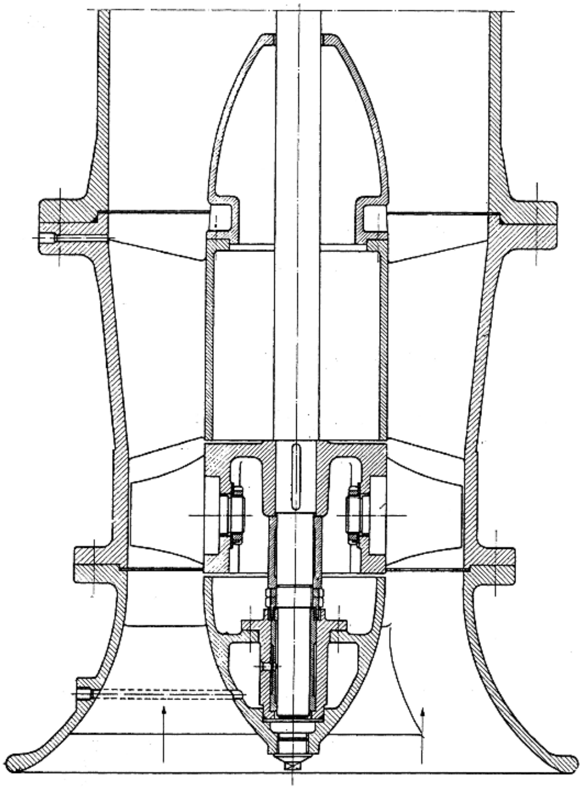


Fig. 1.14 Axial flow pump [4]

1.4 Construction Elements

The important parts of an impeller pump are

- suction element
- rotor or impeller
- diffuser
- volute or casing

In Fig. 1.20, different construction elements of a centrifugal pump are presented. The suction element has a great influence on the velocity distribution at the impeller eye and on the performance of the impeller. Different shapes of suction elements are used depending on the pump type and operational conditions.

The rotor increases the pressure and velocity of the liquid passing through the pump. A diffuser with or without blades can be used for transforming the kinetic energy of fluid to the potential energy (pressure) at the outlet of the impeller. Diffuser is an optional part of a pump and depending on the size and the degree of

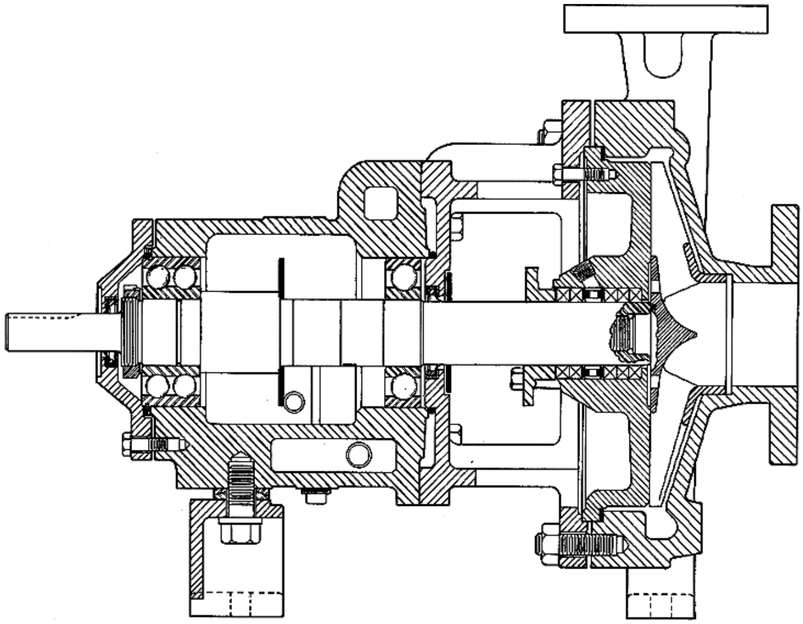


Fig. 1.15 Horizontal chemical pump (www.peerlesspump.com)

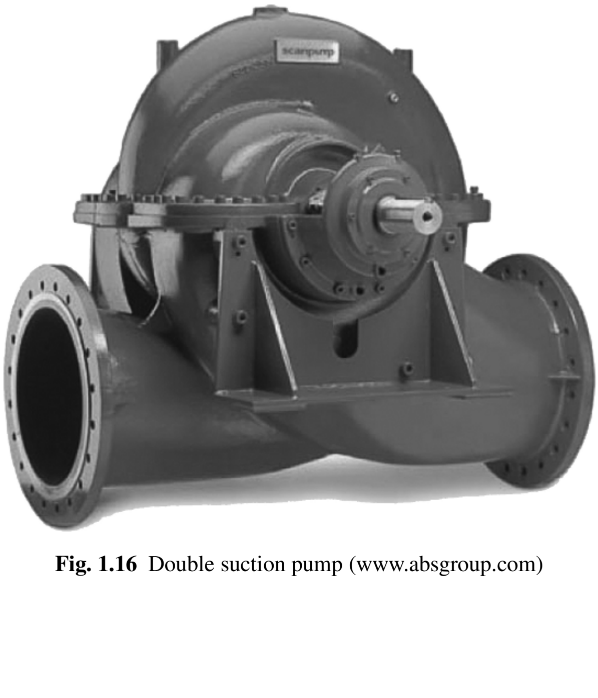


Fig. 1.16 Double suction pump (www.absgroup.com)

Fig. 1.17 Submersible vertical pump (www.gouldspumps.com)

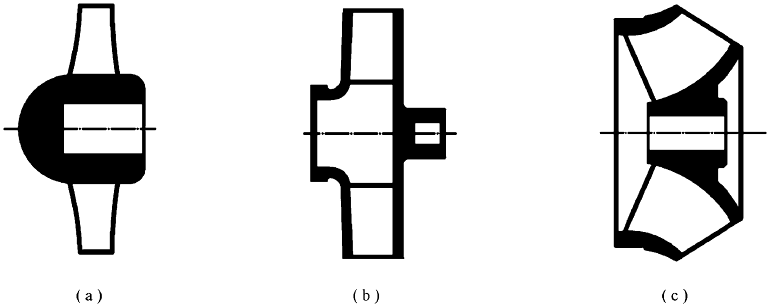


Fig. 1.18 Axial (**a**); centrifugal (**b**); and mixed flow impellers (**c**) [5]

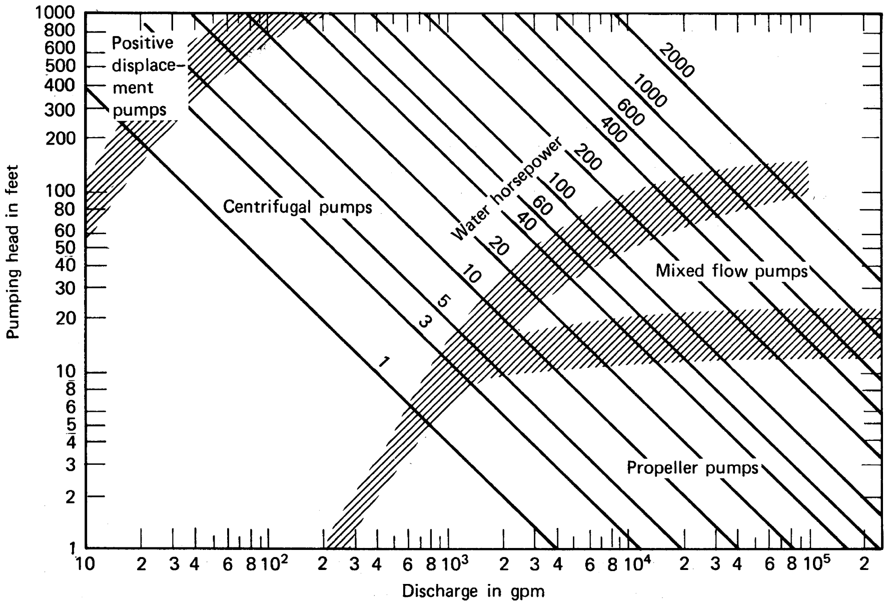
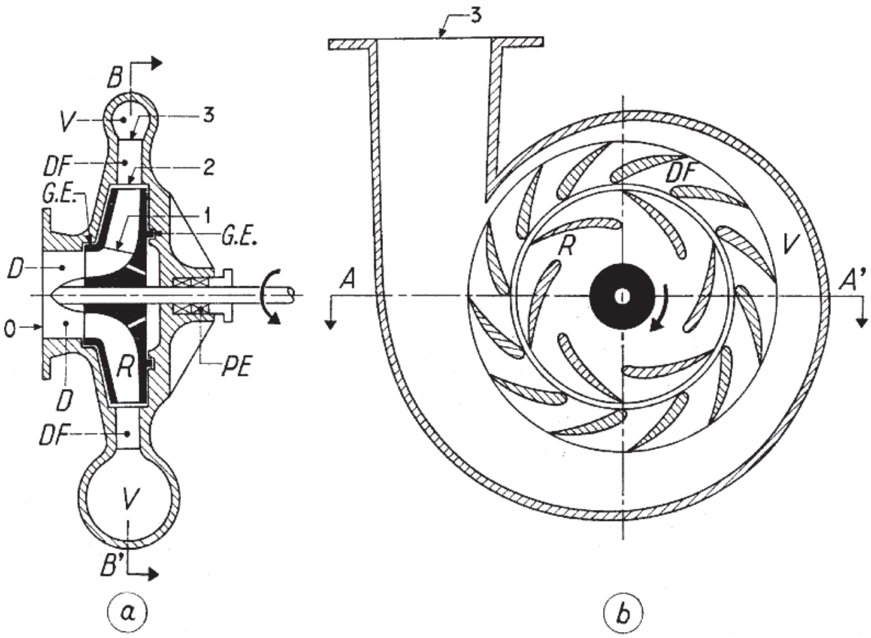


Fig. 1.19 Pump classification chart (English Units) [6]



(**a**) Axial cross section
D Suction element (inlet), (0-1)
R Rotor (impeller), (1-2)
DF Diffuser with blades (2-3)

(**b**) Radial cross section
V Volute (casing) (3)
GE Leakage ring
PE Staffing box

Fig. 1.20 Construction elements of a centrifugal pump [7]

the reaction required this part may or may not be present. Finally, after leaving the impeller or diffuser, fluid moves into the volute in centrifugal pumps. More details about the constructional elements of turbopumps are presented in Appendix.

References

1. Karassik Igor J. et al., "Pump Handbook", McGraw-Hill, 2001.
2. Karassik Igor J. et al., "Pump Handbook", McGraw-Hill, 1976.
3. "Planning of Centrifugal Pumping Plants", Pump Division, Sulzer Brothers Limited, Winterthur-Switzerland, 1987.
4. Lazarkiewicz Stephen , "Impeller Pumps", Pergamon Press, 1965.
5. "Centrifugal Pump", Lexicon, KSB, Frankenthal; Germany, 1980.
6. Simon Andrew L., "Practical Hydraulics", John Wiley & Sons, 1981.
7. Jaumotte A. L., Decock P., "Turbomachines a fluid incompressible. Generalities", Techniques de l'ingenieur, Paris, 3- 1974.

Chapter 2
Fundamental Concepts

The main purpose of this chapter is to introduce the terminologies, the fundamental concepts, and basic definitions that are usually used and applied in turbopump design and application. The chapter is divided into six major sections. In the first two sections, basic hydraulic parameters in a pump are introduced and their relations which are demonstrated in terms of the "characteristic curves" of a pump will be presented. The main purpose of Sect. 2.3, pump testing, is to show the relevance between the measured parameters in a testing system and the characteristic curves of a pump. After demonstrating the velocity triangle in a pump and introducing parameters such as blade angles and the absolute and relative velocities of liquid in an impeller, the one-dimensional governing hydraulic equations in the impeller are presented. In the last section, it will be shown that the hydraulic losses in a turbopump will change the shape of the actual characteristic curves of the pump.

2.1 Basic Hydraulic Parameters

Volumetric flow rate (Q)

The flow rate or capacity is the volume of liquid delivered by the pump per unit of time.

Mass flow rate ($\dot{m}$)

The mass flow rate is the mass of liquid delivered by a pump per unit of time. The following relation exists between the mass flow rate and the volumetric flow rate:

$$\dot{m} = \rho Q$$

Total head (H)

The monometric head or pump head represents the mechanical energy transferred by the pump to the liquid which is pumped. This work is expressed per unit of weight of the pumped liquid and is expressed by the height of liquid columns.

Power (P)

Different powers can be defined in a pump (see Sect. 2.2). The two most used definitions are

1. P_Q, useful power ($P_Q = \rho.g.Q.H$): the useful power transferred by the pump to the liquid.
2. P, input power or shaft horsepower.

Overall efficiency (η)

The overall efficiency of a pump is defined by

$$\eta = \frac{\rho \cdot g \cdot Q \cdot H}{P} \tag{2.1}$$

*Net positive suction head (*NPSH*)*

There is another important parameter called net positive suction head that affects the pump performance. NPSH defines the cavitation characteristic of a pump and will be discussed in more detail in Chap. 4.

2.2 Characteristic Curves

For pump users, there are several important variables which define the operating conditions of a pump. These variables can be divided into two categories:

1. Hydraulic variables, including flow rate, Q, and total head, H.
2. Mechanical variables, including rotational speed, n, and shaft power, P.

These variables are related to each other by the overall efficiency of the pump, (2.1). In order to find the characteristic curves of a pump, a hydraulic and a mechanical variable must be chosen as independent variables. Then other variables can be defined as functions of these two independent variables. Usually, flow rate Q and rotational speed n or angular velocity ω are selected as independent variables.

The variation of other variables versus Q and ω are called "characteristic surfaces" of a pump and are defined as follows:

$$H = f_1(Q, \omega) \tag{2.2}$$

$$P = f_2(Q, \omega) \tag{2.3}$$

$$\eta = f_3(Q, \omega) \tag{2.4}$$

Normally, characteristic surfaces are not easy to use and therefore they are not presented in the pump catalogs. In practice, the "characteristic curves" are presented by manufacturers for different constant speeds ($\omega = cte$):

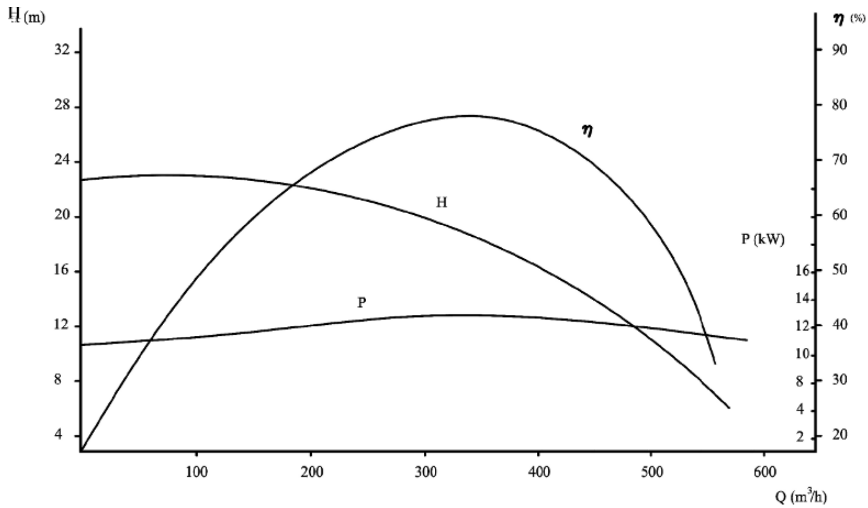


Fig. 2.1 Typical pump characteristic curves [1]

$$H = f_1(Q) \tag{2.5}$$

$$P = f_2(Q) \tag{2.6}$$

$$\eta = f_3(Q) \tag{2.7}$$

Figure 2.1 shows a sample of characteristic curves for a constant rotational speed.

The $H = f_1(Q)$, $P = f_2(Q)$, and $\eta = f_3(Q)$ are called manometric head, power, and efficiency characteristic curves of a pump, respectively. For centrifugal pumps, $H = f_1(Q)$ normally has a parabolic form with its maximum on or close to the vertical axe. $\eta = f_3(Q)$ also has a parabolic form and has a maximum, which denotes the "design point" of the pump. $P = f_2(Q)$ is almost a straight line with a very flat maximum. The power of driving motor should be more than the maximum power used by the pump.

2.2.1 Iso-efficiency Curves

The complete characteristic curves of a pump also include the curves that are called the *iso-efficiency* curves. In Fig. 2.2 the two-dimensional complete characteristic curves, with iso-efficiency curves for a turbopump, are shown. These curves are plotted as follows. First the experimentally head-discharge characteristic curve at different speeds of 0.5–1.3 of nominal speed are determined and plotted (solid parabolic lines in the upper plot). This will cover the whole operation range of a pump. The curves are plotted in a non-dimensional system of coordinates, where the non-dimensional characteristic curves of $H/H_n = f(Q/Q_n)$ is obtained as $H\% = f(Q\%)$. Subscript n notifies the characteristic curve at normal speed. Now

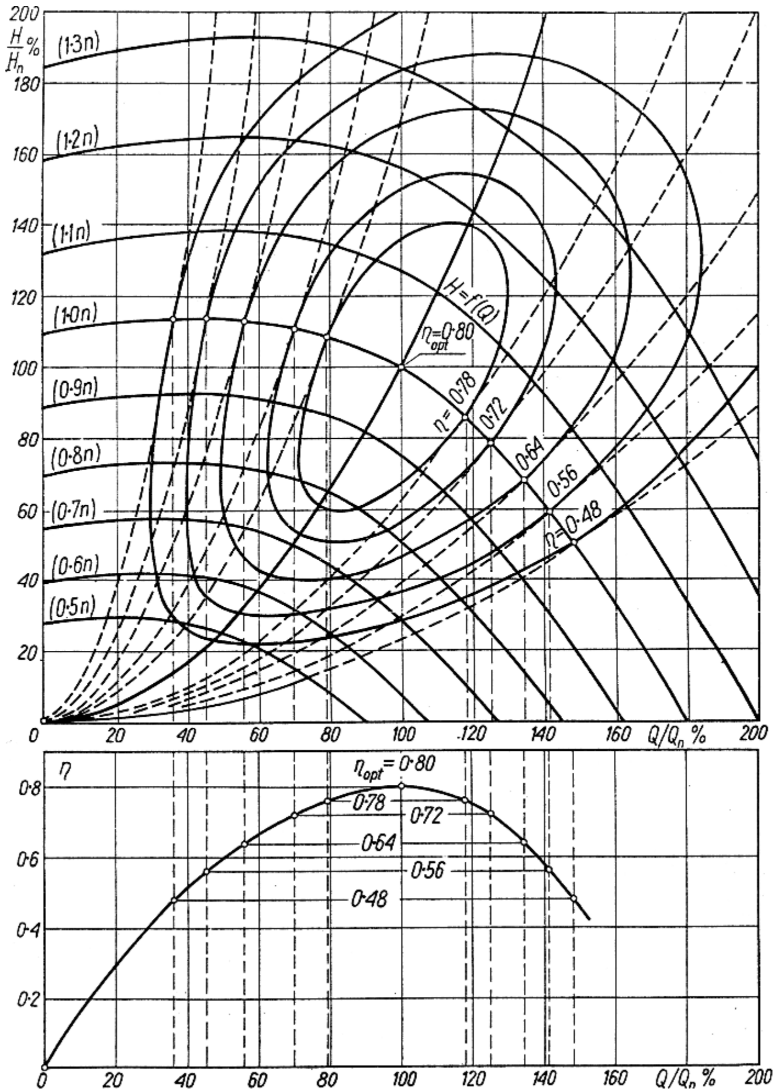


Fig. 2.2 Characteristic curves for different rotational speeds [2]

the efficiency curves of $\eta = f_3(Q)$ which are obtained from the experiment at the same rotational speeds are plotted, using the same percentage scale for the discharge axis (lower plot). For clarity, only the efficiency curve at nominal speed is shown in this figure. The intersection of this curve with the straight lines of the same efficiency, e.g. $\eta = 0.48, 0.56, 0.64, 0.72, 0.78,$ and 0.80 (maximum efficiency), are then obtained (see the lower plot).

Now, from each efficiency curve, these points are projected to the non-dimensional head-discharge curve on the top plot. Once this procedure is completed for all efficiency and head-discharge curves at different speeds, the point with equal efficiency values are connected to each other, forming a series of closed oval iso-efficiency curves. These curves are very important during the pump selection, since they would show the efficiency of the pump at different conditions and rotational speeds.

If the points of the $H = f(Q)$ curves corresponding to the best operating conditions (maximum efficiency) at the respective speeds are joined together, a parabolic curve is obtained whose apex is at the origin and its axis coincides with the H-axis (solid line in the upper plot). This curve represents the optimum operating conditions for the impeller. The other similar curves, dashed lines, show the similar flow conditions, but at different efficiencies.

More information about the characteristic curves at different rotational speeds can be found in Sect. 3.5.1.

In some cases, a set of the characteristic curves for a pump is presented at a constant speed but for different impeller diameters, as shown in Fig. 2.3. The top part shows the head and efficiency curves at different impeller diameters from 140 to 175 mm. The middle part shows the required NPSH for the pump which is independent of the impeller diameter. (More details about this parameter are explained in Chap. 4.) The bottom part shows the absorbed power by pump at the same diameters. Both SI and English units are shown in the figures. The power characteristic curve for this pump is shown in the lower part of this figure.

The characteristic curves of a pump can be obtained by experiments or through theoretical methods.

2.2.2 Important Points on the Characteristic Curves

There are some points on the characteristic curves of the pumps that are important:

- Q_n, nominal flow rate: flow rate for which the pump has been ordered.
- $Q_{\min}$, minimum flow rate: minimum permissible flow rate for a pump.
- $Q_{\max}$, maximum flow rate: maximum permissible flow rate for a pump.
- $Q_{D,P}$, optimum flow rate: flow rate at the best efficiency point.
- Q_{Opt}, operation flow rate: flow rate at which the pump is normally operating.
- H_n, nominal head: total head for which the pump has been ordered.
- H_0, shut-off head: total head at zero flow rate.
- $H_{D,P}$, optimum head: total head at the point of maximum efficiency.
- P_Q, useful power ($P_Q = \rho \cdot g \cdot Q \cdot H$): the useful power transferred by the pump to the liquid.
- P, input power or shaft power: power absorbed by the pump.
- P_D, shut off power: power required at zero flow rate.
- P_n, nominal input power: power required at Q_n, H_n, and N_n.
- P_N, optimum input power: power required at best efficiency point.
- P_M, driver output power: power for all possible operating conditions of the pump driver.

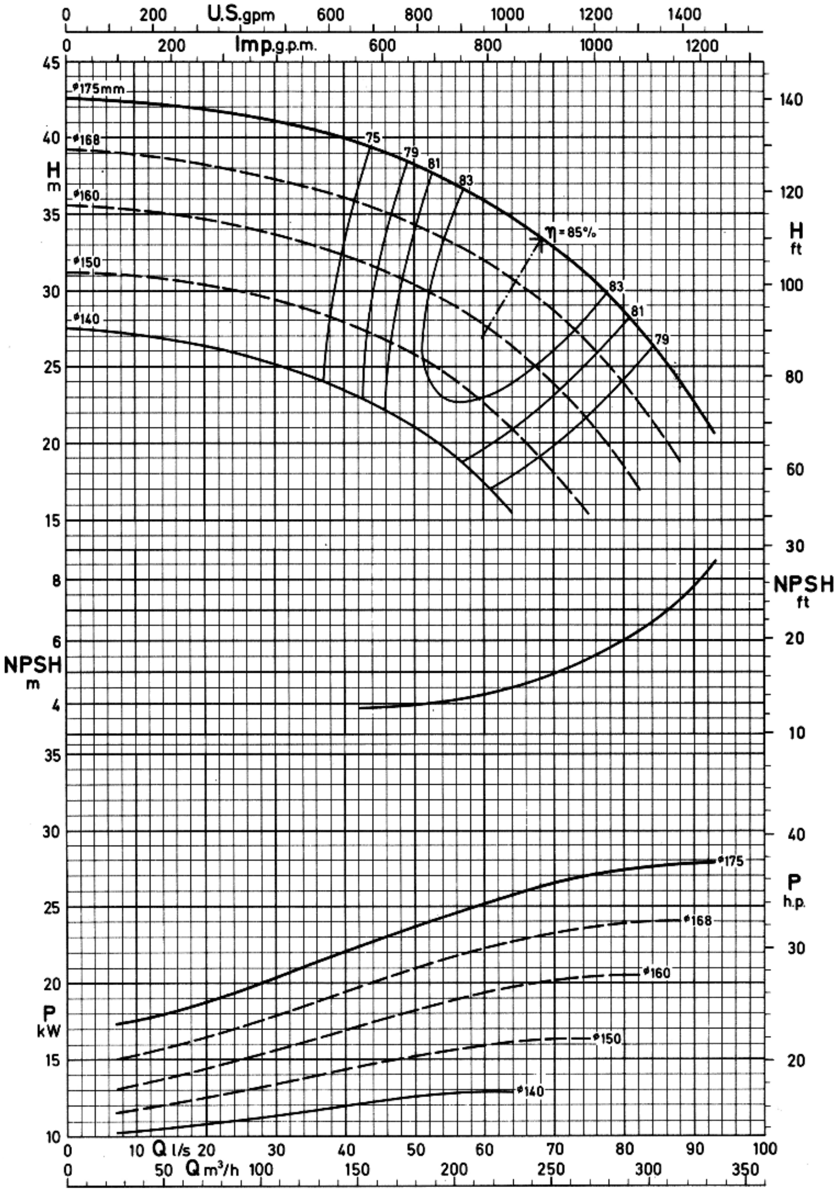


Fig. 2.3 Characteristic curves for different impeller diameters (KSB product catalog (1312.4050/3-90 G3)

Table 2.1 Power margin for drivers [3]

Power consumption	Driver power
$P < 1.5\,\mathrm{kW}$	$P_\mathrm{M} = 1.5\,\mathrm{P}$
$1.5 < P < 4\,\mathrm{kW}$	$P_\mathrm{M} = 1.25\,\mathrm{P}$
$4 < P < 7.5\,\mathrm{kW}$	$P_\mathrm{M} = 1.2\,\mathrm{P}$
$7.5 < P < 40\,\mathrm{kW}$	$P_\mathrm{M} = 1.15\,\mathrm{P}$
$P > 40\,\mathrm{kW}$	$P_\mathrm{M} = 1.1\,\mathrm{P}$

The power of the pump driver, like electrical motor, P_M should be always more than the power absorbed by the pump P. It is recommended that P_M is selected higher than the maximum value of P on the characteristic curve (not based on $Q_{D,P}$). In some special cases where the flow rate remains constant or the power of the motor is very high, the power P_M can be selected based on (higher than) the power at the design point. However, a special control system should be installed on the motor to prevent overload.

Table (2.1) shows some suggestions for selecting the driver power for radial flow pumps based on the pump power consumption [3].

Depending on the shape of the power characteristic curve and the position of the operating point, deviation from the above-suggested power margins are possible in special cases. Power margins for mixed flow and axial flow pumps are a function of the shape of the pump input power (for more details see Chap. 6).

2.3 Pump Testing

The characteristic curves of a pump can be determined from experiments by measuring the pressure, flow rate, rotational speed, and absorbed power in a test rig, (Fig. 2.4).

The test rig and measuring equipments should be made and selected according to an acceptable standard for pump users (such as ISO, DIN, and BS Standards). The class of standard should be agreed upon before starting the test (class A, B, or C). The test starts normally by calibrating the measuring equipment. Later, at each constant speed, the flow rate is changed by means of a valve. At each flow rate, the liquid pressure at outlet and inlet of the pump, P_3 and P_0, are measured by a pressure gauge or a pressure transducer. The head of a pump can be obtained by the following relation (see (2.20)):

$$H = \frac{P_3 - P_0}{\rho g} + \frac{V_3^2 - V_0^2}{2g} + Z_3 - Z_0 \tag{2.8}$$

V_3 and V_0 are velocities and Z_3 and Z_0 are geometrical elevations at outlet (subscript 3) and inlet (subscript 0) of the pump. Normally, the inlet and outlet diameters of the pump are equal and one can assume $V_3 = V_0$. Difference of $(Z_3 - Z_0)$ can be measured easily.

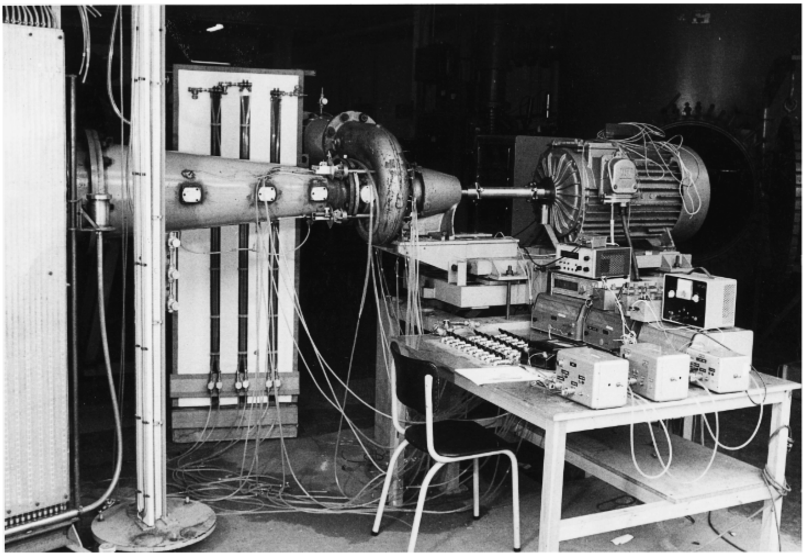


Fig. 2.4 Pump test rig [4]

The flow rate is usually measured by a Venturi, an orifice, a nozzle, or a magnetic flow meter that is located on the delivery pipe of the pump. There are different ways for measuring the power: two watt meter method, dynamometer, or torque meter.

Once these variables are measured, by using the relation $\eta = \frac{\rho \cdot g \cdot Q \cdot H}{P}$, the efficiency can be calculated at each flow rate. Therefore, the characteristic curves of the pump $H = f_1(Q)$, $P = f_2(Q)$, and $\eta = f_3(Q)$ can be obtained for a whole range of flow rates. The same procedure can be repeated for different rotational speeds.

2.4 Velocity Triangles

Figure (2.5) shows a schematic illustration of a pump impeller with a radius R and an angular velocity ω. In any point of the flow path between two blades of an impeller, three velocities can be defined:

1. Local peripheral velocity of the impeller, $\vec{U}$, which is equal to $\omega\ R$. $\vec{U}$ is perpendicular to the radius of the impeller.
2. Relative velocity of flow, $\vec{W}$, which is tangent to the streamlines in the relative system of coordinates (impeller).
3. Absolute velocity of flow, $\vec{V}$, which is equal to the vector sum of the relative velocity, $\vec{W}$, and the peripheral impeller velocity, $\vec{U}$.

The three velocities form a triangle which is called the "velocity triangle," as shown in Fig. 2.5. The velocity triangle can be defined in any point between two adjacent blades of an impeller. However, the most important velocity triangles are those that are defined at the entrance and exit of an impeller and are used for flow calculations. Subscripts "1" and "2" refer to the components at the entrance and the exit (discharge) of the impeller, respectively.

Tangential and radial components of the absolute velocity are designated as V_u and V_m. The subscript "m" stands for the meridional velocity and its direction depends on the type of impeller; in a radial impeller it is radial and in an axial impeller it is axial.

As shown in Fig. 2.5, α and β are the absolute and relative angles of flow with respect to the peripheral velocity. The "blade angle" β' is the angle between the tangents to the blade and the peripheral velocity in the opposite direction. If all streamlines remain parallel to the blades (which is not the case in practice) then, in any point, one can write $\beta = \beta'$. Usually, for theoretical studies and practical designs, the flow velocity is considered to be the average velocity of the flow at each plane normal to the general direction of the flow. This is not true in practice, but can help to simplify the problems when studying the flow pattern between two blades.

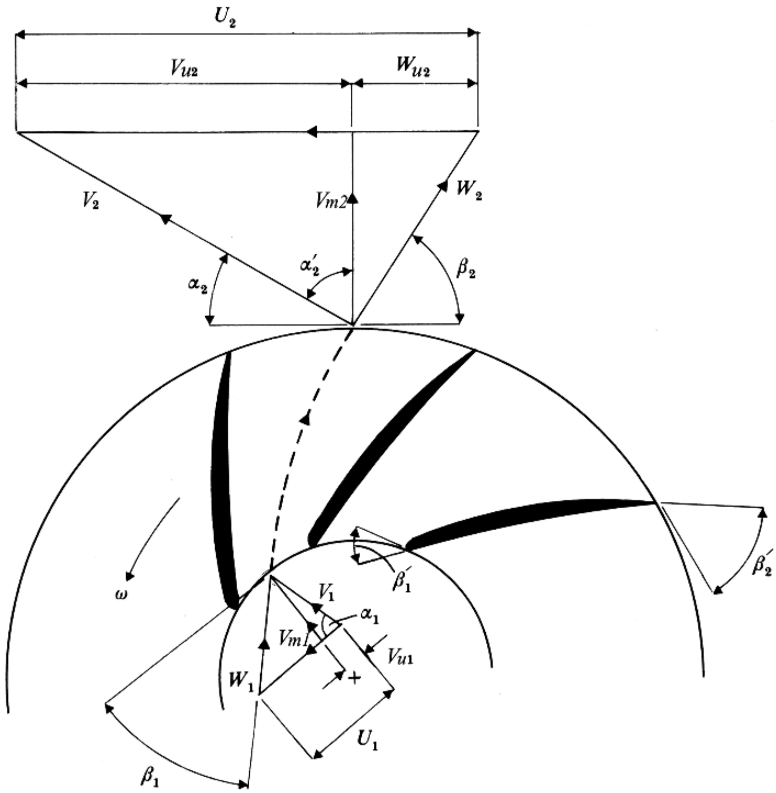


Fig. 2.5 Velocity triangles at the inlet and outlet of the impeller [5]

2.5 Basic Laws and Governing Equations

2.5.1 Continuity Equation

The continuity equation for a steady flow inside an impeller in a one-dimensional domain can be written as

$$\dot{m}'' = \rho_1 V_{m1} A_1 = \rho_2 V_{m2} A_2 \tag{2.9}$$

For incompressible flows

$$Q'' = V_{m1} A_1 = V_{m2} A_2 \tag{2.10}$$

where $\dot{m}''$ and Q'' are the mass and volumetric flow rates of the impeller, respectively. Note that due to the internal leakage losses, these two parameters are different with the mass and volumetric flow rates of the pump itself. A_1 and A_2 are effective areas of the impeller at inlet and outlet, perpendicular to the absolute velocities V_{m1} and V_{m2}.

2.5.2 Equation of Moment of Momentum

For any control volume of the liquid, the net angular momentum of the content in the control volume at any time plus the net rate of flow of the mass flow through the control surface is equal to the external torques acting on the control volume. For a one-dimensional assumption, the torque, power, and head exchanged in the impeller can be simplified to [6, 7, 8]:

$$T'' = \dot{m}''(R_2 V_2 \cos\alpha_2 - R_1 V_1 \cos\alpha_1) \tag{2.11}$$

$$P'' = \dot{m}''(U_2 V_2 \cos\alpha_2 - U_1 V_1 \cos\alpha_1) \tag{2.12}$$

$$H'' = \frac{1}{g}(U_2 V_2 \cos\alpha_2 - U_1 V_1 \cos\alpha_1) \tag{2.13}$$

From velocity triangle, Fig. 2.5,

$$W^2 = U^2 + V^2 - 2U\,V \cos\alpha \tag{2.14}$$

Therefore, the total head (power per unit weight) in the impeller can be written as

$$H'' = \frac{U_2^2 - U_1^2}{2g} + \frac{V_2^2 - V_1^2}{2g} + \frac{W_1^2 - W_2^2}{2g} \tag{2.15}$$

For the axial impellers, $U_2 = U_1$, therefore

$$H'' = \frac{U}{g}(V_2 \cos\alpha_2 - V_1 \cos\alpha_1) \tag{2.16}$$

or

$$H'' = \frac{V_2^2 - V_1^2}{2g} + \frac{W_1^2 - W_2^2}{2g} \tag{2.17}$$

2.5.3 Energy Equations

The energy equations with a one-dimensional assumption for an impeller (2.18) and a pump (2.19) can be written as follows:

$$H'' = \frac{P_2 - P_1}{\rho g} + \frac{V_2^2 - V_1^2}{2g} + (Z_2 - Z_1) + H_{f_1^2} \tag{2.18}$$

$$H'' = \frac{P_3 - P_0}{\rho g} + \frac{V_3^2 - V_0^2}{2g} + (Z_3 - Z_0) + H_{f_0^3} \tag{2.19}$$

where subscripts 0 and 3 denote the inlet and outlet of the pump, $H_{f_1^2}$ and $H_{f_0^3}$ are the hydraulic losses in the impeller and the pump, respectively, and Z is the geometric height and is defined from an arbitrary reference level to the axis of the pump. The hydraulic efficiency can be defined as $\eta_\eta = \frac{H}{H''}$, where the actual manometric head of the pump is

$$H = \frac{P_3 - P_0}{\rho g} + \frac{V_3^2 - V_0^2}{2g} + (Z_3 - Z_0) \tag{2.20}$$

2.6 Losses

In the following sections, the origins of main losses which occur in different sections of a turbopump will be discussed in more details.

2.6.1 Pressure or Head Losses ($H_{f_0^3}$)

Pressure losses in a pump are basically caused by different sources:

1. Friction between the fluid passing through different elements of a pump and the walls (friction losses).

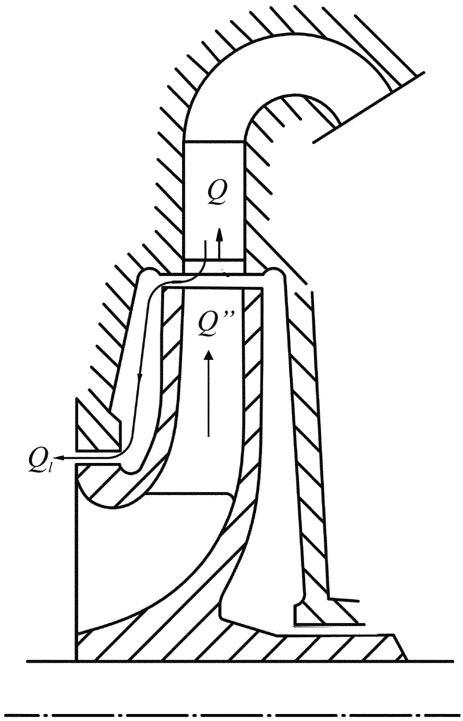


Fig. 2.6 Internal leakage losses [2]

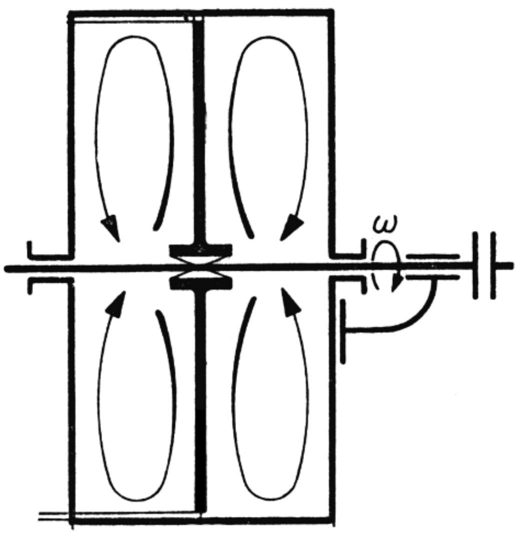


Fig. 2.7 (similar to a losses caused by a rotating disk in a fluid) Disk friction losses [2]

2. "Incidence losses" that occur when there is a difference between the flow direction and the blade angles at entrance of an impeller and entrance of diffuser with blade.
3. Other pressure losses caused by separation of boundary layers, back flow, prerotation, and recirculation losses (hydraulic losses).

The first two pressure losses change with the flow rate with a parabolic shape.

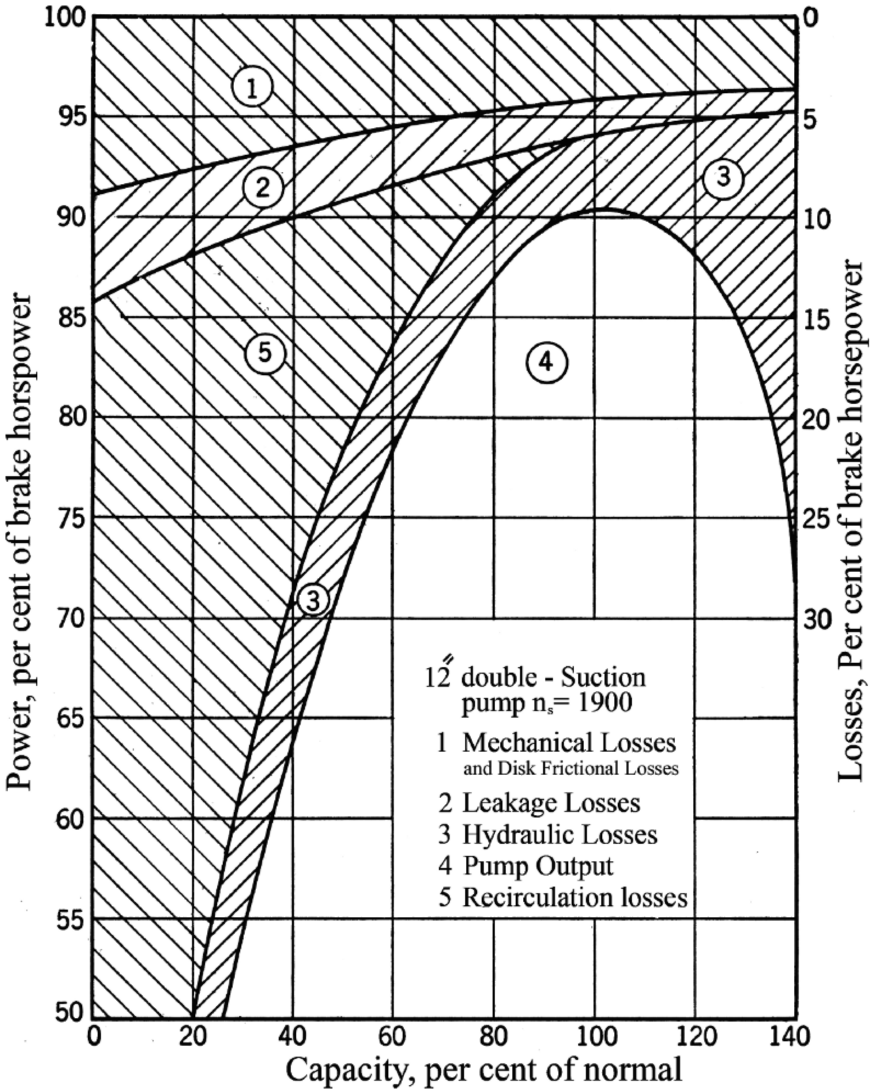


Fig. 2.8 Power characteristic curve [9]

2.6.2 Internal Leakage Losses (Q_l)

Due to the difference between the outlet pressure P_2 and the inlet pressure P_1 of an impeller, some portion of outlet fluid returns to the inlet from the existing clearances between the impeller and the casing. This "internal leakage" Q_l, causes some losses, (Fig. 2.6). Therefore, the flow passing through the impeller Q'' is larger than the useful flow rate of pump, i.e. $Q'' = Q + Q_l$

2.6.3 Disk or Wall Friction Losses (P_{01})

The fluid between external surfaces of the impeller and internal walls of the casing rotates due to viscous effects (Fig. 2.7). The power absorbed by liquid for rotating this portion of fluid is called disk friction loss. The value of this loss depends mainly on viscosity of fluid, rotational speed, and the diameter of the impeller.

2.6.4 Mechanical and External Leakage Losses

Mechanical losses in a pump include the frictional losses in the bearing and packing losses. These losses are nearly constant for a given rotational speed. The external leakage losses can appear due to flow coming out of the pump (between the shaft and the stuffing box). However, these losses are usually very small and can be neglected.

In Fig. 2.8 the power characteristic curve is presented, considering different type of losses. The numbers on the figure correspond to the losses indicated in the insert.

References

1. Nourbakhsh S. A., "Theoretical and Real Slip Factor in Centrifugal Pumps", Von Karman Inst., Technical Note 93, 1973.
2. Lazarkiewicz Stephen, "Impeller Pumps", Pergamon Press, 1965.
3. "Basic Principles for the design of Centrifugal Pump Installations", SIHI Group, 1980.
4. Nourbakhsh S. A., "Le facteur de glissement reel et sa relation avec le comportement del'ecoulement dans les machines radiales", Pumps and Pumping Systems for Liquids in Single or Multiphase Flow, Worthington European Technical Award Hoepli, Vol. III, 1974.
5. Sayers T., "Hydraulic and compressible flow turbomachines", McGraw Hill Book Company (UK) Limited, 1990.
6. Shames I. H., " Mechanics of Fluids", McGraw-Hill, 1992.
7. Streeter V. L., "Fluid Mechanics", 9th ed., McGraw-Hill, 1997.
8. Fox, R.W. and McDonald, A.T., "Introduction to Fluid Mechanics", John Wiley & Sons, 1994.
9. Stepanoff A. J., "Centrifugal and Axial Flow Pumps", John Wiley Sons, 1957.

Chapter 3
Similarity Laws

Similarity laws provide a great tool for classifying turbopumps in similar families. As will be shown in this chapter, this classification would help designers to design similar pumps with the same set of information. It also provides a practical tool for pump users to select the best pump for their application and adjust the pump to best suit to a specific application condition. The concept of the specific speed is also introduced in this chapter which is one of the fundamental definitions for the turbopump and is derived from the similarity laws.

3.1 Conditions for Similarity

Two pumps are geometrically *similar* when all corresponding dimensions have the same constant ratio and all angles such as blade angles, β_1' and β_2', are equal. Two geometrically similar pumps belong to the same *family* and are from the same *class*.

The similarity in working conditions exist between two pumps from the same class when, at all corresponding points in the two pump impellers, the velocity triangles are similar. This means all velocity vectors at those points are parallel and have the same constant ratio. In other words, if one divides the velocity vectors by the corresponding local peripheral speeds at two corresponding points, the velocity triangles become equal.

Considering the above definitions, if two pumps A and B, also from the same class, have similar working conditions, for each pair of the corresponding points the following relations are applicable:

$$\begin{gathered} \alpha_{\mathrm{A}} = \alpha_{\mathrm{B}} \qquad\qquad \beta_{\mathrm{A}} = \beta_{\mathrm{B}} \\ \left(\frac{V}{U}\right)_{\mathrm{A}} = \left(\frac{V}{U}\right)_{\mathrm{B}} \quad \left(\frac{W}{U}\right)_{\mathrm{A}} = \left(\frac{W}{U}\right)_{\mathrm{B}} \end{gathered} \tag{3.1}$$

Similarity laws, as will be shown later, can be used to find the characteristic curves of pumps in the same family and to help designers to design pumps within a family.

3.2 Dimensionless Coefficients

Consider two pumps from the same class that work under similar conditions. For these two pumps, a series of dimensionless coefficients can be defined that will be discussed in more details in the following sections.

3.2.1 Dimensionless Head Coefficient

From Euler equation, one can see that

$$H'' = \frac{1}{g}(U_2 V_2 \cos\alpha_2 - U_1 V_1 \cos\alpha_1) \tag{3.2}$$

Multiplying both sides of (3.2) by g/U_2^2 one obtains

$$\frac{gH''}{U_2^2} = \frac{V_2}{U_2}\cos\alpha_2 - \frac{U_1}{U_2}\frac{V_1}{U_2}\cos\alpha_1 \tag{3.3}$$

As was shown in (3.1), all terms in the right-hand side of (3.3) are equal for the two pumps. Therefore, for these two pumps one can write

$$\frac{gH''}{U_2^2} = Cte \tag{3.4}$$

It can be shown that under similar working conditions and when the relative roughness of the internal surfaces of two pumps and the Reynolds numbers are the same (see Sect. 7.2.2), the following relation is also valid for two pumps:

$$\frac{gH_{f0}^3}{U_2^2} = Cte \tag{3.5}$$

where H_{f0}^3 is the hydraulic loss. Therefore, since $H = H'' - H_{f0}^3$, one can write:

$$\psi = \frac{gH}{U_2^2} = Cte \tag{3.6}$$

where ψ is called the head coefficient. Therefore, two pumps from the same class, working under similar working conditions, have the same head coefficient.

3.2.2 Dimensionless Flow Rate Coefficient

Since for two pumps A and B the velocity triangles at corresponding points are similar, one can write:

$$\frac{V_2 \sin \alpha_2}{U_2} = cte \tag{3.7}$$

By multiplying both numerator and denominator of (3.7) by $A_2 = 2\pi K_2 R_2 b_2$, one can write:

$$\frac{Q''}{2\pi K_2 R_2 b_2 U_2} = cte \tag{3.8}$$

where A_2 is the effective flow cross section area at the exit of the impeller, R_2 is the outlet radius of the impeller, b_2 is the outlet width of the impeller, and k_2 is the blockage coefficient. Since the term $2\pi K_2\,(b_2/R_2)$ is same for the both pumps, one can see that:

$$\frac{Q''}{U_2 R_2^2} = cte \tag{3.9}$$

It can be shown that under similar working conditions, the following relation applies for the internal leak losses in the pump:

$$\frac{Ql}{U_2 R_2^2} = cte \tag{3.10}$$

Since $Q = Q'' - Ql$, it is clear that

$$\frac{Q}{U_2 R_2^2} = cte \tag{3.11}$$

The above term $\delta = \frac{Q}{U_2 R_2^2}$ is called flow rate coefficient and is constant for two pumps from the same class that work under similar working conditions.

3.2.3 Dimensionless Power Coefficient

Using the Euler relation, the power absorbed by the impeller can be calculated from

$$P'' = \rho g Q'' H'' \tag{3.12}$$

Both sides of the above equation can be divided by $\rho U_2^3 R_2^2$, i.e.

$$\frac{P''}{\rho U_2^3 R_2^2} = \frac{Q''}{U_2 R_2^2} \frac{gH''}{U_2^2} \tag{3.13}$$

All terms in the right-hand side of (3.13) are constant for two pumps A and B. Therefore

$$\frac{P''}{\rho U_2^3 R_2^2} = cte \tag{3.14}$$

It can be shown that under similar working conditions, for the power losses due to friction between the liquid and the pump casing, one can write

$$\frac{P_{01}}{\rho U_2^3 R_2^2} = cte \tag{3.15}$$

Since $P' = P'' - P_{01}$ (P' is the internal power), the dimensionless power coefficient can be defined as

$$\pi = \frac{P'}{\rho U_2^3 R_2^2} = cte \tag{3.16}$$

Therefore, for two pumps from the same class, working under similar conditions, the power coefficient is equal. Note that since the mechanical losses are different with hydraulic, volumetric, and side disk losses, the power coefficient is defined only for the internal power.

3.2.4 Efficiency

The internal efficiency in a pump is defined by

$$\eta_i = \frac{\rho g Q H}{P'} \tag{3.17}$$

After dividing both sides by $\rho U_2^3 R_2^2$, one can re-write the expression as

$$\eta_i = \frac{\frac{Q}{U_2 R_2^2} \frac{gH}{U_2^2}}{\frac{P'}{\rho U_2^3 R_2^2}} \tag{3.18}$$

$$\eta_i = \frac{\delta \psi}{\pi} \tag{3.19}$$

The right-hand side of the above relation is the same for two pumps A and B; therefore, the internal efficiency η_i is also constant for both pumps, when working under similar working conditions.

As mentioned before, the mechanical losses and the total efficiency of a pump do not follow the similarity rules. Therefore, (3.16) and (3.19) are not valid for total power consumption and total efficiency of the pumps. However, since the mechanical losses are rather small compared to the total power, practically the similarity laws can be extended to the total power and efficiency by accepting some error margins.

Note that the head, capacity, and power coefficients may be defined differently in some cases. Therefore, when using the empirical charts presented in different references, one must carefully examine the definitions and the conditions for the corresponding parameters.

3.3 Affinity Laws

Consider the following three dimensionless coefficients for a pump:

$$\psi = \frac{gH}{U_2^2}, \quad \delta = \frac{Q}{U_2 R_2^2}, \quad \pi = \frac{P'}{\rho U_2^3 R_2^2} \tag{3.20}$$

Using these coefficients for a single pump that works with two different rotational speeds, n_1 and n_2, under two similar working conditions, one can write:

$$\frac{Q_1}{Q_2} = \left(\frac{n_1}{n_2}\right), \quad \frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2, \quad \frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{3.21}$$

where quantities denoted with subscripts 1 and 2 correspond to quantities under rotational speeds n_1 and n_2, respectively. The equations of (3.21) are called "Rateau" equations or Affinity Laws. Based on affinity laws, for the corresponding working points in a pump or for two pumps which are geometrically similar, while the ratio of volumetric flow rates is the same as the ratio of the corresponding rotational speed, the ratio of the manometric heads is equal to the square ratio of the rotational speeds. For all these cases, the internal efficiency remains constant. It should be noted that the performance of the pumps based on affinity laws are guaranteed by pump manufacturers for water or pure liquids with kinematic viscosities less than 10 cS [1]. For liquids with higher viscosities, the above assumptions are no longer valid. In addition, the above equations are still applicable if the rotational speed changes are in the range of -5% to $+10\%$.

It is also a usual practice in pump manufacturing to trim the rotor or impeller to adjust its performances. For a centrifugal pump impeller reducing its diameter from D_{2A} to D_{2B} at a constant rotational speed, one can write the following relations [3]:

$$\frac{H_1}{H_2} \cong \left(\frac{D_{2A}}{D_{2B}}\right)^m \quad \text{and} \quad \frac{Q_1}{Q_2} \cong \left(\frac{D_{2A}}{D_{2B}}\right)^m \tag{3.22}$$

where for the impeller trimmings more than or equal to 6%, $m = 2$ and when it is less than or equal to 1%, $m = 3$. For other values of trimming, m changes between 2 and 3. Note that in such cases there is no geometrical similarity between two impellers. However, when the amount of trimming is small, one can use the relations presented in (3.22) and the cavitation conditions will not change considerably.

3.4 Dimensionless Characteristics

In the previous section, the dimensionless parameters, ψ (head coefficient), δ (flow rate coefficient), π (internal power coefficient), and η_i (internal efficiency) were defined. The characteristic curves (Fig. 3.1), plotted based on these parameters, are called the dimensionless characteristics, i.e.

$$\psi = f_1(\delta) \qquad \pi = f_2(\delta) \qquad \eta_1 = f_3(\delta) \tag{3.23}$$

Based on similarity laws, the dimensionless characteristics of all pumps belonging to the same class are identical. Once the dimensionless characteristics of a class of pumps are known, one can determine the characteristics of the pumps from the same class, knowing their diameters and rotational speeds.

Since the dimensionless coefficients are related to each other by (3.19), only two dimensionless characteristics are needed to represent a family of pumps.

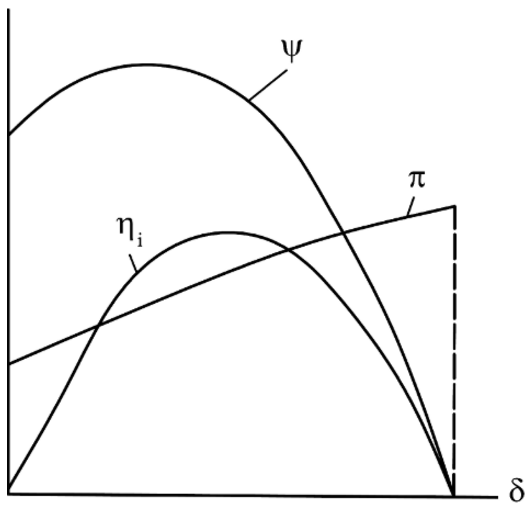


Fig. 3.1 Dimensionless characteristic curves [1]

3.5 Application of Similarity Laws

3.5.1 Pump Characteristic Curves at Different Speeds

When the pump characteristic curves are known for a particular rotational speed, the similarity laws can be used to find the characteristic curves at different speeds.

Figure 3.2 shows the manometric head curve for a pump at rotational speed of n. To obtain the head curve for the same pump at another rotational speed n', one can choose an arbitrary point on the curve, as A, with the manometric head H_A and capacity Q_A. When the rotational speed changes from n to n', the corresponding working point A' can be obtained using similarity laws from:

$$H'_A = \left(\frac{n}{n'}\right)^2 H_A \quad \text{and} \quad Q'_A = \left(\frac{n}{n'}\right) Q_A \tag{3.24}$$

Therefore, the position of point A' on the H–Q plane is specified. This procedure can be repeated for some other points like B, C, ... and the corresponding points B', C', ... can be identified. Now, the new points can be connected to each other to form a new characteristic curve for the same pump at the rotational speed n'.

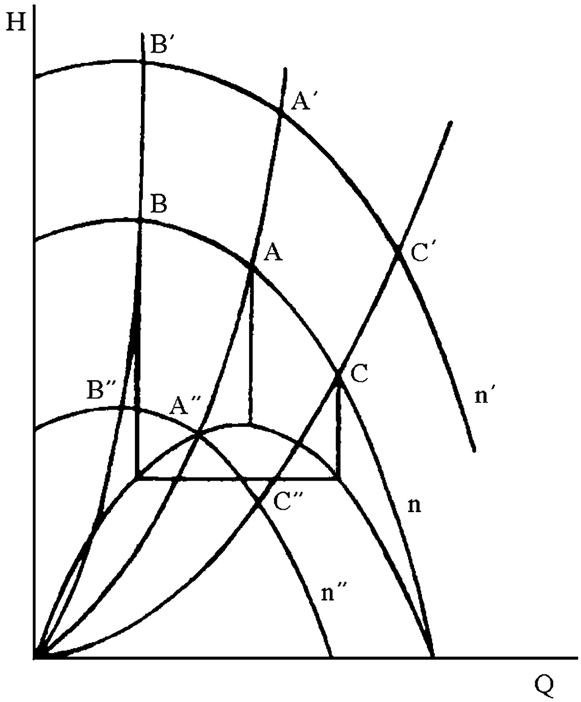


Fig. 3.2 Finding H–Q curve with changing rotational speed [1]

If the same procedure is repeated for the corresponding points for a different rotational speed like n'', another characteristic curve can be obtained. As shown in Fig. 3.2, by connecting the matching points A, A', and A'', a new curve is created that according to the similarity laws is the iso-efficiency curve (since all these points have the same efficiency). Since the flow rate of the pump (the horizontal axis) is proportional to the rotational speed and the manometric head (the vertical axis) is proportional to the square of the speed, the efficiency curves are elliptic. However, in practice, the efficiency curves are elliptic only when the changes in the rotational speed are small. At lower speeds, the effect of Reynolds number on the efficiency is larger and this changes the shape of the efficiency curves. See Fig. 2.2 for more illustration.

3.5.2 *The Approximate Manometric Head of the Centrifugal Pumps*

In many industrial pumps, the head coefficient ψ has a variation between 0.45 and 0.55 at the maximum efficiency point. Because of this small variation, one can assume that

$$\frac{g\,H}{U_2^2} = 0.5 \tag{3.25}$$

This means

$$H = \frac{\left(\frac{\pi D_2 n}{60}\right)^2}{2g} \tag{3.26}$$

This equation can be written in SI unit system as

$$H = \left(\frac{D_2 n}{8450}\right)^2 \tag{3.27}$$

In the above relation, D_2 is the outer diameter of the impeller in centimeters, n is the rotational speed in rpm, and H is the manometric head of the pump in meters. In industrial applications, it can be shown that one may obtain the manometric head of the pump from (3.27) with good approximation.

3.6 Limits of Similarity Laws

In deriving the similarity laws in the previous section, it was assumed that the friction losses are the same in both pumps. This assumption is not valid everywhere. In fact, the friction coefficient, f (see Sect. 7.2), is a function of the Reynolds number,

Re, and the relative roughness of the surface. Although in turbulent flows the effect of Reynolds number on the friction coefficient is negligible, its effect on laminar flows is important and should be considered (e.g., in pumps handling highly viscous liquids). Also, the effect of the surface relative roughness is negligible in the range of working conditions. For these reasons, usually the effects of Reynolds number and relative roughness are neglected and the similarity laws are used in deriving the characteristic curves for pumps in the same family. However, one must always consider that the calculations are within some error margin.

Another type of limitation for using the similarity laws is the liquid to be pumped. In the derivation of similarity laws, it was assumed that the liquid is single phase, i.e. the amount of dissolved gases or air in the liquid is negligible and the liquid is not changing phases while being handled. If a phase change occurs in the flow, the properties of the flow change and there would be no similarity in working conditions. One of the possible cases for the phase change is the cavitations phenomenon which occurs when a pump works with high rotational speed. In those cases, the liquid evaporates and vapor bubbles are produced.

3.7 Specific Speed

The head and flow rate coefficients can be re-written as follows:

$$\psi = \frac{gH}{U_2^2} = \frac{gH}{R_2^2\omega^2} \quad \text{and} \quad \delta = \frac{Q}{U_2 R_2^2} = \frac{Q}{R_2^3\omega} \tag{3.28}$$

The outer radius of the impeller, R_2, can be obtained by rearranging the above equations:

$$R_2 = \left(\frac{gH}{\psi\omega^2}\right)^{1/2} = \left(\frac{Q}{\delta\omega}\right)^{1/3} \tag{3.29}$$

which leads to the following equality:

$$\delta\left(\frac{g}{\psi}\right)^{3/2} = \frac{\omega^2 Q}{H^{3/2}} \tag{3.30}$$

The left side of (3.30) is the same for all pumps from the same family, working under similar conditions. Therefore, one can write:

$$\omega\frac{Q^{1/2}}{H^{3/4}} = Cte \tag{3.31}$$

With the help of (3.31) one can define the specific speed. The specific speed, ω_s, of a pump is defined as the rotational speed of a pump in the same family that can produce a manometric head equal to one with the flow rate equal to one (in the same system of units). This means

$$\omega_s \frac{1^{1/2}}{1^{3/4}} = \omega \frac{Q^{1/2}}{H^{3/4}} \tag{3.32}$$

i.e.

$$\omega_s = \omega \frac{Q^{1/2}}{H^{3/4}} \tag{3.33}$$

or

$$n_s = n \frac{Q^{1/2}}{H^{3/4}} \tag{3.34}$$

It is obvious that n_s is not dimensionless and its unit depends on the units used for the other variables. In SI units, specific speed is defined by

$$n_s = n(rpm) \frac{\sqrt{Q(m^3/s)}}{H^{3/4}(m)} \tag{3.35}$$

Specific speed, n_s, has an important application in turbopumps design. It should be noted that the specific speed is defined only at the point of maximum efficiency. Therefore, two pumps from the same class, working under similar conditions, with the same specific speed have the same maximum efficiency. Thus, if all design documents for a pump are available, by using the specific speed and the similarity

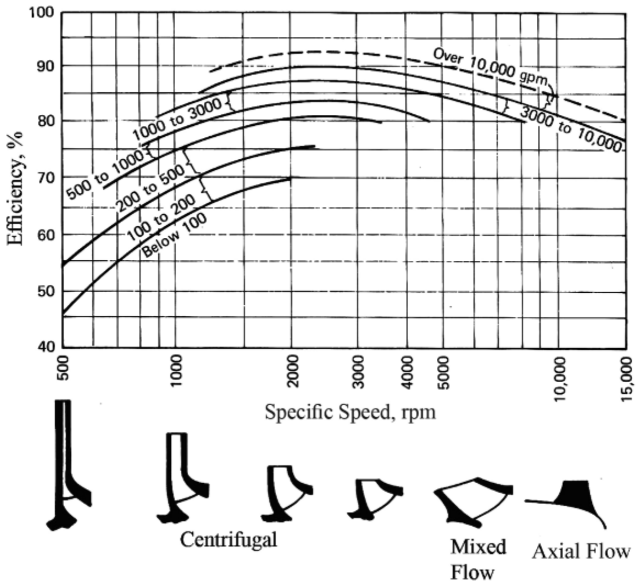


Fig. 3.3 Classification of turbopumps based on the specific speed (English Units) [4]

laws, the design for other pumps in the same class can be done easily. For this reason, usually all the characteristic curves and working tables for the pumps are given based on the specific speed (Fig. 3.3). In English units, Q and H are defined in GPM and ft, respectively. n_s (English units) = 51.6 n_s (Metric units).

References

1. Jaumotte A. L., “Description et Construction des Machines, Turbomachines, Turbopumpes Centrifuges”, Presses Universitaires de Bruxelles, 1984–1985.
2. Lazarkiewicz Stephen, “Impeller Pumps”, Pergamon Press, 1965.
3. “Centrifugal Pump Hand Book”, Sulzer Brothers Limited, Winterthur Switzerland.
4. Simon Andrew L., “Practical Hydraulics”, John Wiley & Sons, 1981.

Chapter 4
Cavitation

Water or any other liquid vaporizes at certain temperature and pressure (vapor pressure). For example, at standard atmospheric pressure and 100°C water evaporates. Now, if the pressure is reduced to 0.02 atm, water evaporates at 20°C. As the liquid flows inside the pump impeller, if the liquid pressure reduces to a level lower than the vapor pressure of the liquid at the working temperature, vapor bubbles are formed. These bubbles will move along with the flow to the other parts of the impeller with higher pressures. If the pressure at the new location is high enough, these vapor bubbles collapse. In this case, cavities are formed in the liquid, causing other liquid particles to deviate from their paths and hit other surfaces like impeller blades with a very high impact speed. In these locations on the surfaces, depending on the severity of the impact, erosion occurs on the blades and the surface becomes porous. This phenomenon is called cavitation. The main objective of this chapter is to explain why cavitation occurs in a pump and what parameters must be considered during pump selection to avoid this phenomenon during pumping operation.

4.1 Cavitation Process in Impeller Pumps

Cavitation is very dangerous for pumps and may erode the impeller after some period of time. Therefore, it must be avoided. The erosion mechanism due to cavitation is very complex and the details about it can be found in other textbooks. Here, only the signs of cavitation and the practical remedies for avoiding this phenomenon will be discussed.

In centrifugal pumps, during the flow suction to the impeller, the velocity of the flow increases and the local pressure decreases, up to some point that, close to the impeller entrance, the pressure reaches to its minimum. If at this point (minimum pressure) the liquid pressure remains higher than the vapor pressure, the liquid keeps moving as a single-phase flow. Otherwise, cavitation occurs. Therefore, it is very important to find this minimum pressure (see Sect. 4.2).

Cavitation always starts with a noise and in the case of more decrease in the entrance pressure the intensity of noise increases. The noise caused by cavitation is very distinguishable and recognizable and sounds like hitting bullets at a surface. At the same time, when noises are generated, vibrations also start in the pump structure. As the cavitation progresses, the sound changes to a continuous and loud noise and because at this time the liquid vapor blocks the impeller passages, the flow rate of the pump is reduced significantly.

As mentioned before, the location of harsh erosion on the blades is different from the location at which the bubbles are generated. In fact, when these bubbles reach to the exit of the pump, they rapidly collapse, causing pressure fluctuation with very large frequency. The frequency of cavitation is between 20 and 25 kHz and the maximum pressure is few times higher than the dynamic pressure of the liquid (in some cases to seven times). The liquid particles hit the surface with high velocity and frequency and this causes erosion. In Fig. 4.1 a photo of an impeller that is eroded due to cavitation is shown.

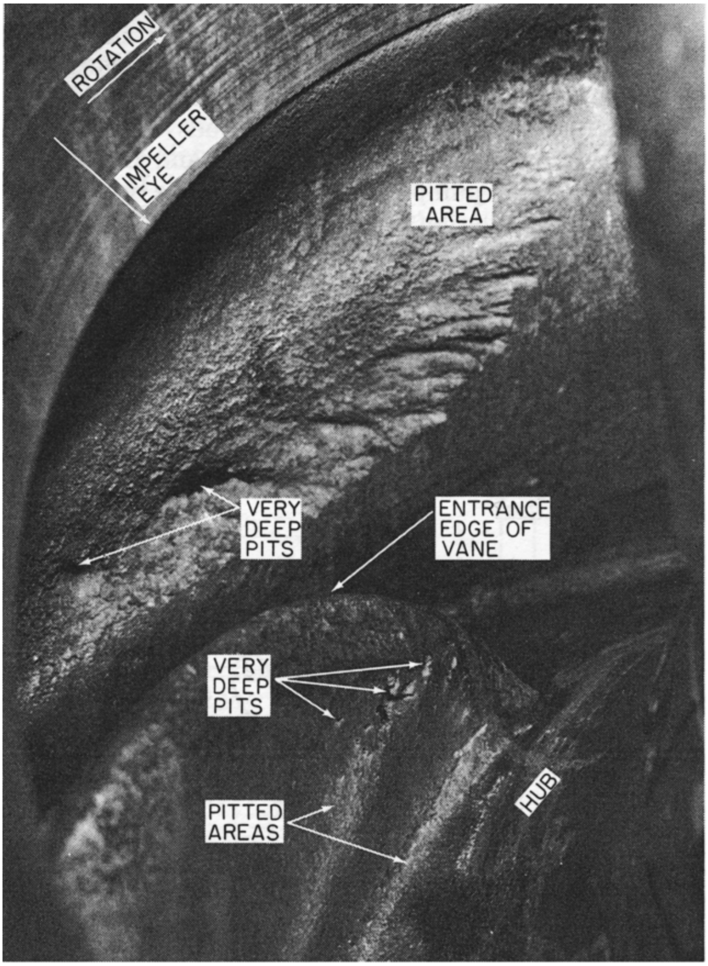


Fig. 4.1 Erosion due to cavitation [1]

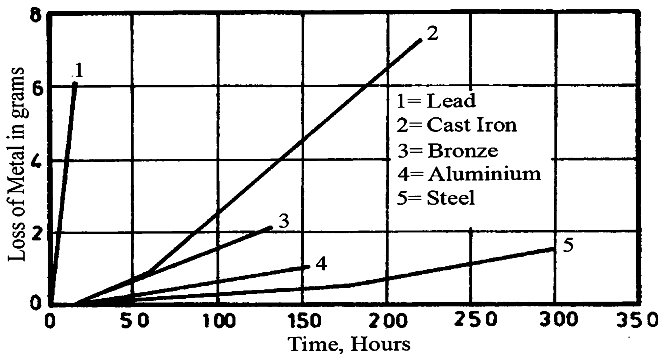


Fig. 4.2 The strength of different metals to cavitation [2]

Additives in the liquid which increase vapor pressure, as well as corrosive properties of the liquid, can increase cavitation damage. For example, cooling tower water treatment agents may accelerate the damage. Also solid/abrasive particles in the liquid in addition to the high implosive velocities from the collapsing vapor bubbles increase the wear rate.

Different materials show different resistance to the cavitation. In general, there is no material that can be completely resistant to this phenomenon. Rigid plastics and composites are normally the least cavitation resistant materials. Cast iron and brass will experience the most damage of commonly used metals, while stainless steel, titanium, nickel, aluminum, and bronze will have much less damage, under the same cavitation conditions.

Both erosion and corrosion are possible in the pumps, although the importance of the former is much more prominent. The resistance of different materials to cavitation also depends on the way the material is formed, the roughness of the surface, the type of alloys, and the uniformity of the surface in casting or machining process, its strength properties against fatigue, as well as whether it has gone under heat treatment or not.

In Fig. 4.2 the test results of different metals are shown. These tests have been performed in a special Venturi and with the liquid velocities at around 60 m/s. As one can see, steel has the best performance with respect to other common metals.

4.2 Minimum Pressure Point

Figure (4.3) shows a section of an impeller in a turbopump. Point B in this figure shows the point at which the liquid pressure is minimal. This point is located on the low-pressure side, close to the leading edge of the blades. As was mentioned in the previous section, cavitation starts from this point. Therefore, to avoid cavitation, the liquid pressure at this point must be kept higher than the vapor pressure. In other words, $P_{\text{min}} > P_{\text{v}}$.

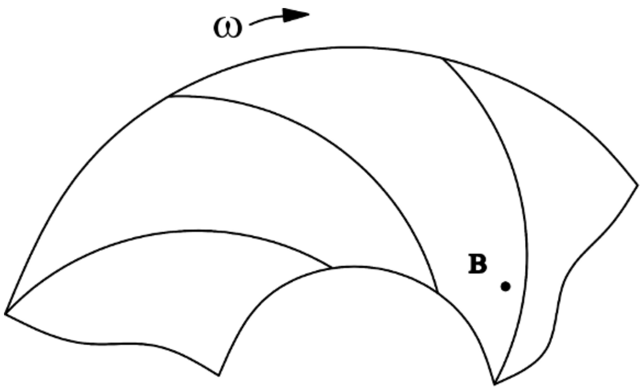


Fig. 4.3 The point of minimum pressure in an impeller

The Bernoulli's equation between the entrance point of the impeller (point 1) and the minimum pressure point can be written as:

$$\frac{P_1}{\rho g} + \frac{W_1^2}{2g} - \frac{U_1^2}{2g} = \frac{P_{\min}}{\rho g} + \frac{W_{\max}^2}{2g} - \frac{U^2}{2g} \tag{4.1}$$

In the above equation, the pressure losses between two points are neglected (note that since at the point with minimum pressure the velocity is at its maximum, the index "max" has been used for the relative velocity). Also, since two points are close together, it can be assumed that the impeller radius is the same for two points and therefore $U_1 = U$. Thus,

$$\frac{P_1 - P_{\min}}{\rho g} = \frac{W_{\max}^2 - W_1^2}{2g} \tag{4.2}$$

or

$$\frac{P_1 - P_{\min}}{\rho g} = \frac{W_1^2}{2g}\left[\left(\frac{W_{\max}}{W_1}\right)^2 - 1\right] \tag{4.3}$$

The term inside the bracket is a non-dimensional coefficient which depends on the type of impeller profile, pump class, and working conditions. By defining $\lambda = \left(\frac{W_{\max}}{W_1}\right)^2 - 1$ and $\Delta p = P_1 - P_{\min}$, one can show

$$\frac{\Delta p}{\gamma} = \lambda \frac{W_1}{2g} \tag{4.4}$$

λ changes as the pump flow rate changes and has its minimum value at the design point. The values of λ for the design point of axial pumps vary between 0.25 and 0.3, and for centrifugal pumps between 0.16 and 0.2.

4.3 Maximum Suction Head

The cavitation phenomenon imposes limitations on the pump suction head. Figure 4.4 shows a simple pump set-up. This pump has to get the water from a lower source with pressure P_A. The Bernoulli's equation between point A on the surface of water and point 1 at the entrance to impeller can be written as

$$\frac{P_A}{\gamma} + \frac{V_A^2}{2g} + Z_A = \frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 + H_{f_A^1} \tag{4.5}$$

where Z_A and Z_1 are the geometrical heights of the water surface and pump axis, respectively, from an arbitrary reference level. By denoting the suction height as $Z_s = Z_1 - Z_A$ and neglecting the flow velocity on the surface of the water ($V_A = 0$), the above equation can be rewritten as:

$$Z_s = \frac{P_A}{\gamma} - \left(\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + H_{f_A^1}\right) \tag{4.6}$$

After substituting (4.2) into (4.6), one can see that

$$Z_s = \frac{P_A}{\gamma} - \left(\frac{P_{\min}}{\gamma} + \lambda\frac{W_1^2}{2g} + \frac{V_1^2}{2g} + H_{f_A^1}\right) \tag{4.7}$$

The above equation is valid for any working condition, including the condition under which cavitation occurs. Under that condition, $P_{\min}$ reduces to vapor pressure of the liquid, P_v, and therefore, the maximum allowable suction pressure can be obtained from

$$Z_{S\,\max} = \frac{P_A}{\gamma} - \left(\frac{P_v}{\gamma} + \lambda\frac{W_1^2}{2g} + \frac{V_1^2}{2g} + H_{f_A^1}\right) \tag{4.8}$$

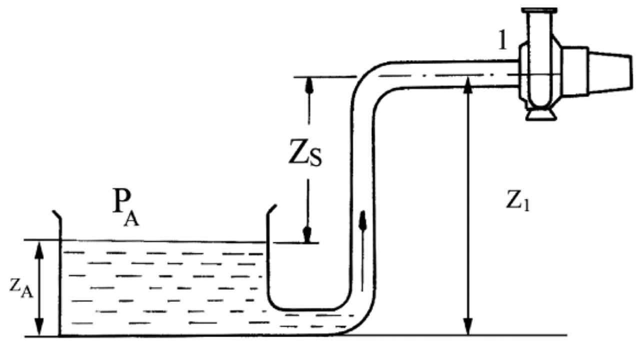


Fig. 4.4 Suction head

The following conclusions may be derived from (4.8):

1. As the pressure in the suction reservoir, P_A, increases, the maximum allowable suction head increases as well. In open reservoirs, P_A is the atmospheric pressure. At the sea level, $\frac{P_{\text{atm}}}{\gamma}$ is equal to 10.33 m of water and its value decreases with increasing altitude from the sea level.
2. As the temperature of the liquid increases, the vapor pressure, P_v, increases and as a result the maximum allowable suction head decreases. For this reason the suction elements of the feeding pumps of boilers in the power plants must be pressurized.
3. As the velocity V_1 increases in the pump suction, the maximum allowable suction pressure decreases. For this reason, one cannot design a turbopump with a very high liquid velocity at the entrance. For the same reason, even the rotational speeds are limited in these types of pumps, because the absolute and relative liquid velocities depend on this parameter.
4. As the pressure losses increase in the suction pipe, the maximum allowable suction head decreases. Therefore, to avoid cavitation in the pumps, fittings with high-pressure loss coefficients like valves are not located on the suction pipe. Also, the pipe diameters at the suction side are usually larger than the diameters at the discharge side.
5. Stepanoff [5] has proposed an empirical relation to estimate the maximum allowable suction head:

$$Z_{S\ \max} = \frac{89.5}{g} - 3.5\frac{V_1^2}{2g} \tag{4.9}$$

The above equation can be used as a first approximation for estimating the maximum allowable suction head for centrifugal pumps working with water at 20 °C, with a reasonable rotational speed, and when the suction reservoir is open under the atmospheric pressure.

4.4 Net Positive Suction Head

Since the value of λ that has been defined in the previous section is usually unknown and because of its dependency to flow rate, (4.9) cannot be used practically. For this reason, most of the pump manufacturers define another parameter, NPSH, that is defined by:

$$\text{NPSH} = \frac{P_1}{\gamma} + \frac{V_1^2}{2g} - \frac{P_v}{\gamma} \tag{4.10}$$

There are two types of NPSH:

1. Net positive suction head for the system or NPSH_R which is defined as the total energy that exists in the liquid which can be calculated from (4.10).

2. Net positive suction head for the pump or $NPSH_P$ which is defined as the suction head that is required in the liquid at the inlet of the pump in order to avoid cavitation. To calculate $NPSH_P$, one can use (4.10), substituting velocity and pressure for the cavitation in the equation.

Pump manufacturers usually present the $NPSH_P$ characteristic curves for the pump in their catalogs. However, the $NPSH_R$ shall be calculated for any specific piping system. It is obvious that the condition for avoiding cavitation is

$$\mathrm{NPSH}_R(system) > \mathrm{NPSH}_P(pump) \tag{4.11}$$

There are several factors that would affect the degree of cavitation erosion damage (and sometimes noise) within a pump when sufficient NPSH margin is not provided above the NPSH of the pump. Some of these factors are as follows:

- Pump size: Large pumps (impeller inlets over 450 mm) in diameter can be more prone to cavitation damage than smaller pumps.
- The gas content of the liquid: Small amounts of entrained gas (1% to 2%) cushion the forces from the collapsing cavitation bubbles, and can reduce the resulting noise, vibration, and erosion damage. The lack of any entrained gas can have the opposite effect. Warmer liquids tend to release less dissolved gas, which increases the noise level of a pump. On the other hand, gas can accumulate in the inlet of a pump and block portions of the flow area, thus increasing the inlet velocity of the liquid and creating even more cavitation. This increases the apparent $NPSH_R$ of the pump. The net result of these two counter effects of gas content on pump noise and vibration will vary based on the suction energy level of the pump.
- The duty cycle of the pump: Cavitation damage is time related. The longer a pump runs under cavitation conditions, the greater the extent of damage. Fire pumps, which run intermittently, rarely have a problem with cavitation damage for this reason.

4.5 The Allowable Suction Head and NPSHP

The allowable suction head can be obtained from

$$Z_{S\,\max} = \frac{P_A}{\gamma} - \left(\frac{P_v}{\gamma} + \frac{\Delta P_{\min}}{\gamma} + \frac{V_1^2}{2g} + H_{f_A^1}\right) \tag{4.12}$$

where $\Delta P_{\min} = P_{1\,\min} - P_v$. Therefore,

$$Z_{S\,\max} = \frac{P_A}{\gamma} - \left(\frac{P_v}{\gamma} + \frac{P_{1\,\min}}{\gamma} - \frac{P_v}{\gamma} + \frac{V_1^2}{2g} + H_{f_A^1}\right) \tag{4.13}$$

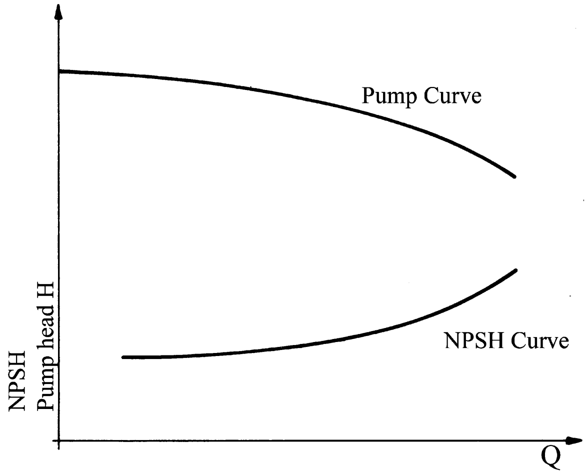


Fig. 4.5 NPSH characteristic curves for pumps [3]

or

$$Z_{S\,\max} = \frac{P_A}{\gamma} - \left(\frac{P_v}{\gamma} + \mathrm{NPSH}_P + Hf_A^1 \right) \tag{4.14}$$

In many cases the pressure in the suction reservoir is atmospheric pressure and the liquid temperature (water) is the same as the ambient temperature. In this case the maximum allowable suction head at which the pump can be installed would be approximately equal to

$$Z_{S\,\max} \cong 10.33 - \mathrm{NPSH}_P \tag{4.15}$$

One can deduct about 0.5 m inside the suction pipe for the pressure losses, vapor pressure for water, and safety margin to find the installation height. Note that the vapor pressure of liquids also depends on the altitude of the location. Therefore, the number 10.33 that was calculated at the sea level shall be modified if necessary.

As mentioned before, the NPSH_P is presented in the pump catalogs; therefore, one can easily estimate the pump installation height. Also, note that the NPSH_P is a function of the flow rate. Therefore, the characteristic curve $\mathrm{NPSH}_P = f(Q)$ is presented as one of the pump's characteristic curves by the pump manufacturer. Figure (4.5) shows the NPSH characteristic curves for a centrifugal pump.

4.6 Head Performance Drop due to Cavitation

Due to cavitation a significant drop in head is observed. This can be seen in Fig. 4.6 where the numbers 1–3 correspond to decrease in values of $\mathrm{NPSH}_{R.}$ of the system. For example, if the cavitation appears at point 3, the pump head suddenly drops and it is not possible to have higher flow rate after this point [4, 5].

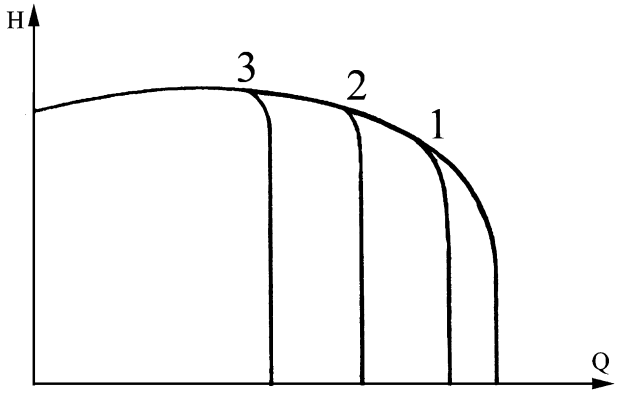


Fig. 4.6 Head drop in a pump due to cavitation

References

1. Karassik Igor J. *et al.*, “Pump Handbook”, McGraw-Hill, 2001.
2. Lazarkiewicz Stephen, “Impeller Pumps”, Pergamon Press, 1965.
3. “Centrifugal Pump Design”, Technical Appendix, KSB, Frankenthal, Germany.
4. Nourbakhsh S. A., “Turbomachinery”, University of Tehran Press, 2005.
5. Stepanoff A. J., “Centrifugal and Axial Flow Pumps”, John Wiley Sons, 1957.

Chapter 5
Axial and Radial Thrusts

The axial and radial thrusts in a turbopump are produced by flow movement inside the impeller. These forces must be balanced hydraulically or by other proper methods. Otherwise, they will insert unwanted forces to the pump shaft, damage the pump structure, and interfere with the pump operation. Because of the importance of these forces and since the balancing devices that are used to hydraulically balance these forces are part of the pump elements, this chapter is dedicated in Part I to introduce these forces and to describe the appropriate balancing methods. While in Sects. 5.1 and 5.2 the axial thrust and the balancing methods, especially in multi-stage pumps, are discussed, in Sect. 5.3 the methods for determining the magnitude and direction of this force is described in more detail.

5.1 Axial Thrust

In practice, even after all design considerations, the pressure distributions on two sides of an impeller inside a turbopump are not equal. Also, because of the difference in the magnitude and direction of the flow velocity at the inlet and outlet of the impeller, the flow momentum would not be the same in these two locations. For this reason, there is always an acting force on the impeller, in the direction of the pump axis. This axial thrust must be supported by the bearings and be reduced as much as possible.

5.2 Balancing the Axial Thrust

5.2.1 Single-Stage End Suction Pumps

In small pumps, usually the bearings can handle the axial thrust adequately. However, in medium and large pumps, the axial thrust must be balanced hydraulically. One of the usual methods is to provide a balancing chamber. For this purpose, as

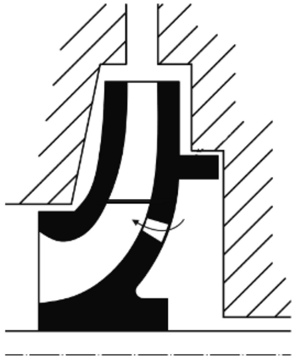


Fig. 5.1 Impeller with balancing chamber [1]

shown in Fig. 5.1, small holes are made on the back wall of the impeller, to balance the pressures at both sides of the impeller.

In this method, a circular ring is also provided on the back wall of the impeller as shown in Fig. 5.2. This ring would keep the liquid inside the balancing chamber. The number of holes and their diameters should be enough in order for the flow to pass through with minimum pressure loss. But even under the best conditions, the pressure inside the balancing chamber is always less than P_0 (suction pressure). For this reason, the radius of the back wall ring must be larger than the inlet radius, such that the resulting forces cancel each other. This ring is usually added to the impeller during casting process, as one piece. In some cases, in order to make the fabrication easier, the inlet radius and the ring radius are taken to be the same. In this way, only 90% of the axial thrust is balanced. Figure 5.2 shows a balancing chamber with identical balancing rings in front and back.

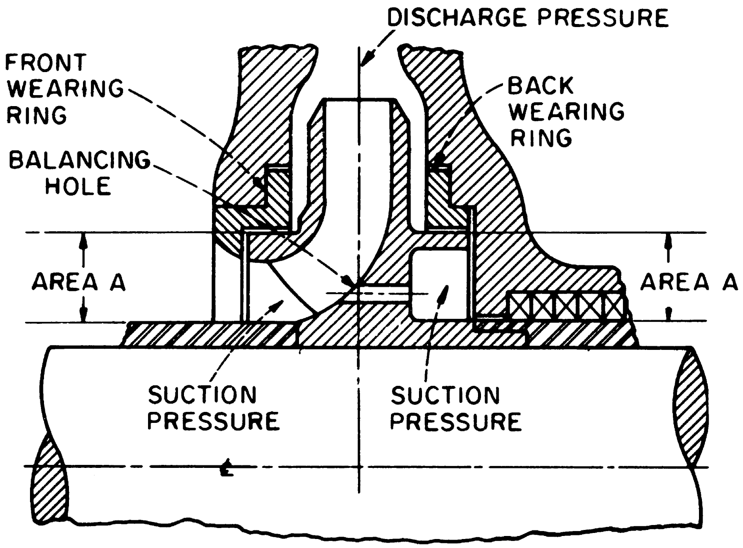


Fig. 5.2 The balancing chamber [2]

In Fig. 5.3, the change in axial thrust with flow rate for a centrifugal pump with an impeller diameter of 215 mm and inlet diameter of 75 mm is shown. Curve number (1) shows the axial force, Fa, when there is no provision for balancing the axial thrust. Curve number (2) shows the axial thrust after a balancing chamber is installed. The radii of the rings at two sides of the impeller are the same and equal to 105 mm. The number of the balancing holes is 9, each with a diameter of 10 mm.

By adding a balancing chamber to an impeller, the efficiency of the pump will change, because it would almost double the leaking losses. A more efficient way of building a balancing chamber is to connect the chamber to the inlet through a small pipe, as shown in Fig. 5.4 by pipe B, instead of drilling holes in the impeller. This pipe connects the suction part 1 to the balancing chamber 2. With this provision, the flow pattern at the inlet of the impeller would not be disturbed.

Another method of balancing the hydraulic thrust is to install radial vanes at the back of the impeller (like a two-sided impeller) (Figs. 5.5 and 5.6). These radial vanes would push the liquid toward the volute and would reduce the pressure at this location. This method would not eliminate all the axial thrust. The disadvantage of this method is that the manufacturing of the two sides of the impeller would be more complicated.

Another solution to balance the axial thrust is to use an automatic balancing method (Fig. 5.7). In this method, the impeller can move freely on the axis. When the impeller moves in one direction, the distance between the moving and stationary parts would change. This will change the pressure difference on the impeller walls and therefore would change the direction of the axial thrust and would limit the

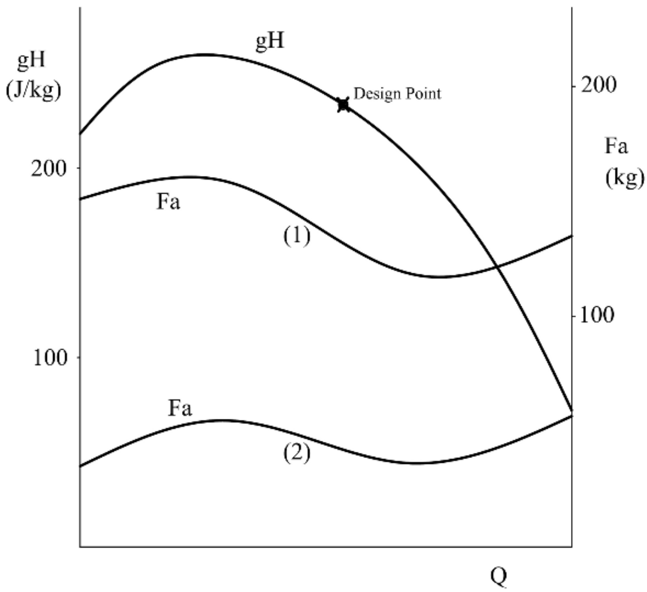


Fig. 5.3 Variation of the axial thrust, Fa, with flow rate, with (2) and without (1) a balancing chamber [1]

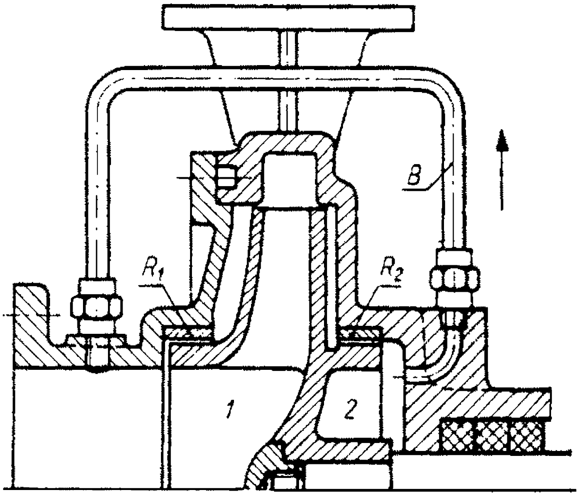


Fig. 5.4 Connecting the balancing chamber, 2, to the inlet of the impeller, 1, with pipe B [3]

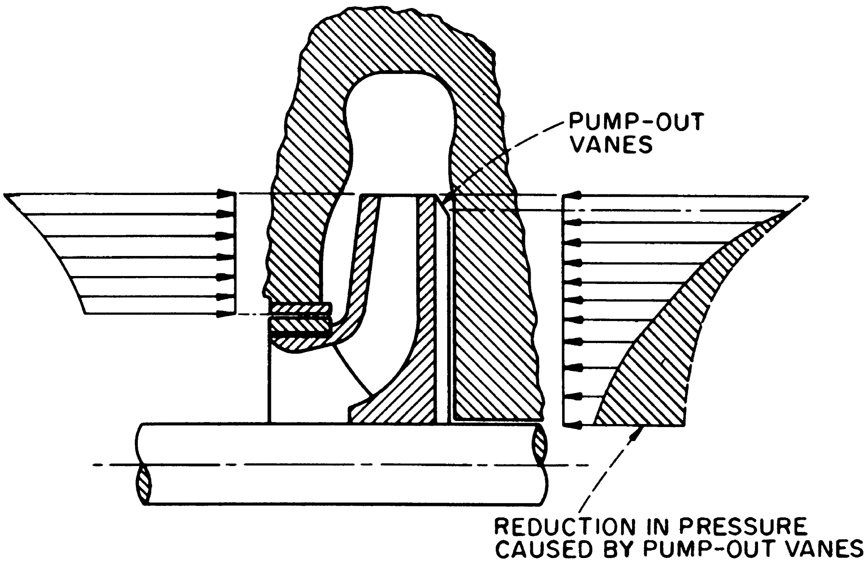


Fig. 5.5 Radial vanes at the back of the impeller and the resulting force balance [2]

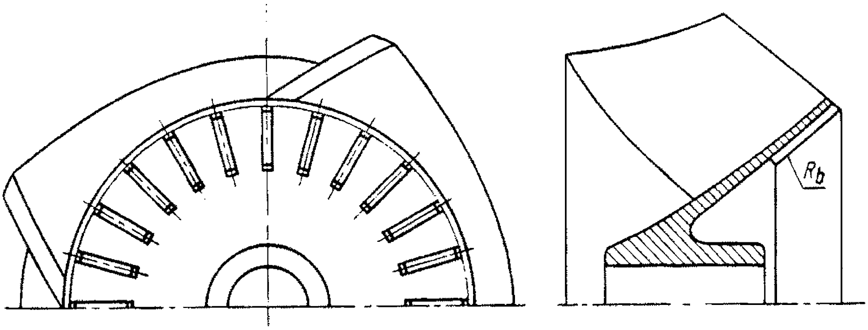


Fig. 5.6 Radial vanes at the back of a mixed flow impeller. R_b is the radius of the radial vanes [3]

movement of the impeller. This method is expensive and therefore is not used in single stage pumps.

There are other methods like manufacturing impellers with two walls, each having different diameters (Fig. 5.8), which is not commonly used. It should be noted that even when the axial thrust is completely balanced, the mechanical parts of the pump must be chosen with care and especially the bearings must be such that they can handle the unbalanced portion of the axial force.

5.2.2 Single-Stage Double Suction Pumps

The impeller of these pumps is located vertically on the shaft and basically all axial forces must balance each other (Fig. 5.9). However, because of the minor defects during the manufacturing process and also uneven distribution of flow in both sides of the impeller, there is always the possibility of axial thrust residual. To balance the force, a vertical space is provided between the rotor and the stationary part of the pump. For example, if the axial thrust is from right to left, the impeller would move

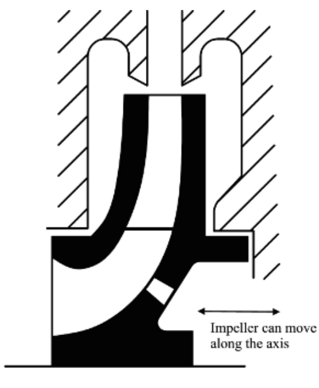


Fig. 5.7 Automatic balancing of the axial thrust [1]

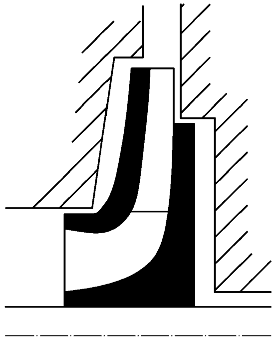


Fig. 5.8 Impellers with walls with two different diameters [1]

toward the left direction. This would decrease the axial clearance on the left side and thus increase the clearance in the right side. Consequently, the liquid pressure acting on the left side of the impeller would increase and would form an axial thrust in the opposite direction (automatic balancing).

5.2.3 Multi-stage Pumps

The axial thrust acting on multi-stage pumps is considerably higher and therefore it is important to correctly estimate the order of magnitude of this force and provide an efficient balancing mechanism. For example, in a small ten-stage pump with an inlet radius of $R_0 = 4\,\text{cm}$ and shaft radius of $R_a = 2\,\text{cm}$, which rotates at 2900 rpm and delivers a head of 350 m, the axial thrust is about 1200 kgf.

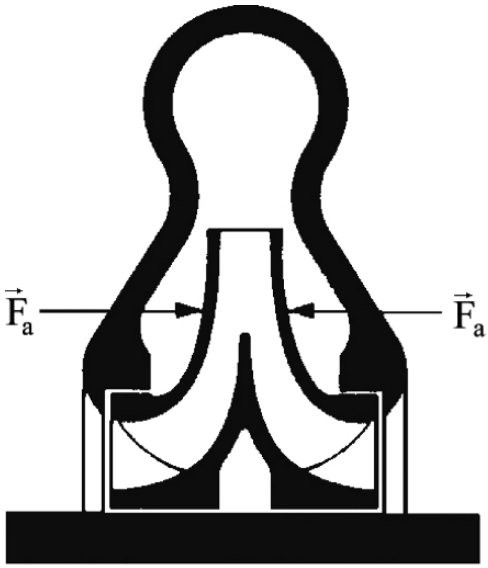
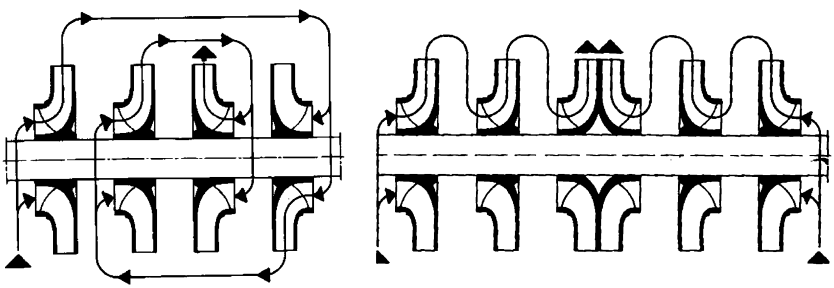


Fig. 5.9 A double suction impeller [4]

In small pumps, it is possible to balance the axial thrust mechanically and in fact these methods would be economically more feasible. However, in larger pumps, only hydraulic methods must be used. To do so, there are several methods that are common in practice:

1. One method is to divide impellers into two groups and assemble each group symmetrically. One group has all entrances on the left, whereas in the other group the entrances of the impellers are on the right. Figure 5.10a shows an example of this method along with the flow path. As it can be seen from Fig. 5.10b, in this arrangement, the impellers can be positioned in series with only one flow entrance.
2. In the second method, the impellers are divided into groups of two. In each group, the impellers are located in front of each other, such that the entrance of one impeller locates at the opposite side of the other one (Fig. 5.11). This method is more expensive than the first method and sometimes it is harder to implement due to difficulty of manufacturing more complex parts. Nonetheless, this method is used more often in high-pressure multi-stage pumps.
3. An old method is to position all impellers with entrances in one side and to balance the axial thrust in each impeller independently, like in the single stage pumps. The problem is that the leakage flow rate would be increased and consequently the efficiency would decrease. This method is not used any more in modern pumps.
4. In the fourth method, where the entrances of all impellers are in one side, the balancing of the axial thrust will be provided by installing a balancing piston on one side of the shaft. In Fig. 5.12, a sample of such arrangement is shown. The balancing piston in connected to the shaft and is rotating with it inside a chamber. The distance between the surface of the cylinder and the stationary part is very small.

   On the one side of the piston, the flow exiting from the last stage is acting with a pressure equal to P_R. On the other side of the cylinder, the inlet pressure P_a is acting. The effective area of the piston, A, must be chosen such that $F_a = A(P_R - P_a)$.



(b) (a)

Fig. 5.10 Multi-stage pumps, arrangement [4]

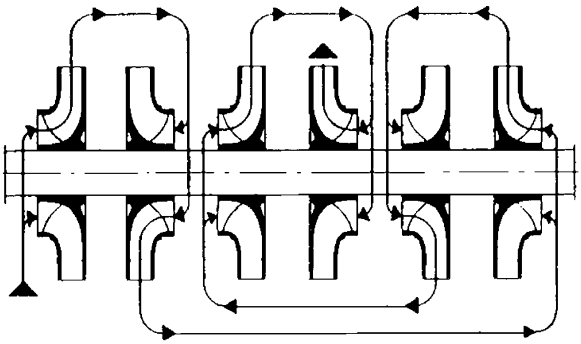


Fig. 5.11 Multi-stage pumps, arrangement [4]

Considering that there is a limit for the minimum distance between the other surface of the cylinder and the stationary part, there would always be some leaking fluid. In addition, the balancing cylinder is designed for a specific flow rate; therefore, a mechanical system must be always provided to balance the axial thrust at flow rates other than the design flow rate. For this reason, this method is not used very often.

5. In modern pumps, usually the balancing disks are used for balancing the axial thrust (Fig. 5.13). On one side of this disk, the pressure inside the chamber, i.e. P_C, is acting (this pressure is lower than the exit pressure of the liquid from the last stage of the pump) and on the other side of the disk, the suction pressure P_a is acting. If the axial thrust moves the rotating parts to the left side, the axial clearance S would decrease. This would cause the pressure in the chamber P_C to increase to a level close to P_R until the necessary balance to limit the movement of the impeller assembly is reached. If the impeller moves toward the right side, the gap would become larger and P_C decreases until the forces balance each other. In this way, by variation of P_C from its minimum, P_a (when S is completely open), to its maximum, P_R (when the gap is completely close), the axial thrust is balanced automatically.

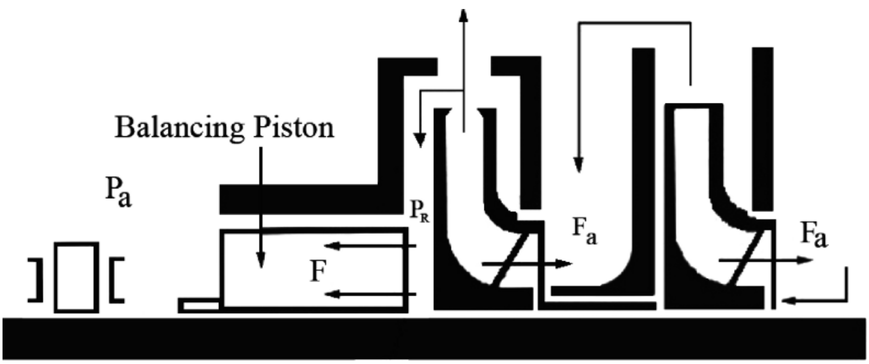


Fig. 5.12 Balancing piston [4]

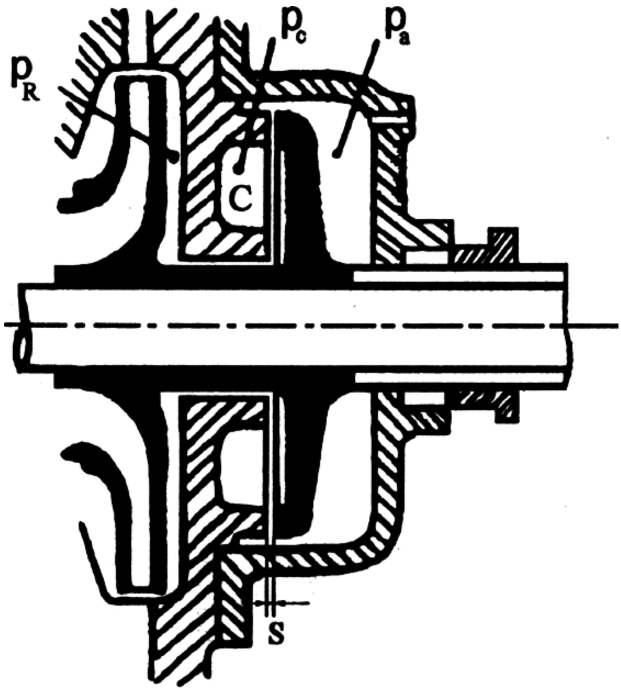


Fig. 5.13 Balancing disk [1]

5.3 Radial Thrust

One of the most important design aspects in the centrifugal pumps is to synchronize the design points of three important components of the pump, i.e. entrance, impeller, and the volute casing. If these three design points coincide, the fluid flow at the outlet of the impeller and around it would be symmetric. This ensures that the static pressure around the impeller is uniform. If these three design points do not match, the fluid flow around the impeller and inside the volute would not be uniform, and the flow velocity would change in different points. This would cause a non-uniform azimuthal static pressure distribution. This pressure difference would impose a radial force or thrust which acts on the pump shaft (Fig. 5.14). The magnitude and direction of this force would change by time and as the flow rate changes. The pump shaft has to be designed considering this radial thrust.

The non-uniform distribution of pressure would create flow recirculation and return flow at the volute nose area (Fig. 5.15), which is more prominent at lower flow rates, Fig. 5.15a, than higher flow rates, Fig. 5.15c. The effect of the return flow rate is more serious at lower flow rates, causing a non-uniform azimuthal velocity distribution not only at the outlet, but also at the inlet. The maximum radial thrust, then, is practically produced at zero flow rate.

The vibration caused by radial thrust fluctuation is very harmful, especially in double suction pumps, in which the distance between two bearings is relatively

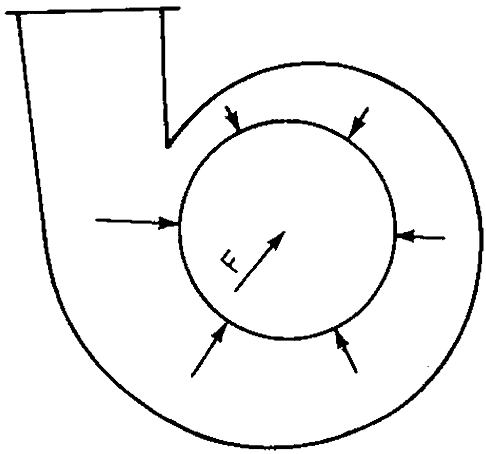


Fig. 5.14 Radial thrust on the impeller [2]

large. If a pump works for a long time at zero flow rate, the pump shaft may even break. The fracture usually happens at the connection point of impeller and the axis.

5.3.1 Variation of the Radial Thrust with Flow Rate

Experiments have confirmed that under off design conditions, the distribution of the static pressure around the impeller is non-uniform. Figure 5.16 shows the azimuthal distribution of the static pressure for three different flow rates: near zero, design, and maximum. It can be seen that at zero flow rate, the pressure change is quite high and the difference between the maximum pressure (at the throat of the volute) and the minimum pressure (at the volute nose) can be substantial.

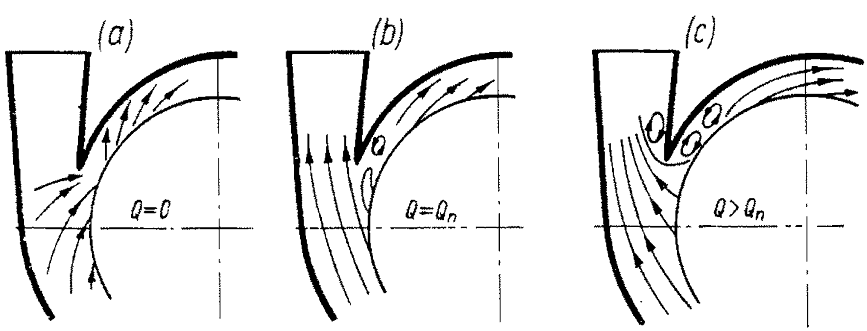


Fig. 5.15 Flow circulation in around volute nose as the flow rate changes [3]

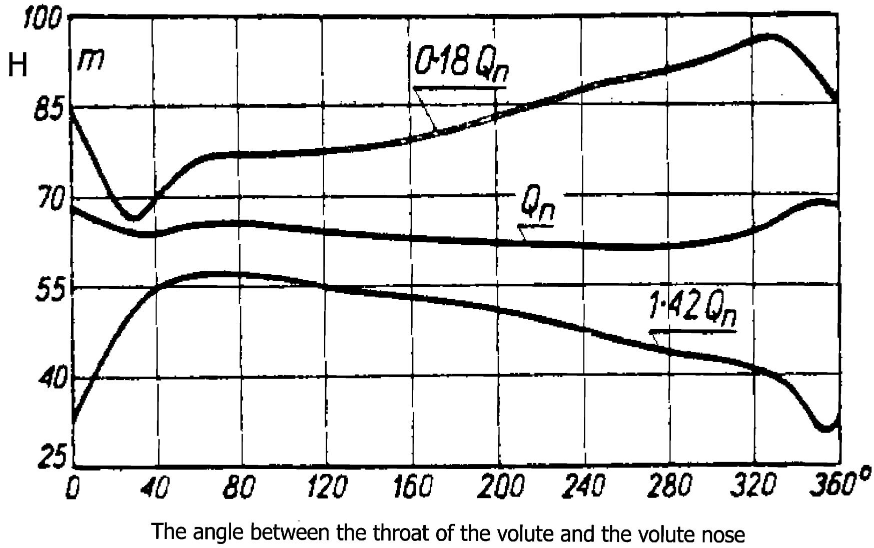


Fig. 5.16 Variation of the static pressure on the perimeter of the impeller [6]

5.3.2 Radial Thrust and the Shape of the Volute Casing

The design of the volute casing has a major effect on the radial thrust. In Fig. 5.17, four types of volute casings with different designs as well as the variation of the radial thrust versus flow rate for each of them is shown. In design (a), single volute casing, the minimum radial thrust occurs at the design point, whereas in design (c),

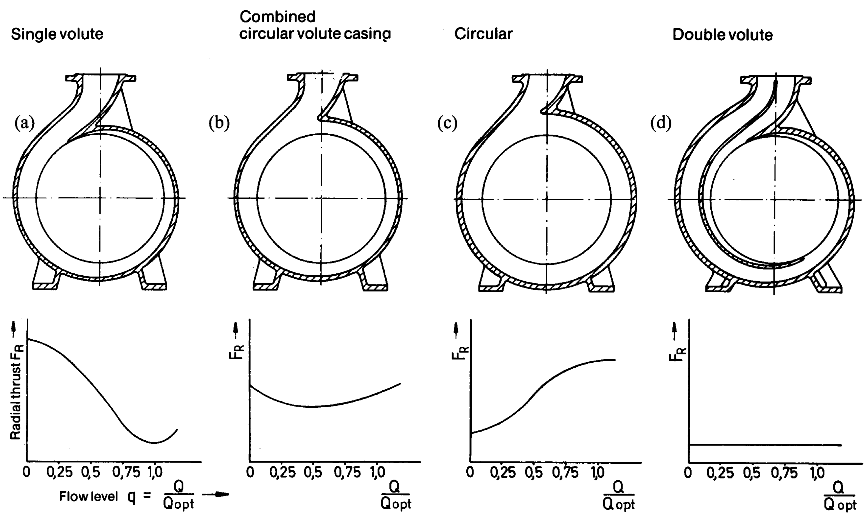


Fig. 5.17 The shape of the volute casing and radial thrust [6]

circular design, the radial thrust is maximum at design point. The more uniform radial thrusts can be obtained with designs (b) and (d), where the double wall volute casing design, (d), has the best performance in terms of producing a uniform radial thrust. However, this design is also more complicated for casting.

Experiments performed for pumps with different specific speeds have shown that the double wall volute casing can produce the minimum radial thrust, see Fig. 5.17. Nevertheless, in the design points, most volute casings have similar performances.

5.3.3 Radial Thrust and Specific Speed

Figures 5.18–5.20 show variations of the radial thrust with specific speed for different volute types and flow rates. The flow rate changes from zero to maximum. At specific speeds from 1165 to 3500 (English Unit System), the radial thrust reaches its maximum at flow rate equal to zero for standard volute.

5.3.4 Radial Thrust Calculation

The magnitude of radial thrust depends on the shape of the volute casing as well as the specific speed and will change as the flow rate changes. Different empirical relations have been proposed for calculating the radial force. In this book, only the relation for a simple volute casing will be presented.

Stepanoff [5] has proposed an empirical relation for calculating the magnitude of the radial thrust. However, the direction of the radial thrust cannot be found from his relation. According to Stepanoff, the radial thrust is equal to

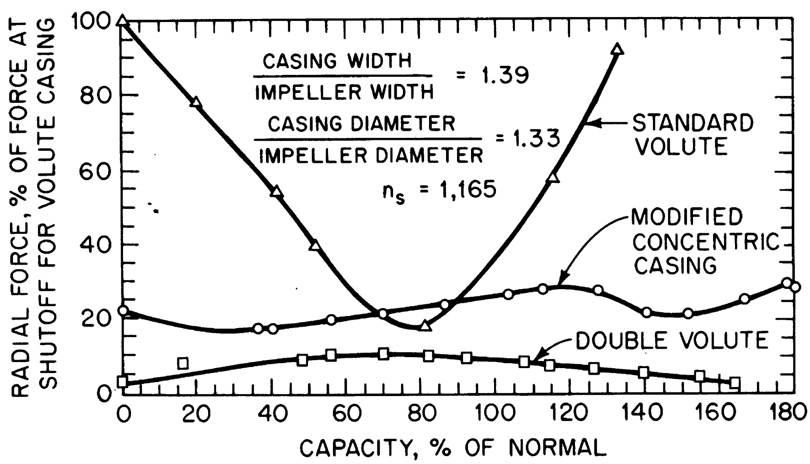


Fig. 5.18 Radial thrust in lower specific speed (English Units) [2]

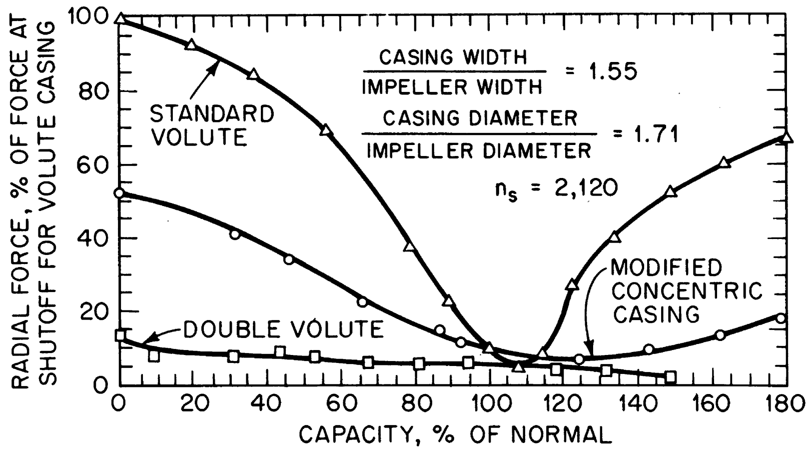


Fig. 5.19 Radial thrust in medium specific speed (English Unit) [2]

$$F_R = K_R \rho g D_2 b' H \tag{5.1}$$

where F_R is the radial thrust, ρg is the specific weight of the liquid, D_2 is the outside diameter of the impeller, b' is the width of the impeller including the thickness of the walls, and K_R is the empirical radial thrust coefficient which is a function of the flow rate. The value of the radial thrust coefficient can be obtained from Fig. 5.21. (Note that in this figure, the specific speed is in English Unit System.)

Determining the radial thrust direction, however, needs more work. The magnitude and direction of radial thrust for different specific speeds are shown in Fig. 5.22 for a simple volute casing. Each curve represents a specific speed. The numbers on

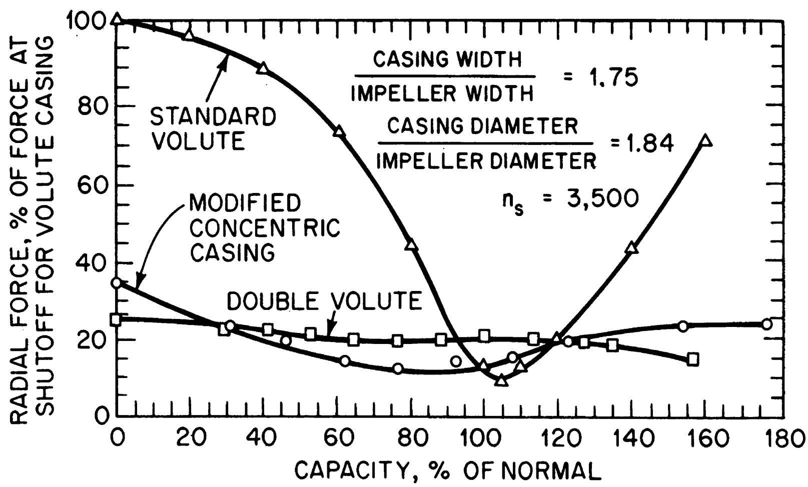


Fig. 5.20 Radial thrust in high specific speed (English Units) [2]

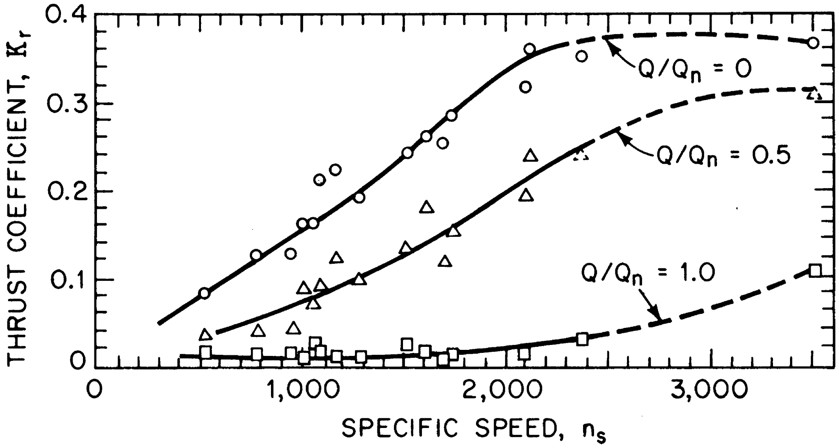


Fig. 5.21 Radial thrust coefficient and specific speed for simple volute casing [2]

the curves show the ratio of the flow rate to the design flow rate for the pump. By knowing the specific speed and flow rate, the working point on the corresponding curve is specified. From this point, a line can be connected to the point of origin on the curve (Figs. 5.22 and 5.23). The length of the line represents the magnitude of the radial thrust, whereas its angle shows the direction of the force (see the insert in the Fig. 5.22). It is recommended that the magnitude of the thrust is calculated

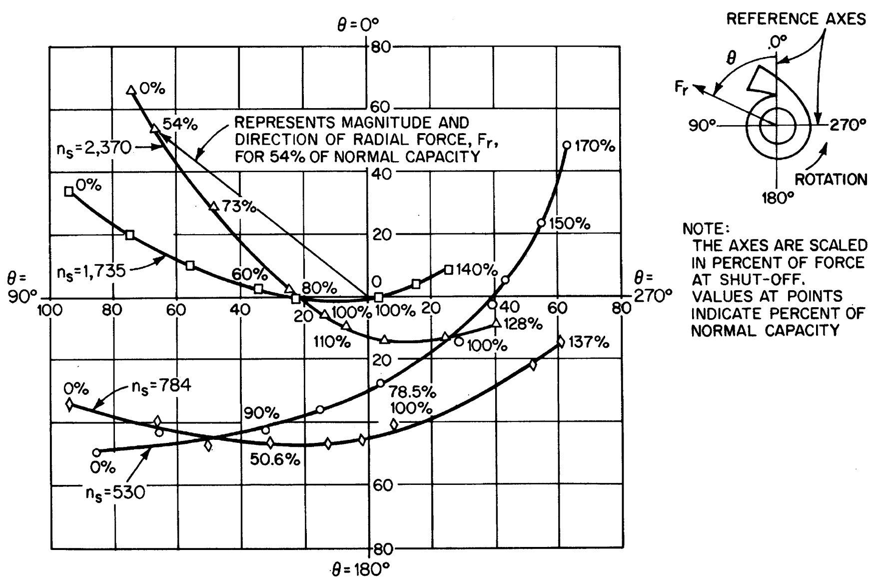


Fig. 5.22 Magnitude and direction of the radial thrust in a simple volute casing [2]

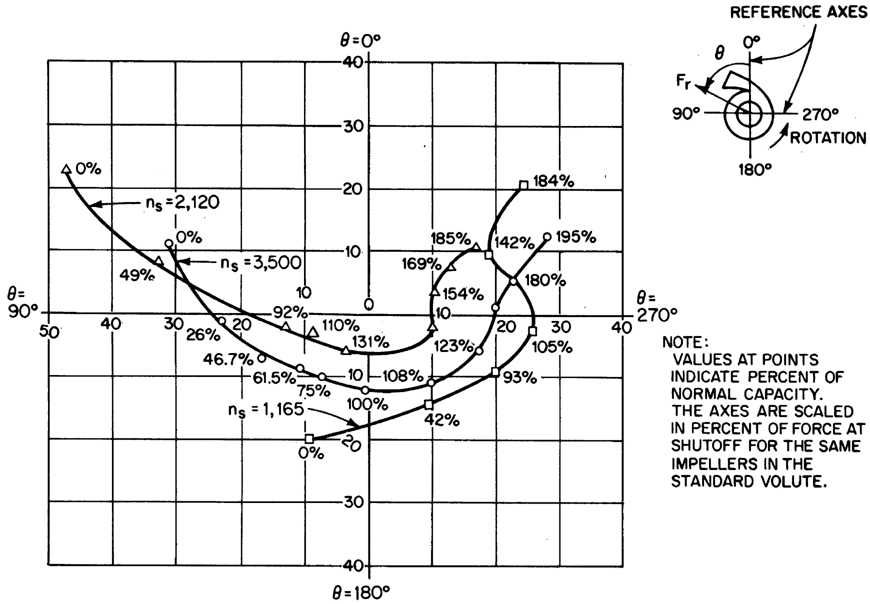


Fig. 5.23 Magnitude and direction of the radial thrust in symmetrical volute [2]

from (5.1). Figure 5.23 shows the same type of diagram for the symmetrical volute casing (the casing has the same center as the impeller at 270°).

It should be noted that the radial thrust has two components. A constant component that can be calculated from (5.1). The magnitude and direction of this component remain constant if the flow rate does not change. Another component of the radial thrust is a variable component, which is also called the rotating radial thrust. The variable component of thrust can have different sources and depending on its direction may be added to the constant part or be subtracted from it.

References

1. Jaumotte A. L., “Description et Construction des Machines, Turbomachines, Turbopumpes Centrifuges”, Presses Universitaires de Bruxelles, 1984–1985.
2. Karassik Igor J. et al., “Pump Handbook”; McGraw-Hill, 2001.
3. Lazarkiewicz Stephen, “Impeller Pumps”, Pergamon Press, 1965.
4. “Centrifugal Pump”, Lexicon, KSB, Frankenthal, Germany, 1980.
5. Stepanoff A. J., “Centrifugal and Axial Flow Pumps”, John Wiley & Sons, 1957.
6. “Centrifugal Pump Design”, Technical Appendix, KSB, Frankenthal, Germany.

Part II
Pumping Systems

In Part I of this book, the major elements of a pump and the important hydraulic parameters in the rotor and stationary parts of a pump were discussed in detail. Turbopumps are one of the mechanical machineries in which their performances are closely dependent on the system they are operating in. For this reason, it is essential that pump(s) selections for a system be performed very carefully considering the hydraulic parameters of the system it is intended to be installed in.

Usually, the pump selection procedure is a trial-and-error method in which at the beginning the hydraulic characteristics of the piping system are determined using the required flow rate and pressure. Then, based on this information, the most suitable pumps, the number of pumps required, and the method of connecting the pumps are decided. Later, the performance of the pump(s) in the system is analyzed and necessary modifications in the piping system, pump types and size, and the number of pumps will be carried out. This procedure may lead to the partial redesigning of the system.

At some point the hydraulic design of the pump station will be determined which may affect the complete pump selection procedure. As will be discussed later, in all this process the economical, geographical, and human interaction factors with the system must be considered and be involved in the design procedure.

In the next chapters, these steps are discussed in more details and the information required by engineers to design the piping system and to make the overall decision about the number, type, and size of pumps and the pumping station will be presented.

Chapter 6
Operational Properties of Characteristic Curves

In Part I, Sect. 2.2, general information about characteristic curves of turbopumps has been presented. In this chapter, more detailed information about the shape of these curves and their dependencies on the pump types and the specific speed will be discussed. Also, in this chapter, the effect of liquid properties, specifically its viscosity, on the characteristic curves of a turbopump will be presented.

6.1 H–Q Characteristic Curves

The shapes of H–Q characteristic curves for centrifugal, mixed, and axial flow pumps are different. In Fig. 6.1, three types of H–Q characteristic curves for these pumps are shown.

In general, in centrifugal pumps the head characteristic curve has a parabolic shape with maximum point usually on or very close to the vertical axis (head coordinate) (Fig. 6.1, top). In these pumps the slope of the curve depends on the angle of blades. With decreasing outlet blade angle, the slope of the curve increases and as a result any changes in the flow rate would cause a sharper change in the pump head.

One of the important issues in utilizing the turbopumps is their performances at different flow rates. When using turbopumps it is recommended that the working region be always situated on that part of the H–Q curve which has a negative slope. In the region of $\frac{dH}{dQ} > 0$, the total head increases with increasing flow rate, making the operation unstable. In the region of $\frac{dH}{dQ} < 0$, the total head decreases with increasing flow rate which is a type of protection for the pump. For this reason, pumps that have characteristic curves with the maximum point on the H coordinate are more preferable. These pumps would work with no problem at all possible working conditions [2].

In axial flow pumps, the maximum head is not usually located on the H axis and therefore only that part of the $\frac{dH}{dQ} < 0$ curve should be used in the selection process. In mixed flow pumps, depending on the specific speed, the characteristic curves are similar to those of the centrifugal pumps at higher flow rates. At lower flow rates, the curves are more similar to the characteristic curves of the axial flow pumps.

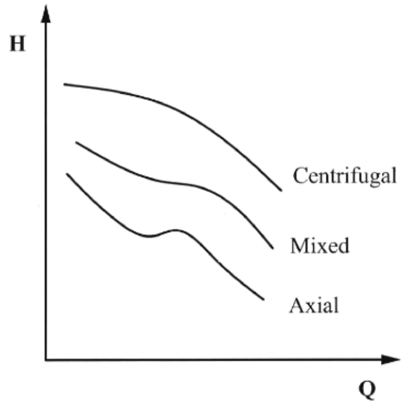


Fig. 6.1 Three types of $H-Q$ characteristic curves for centrifugal, mixed, and axial flow pumps

6.2 $P-Q$ Characteristic Curves

The shape of the characteristic curves for power also changes with the specific speed. In pumps with low specific speeds (centrifugal pumps), the slope of the curve increases by increasing the flow rate. In some centrifugal pumps, this curve even has a maximum point. In pumps with high specific speed (axial flow pumps), the $P-Q$ curve has a negative slope and in mixed flow pumps, depending on the specific speed of the pump, the shape of the curve can be a combination of these two slopes (Fig. 6.2).

As was mentioned in Part I, one of the most important issues in selecting turbopumps for a specific application is to select the pump driver and the required electrical or mechanical power based on the characteristic curves of the turbopumps. Although the pump selection is usually done based on its performance at the maximum efficiency point, the required power to drive the electromotor cannot

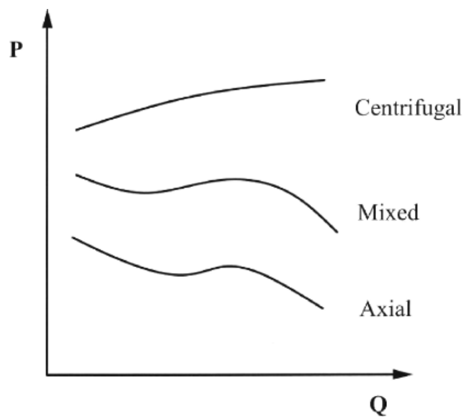


Fig. 6.2 The power characteristic curve for centrifugal, mixed, and axial flow pumps

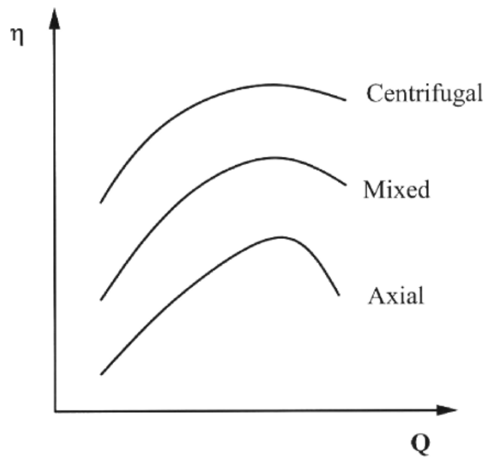


Fig. 6.3 The efficiency characteristic curves, η, for centrifugal, mixed, and axial flow pumps

be determined with the same criteria. The reason is obvious because under conditions in which the flow rate increases in the system, the required power for the pump would increase beyond the nominal power and this may damage the electrical wirings inside the motor. For this reason, depending on the working conditions of the pump and the possibility of flow rate increase, the motor power can be selected based on the maximum point of the P–Q characteristic curve.

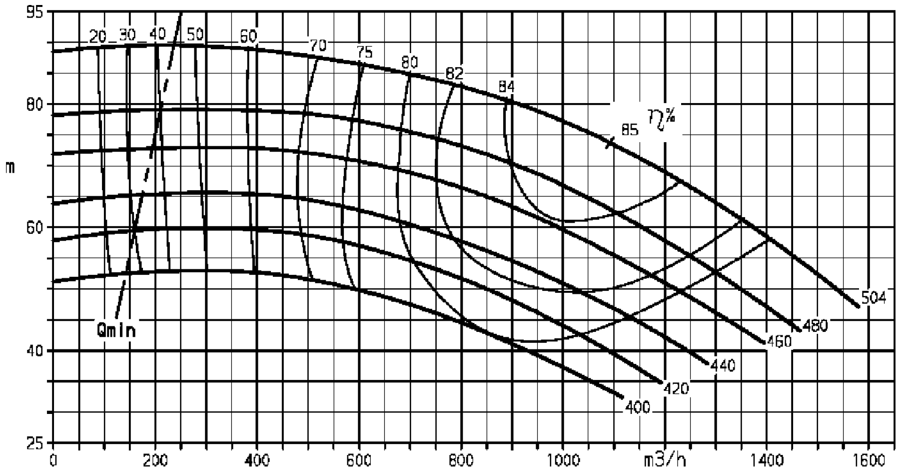


Fig. 6.4 The iso-efficiency curves for a centrifugal pump [KSB product catalogue (1312.4050/3-90 G3)]

6.3 η–Q Characteristic Curve

The characteristic curve for the pump efficiency has usually a parabolic shape and intersects with the Q coordinate at two points, $Q = 0$ and $Q = Q_{\max}$. In between these two points, there is a maximum efficiency point which represents the optimum pump performance condition. It is recommended that the working condition of the pump is selected based on this point. Figure (6.3) shows three types of η–Q curves for three different pump types.

It is a common practice of pump manufacturers to show the (iso-efficiency) lines within the H–Q curves. Samples of such curves are shown in Fig. 6.4.

6.4 Cavitation Characteristic Curve NPSH_{req}–Q

The NPSH_{req} curve in centrifugal pumps usually has a monotonically positive slope. Therefore, with increasing flow rate NPSH_{req} increases as well. This indicates that the probability of cavitation occurring increases as the flow rate increases. In axial flow pumps, however, the NPSH_{req} characteristic curve has a minimum which indicates that the risk of cavitation exists both at low and at high flow rates.

In mixed flow pumps at lower specific speeds, the characteristic curves exhibits the same behavior as the centrifugal pumps whereas at higher specific speeds it is more like the axial flow pumps. In Fig. 6.5 the $\mathrm{NPSH}_{\mathrm{req}}$ characteristic curves for these three types of pumps are shown.

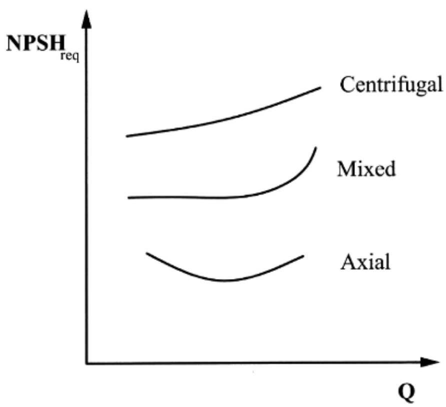


Fig. 6.5 $\mathrm{NPSH}_{\mathrm{req}}$ characteristic curves for centrifugal, mixed, and axial flow pumps

6.5 Special Points on the Characteristic Curves

There are several special points on the characteristic curves of turbopumps that are important and must be discussed in more detail (Fig. 6.6). These points are as follows:

1. The maximum efficiency point (B) on the η–Q curve: the selection of a pump to work under specific conditions is made based on this point, Point B in Fig. 6.6.
2. The maximum head point of the H–Q curve: this point represents the limit of pump operation on the negative slope region of the curve ($\frac{dH}{dQ} < 0$).
3. The maximum flow rate point on the H–Q curve which limits the pump operation with regard to cavitation, point A in Fig. 6.6. In axial flow pumps, the minimum flow rate is also important since it corresponds to the maximum required power.

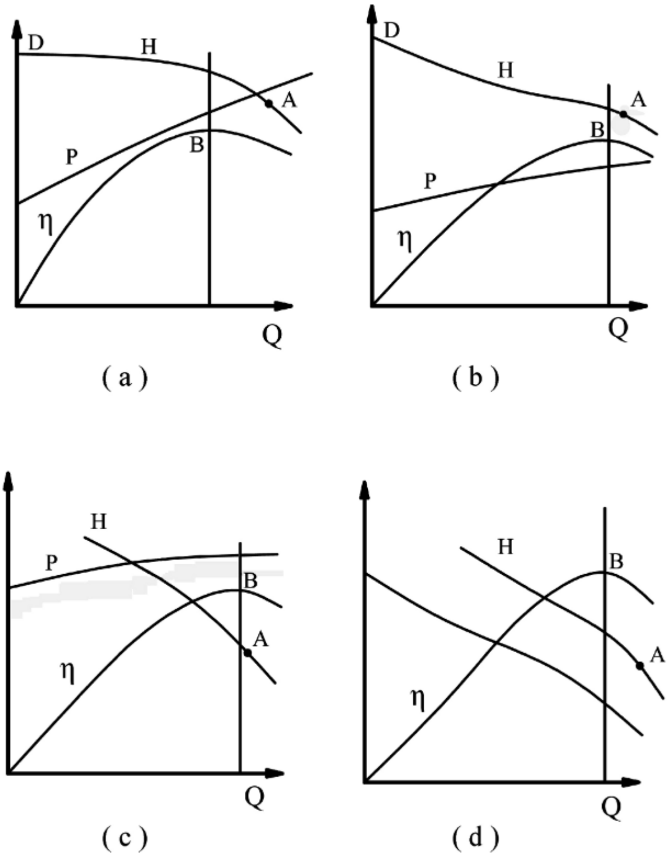


Fig. 6.6 Specific points on the characteristic curves. (**a**) centrifugal pump; (**b**) and (**c**) mixed flow pumps; (**d**) axial flow pump

4. The zero flow rate point: the start up torque and required power to start the centrifugal pump are calculated at this point, point D in Fig. 6.6.
5. The maximum power point on the P–Q curve at which the power of pump driver is calculated.

6.6 Characteristic Curves and Specific Speed

The shapes of characteristic curves of pumps are very dependent on the specific speed as shown in Fig. 6.7. For this reason, by looking at the characteristic curves of a pump one can identify the pump type. For example, in centrifugal pumps (low

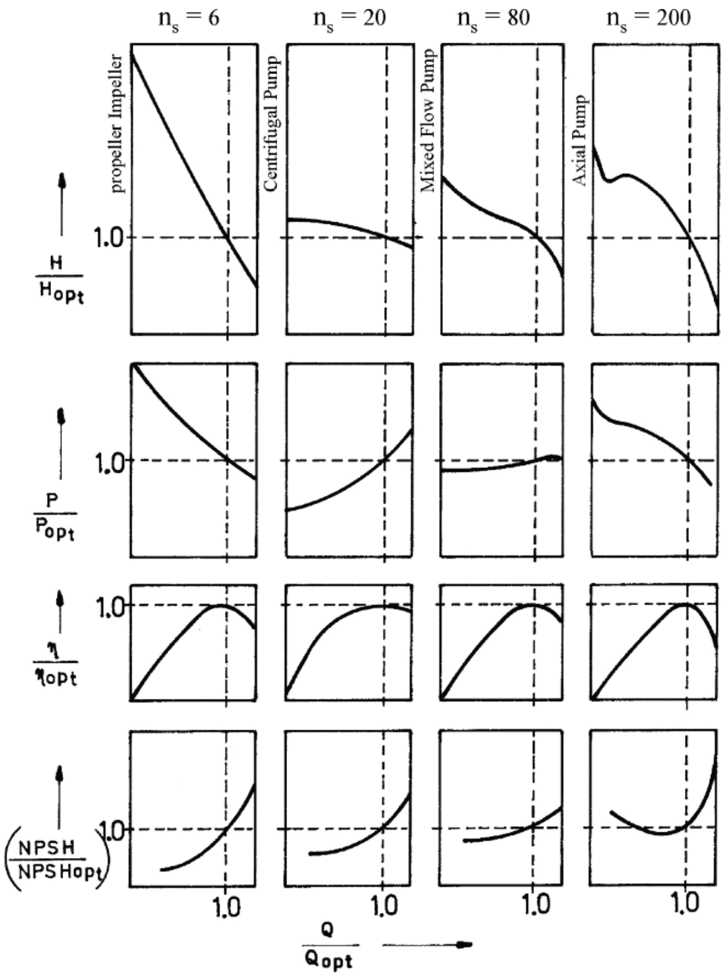


Fig. 6.7 Variation of the characteristic curves with specific speed (in dimensionless system) [1]

specific speed), the $H-Q$ curve has a parabolic shape (with maximum point on the H axis) while both power and $NPSH$ curves have positive slopes. As can be seen in Fig. 6.7, as the specific speed, N_s, increases, the shapes of the characteristic curves gradually change.

6.7 Characteristic Curves and Pump's Rotational Speed

By looking at the Euler equations (Sect. 2.5), one can see that the input power and the manometric head of a pump are dependent on the peripheral speed of the impeller. For this reason, one method to change the characteristic curves of a pump is to change the peripheral speed by changing either the rotational speed or the impeller diameter.

Usually, when it is possible to change the rotational speed of the driving motor, the pump manufacturers present the characteristic curves at different rotational speeds on a same system of coordinates. To do so, the $H-Q$ curves at constant rotational speeds n_1, $n_2, \ldots$ are plotted and the constant efficiency curves are obtained and presented on the same plot. Figure 6.8 shows a sample of such $H-Q$ and iso-efficiency curves for a centrifugal pump. The characteristic curves for power and NPSH_{req} are also plotted separately for different rotational speeds.

When it is not possible to change the rotational speed of a pump during the operation, the pump manufacturers present the characteristic curves at few specific rotational velocity which are factors of the voltage frequencies. The maximum rotational speed for 50 Hz frequency is usually 3000 rpm.

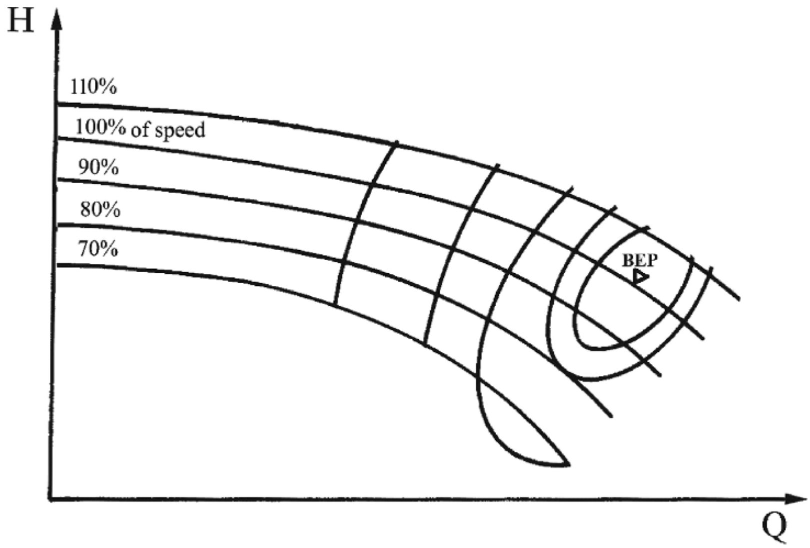


Fig. 6.8 A sample of characteristic curves for a centrifugal pump at different rotational speeds

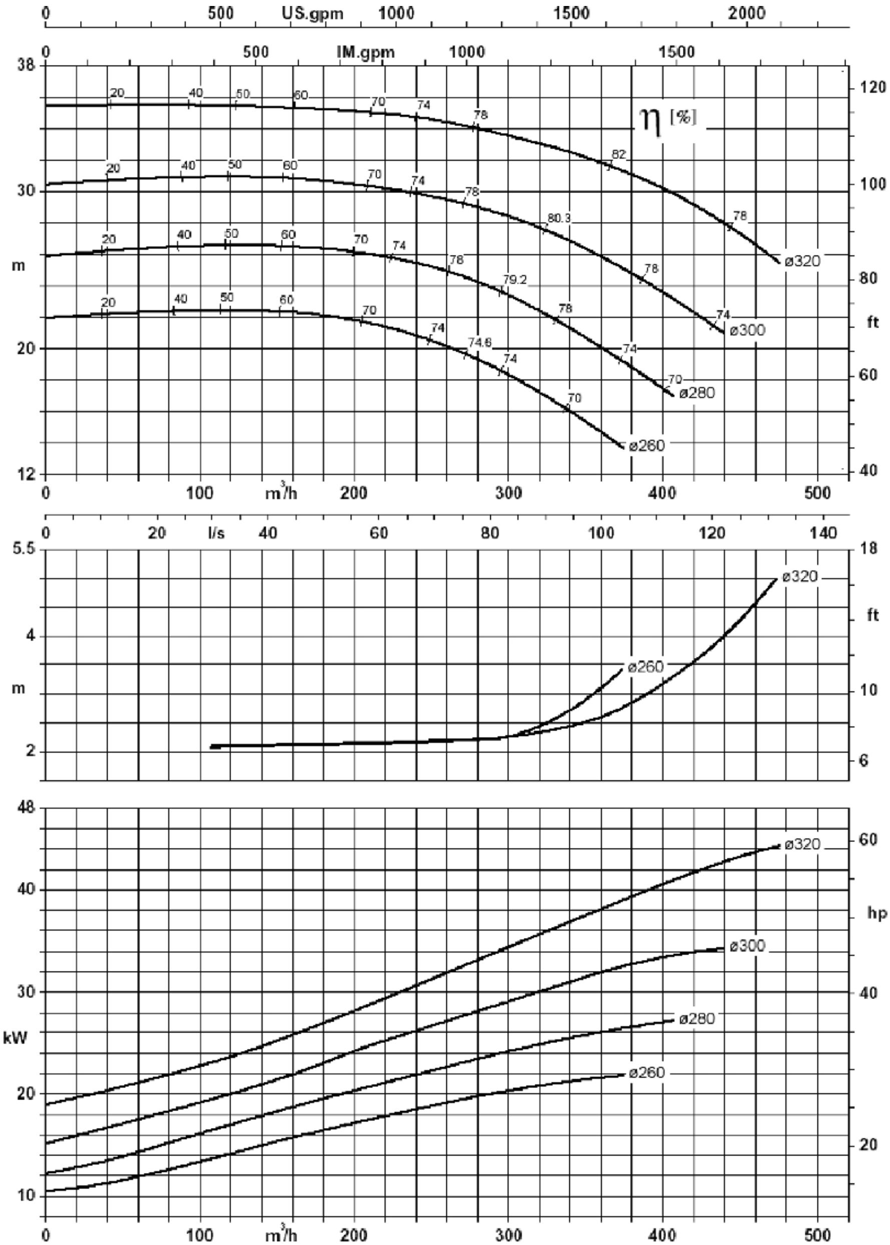


Fig. 6.9 Sample of characteristic curves for a pump with different impeller diameter [KSB product catalogue (2721.450/11-90 G2)]

6.8 Characteristic Curves and Impeller Diameter

In many cases, it is not possible to change the rotational speed of a pump. Since the change in the impeller diameter has the same effect the rotational speed has on the characteristic curves of the pump, the H–Q and P–Q curves can be presented in a constant speed but at different impeller diameters. These curves are also plotted on the same system of references (Fig. 6.9). The iso-efficiency curves are also obtained in the same method as before.

The change in the impeller diameter is done by trimming the impeller or the blades. Usually, for pumps with relatively low specific speeds, the impeller diameter can be reduced by 15% to 20%. However, for mixed flow pumps, this change is limited to 3% or to a maximum of 4%. As mentioned in Chap. 3, the new flow rate and head can be obtained from the similarity laws for the new impeller diameter. In Fig. 6.9, samples of characteristic curves for different impeller diameters are shown.

It should be mentioned that only the impeller diameter is changed in this method and all other parts of the pump are kept at the same size. Different impellers with different diameters are installed within the same casing.

6.9 Characteristic Curves and Liquid Viscosity

Depending on their specific designs, turbopumps are usually capable to transfer liquids with viscosities lower than 520–760 cS. The viscosity can be increased to 1000 cS by using specific impellers. However, for a pump to be economically efficient, the maximum recommended liquid viscosity is 150 cS (water has a viscosity of 1 cS at 20°C).

The characteristic curves of a turbopump would change if the viscosity of the pumped liquid is different from water (all characteristic curves presented by pump manufacturers are obtained for water). By increasing the liquid viscosity, the pump head and its efficiency decrease and the absorbed power increases (Fig. 6.10). In the H–Q curve, the head at starting point (flow rate equals to zero) remains the same.

When a pump is being selected or used for pumping liquids with viscosities higher than the viscosity of water two major steps must be taken:

1. Selecting the proper pump to handle the required flow rate and head for the viscous liquid.
2. Obtaining the new characteristic curves for the pump that is used to pump the viscous liquid.

For both purposes, one can use the conversion factors, such as those presented in Fig. 6.11, to obtain the corresponding flow rates and heads for water or for a highly viscous liquid.

The following two examples show how to use these conversion factors.

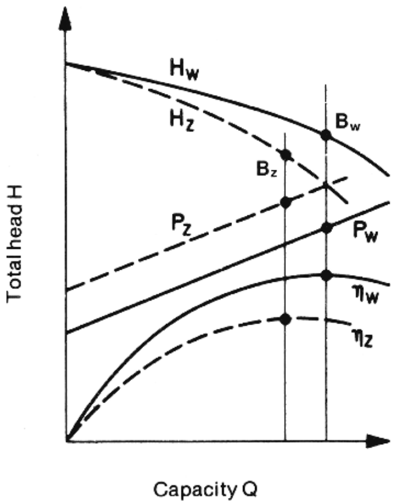


Fig. 6.10 The change in characteristic curves with liquid viscosity. W: water; Z: higher viscous liquid [2]

Example 1

The required flow rate and head for pumping a liquid with viscosity of 206 cS and a density of 0.9 are 170 m^3 and 30 m, respectively. After selecting the pump, calculate the required power and pump efficiency [2].

First, find the flow rate in Fig. 6.11 and draw a line to intersect with corresponding H curve in the bottom part of the Figure. From the intersection point draw a horizontal line to the corresponding viscosity line. From this point a vertical line is drawn toward the top part of the figure until it passes through the correction curves for flow rate, efficiency, and head. The corresponding coefficients for this example, then, are:

$$K_Q = 0.94,\ K_\eta = 0.64,\ K_H = 0.92 \ \ (\text{at } Q = Q_{opt})$$

Using these coefficients, the equivalent flow rate and head for water are obtained as follows:

$$Q_W = \frac{170}{0.94} = 181\, m^3/h$$

$$H_W = \frac{30}{0.92} = 32.6\, m$$

Now using these values, one can select the appropriate pump. If the selected pump has efficiency equal to 0.8 at the working point, its efficiency when pumping the viscous fluid would be

$$\eta_v = 0.8 \times 0.64 = 0.51$$

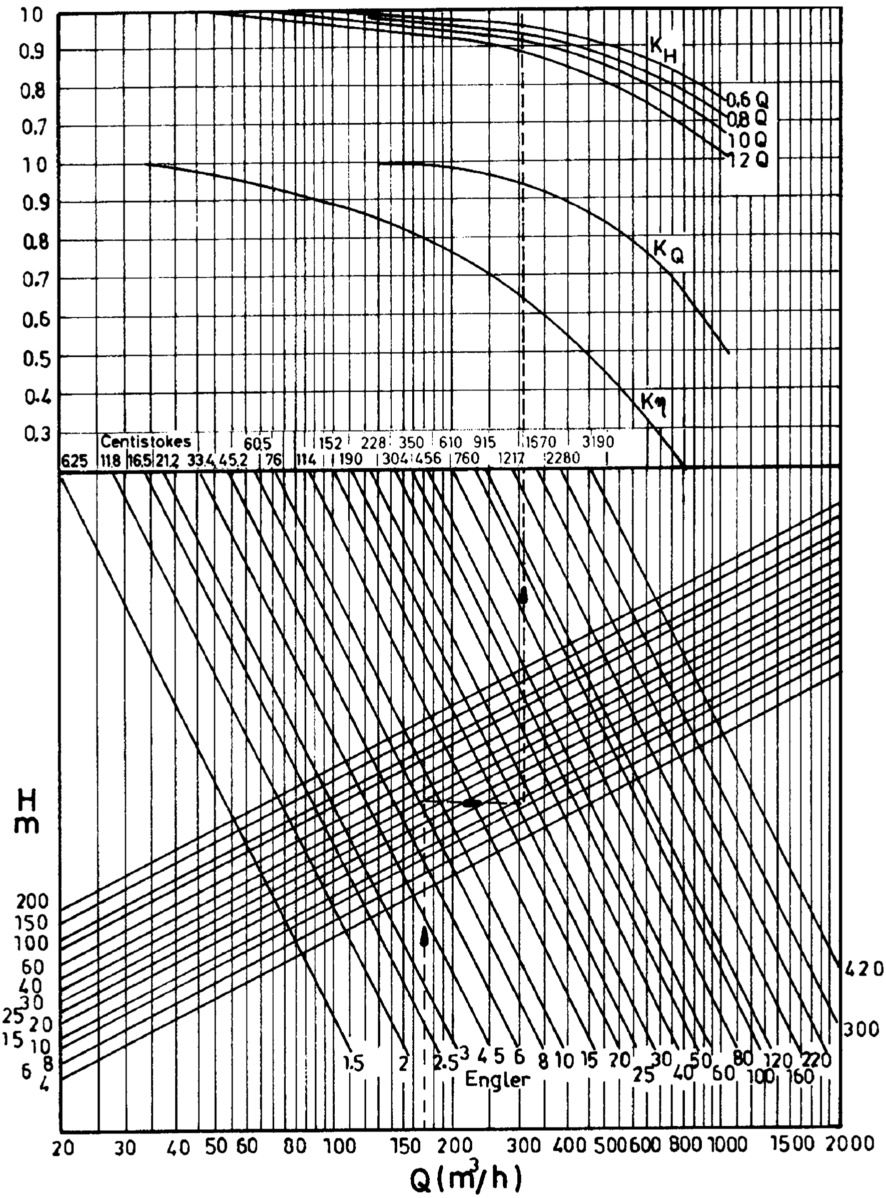


Fig. 6.11 The conversion curve for determining the centrifugal pump characteristic curves used for pumping the viscous liquids. K_H, K_Q, and K_η are the head, flow rate, and efficiency coefficients. The application of these coefficients is explained in Example 1 [3, from Hydraulic Institute-New York-1955]

The absorbed power by the pump can then be calculated from

$$P_{consumed} = \frac{QH\gamma}{367 \times \eta} = \frac{170 \times 30 \times 0.9}{367 \times 0.51} = 24.52\,kW$$

Example 2

The characteristic curves for a pump are available (Fig. 6.12). Modify these curves for a liquid with viscosity of 206 cS and a density of 0.9.

Since the correction coefficients of Fig. 6.12 correspond to the maximum efficiency point, first the flow rate and head at the maximum efficiency of this pump must be found. From Fig. 6.12, one obtains

$$Q_W = 170 \ \ \mathrm{m^3/h}$$

$$H_W = 30\mathrm{m} \quad \eta = 82\%$$

Once these values are obtained, the corresponding coefficients for head, K_H, flow rate, K_Q, and efficiency, K_η, can be obtained from Fig. 6.11. Then the equivalent heads and efficiencies for other flow rates like $0.6Q$, $0.8Q$, and $1.2Q$ can be obtained from Fig. 6.12. These numbers, then, are multiplied by the corresponding coefficients. The results are summarized in Table (6.1) and plotted in broken lines in Fig. 6.12 [3].

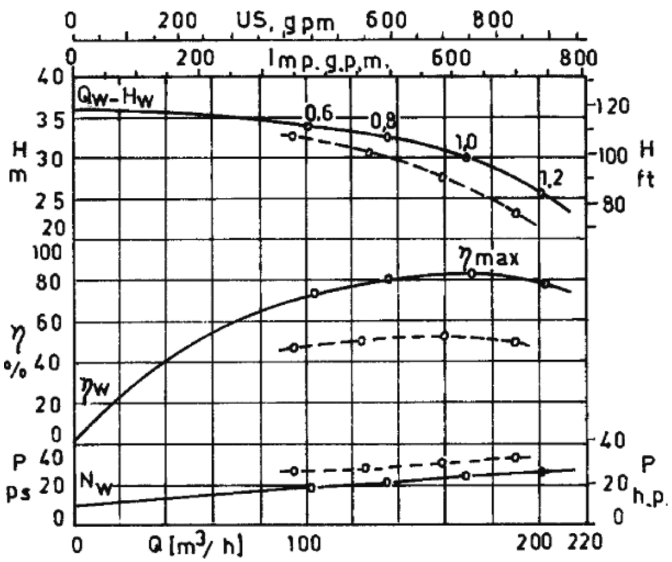


Fig. 6.12 Correction of the characteristic curves for a viscous liquid [3]

Table 6.1 The transformation table for the viscous liquid [3]

	Q/Q_{op}		0.6	0.8	1	1.2
Water	Q_W^*	m^3/h	102	136	170	204
	H_W	m	34	32.5	30	26
	η_W	%	72.5	80	82	79
		K_Q		0.94		
	Correction factors	K_η		0.635		
		K_H	0.96	0.94	0.92	0.89
Viscous liquid	$Q_\nu{}^* = Q_W \times K_Q$		96	128	160	192
	$\eta_\nu = \eta_W \times K_\eta$		46	50.8	52	50
	$H_\nu = H_W \times K_H$		32.6	30.5	27.6	23.1
	$P_\nu = \dfrac{\rho_\nu\, g\, H_\nu\, Q_\nu}{367\,\eta_\nu}$		16.9	18.97	20.95	21.91

* Subscripts W and ν correspond to water and viscous liquid, respectively.

References

1. "Basic Principles for the design of Centrifugal Pump Installations", SIHI Group, 1980.
2. "Centrifugal Pump Design", Technical Appendix, KSB, Frankenthal, Germany.
3. "Pumps Hand Book", KSB, Frankenthal, Germany, 1968.

Chapter 7
Pipeline Calculations

7.1 Pipeline Calculations

The performance of a pump in a piping system is determined by the pipeline characteristics. A single pump will deliver different flow rates and heads when installed in two different pipelines. Therefore, the characteristic behavior of a piping system on which the pump(s) are installed must be determined before any pump selection is started. In the next section, the methods for calculating the pressure losses in the straight pipes as well as in different common fittings installed in a piping system will be presented.

7.1.1 Bernoulli's Equation

Bernoulli's equation which basically describes the energy conservation law for inviscid flows in a piping system is used to obtain the characteristic curve of a system. According to this law,

> if in a non-compressible flow, the pressure losses in the system are neglected, no external energy is given to the liquid, and liquid does not transfer energy to an external source, the total energy of the system between two points of the flow on one streamline remains constant.

Therefore, between any two points in the flow, e.g. points 1 and 2, one can write

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2 = cte \tag{7.1}$$

However, for a system of pipelines, there are always energy losses in the pipes and fittings. Thus, Bernoulli's equation for two different points of the pipeline can be written as

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2 + H_{L12} \tag{7.2}$$

where H_{L12} is the pressure loss between points 1 and 2 with a unit of *meter* in SI unit system.

7.1.2 Pressure Loss

The pressure losses occur due to the friction of the liquid particles against each other (shear forces) and the pipe walls. Due to this friction, there would be a significant energy loss that could have been transferred to the useful work. In other words, there would be always a static pressure loss, in the direction of flow that can be measured with a pressure gauge.

In general, the pressure losses appear in two forms in a piping system:

1. Pressures losses in straight pipes, which are also called linear pressure losses.
2. Pressure losses in valves, bends, and other pipe fittings, which are called local pressure losses.

As will be presented later, if the pipe diameter and the flow rate remain constant in a straight piping system, the pressure loss in that straight pipe will change linearly with increasing pipe length, i.e. the slope of the pressure loss remains constant.

The pressure losses in the valves and other fittings in the pipeline are generated from two sources:

- The pressure losses due to the liquid friction at the walls of the fittings. When the fittings also change the direction of the fluid flow, like bends or some types of valves, the flow would be disturbed and more pressure loss is generated.
- The pressure losses that are generated in the straight parts of the pipe due to the existence of a fitting or change in the pipe diameter. Because of the fittings, the fluid flow is disturbed and that would increase the pressure loss in the straight part of the pipe.

7.2 Pressure Loss Calculation in Straight Pipes

7.2.1 Darcy-Weisbach Formula

The Darcy-Weisbach relation is one of the basic equations for calculating the pressure loss and can be written as

$$H_L = f\frac{L}{D} \times \frac{V^2}{2g} \tag{7.3}$$

In this relation, H_L is the pressure loss, L is the length of the pipe, D is the internal diameter of the pipe (all in meters), V is the average velocity of the flow

(Flow rate/Pipe Cross Section) in m/s, f is the friction coefficient, and g is the gravitational acceleration in m/s^2. The above equation can be written as

$$\Delta P = f\frac{L}{D} \times \rho\frac{V^2}{2} \tag{7.4}$$

where ΔP is the pressure loss, and ρ is the density of the liquid. The Darcy equation can be used for any flow regime, laminar or turbulent flow. Equation (7.4) can be used for calculating the pressure loss in the pipes with constant diameter and liquids with constant velocity and density. For cases in which the liquid conditions are different in different sections of the pipe, the pressure loss can be calculated for each section, using the diameter and flow conditions for that section of the pipe. These losses can then be added together to obtain the total pressure loss.

7.2.2 Reynolds Number

The Reynolds number is the ratio of inertial forces ($V\rho$) to the viscous forces (μ/L) in a flow and is used for determining whether a flow will be laminar or turbulent. It is named after Osborne Reynolds (1842–1912), who in 1883 after studying different fluid flow regimes presented two distinctive types of flows.

1. *Laminar flow*: in this flow regime, the liquid particles move along parallel lines and do not get mixed.
2. *Turbulent flow*: in this flow regime, the flow lines are not parallel and the liquid particles get mixed as they move.

To determine whether a flow is laminar or turbulent in a pipeline, one should calculate the Reynolds number as defined below:

$$Re = \frac{\rho V D}{\mu} \tag{7.5}$$

where V is the flow velocity in m/s, D is the pipe diameter in meter, and μ is the dynamic viscosity of the liquid in *kg/ms*. The Reynolds number Re is non-dimensional.

For laminar flows $Re \leq 2000$ and for turbulent flows $Re > 4000$. It is possible that with higher Reynolds numbers, more than 4000, the flow remains laminar. But these conditions apply only in the labs and in most industrial applications the two above limits really exist. The flow regime corresponding to Reynolds number between 2000 and 4000 is called the transitional flow.

7.2.3 Friction Factor

The friction factor for different flow regimes is different:

a. For laminar flows, Re $\leq$ 2000, the friction factor is only a function of the Reynolds number and can be obtained from

$$f = \frac{64}{\mathrm{Re}} = \frac{64\nu}{DV} \tag{7.6}$$

By using this value in the pressure drop (7.4), one can see that

$$\Delta P = 32\frac{LV\mu}{D^2} \qquad or \qquad H_L = 32\frac{\nu LV}{g\,D^2} \tag{7.7}$$

b. For turbulent flows, Re $>$ 4000, the friction factor is a function of the Reynolds number, and also depends on the relative roughness of the pipe walls, i.e. $\frac{\varepsilon}{D}$. ε is the roughness of the walls and D is the internal diameter of the pipe (in some references symbol K is also used to specify the wall roughness). One of the most commonly used theoretical relation for calculating the friction factor in turbulent flows is Colebrook-Prandtl relation. This relation which also has a good agreement with experiments is presented as

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{2.51}{\mathrm{Re}\sqrt{f}} + \frac{\varepsilon}{D} \times \frac{1}{3.71}\right) \tag{7.8}$$

In those cases where the Reynolds number is much higher compared to the relative roughness, the first term inside the bracket can be ignored. This means for very rough pipes, the friction factor is only dependent on the relative roughness (in other words, the wall roughness would be larger than the boundary layer which affects the fluid flow). For these cases the friction factor can be obtained from

$$\frac{1}{\sqrt{f}} = 1.14 - 2\log\frac{\varepsilon}{D} \tag{7.9}$$

In those pipes that are hydraulically smooth ($\varepsilon \approx 0$), such as glass or brass pipes, the effect of the wall roughness is much smaller than the Reynolds number; therefore, the friction factor can be obtained from (Darcy equation)

$$\frac{1}{\sqrt{f}} = 2\log\left(\frac{\mathrm{Re}\sqrt{f}}{2.51}\right) \quad or \quad f = 0.36\,\mathrm{Re}^{-1/4} \tag{7.10}$$

c. In the transient regime, $2000 < \mathrm{Re} < 4000$, the friction factor is unknown and its value depends on the laminar flow conditions for the low Reynolds numbers and on the turbulent flow conditions at higher Reynolds numbers.

In pipes with non-circular cross sections, the equivalent or hydraulic diameter must be used in all calculations. The hydraulic diameter can be obtained from

$$D_h = \frac{4A}{S} \tag{7.11}$$

where D_h is the hydraulic diameter, A is the cross-sectional area of the pipe, and S is the wetted perimeter. In open canals, the hydraulic diameter can be used with good accuracy.

Since the application of (7.6–7.10) for calculating the friction factor is not straightforward in many cases, usually friction factors are presented in the form of charts or diagrams. One of the best and fundamental charts for calculating the friction factor is the Moody Diagram, presented in Fig. 7.1. In this diagram the friction factor f is presented based on the Reynolds number and the relative roughness.

The relative roughness, $\frac{\varepsilon}{D}$, for new pipes and different materials can be also obtained from Fig. 7.2.

There is also another diagram for calculating the pressure loss directly from the friction factors of Colebrook, which is presented in Fig. 7.3. From this curve, the pressure losses for every 100 m of a straight pipe based on the mass flow rate and pipe diameter (velocity) can be obtained. This diagram is prepared for new cast-iron pipes. When the wall roughness and material of the pipe are different with these conditions, the pressure loss can be corrected by multiplying it by the numbers presented in Table (7.1).

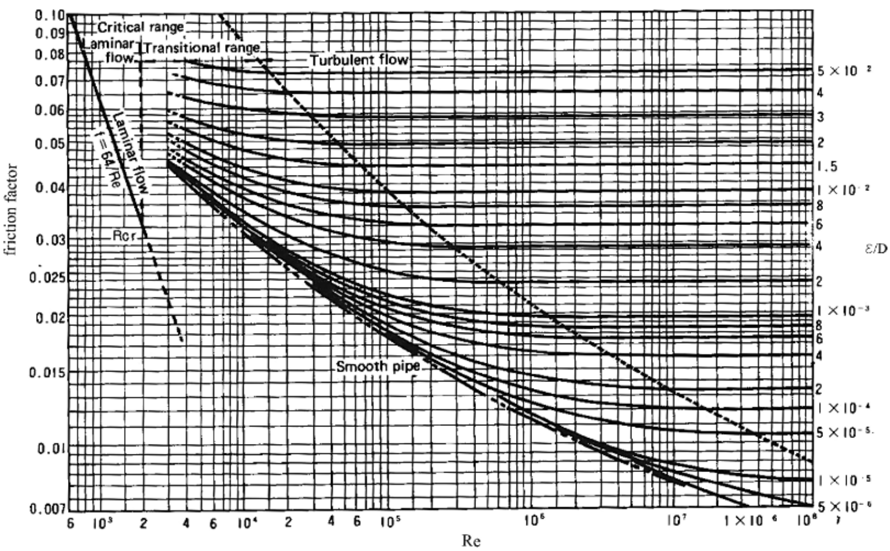


Fig. 7.1 The friction factor in the pipes (Moody Diagram) [1]

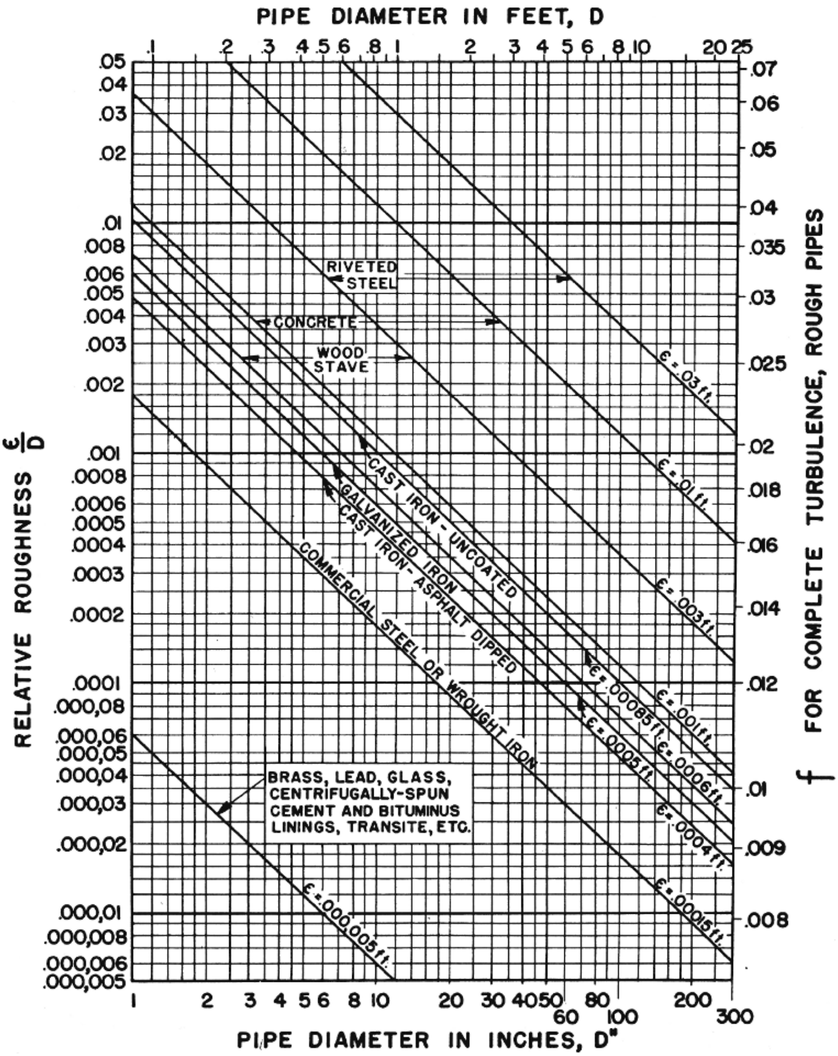


Fig. 7.2 The relative roughness and the friction factor for commercial pipes for fully turbulent flows. Taken from ASME and prepared by L.F. Moody. (English Units) [2]

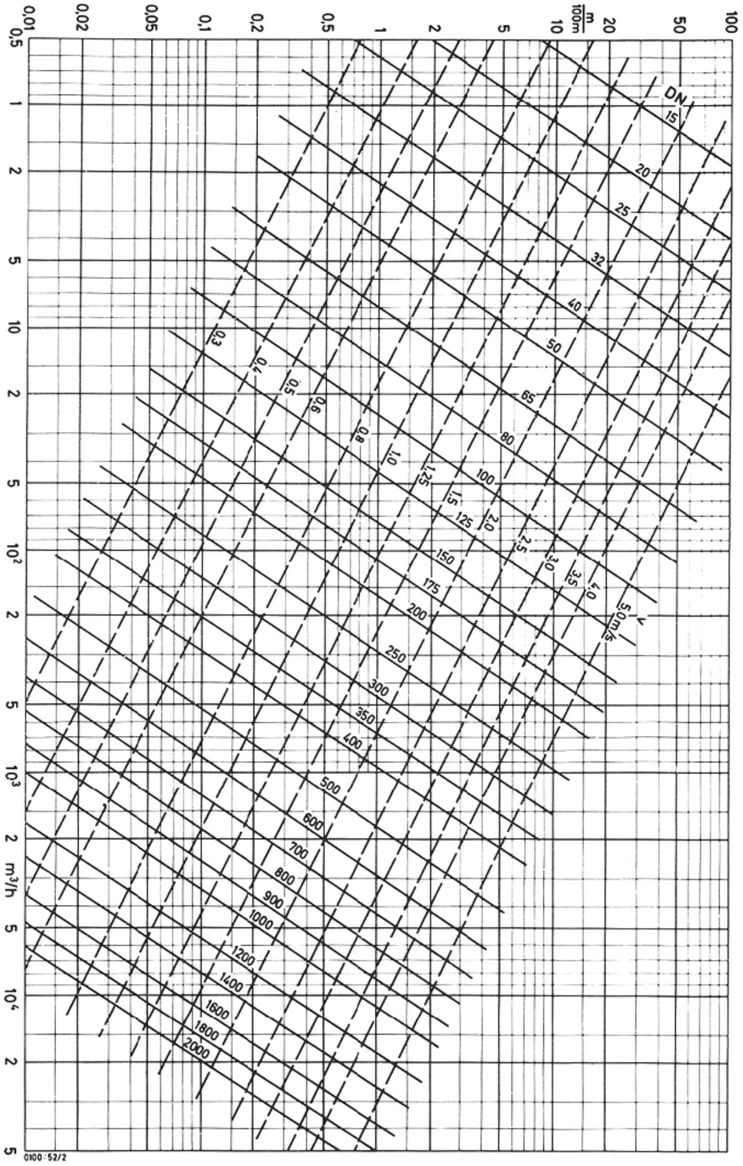


Fig. 7.3 Pressure loss in new cast iron straight pipes. Water temperature is 20 °C [3].

Table 7.1 Pressure loss factor for different pipe roughness [3]

Pipe type	Multiple by
For new steel pipes	0.8
For very rough pipes	1.7
For old and rusty steel pipes	1.25

When the viscosity of the liquid is different from the viscosity of water, the actual pressure loss can be obtained by applying the corresponding correction factors presented in Fig. 7.4 [3]. To use this figure, first the pressure loss for the water is calculated. Then, the friction factors for water and the viscous liquid is obtained from the diagram and finally the pressure loss is calculated from the following relation:

$$H_{LV} = \frac{f_V}{f_W} \times H_L \tag{7.12}$$

where f_W and f_V are the corresponding friction factors for water and viscous flow, respectively, and H_L and H_{LW} are the pressure losses for water and for the viscous liquid.

Example

The flow rate for a liquid with viscosity of $2 \times 10^{-4}\ \mathrm{m^2/s}$ in a pipe with diameter of 250 mm is $100\ \mathrm{m^3/h}$. Find the pressure loss for 100 m of this pipe.

From Fig. 7.3, the pressure loss for $Q = 100\ \mathrm{m^3/h}$ for water is $H_L = 0.14\mathrm{m}/100\ \mathrm{m}$.

In Fig. 7.4, for the same flow rate and diameter, following the direction of arrows, the friction factors for water, f_W, and viscous flow, f_V, can be obtained and the pressure loss, then, is calculated from

$$H_{LV} = \frac{0.08}{0.021} \times \frac{0.14}{100} = 0.53 \quad m/100\,m \tag{7.13}$$

7.2.4 *Empirical Relation for Calculating Pressure Loss*

Among the numerous empirical relations suggested for calculating the pressure loss in the straight pipes, the most used relation is Hazen-Williams relation. This relation can be used for pressure loss calculation in the water pipes [5]:

$$H_L = 10.64\,L \left(\frac{Q}{C}\right)^{1.85} \frac{1}{D^{4.87}} \tag{7.14}$$

In which Q is the flow rate in $\mathrm{m^3/s}$, L and D are the length and diameter of the pipe in m, H_L is the pressure loss in 1 m length of the pipe, and C is the Hazen-Williams constant which depends on the relative roughness of the pipe. The values for C for some pipe materials are given in Table (7.2).

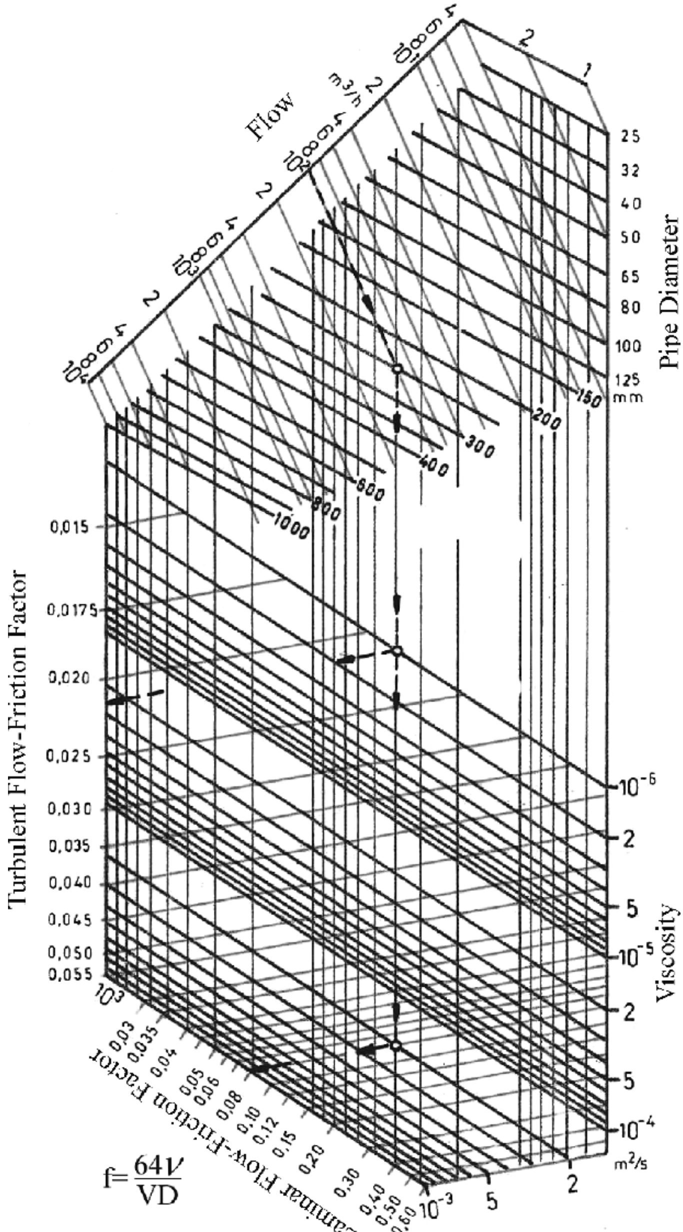


Fig. 7.4 Friction factor for viscous liquids, straight pipes [3]

Table 7.2 Values for C, used in Hazen-Williams relation [2]

Pipe type	Average value for C	Design values of C
Very smooth pipes, like asbestos pipes	150	140
Smooth pipes like copper and brass pipes	140	130
Seamless steel pipes	140	110
Steel or commercial pipes	130	100
Cast iron pipes	130	100
Cement pipes	120	100
Rough iron pipes with many years in service	100	80

In Fig. 7.5, a diagram that has been made based on Hazen-Williams relation is shown. This diagram is in English units and can be easily used to directly find the pressure loss in pipes [4]. First, a straight line is drawn between the capacity value and the pipe diameter as shown in the figure. This line intersects with the turning line. From the intersection of this line and the turning line, another line is drawn to connect this point to the value of the Hazen-Williams coefficient, C. The intersection point of this line and the "loss of head" line will determine the friction loss per 1000 ft of the pipeline.

As it can be seen, from the figure, this diagram also determines the value of the liquid velocity in the pipe in ft/s. Alternatively, the liquid velocity can be used to determine the friction loss instead of the flow capacity.

7.3 Pressure Loss in Valves and Other Fittings

The pressure loss in valves, bends, and other fittings in the piping system can be determined with two methods that are described in the next two sections.

7.3.1 Equivalent Length Method

In this method, the equivalent length of a straight pipe, corresponding to a specific fitting is determined. When the equivalent length is obtained, it is added to the total pipe length and the new length is, then, used in calculating the pressure loss in the whole piping system. The equivalent length of a fitting is the length of a straight pipe that produces the same pressure loss as the fitting.

The diagram presented in Fig. 7.6 can be used to determine the equivalent length for different valves and fittings.

7.3.2 Direct Method

In this method the pressure loss in each fitting in the piping system is obtained from the following relation:

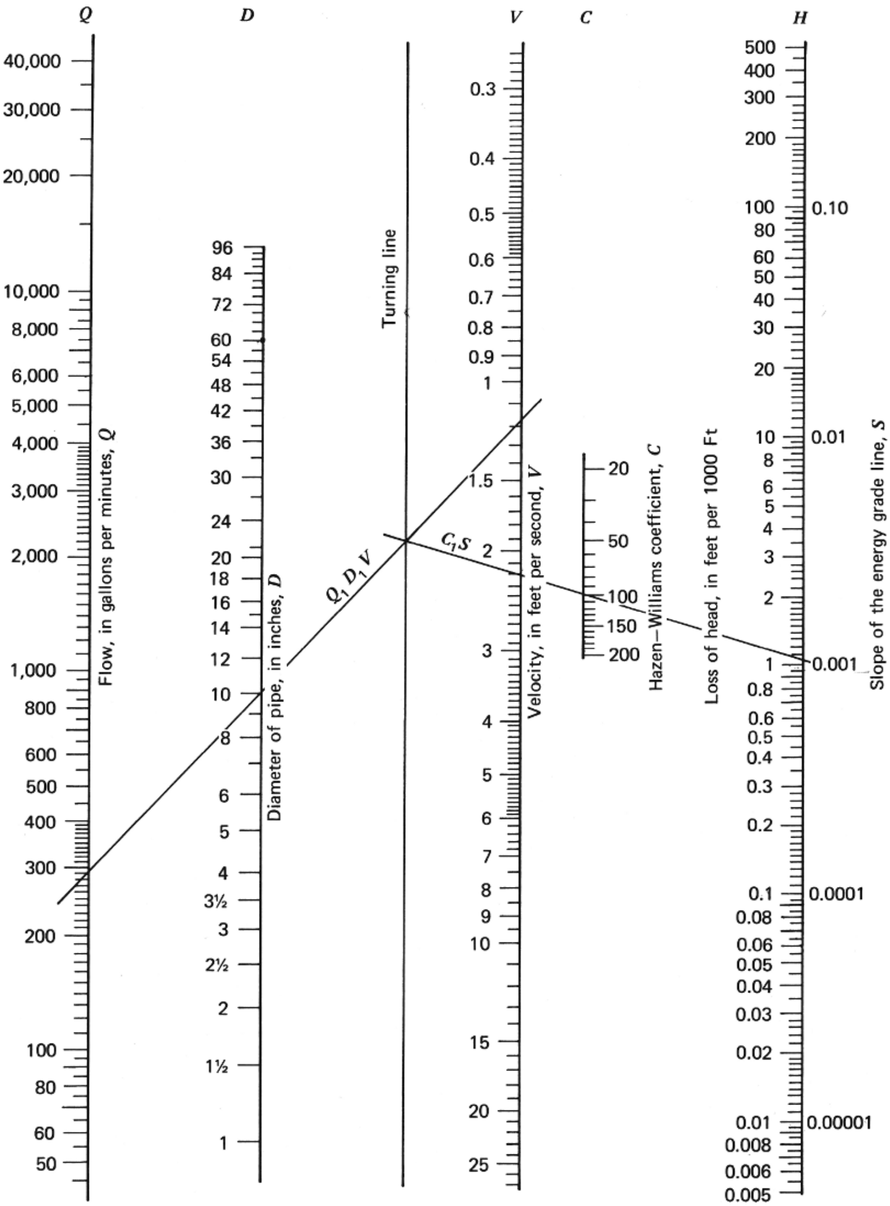


Fig. 7.5 The diagram obtained from Hazen-Williams relation (English Unit System) [4]

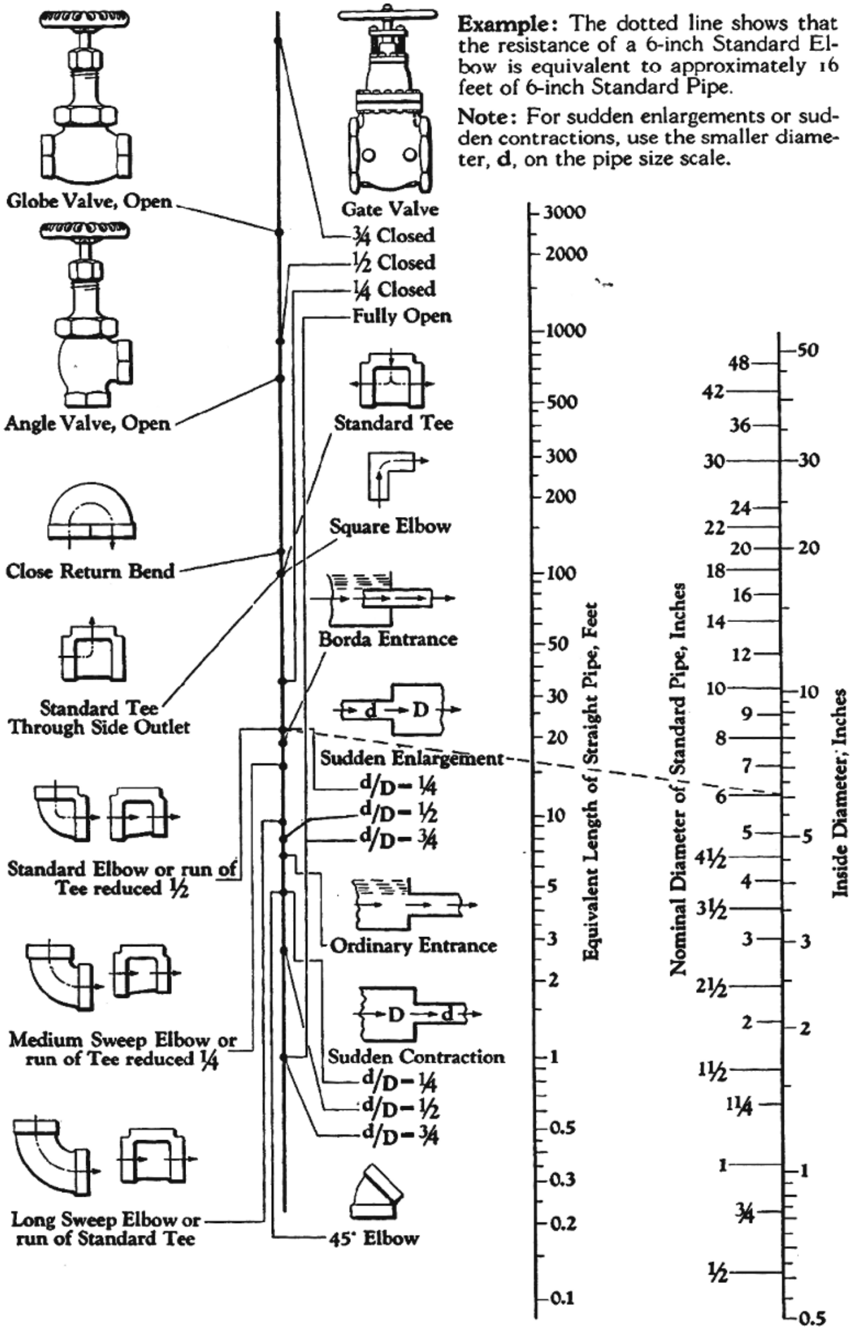


Fig. 7.6 Determining the equivalent length. First, the pipe diameter is selected from the "Inside Diameter" indicator, right bar. Then, a straight line is drawn from that point to the particular fitting. The intersection of this line with the "equivalent Length Indicator," second line from the right, would determine the equivalent length, (English Unit System) [5]

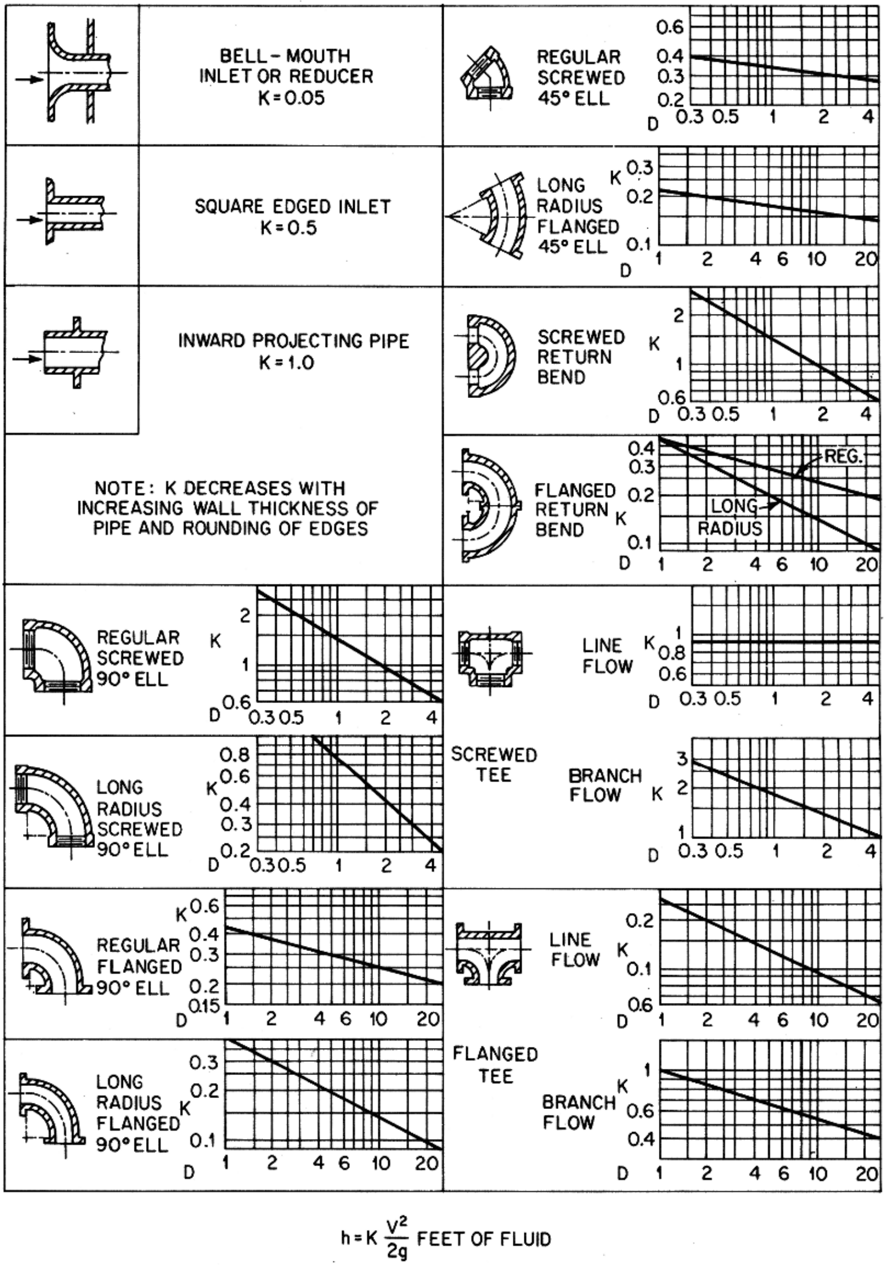


Fig. 7.7 Resistance coefficient *K* for different fittings to be used in (7.15) [6]. *D* is the pipe diameter in inches

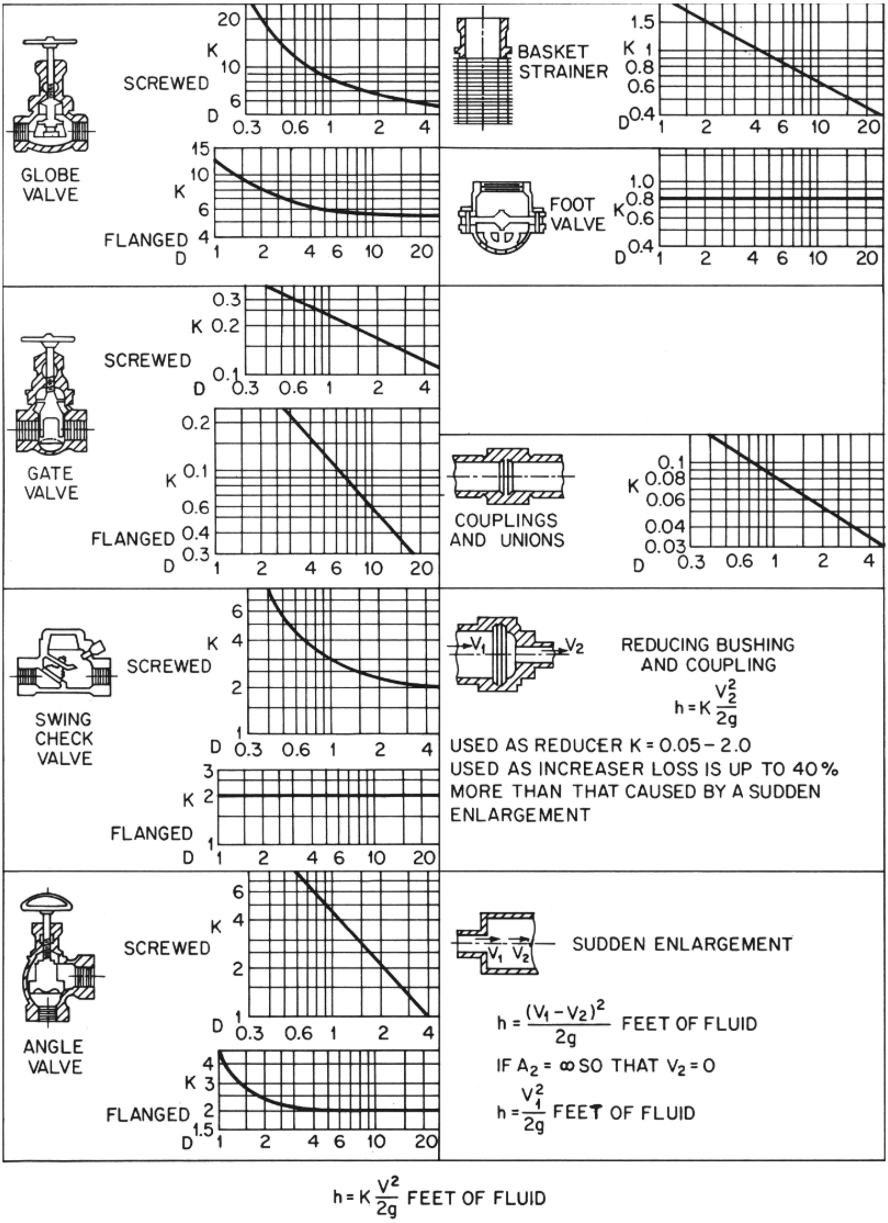


Fig. 7.8 Resistance factor K in different fittings to be used in (7.15). D is the pipe diameter in inches (English Units) [6]

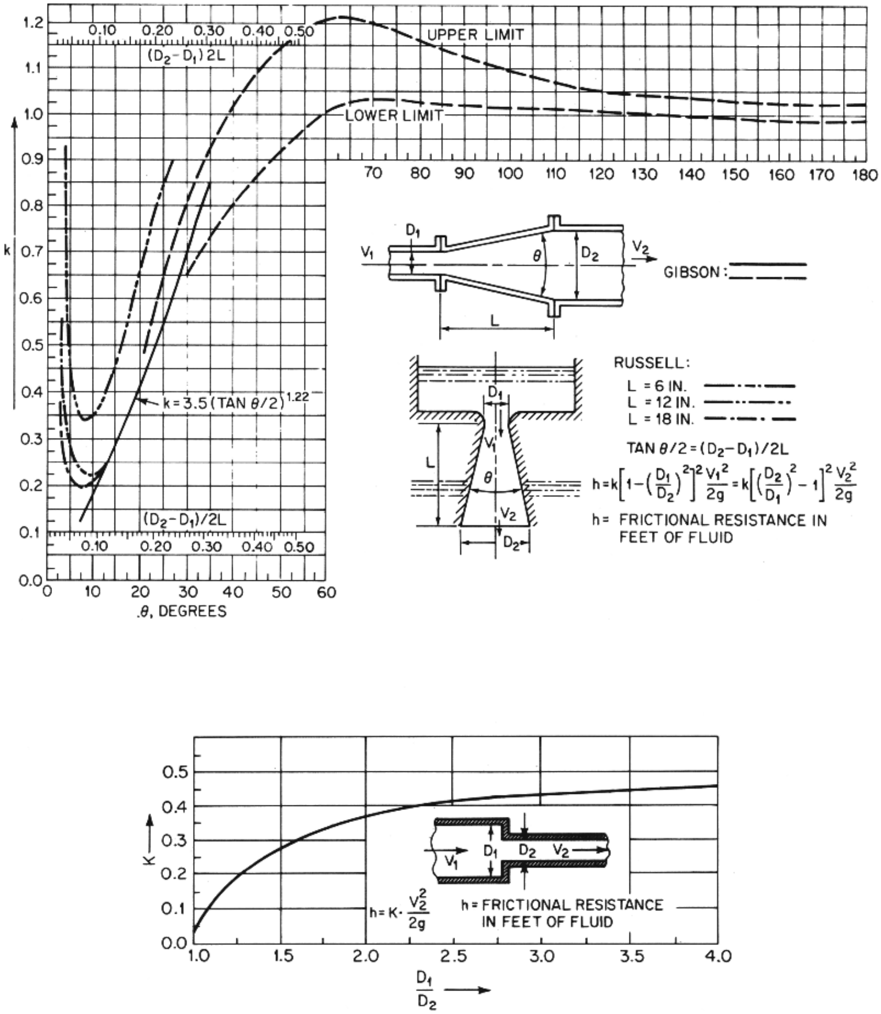


Fig. 7.9 Resistance coefficient K in different fittings. All parameters are defined in the inserts (English Unit System) [6]

$$H_L = K\frac{V^2}{2g} \tag{7.15}$$

Where K is the resistance coefficients and its value for different fittings has been presented in many references. Once K and the flow velocity are known, the pressure loss can be calculated and be added to the pressure loss for straight pipe. Figs. 7.7–7.9 present different K coefficients [6].

For more information about pipeline systems, see [7, 8, 9, 10, 11, 12, 13, 14].

References

1. "Kubota pump handbook", Vol. 1, Technical Manual, Kubota Ltd., 1972.
2. Karassik Igor J. et al., "Pump Handbook", McGraw-Hill, 2001.
3. "Centrifugal Pump Design", Technical Appendix, KSB, Frankenthal, Germany.
4. Simon Andrew L., "Practical Hydraulics", John Wiley & Sons, 1981.
5. "The Planning of Centrifugal Pumping Plants", Sulzer Brothers Limited, Pump Division, Winter Thur Switzerland.
6. http://www.gouldspumps.com/
7. Mohinder L. Nayyar, "Piping Handbook", 7th ed., McGraw-Hill, New York, 2000.
8. Frankel M., "Facility Piping Systems Handbook", McGraw-Hill, 2002.
9. Smith P. "Piping Materials Guide Selection and applications", Elsevier, 2005.
10. Sanks R. L.,"Pumping Station Design", Butterworth-Heinemann, 1998.
11. Tullis Paul. , "Hydraulics of Pipelines: Pumps, Valves, Cavitation, Transients", John Wiley & Sons, 1989.
12. Shames I. H. "Mechanics of Fluids", McGraw-Hill, 1992.
13. Streeter V. L., "Fluid Mechanics", 9th ed., McGraw-Hill, 1997.
14. Nourbakhsh S. A., "Pump and Pumping", University of Tehran Press, 2004.

Chapter 8
Pump Performance in a Piping System

As mentioned before in this book, the performance of a turbopump depends on the system on which it is installed. In other words, the intersection of the system curve and characteristic curves of a turbopump determines its operation point. For systems with complicated piping arrangement and for pumps that are connected in a parallel or serial arrangement, the procedure of finding this working point of the system is very crucial.

In the next sections, the performances of turbopumps in different piping systems are described in more details. Also covered in this chapter are the start up and shutdown procedure of pumps in the system.

8.1 System Characteristic Curve

Consider a piping system consisting of a pump and a set of pipes and fittings, as shown in Fig. 8.1. The pump transfers water from the lower tank A, with pressure P_A and velocity of V_A, to the upper reservoir R, with a pressure of P_R and velocity of V_R. For this system, one can write the Bernoulli's equation between Point A and the pump inlet, point 0, also between the pump exit, point 3, and the upper reservoir, point R, i.e.

$$\frac{P_A}{\rho g} + \frac{V_A^2}{2g} + Z_A = \frac{P_0}{\rho g} + \frac{V_0^2}{2g} + Z_0 + H_{L1} \tag{8.1}$$

$$\frac{P_3}{\rho g} + \frac{V_3^2}{2g} + Z_3 = \frac{P_R}{\rho g} + \frac{V_R^2}{2g} + Z_R + H_{L2} \tag{8.2}$$

where H_{L1} and H_{L2} are the pressure losses in the suction and delivery pipes, respectively. By adding two equations together, one has

$$\frac{P_3 - P_0}{\rho g} + \frac{V_3^2 - V_0^2}{2g} + (Z_3 - Z_0) = \frac{P_R - P_A}{\rho g} + \frac{V_R^2 - V_A^2}{2g} + (Z_R - Z_A) + H_L \tag{8.3}$$

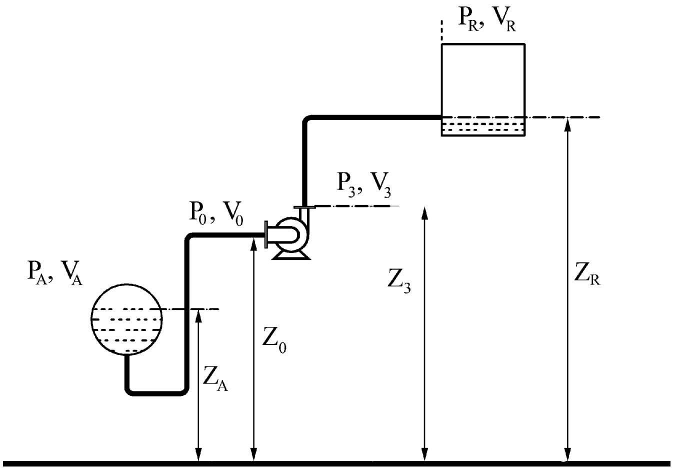


Fig. 8.1 Pump in a system

where H_L is the total pressure loss in the system. As it is seen, the left-hand side of (8.3) is representative of the total manometric head of the pump, i.e. H_P (see Sect. 2.2). The right-hand side of the equation is called the *system resistance* and is shown by H_R. Therefore, the condition under which a pump would work in a piping system is

$$H_P = H_R \tag{8.4}$$

The above equation states that the total manometric head supplied by a pump is equal to the total resistance of the system in which it is working. This means that by changing the system resistance, the manometric head of the pump also changes. This is an important aspect of pump operation and one of the fundamental characteristics of the turbopumps. For this reason, it is impossible to indicate what is the total head or flow rate of a pump without knowing the specification of the system in which it is working. The flow rate of a pump is changed based on the system resistance. Of course each pump has a specific flow rate and head at its maximum efficiency point which are called the design head and flow rate. However, these flow rate and head are not necessarily supplied in a system [1].

In order to find the working point of a particular pump in a system, the characteristic curve of the system must be obtained. This curve which is also called the system head or resistance curve, H_R, consists of two components: dynamic resistance and static resistance, i.e.

$$H_R = H_{dy} + H_{st} \tag{8.5}$$

The dynamic resistance of the system is proportional to the square of the flow rate and can be obtained from

$$H_{dy} = \frac{V_R^2 - V_A^2}{2g} + H_L = f\left(Q^2\right) \tag{8.6}$$

The static resistance of the system is independent of the flow rate and is the constant component of the system curve:

$$H_{St} = \frac{P_R - P_A}{\rho g} + (Z_R - Z_A) \tag{8.7}$$

If the characteristic curve of the system, H_R, and the characteristic curve of a pump are plotted in the same system of coordinates, the intersection point of two curves would determine the working point of the pump (Fig. 8.2). By specifying this point, the flow rate, head, efficiency, absorbed power, and NPSH of the pump are determined. It is obvious that for a pump to work efficiently this point should be very close to the design point of the pump.

When a pump is working in a closed system (like hot water pumps in heating systems), H_{st} is equal to zero. That means the system characteristic curve is passing through the origin ($H = 0$, $Q = 0$). Now if $V_R \approx V_A$ and $H_R = H_L$, it means that the manometric head of the pump is used only to overcome the pressure losses in the system. For this reason the manometric heads of the pumps used in the closed systems are in the order of few meters [1].

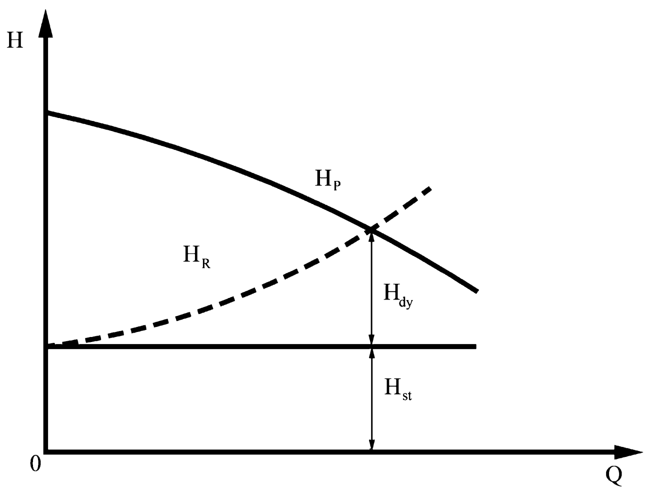


Fig. 8.2 Working point of a pump

8.2 Variations of System Characteristic Curves

As seen in the last section, the characteristic curve of a system consists of two components: the dynamic resistance and the static resistance of the system. Therefore, any changes in either of these two components would change the system curve.

8.2.1 Variation in Dynamic Resistance

8.2.1.1 Opening and Closing of Valves in the System

In a pumping system, in order to adjust the flow rate, to shutdown, or to start up the pump, different valves may be used. Any change in the operating condition of a valve would change the dynamic resistance of the system (because of its effect on the pressure loss of the valve). In Fig. 8.3, these changes are shown schematically. By closing a valve gradually from positions 1 to 2 and 3, the resistance curve of the system would change from H_{R1} to H_{R2}, and finally to H_{R3}. Consequently, the operating point of the pump would change from Point A_1 to A_2 and A_3. As a result, the flow rate of the pump decreases from Q_1 to Q_2 and to Q_3. As one can see, by further closing the valve, the pressure loss increases and the slope of the system curve increases as well.

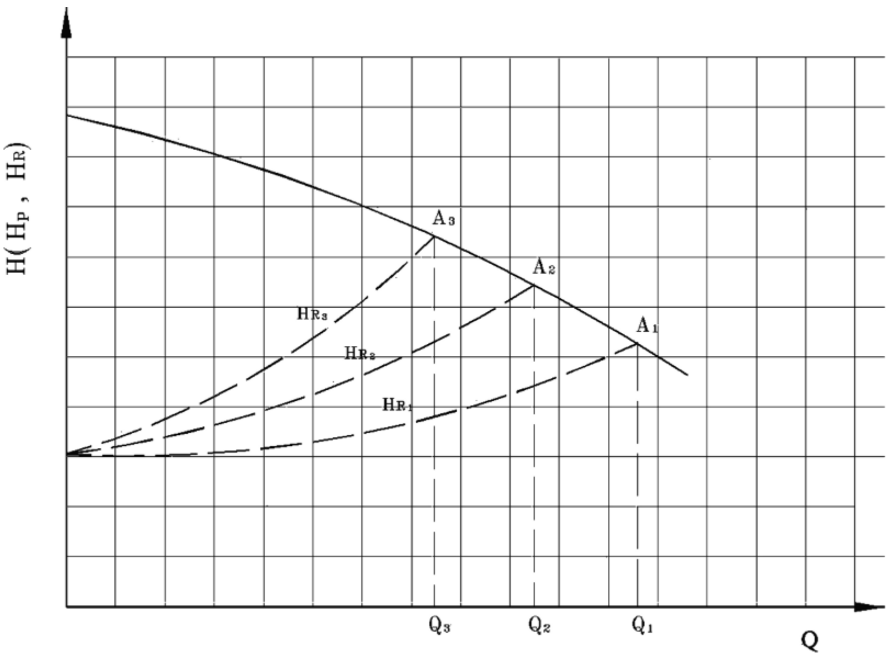


Fig. 8.3 The effect of closing a valve on system characteristic curve

In Fig. 8.3, the pressure loss in the system due to the closing action of the valve is also shown. This figure shows that the flow rate adjustment of the pump should be done on the delivery line of the pump. Since if the control valve is located on the suction pipe, due to the high pressure loss in the valve, the available NPSH_{avail} decreases and the risk of cavitation increases.

8.2.1.2 Service Life of the Piping System

When a piping system is in service for long time and following many years of liquid flow passing through the pipes, there would be always some deposit built up on the inner walls of the pipes. These deposits are usually the result of the hardness of the liquid and/or the existence of the suspended particles in it. For this reason, after a while, the roughness of the walls and also internal pipe diameter would change, resulting in more pressure losses in the system. In this case, the dynamic resistance of the system changes and its slope increases, leading to a decrease in delivered flow rate. Therefore, when designing a pumping system, the anticipated service life of the system must be considered and the design of the piping system should be done accordingly.

8.2.1.3 Broken Pipes in the System

If for any reason (like constructional works) the delivery pipe of a pump is broken, the system resistance would drop abruptly. This leads to a sudden increase in the pump flow rate and consequently the absorbed power of the pump increases. If the pump driving motor cannot handle this sudden power increase, the risk of serious damages to the motor would be highly possible.

8.2.1.4 Orifice in the Piping System

In some pump stations, it is observed that at the beginning of the system operation the pump is not working close to its maximum efficiency point. This usually happens if the system is designed based on the anticipated life of the system and therefore the initial working conditions are far from the optimum conditions. In order to bring the working point closer to the design point of the pump, an orifice plate can be installed at the end of the delivery pipe to make artificial pressure losses. By selecting a proper orifice, the working point of a pump can be adjusted by this method.

8.2.2 Variations in Static Resistance

If for any reason the static pressure of the system or the geometric height between the suction and delivery reservoirs is changed, the static resistance of the system is changed accordingly. This is shown in Fig. 8.4, where the initial height of the reservoir Z_A has been changed to Z_B during the pump operation. In this case, the

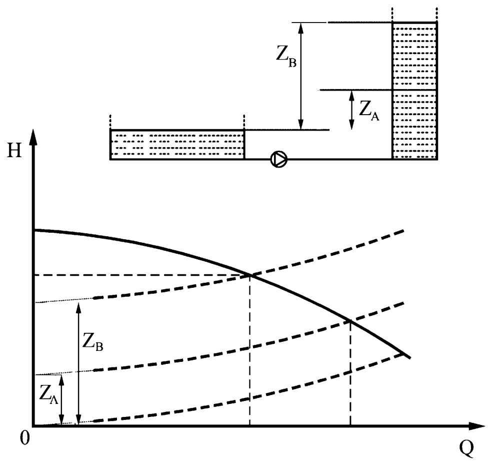


Fig. 8.4 The variation of the system characteristic curve due to change in static head of the system

characteristic curve of the system will shift to a lower or higher position on the system of coordinates (Fig. 8.4).

This problem usually exists for pumps that are used to transfer water from wells or similar reservoirs. During different seasons of the year, the level of the underground water may change; therefore, the static head of the system is changed accordingly. This will lead to a change in the operating point of the pump. The same problem exists for those pumps that deliver water from a lower reservoir to an upper reservoir. After a while, the upper reservoir could become full and/or the lower reservoir is emptied, modifying the geometrical elevation of the system.

8.3 Connecting Pumps

When a single pump cannot produce the required head or deliver the flow rate in a pump station, two or more pumps can be used in the system. Depending on the requirement of the station, the connection of the pumps can be done in series or in parallel.

8.3.1 Pumps in Parallel

When a relatively high flow rate is required from a station, the total flow rate of the system will be supplied through several pumps that are connected to each other in a parallel arrangement. The performance of such system is similar to that of an electrical circuit containing different elements in parallel, i.e., the total flow rate passing through the pump station is the sum of the flow rates of each pump, while the manometric heads of all pumps remain the same (Fig. 8.5).

Connecting pumps in parallel affects the performance of each pump. To obtain the resulting characteristic curve for the whole pumping system, at one constant

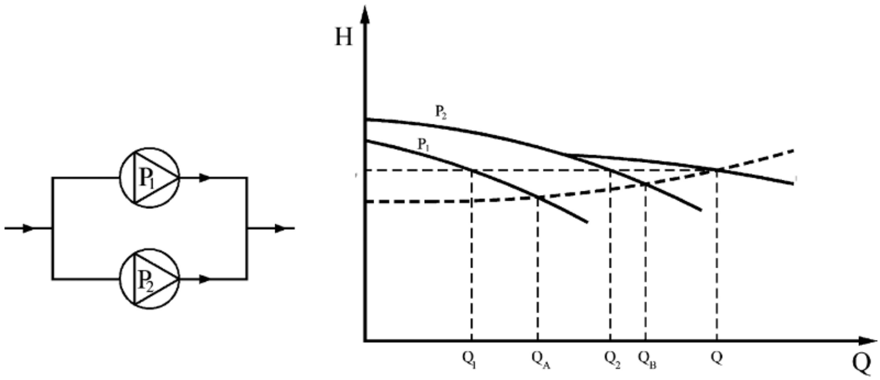


Fig. 8.5 Pumps in a parallel system and the characteristic curves

head, the corresponding flow rates of the pumps must be added on the same system of coordinates as shown in Fig 8.5. The resulting points, then, are connected to each other to form the new characteristic curve, see Fig. 8.5. The intersection point of this curve and the system curve would determine the working point of the pump station (Q, H_P).

To obtain the working point of each pump, a line can be drawn from the new working point parallel to the Q axis until it passes through the characteristic curves of all pumps. The corresponding points determine the working point for each pump (Q_1 and Q_2).

Pumps are usually selected to operate close to their maximum efficiency points. When only one pump, either P_1 or P_2, has to work in the same system, the corresponding supplied flow rate of each would be Q_A or Q_B, respectively.

Two or more pumps are connected in a parallel arrangement when:

1. The required flow rate is very high such that in order to deliver it a very large pump is needed. In this case, by dividing the flow rate between two or more smaller pumps, the same manometric head can be supplied.
2. In some designs, due to very large required flow rate, the NPSH_{req} for a single pump would become larger than the $\text{NPSH}_{\text{avail}}$. In these cases, by installing two or more smaller pumps in a parallel arrangement, the flow rate of each pump would be smaller and therefore the cavitation can be prevented in the system.
3. In many pump stations, it is desired to use more numbers of smaller pumps to increase the flexibility of the operation and to achieve better performances (there is always an optimum number). By increasing the number of pumps in a pump station, the safety factor of operation is increased because of the following reasons:

    - There would be more spare pumps in emergency situations.
    - There is always the opportunity to perform more regular inspections, repairs, and part exchanges.
    - It is easier to obtain the spare parts (smaller pumps).
    - It is easier to transport and move the units.

When connecting two or more pumps in parallel, one should always consider the following issues:

a. When two different pumps are connected together in parallel, during start up or shutdown period, there is a possibility of water flowing from one pump to another. This is because the characteristic curves of the pumps are different and one pump delivers a higher manometric head. For example, as can be seen in Fig. 8.5, when the flow rate is zero, pump number 2 can produce higher pressure and there is a chance of water returning from pump 2 to pump 1. To prevent this problem, it is essential to install one-way valves on the discharge pipes of all pumps. More importantly, it is recommended to use similar pumps in all parallel arrangements.
b. Due to the shape of the system characteristic curve, i.e. as the flow rate increases the required pressure is also increased, connecting pumps in a parallel arrangement would not increase the supplied flow rate by the same factor as the pumps work independently in the same system. This factor must be considered when selecting pumps to work in parallel. The selection process must be done as described in Fig. 8.5.
c. In a pump station with pumps installed in parallel, one must try to minimize the dynamic resistance of the piping system. Because, as is seen in Fig. 8.6, in systems with steeper characteristic curves, H_{R2}, the delivered flow rate of the pumps in the system is lower than a more flat system curve, H_{R1}. Obviously, the whole purpose of connecting two or more pumps in a parallel arrangement would be questionable in this case, since as can be seen the ΔQ in system with steep curve is not much more than a pump working alone in the system. This is another reason for choosing larger diameters for pipes at the delivery side of the pumps.
d. As mentioned in (b), the delivered flow rate of each pump working in parallel is less than the flow rate of the same pump if it works independently in the system, see Fig. 8.6. Therefore, when one pump stops working in the system, the flow

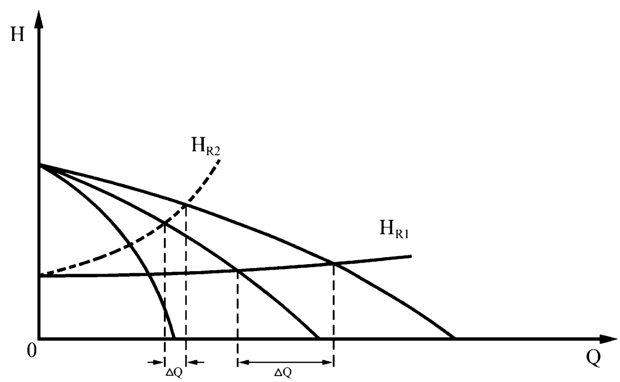


Fig. 8.6 Effect of increasing the slope of the system curve in performance of pumps working in parallel

rate of other pump increases suddenly. This possibility should be considered when selecting the driving motor and determining the $NPSH_{req}$ for each pump.

8.3.2 Pumps in Series

Connecting pumps in series (serial connection) in a system is recommended to increase the supplied pressure at a constant flow rate. In this case as the total flow rate of the system passes through each pump in a serial arrangement, its pressure is increased at each step.

To plot the resulting characteristic curve of the assembly of the pumps, at each constant flow rate, the corresponding heads of the pumps are added together. Sample of such characteristic curves for two pumps in a serial arrangement are shown in Fig. 8.7 and Fig. 8.8. The intersection of this curve with the characteristic curve of the system determines the working point of the pump station.

To obtain the working point of each pump in the system, one can draw a line parallel to the H axis from the intersection point to cross each characteristic point. These points are the working points of each pump (H_1 and H_2). When connecting pumps in a serial arrangement the following issues must be considered:

- When the pumps are not geometrically similar, it is better to install the pump with lower $NPSH_{req}$ as the first pump in the system.
- Because the liquid entering the second pump has a higher pressure, the liquid pressure in the casing and stuffing box for this pump is also higher. Thus, the leakage protection in the second pump must be performed better. Also, the casing

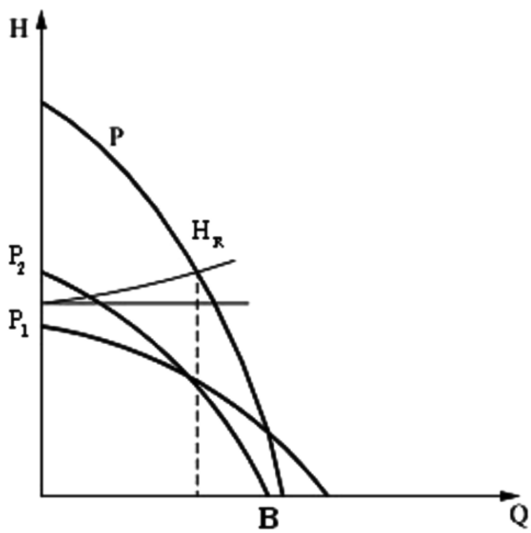


Fig. 8.7 Connecting pumps in a serial arrangement and the resulting characteristic curve

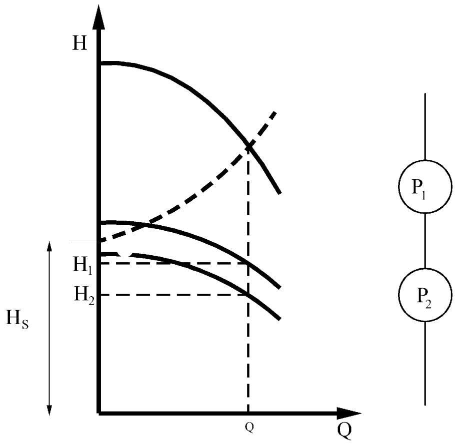


Fig. 8.8 Connecting non-similar pumps in a serial arrangement

of the pump and all other mechanical elements should be able to handle this high pressure.

- When two similar pumps are installed in a serial arrangement, the total supplied head would be twice of the supplied head of each pump in the system. However, the total supplied head is less than twice of the supplied head of each pump when it is working independently in the system.
- When two different pumps are connected in series, it is possible that at a specific flow rate, the supplied head of one pump becomes zero, while the other pump still supplies a positive head. If the flow rate of the station is set to be at this value or higher, the first pump would not supply a positive pressure. As a matter of fact from this point on, the pump acts as a system resistance. This example is shown in Fig. 8.7, where at flow rates higher than point B, pump P_2 would act as a resistance for pump P_1. Also, the supplied head from pump P_2 would be negative and total head would become $H_P = H_{P1} - H_{P2}$. For this reason, it is recommended to use similar pumps when a serial connection is required unless proper safety systems are used on the pumps to stop them from working after one reached its maximum flow rate point.

8.4 Effect of Different Parameters on Pump Performance

8.4.1 Life Time of the Pump Station

As was mentioned in Sect. 8.2.1, the characteristic curve of a system will change through years of a pump being in service. This would affect the working point of a pump and would change its delivered flow rate through the years. This fact must be considered when designing a pump station.

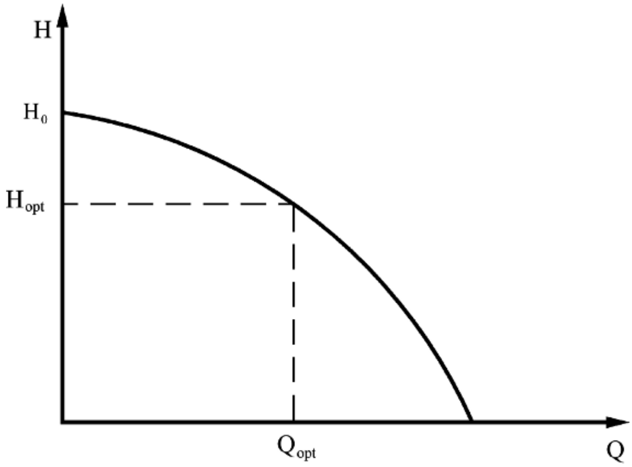


Fig. 8.9 Slope of the H–Q curve

8.4.2 Slope of H–Q Curve

The slope of the H–Q curve is represented by $(H_0 - H_{opt})/Q_{opt}$. In this definition, H_0 is the supplied head of the pump at $Q = 0$ and H_{opt} and Q_{opt} are the head and flow rate of the pump at the maximum efficiency point (Fig. 8.9). Slope of the H–Q curve is a function of the specific speed and has generally the values presented in Table (8.1).

In general, the slope of the H–Q curve would affect its working point and is an essential parameter in selecting different pumps for different applications. In those systems where it is necessary to maintain a uniform constant flow rate when the resistance of the system undergoes many fluctuations, the pumps with steep slopes are the best choices (Fig. 8.10). This will minimize the flow rate fluctuation. However, in pumps with flat characteristic curves, larger changes in the flow rate would cause smaller changes in the manometric head. This feature makes these pumps more suitable for pump stations with anticipated high flow rate fluctuations. This feature is also very important when the flow control is made by means of a valve, because the opening and closing action of the valve would not create much head drop in the pump, see Fig. 8.10.

Slope of the pump characteristic curves also affects the way it can be used in a parallel or serial arrangement. When two pumps with flat curves are connected in parallel, the total flow rate would not be noticeably higher compared to when

Table 8.1 Slope of characteristic curves for different pumps [8]

For centrifugal pumps	0.1–0.25
For mixed flow pumps	0.25–0.8
For axial flow pumps	More than 0.8

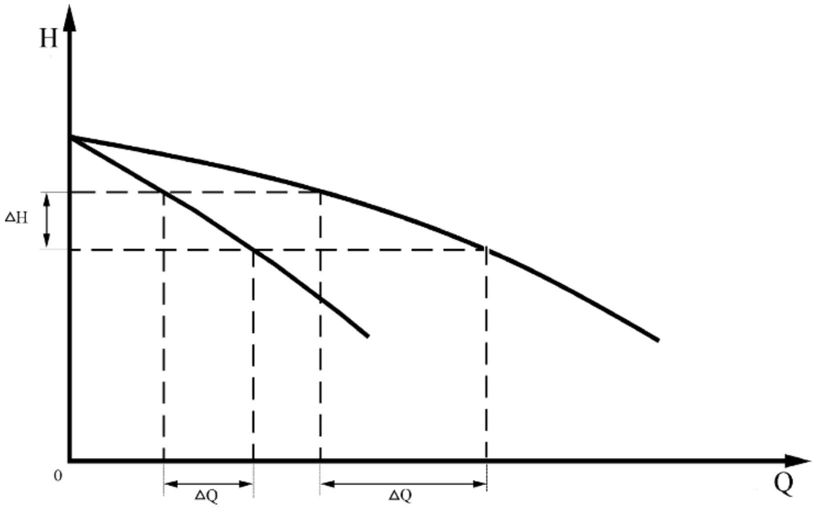


Fig. 8.10 Flat and steep characteristic curves

either of them works independently in the system. This effect would become more significant when the characteristic curve of the system also has a steep slope, see Fig. 8.11.

8.4.3 By-Pass Pipes

By-pass pipes are used widely in the pump stations. One of the applications of the by-pass pipes is to adjust the flow rate. In this arrangement, part of the exit flow rate of the pump is returned to the suction pipe of the pump though a parallel (by-pass) pipe. This would decrease the total delivered flow rate, without

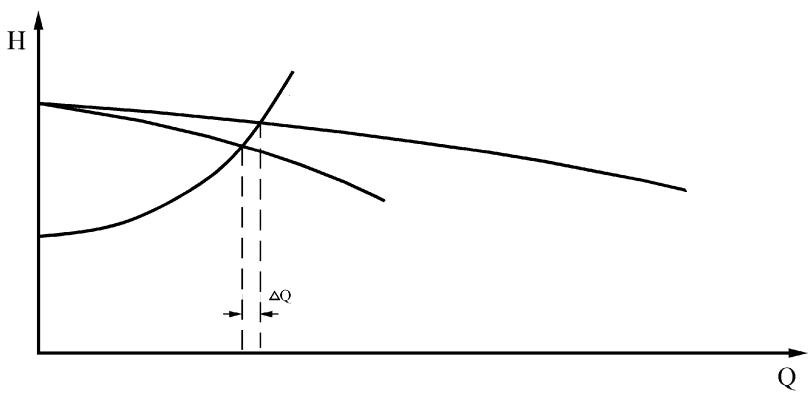


Fig. 8.11 Connecting two identical pumps in parallel with flat characteristic curves

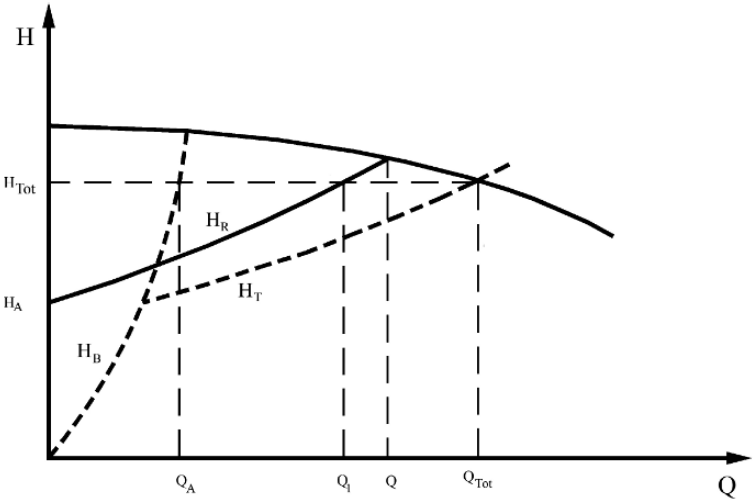


Fig. 8.12 Working point of a pump with a by-pass system

changing the working point of the pump (since the flow rate passing the pump remains constant). However, there would be always loss of energy in this arrangement; thus, this method is used only for systems in which working at the maximum efficiency point of the pump is very critical.

In Fig. 8.12, the characteristic curves of the system, H_R, and by-pass pipe, H_B, are shown separately. The total resistance of the system, H_T, can be obtained by adding these two curves in parallel (adding the flow rates at a constant head). The intersection of the new curve (the dotted line) and the characteristic curve of the pump would determine the working point of the pump, Q_{Tot} and H_{Tot}. To obtain the flow rate in the delivery pipe of the pump (the actual flow rate), a line should be drawn from this new point parallel to the Q axis. The intersection of this line, the system curve, and the by-pass curve would determine the flow rate in the delivery pipe, Q_1, and by-pass pipe, Q_A respectively.

8.4.4 Change in the Rotational Speed

It was shown before that any changes in the rotational speed of a pump would cause a shift in the characteristic curve of the pump, see Sect. 6.7 and Fig. 8.13. Any increase in rotational speed will increase the flow rate and the manometric head, whereas a decrease in the speed would decrease these two parameters. This method, therefore, could be used for adjusting the flow rate, if possible.

8.4.5 Instability of the $H-Q$ Curve

In this section the concept of stability and the effect of instabilities on the working point of a pump will be discussed. In general the region of a characteristic curve is

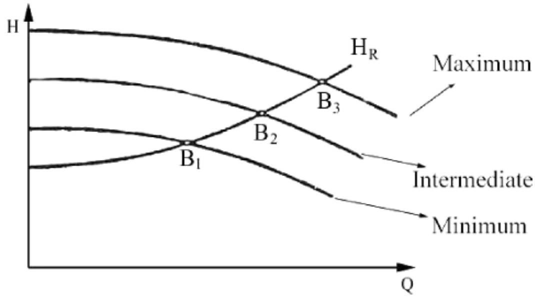


Fig. 8.13 Change in working point caused by variation in the rotational speed

called stable when any small changes in the system characteristic curve would cause only small changes in the working point of the pump. Otherwise, it is said that the working point of the pump is located on the unstable part of the curve.

Usually, in the stable segment of the characteristic curve of a pump with increasing flow rate, the manometric head decreases, i.e. $\frac{dH}{dQ} < 0$ (Fig. 8.14). In the part of the curve in which the manometric head increases with increasing flow rate, there is always a possibility of instability problems in the pump. For this reason, pump manufacturers only present that part of the characteristic curve in which the slope is negative and there is no possibility of instability in working condition of the pump. Otherwise, it is highly recommended that the working point of a pump is not located on the unstable part of the curve. In Fig. 8.14, the stable and unstable parts of a characteristic curve are shown.

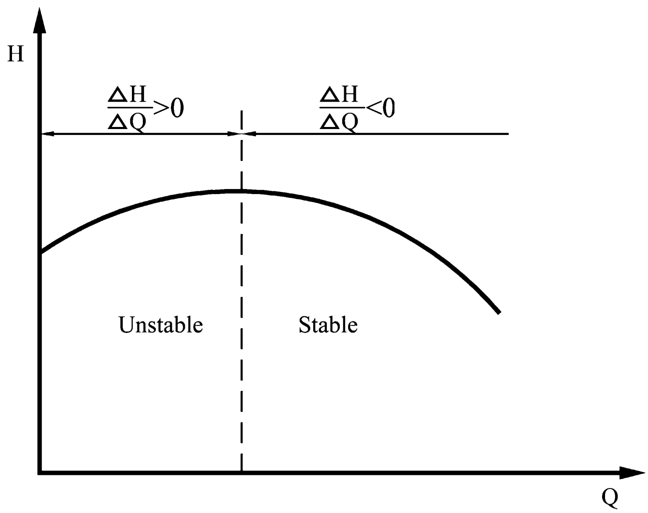


Fig. 8.14 Stable and unstable regions of a characteristic curve

8.5 Performance of Turbopumps in a System

Generally, the performance of the turbopumps can be studied in four different systems. Other systems are usually combinations of these four basic groups and the following examples can be used as guidelines to determine the working point of any pump in any system.

8.5.1 Two Pumps in Parallel in a System with Pressure Loss

In Fig. 8.15, the schematic diagram of such system is shown. The system consists of two pumps P_1 and P_2 that deliver the liquid to a reservoir through necessary pipes and fittings. The corresponding system curves are also shown by H_{R1}, H_{R2}, and H_{R3}. To obtain the working point of the pumping system as well as each pump in the system, the following procedure should be followed.

First, the system curve H_{R1}, with the static resistance of H_1, must be subtracted from the corresponding pump characteristic curve, P_1. The subtraction is done as was described in a serial arrangement of pumps, i.e. at a fixed flow rate, the heads of two curves are subtracted from each other (Fig. 8.16). The same procedure must be repeated for the system curve H_{R2} with static resistance H_s, and pump curve P_2, as shown in the figure. The resultant curves are shown by broken lines.

Next, the resultant curves must be added to each other in a parallel style. The intersection point of this new curve (shown as $P_1' + P_2'$) with the system curve H_{R3} (with static resistance measured from point A to the upstream reservoir) would determine the working point of the system and eventually the working point of each pump, see Fig. 8.16.

To obtain the working point of each pump, a line can be drawn from the working point of the system parallel to the Q axis to intersect the broken lines. The intersection points determine the corresponding flow rates in each line. Then, to obtain the supplied head of each pump, a line can be drawn from each new point, parallel

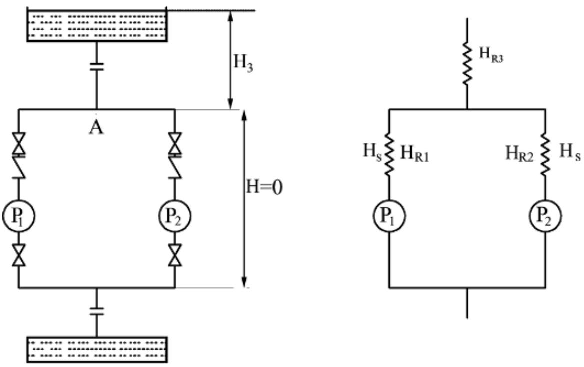


Fig. 8.15 Pumps in parallel arrangement and the corresponding system diagram

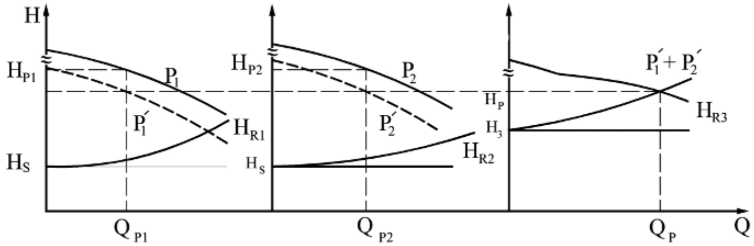


Fig. 8.16 Working point of the pumps working in parallel corresponding to Fig. 8.15

to the H axis to intersect the corresponding characteristic curves of the pumps. In Fig. 8.16, H_P and Q_P are the head and flow rate of the pump station, H_{P1} and Q_{P1} are the head and flow rate supplied by pump P_1, and H_{P2} and Q_{P2} are the head and flow rate supplied by pump P_2.

8.5.2 Single Pump Installed in a Multi-line System

In some pump stations, the flow rate of the pump is distributed to several pipelines. In Fig. 8.17, a schematic diagram of such a system is shown. The equivalent resistance curves are also shown with H_{R1} and H_{R2}, with static resistance of H_1 and H_2, respectively.

To obtain the working point of this system, the total system curve H_R must be obtained by adding two system curves H_{R1} and H_{R2}, in a parallel style (Fig. 8.18). In this example the system resistance between the pump and point A is neglected. If this resistance is large enough, it can be added to the total system curve, H_R, in a serial style (in a constant flow rate the heads are added to each other).

The working point of the pump is the intersection of the system curve, H_R, and the pump characteristic curve. A line that is drawn from this point parallel to the Q axis would intersect with two system characteristic curves, H_{R1} and H_{R2}. These two points would determine the flow rate passing through each branch.

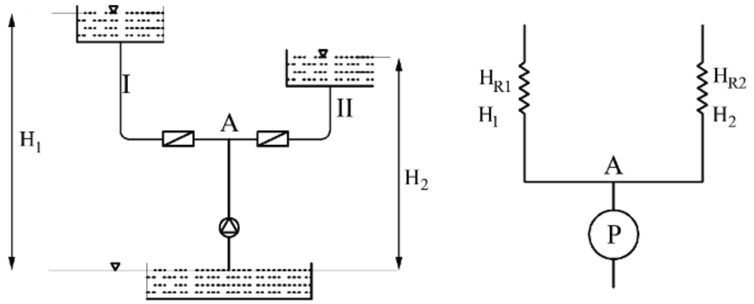


Fig. 8.17 A multiple branch system with corresponding system diagram

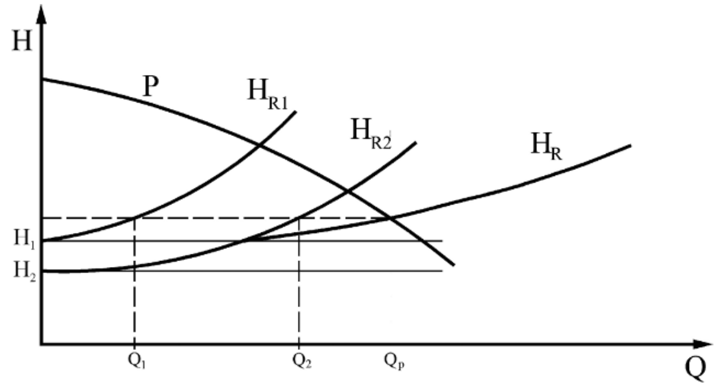


Fig. 8.18 The performance of a pump in a multi-branch system, corresponding to Fig. 8.17

If the static head of two system curves are different, as shown in Fig. 8.18, when the pump stops, flow would run from one reservoir to another, which can be avoided by installing a non-return valve on each branch.

8.5.3 Two Pumps Working in a Serial Arrangement with System Resistance in Between

In some arrangements, especially in systems with very long pipelines, the pumping is done in several stages. i.e. two or more pumps are installed on the same pipeline in various locations. The diagram of such arrangement is shown in Fig. 8.19.

To obtain the working point of the system, the system curve H_{R1} must be subtracted from the characteristic curve of the first pump, P_1, as shown in Fig. 8.20. Then, the resulting curve P_1', which is shown as a broken line, must be added to the characteristic curve of the second pump P_2 with a serial method. The intersection of the new curve $P_1' + P_2$ with the characteristic curve of the system, H_{R2}, would determine the working point of the system.

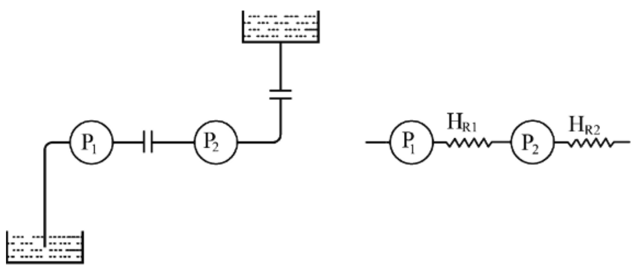


Fig. 8.19 Two pumps working in sequence with resistance

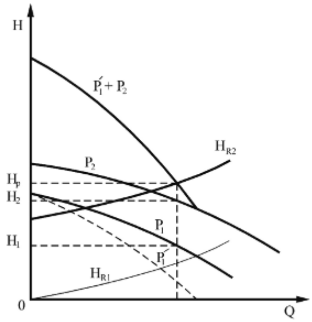


Fig. 8.20 Working point for two pumps working in a serial arrangement, corresponding to Fig. 8.19

To obtain the working point of each pump, a line must be drawn from this point parallel to the H axis to cut the characteristic curve of each pump. The intersection points would determine the head of each pump.

8.5.4 Two Pumps Working in Parallel which are Located at a Distance From Each Other

A schematic diagram of such system and its equivalent system diagram are shown in Fig. 8.21. This arrangement is normally used to provide the flow rate from two different reservoirs.

To simplify the procedure, it is assumed that the system resistance between pump P_2 and point II is negligible. To obtain the flow rate passing through the common pipe and the working point of each pump, the system resistance H_{R1} must be subtracted from the characteristic curve of pump P_1 (Fig. 8.22). This would result in the modified characteristic curve for pump P_1, i.e. P_1', which is shown as a broken line. This new curve, then, must be added to the characteristic curve of pump P_2 in

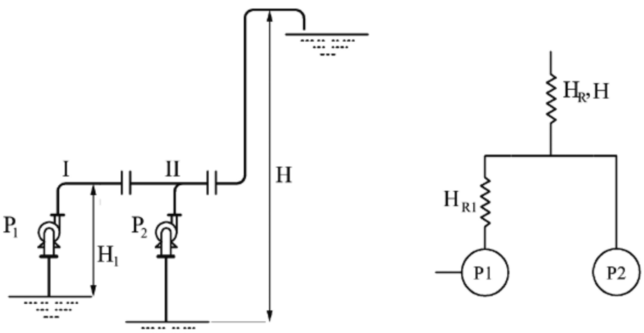


Fig. 8.21 Two pumps working in distance in a parallel arrangement

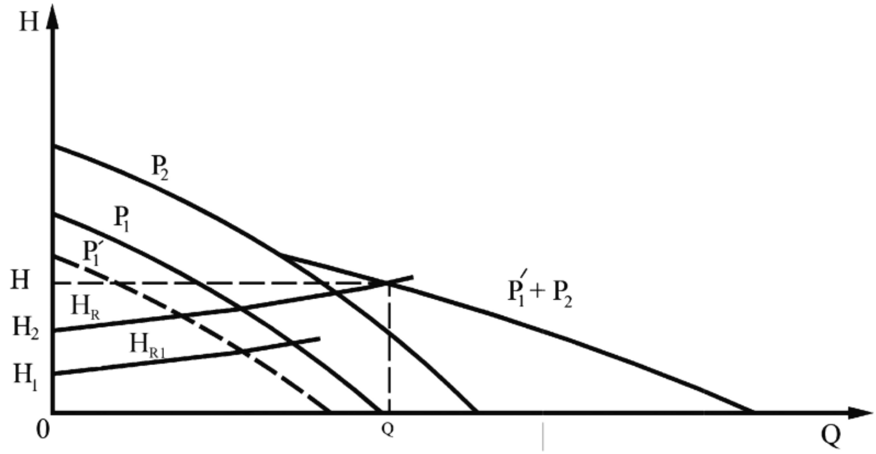


Fig. 8.22 The characteristic curves of pumps working together in parallel, located far from each other, corresponding to Fig. 8.21

a parallel fashion. The intersection point between this new curve, $P_1' + P_2$, and the system curve H_R would give the working point of the pump station. The characteristic curves of the system and the working point for each pump are shown in this figure as well.

For more information about piping systems, see [2, 3, 4]

8.6 Pumps Start Up Procedure

8.6.1 Priming

The centrifugal pumps and their suction pipes must be filled with pumped liquid before they are started. Otherwise, the suction is not done correctly and completely. This procedure is called priming.

In stations where pumps are installed in a lower level than the suction reservoir, pumps are always filled with the liquid. This is a normal practice for pumps with high specific speed and also those that need high values of positive suction head, $NPSH_{req}$ (the impeller of the axial flow pumps are usually submerged in the water). Whereas for pumps with negative suction pressure (i.e., the suction reservoir is located lower than pumps), priming is an essential procedure in start up.

There are different methods for priming. One of the least expensive practices is to install a special non-return valve at the beginning of the suction pipe (sometimes called foot valve) and then to fill the system only one time during the operation. These valves produce high level of pressure loss and leaking is a common problem among them such that after some time in service, the suction pipe must be de-aired each time before start up.

A better and more reliable method is to close the delivery valve of the pump and then take the air out through the air vent that is located at the highest point on the pump casing by a vacuum pump.

Another method for priming is to install and connect a tank filled with the pumped liquid on the casing or very close to the inlet (suction) of the pump. This tank is always filled with liquid before the pump start up and therefore keeps the suction pipe filled with water and prevents air from entering the system. In Fig. 8.23, a sample illustration of such set up is shown. At the start up time, the pump would take the water from this tank and the vacuum that is generated in the tank would cause the water to flow through the suction pipe to the tank. The design of the tank should be done such that before it is emptied completely the filling process starts. The effective volume of the tank, i.e. the distance between the filling and suction pipes, should be at least three times of the volume of the suction pipe (between the tank and the main suction reservoir).

There are also pumps in the market that are called "self-priming" pumps. These pumps have a special impeller that is installed at the suction side of the main impeller. At the beginning of the start up process, this impeller would discharge air from the casing and suction pipe of the pump. After the water fills up the pump casing, the main impeller starts to work. In Fig. 8.24, a sample of such impellers used in self-priming pumps is shown.

8.6.2 Start Up Time

The time that a pump needs to reach its nominal speed n from zero is called the start up time. This time is a function of the moment of inertia of all the rotating

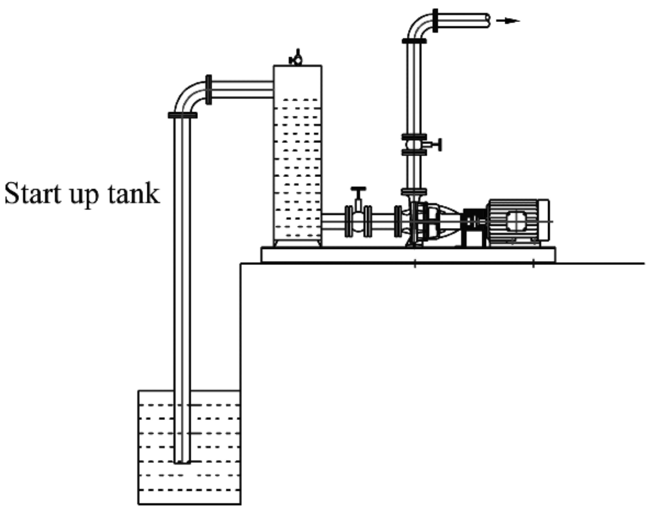


Fig. 8.23 A sample set up for a priming tank used for priming

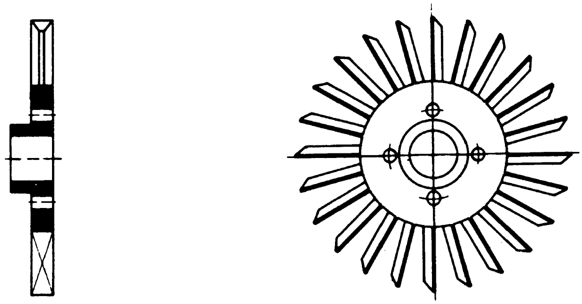


Fig. 8.24 Sample of a special impeller used in self-priming pumps [5]

components of the pump, rotating speed, and the difference between the starting torque of the pump T_S and torque of the driving motor T_m. This torque is used to accelerate the rotating mass of the pump–motor assembly. The start up time can be obtained, with a reasonable accuracy, from the following relation:

$$t_{St} = \frac{\pi W R^2}{30} \sum_{i=1}^{K} \frac{\Delta n_i}{(T_m - T_{Si})} \tag{8.8}$$

Where t_{St} is the start up time in seconds, W is the weight of all rotating parts of the pump, and WR^2 is the equivalent of moment of inertia for all rotating masses with respect to the axis of the rotation in kg m^2, n_i is the change in the rotational speed of the pump at time $t = i$ in rpm, and T_{Si} is the start up moment of the pump at time $t = i$ in N m.

8.6.3 Shutdown Time

The shutdown time of a pump is the time when the pump would stop completely from the nominal rotational speed of n. To obtain this time, one can use the above equation, substituting $T_m =$ zero:

$$t_{St} = \frac{\pi W R^2}{30} \sum_{i=1}^{K} \frac{\Delta n_i}{T_{Si}} \tag{8.9}$$

As it will be shown in the next chapter, the shutdown time is a very important factor in the occurrence of water hammer in the pump station. The longer this time, the longer the pump would reach to a complete stop. Therefore, the pressure waves would not be generated in the system. To increase the shutdown time, the WR^2 term in the above equation can be increased.

8.6.4 Characteristic Curve for Torque-Speed

In start up period for a pump, i.e. until the pump reaches to its nominal speed, it is important to know the relationship between the torque and the rotational speed. This factor is very important in selecting and operating the driving motor as well as the occurrence of the water hammer in the pipes.

The relationship between the torque and the speed for a pump handling a liquid with a specific density and viscosity is

$$P = T\omega \tag{8.10}$$

where P is the power and T is the torque. To plot the characteristic curve for torque-speed, the following parameters must be known:

1. Pump power P at zero flow rate.
2. Pump power P when the valve is open. For this parameter one of the following values should be selected:

    a. P at the point where flow rate is equal to 120% of the flow rate at maximum efficiency point.
    b. Power of the driving motor if it is more than the power of the pump at $Q = \frac{1}{2}Q_{opt}$.
    c. The maximum power on the characteristic curve P–Q, if it is more that the driver motor power.

3. Rotational speed of the pump.
4. Torque at $Q = 0$.
5. Torque at one of a, b, or c.

The two last parameters can be obtained at the nominal speed by using the power at $Q = 0$ and at one of the conditions a, b, or c. Since the absorbed torque is proportional to the square of the rotational speed, it can be easily calculated at different speeds.

Once the values of torques at different speeds are obtained, the characteristic curves for torque-speed, at two flow rates, zero and fully opened valve, can be plotted. To obtain the torque at rotational speed equal to zero, the values reported in Table (8.2) can be used based on the pump power.

In Fig. 8.25, typical characteristic curves of torque-speed obtained using the above method for three different centrifugal, mixed, and axial flow pumps are shown. As can be seen the torque for centrifugal pumps is always lower when the discharge valve is closed. For this reason, centrifugal pumps start up is done with closed valve. The start up of the axial flow pumps, on the other hand, is done with open valve, because the torque is always lower with discharge valve open.

Table 8.2 Coefficients for calculating torque [1]

Range of pump power at the design point		Multiple torque at the full speed by
Horsepower	kW	
Up to 1	Up to 0.746	0.5
1–10	0.746–7.46	0.15
10–50	7.46–37.3	0.1
50–100	37.3–74.6	0.08
Over 100	Over 74.6	0.05

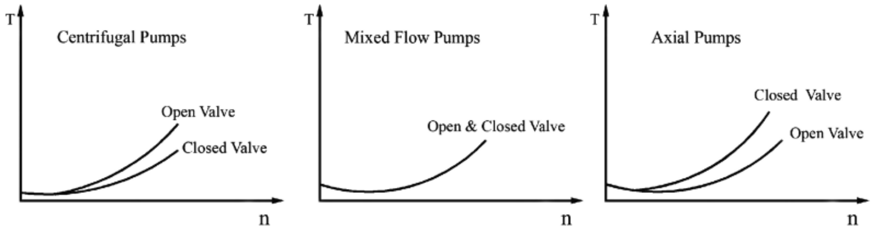


Fig. 8.25 Typical torque-speed characteristic curves

8.7 Temperature Rise at Low Flow Rates

If a pump operates at very low flow rates for a long period of time, the temperature of the liquid would rise inside the pump. The temperature rise would not only cause liquid evaporation inside the pump, but would also affect the performance of gaskets and the stuffing box of the pump. The temperature rise of the water inside the pump can be calculated from the following relation [7]:

$$\Delta T = \frac{H}{778C_p}\left(\frac{1}{\eta} - 1\right) \tag{8.11}$$

Where ΔT is the temperature rise in Fahrenheit, H is the pump head in ft, C_P is the specific heat of the liquid in Btu/lb-F, and η is the total efficiency of the pump.

In pumps that handle hot liquids, like the feeding pumps of boilers, ΔT is limited to 15 F (about 8°C). In pumps that work with liquids at normal temperatures, this temperature rise can be as high as 100 F (about 55 °C). In Fig. 8.26, the allowable temperature rise for pumps working with water is shown.

8.8 Minimum Flow Rate

In centrifugal pumps, by decreasing the flow rate, the absorbed power is also decreased. In some pumps, at flow rate equal to zero, this power would become less than half of the design power. At the same time, at very low flow rates, the tem-

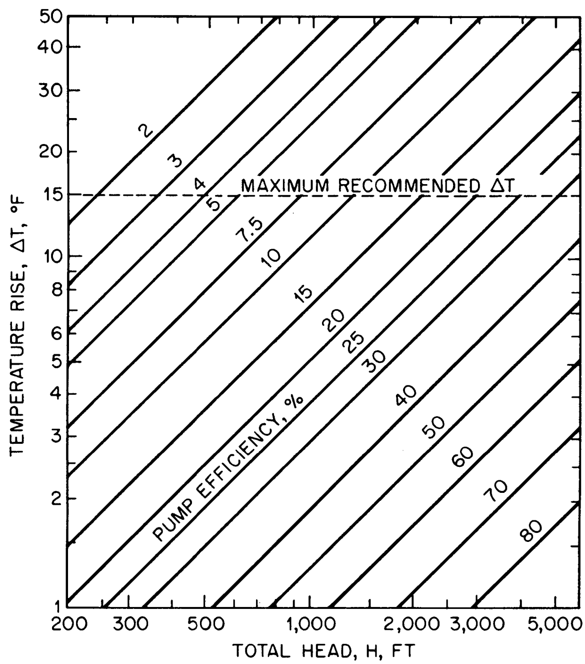


Fig. 8.26 Temperature rise in centrifugal pumps, pumping water (English Unit System) [6, 7]

perature of the pumped liquid would rise such that the evaporation can take place. Under these conditions, pumps would work with a two-phase fluid. To avoid this situation, usually there is a minimum flow rate defined for pumps that determines the maximum temperature rise in the pump. One of the proposed empirical relations for defining this minimum flow rate is as follows:

$$Q_m = 6P_0/\Delta T_m \tag{8.12}$$

Where Q_m is the minimum flow rate in Gpm, P_0 is the power at $Q = 0$ in horsepower, and ΔT_m is the allowable temperature rise in the pump in Fahrenheit.

References

1. Nourbakhsh S. A.,"Pump and Pumping", University of Tehran Press, 2004.
2. Mohinder L. Nayyar, "Piping Handbook", 7th ed., McGraw-Hill, New York, 2000.
3. Frankel M., "Facility Piping Systems Handbook", 2nd ed., McGraw-Hill, 2002.
4. Smith P., "Piping Materials Guide Selection and applications", Elsevier, 2005.
5. "Centrifugal Pump Design", Technical Appendix, KSB, Frankenthal, Germany.
6. Karassik Igor J. et al., "Pump Handbook", McGraw-Hill, 2001.
7. Karassik Igor J. et al., "Pump Handbook", McGraw-Hill, 1976.
8. "Basic Principles for the Design of Centrifugal Pump Installation", SIHI Group, 1980.

Chapter 9
Water Hammer

Pressure surge or water hammer, as it is commonly known, is the formation of pressure waves as the result of a sudden change in liquid velocity in a piping system. Water hammer usually occurs when a fluid flow starts or stops quickly or is forced to make a rapid change in direction. Quick closing of valves and stoppage of a pump can create water hammer. A valve closing in 1.5 s or less depending upon the valve size and system conditions causes an abrupt stoppage of the flow. Since liquid is not compressible, any energy that is applied to it is instantly transmitted. The pressure waves (acoustic wave) created at rapid valve closure can reach five times the system's working pressure. If not considered for, this pressure pulse will rapidly accelerate to the speed of sound in liquid, which can exceed 1200 m/s, causing bursting of the pipeline and pump casing as well as fracture in the pipe fittings. For this reason, it is essential to understand under what conditions these pressure waves are produced and reduce the pressure rise as much as possible in the piping system.

In this chapter the general information about the conditions under which water hammer generates in a piping system and methods to approximately calculate the pressure rise in the pipeline will be presented. The information in this chapter can be used only as guidelines and readers are encouraged to refer to other text books that are specifically dedicated to this complicated phenomenon. Moreover, the parameters that affect the water hammer and the common method for controlling this phenomenon, as well as methods to prevent the harmful effects of the pressure rise in the pipe will be presented in this chapter.

9.1 General Definition

The first significant contributions to the study of wave propagation in incompressible fluids were made by Newton and later by Laplace who related the speed of the sound in air to the pressure and density under the assumptions of isothermal and isentropic compression, respectively. Early attempts to analyze observed surge and water hammer effects were not directly based upon these fundamental equations, but were based upon the one-dimensional wave equation, originally derived and solved by d'Alembert around 1750.

The earliest application of the one-dimensional wave equation to explain observed water hammer effects was made by Joukowsky in 1898. Joukowsky correctly predicted the maximum line pressures and disturbance propagation times in a water distribution system in which sudden valve closures occurred. Joukowsky's equation is expressed as

$$\Delta P = \rho a \Delta V \tag{9.1}$$

where ΔP is the pressure rise due to the water hammer in $\mathrm{N/m^2}$, a is the velocity of impulse waves in m/s, ΔV is the velocity change of liquid in the pipeline in m/s, and ρ is the density of the liquid in $\mathrm{kg/m^3}$. The above relation can also be written as

$$\Delta H = \frac{a\Delta V}{g} \tag{9.2}$$

where ΔH is the pressure increase due to the water hammer in terms of column of water in meters and g is the gravitational acceleration in $\mathrm{m/s^2}$. In deriving the above equations the following assumptions were made:

- The friction losses are much smaller than the static pressure in the pipe.
- Flow is single phase and there are no dissolved gases in the liquid.
- The liquid velocity change occurs in a time less than the critical time. Critical time can be obtained from

$$t_r = \frac{2L}{a} \tag{9.3}$$

where t_r is the critical time which is defined as the time in which the pressure waves would reflect and L is the distance between the point at which the pressure waves are generated and the nearest point at which they would reflect.

The speed of the pressure waves, a, is a function of the following parameters:

1. Specific weight and elasticity module of the liquid.
2. Pipe diameter, wall thickness, and the distance between the support points.
3. The elasticity module of the pipe material.

The derived relation for calculating the pressure wave speed is as follows:

$$a = \sqrt{\frac{1}{\left(\frac{\rho}{K} + \frac{DC_1}{Ee}\right)}} \tag{9.4}$$

Where D is the pipe diameter, e is the pipe wall thickness, E is the elasticity module of the pipe material, K is the elasticity module of the liquid, and C_1 is a constant that can be assumed to be equal to one.

For the preliminary assessment, the speed of the wave can be obtained from Alievi equation which is expressed as follows (see Fig 9.1):

$$a = \frac{9900}{\sqrt{48 + \kappa \frac{D}{e}}} \tag{9.5}$$

Wave velocity, a, in iron pipes is between 800 and 1400 m/s. Using these values, a velocity change of 0.1 m/s in an iron pipe would cause a pressure rise equal to 10 m of water or 1 bar.

Usage of the Joukowsky equation can be limited due to the following reasons:

1. The actual time for the velocity change in the pipe is normally larger than the critical time, t_r. This will cause the actual pressure rise in the pipe to be smaller than the calculated pressure by Joukowsky relation.
2. The pressure losses in the pipes compared to the static pressure of the liquid are not usually negligible. This would normally cause the actual pressure rise to be higher than that from Joukowsky equation.

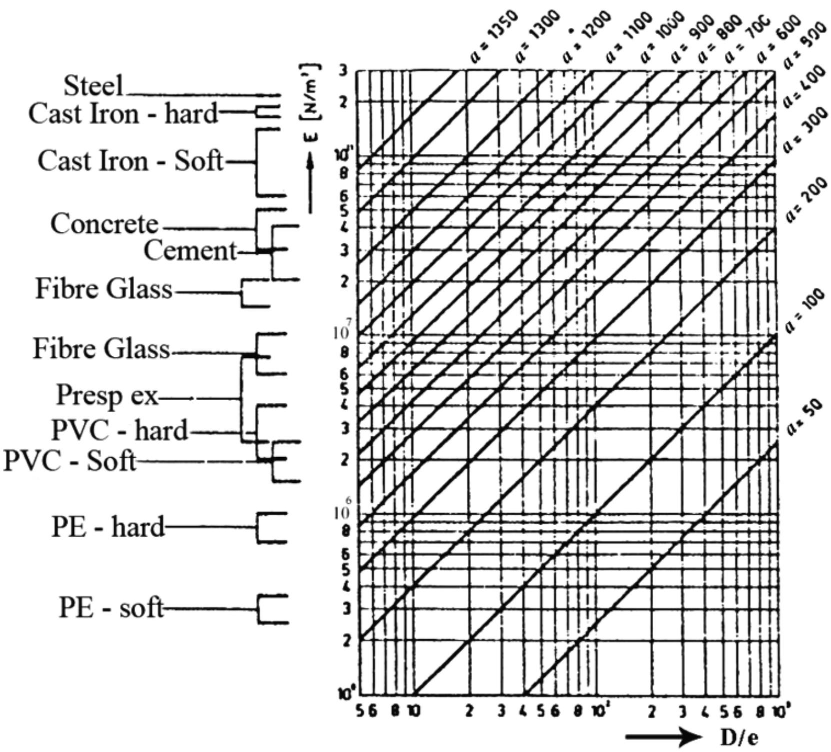


Fig. 9.1 Speed of wave propagation in different pipes carrying water [1]

9.2 Parameters Involved in Water Hammer Formation

Pressure waves in pipelines are generated due to different normal operations in the system such as opening and closing the valves, start up or shutting down the pumps, or any sudden change in the pump rotational speed. Meanwhile, accidental events can also cause water hammer in the pipes such as:

1. Shutdown of the driving motor due to the power failure.
2. Malfunctioning of the non-return valves and other control valves due to the voltage failure, stoppage of the motorized valves, or similar failures.
3. Any mechanical problems on the pump axis such as griping the ball bearings that would cause the pump to stop.
4. Blockage of the fluid flow in the pipe due to existence of solid particles.
5. Malfunctioning of movable parts of the valves and undesired vibrations.
6. Sudden damages to the impeller, such as corrosion due to cavitation.
7. Incomplete priming, such that some air remains in the pipe.
8. Poor design of the piping system.

9.3 Water Hammer in the Delivery Pipe

In this section the unstable hydraulic conditions that would exist in the delivery pipe, after a pump suddenly stops, will be explained. In Fig. 9.2, pressure, flow rate, and rotational speed changes during the formation of a pressure wave in a pipeline are shown.

The horizontal axis of the plot shows time in seconds, which starts from the moment the pressure waves are formed. On the vertical axis the percentage of the velocity, head, and the flow rate with respect to their normal values are shown. All three curves start from the normal working conditions, i.e. time zero and point 100%. As soon as the pump stops, the rotational speed starts to decrease. At this time the only energy that keeps the pump rotating is the kinetic energy (inertia) of the rotating elements of the motor, pump, and the moving liquid. Since this energy is much less than the required energy during normal operation of the pump, flow rate and head of the pump start to decrease at the same time.

As can be seen from Fig. 9.2, the pressure would drop until it reaches to its minimum after few seconds of the motor stoppage. At this point, the negative pressure waves, with pressures much lower than the normal working pressure of the pump (subnormal pressure waves), are formed. These subnormal surged waves would travel with very high speed along the pipeline toward the discharge reservoir. At the same time, the rotational speed continues to decrease to the point that no flow rate is delivered by the pump anymore, point “a” in Fig. 9.2. From this point, two different scenarios can happen.

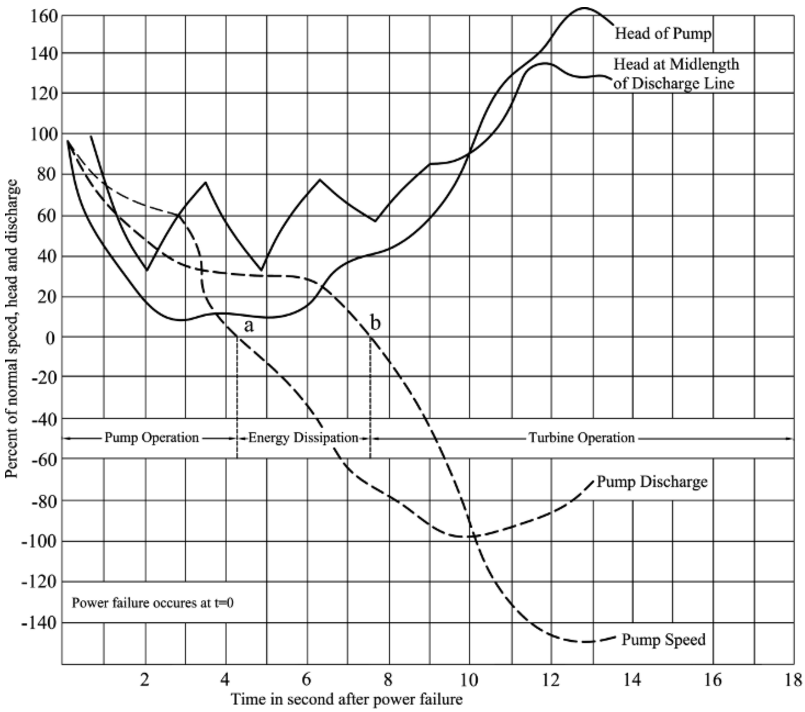


Fig. 9.2 Unstable conditions after pump shutdown. Time in seconds, after the stoppage of the motor [2]

Case-1: When There is No Non-return Valve on the Delivery Pipe

After point "a" since no flow rate is delivered by the pump and fluid flow stops in the discharge pipe and since there is no non-return valve on the line, the liquid would return toward the pump. Meantime, due to the inertia of different rotating parts of the pump, it continues to rotate in the same direction. Once the liquid returns to the pump, the rotational speed of the pump suffers a sharp decrease until it reaches to zero, point "b" in Fig. 9.2. The distance between two points "a" and "b," where the flow rate and the rotational speed become zero is called the "energy dissipation zone." In this region, despite the fact that the pump still rotates in normal direction, no flow rate is delivered by the pump and the manometric head remains at its minimum value.

From point "b," pump acts as a hydraulic turbine and rotates in reverse direction. At the same time, due to flow return to the pump, pressure would increase in the delivery pipe, close to the pump. This pressure rise would continue until pump reaches to its "runaway speed." At this time the maximum pressure due to the water hammer is produced in the delivery pipe which is called "up surge."

The runaway speed of a pump is a function of the maximum static pressure in the system, specific speed of the pump, and the efficiency of the pump. After the pump is reached to its runaway speed, the returned flow rate would remain in its maximum value, then it starts to reduce until it becomes zero. By looking at Fig. 9.2, three different working conditions for the pump can be recognized during the water hammer occurrence in the delivery pipe:

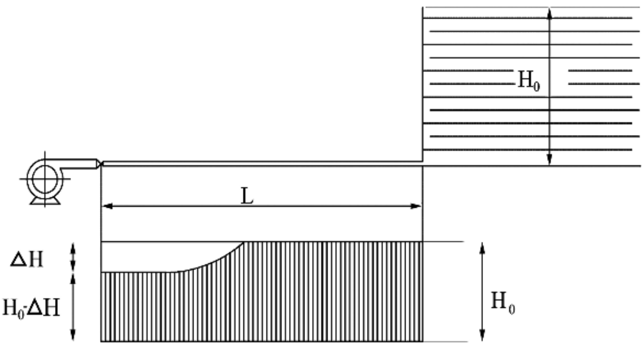
1. Pump operation: this period is between the moment the driving motor suddenly stops and the time in which flow rate becomes zero.
2. Energy dissipation: this period is between the time at which flow rate becomes zero and the pump stops (rotational speed becomes zero).
3. Turbine operation: this is the period between the time the normal rotational speed of the pump becomes zero and the complete shutdown of the pump.

Case-2: When a Non-return Valve is Installed on the Delivery Pipe

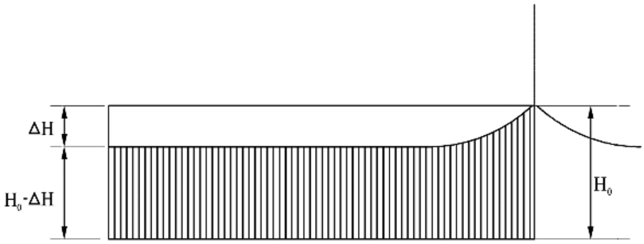
When there is a non-return valve on the delivery pipe (just after the pump), the moment liquid starts returning toward the pump, the valve will be closed and thus prevents the turbine operation of the pump. Also, as will be seen in the next section, in this case there would be no "water column separation," the maximum pressure rise would occur at the valve, and, therefore, the pump would be protected.

In Fig. 9.3, the pressure changes in a simple system, consisting of a pump, a non-return valve, a pipe with length L, and a discharge reservoir, during a motor shutdown are shown [1]. The pressure losses in the system are neglected and no liquid enters or exits the system. The static pressure in the pipe is H_0. The following steps, from (a) to (h), are observed after the motor stops (Fig. 9.3):

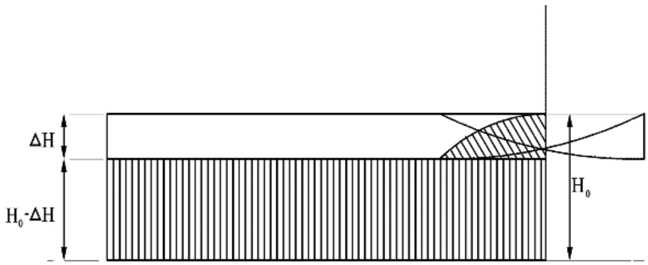
a. The pump is working normally.
b. When the pump stops suddenly, the flow velocity would soon become zero. Based on Jokowsky relation, this velocity change from V_0 (the initial velocity)



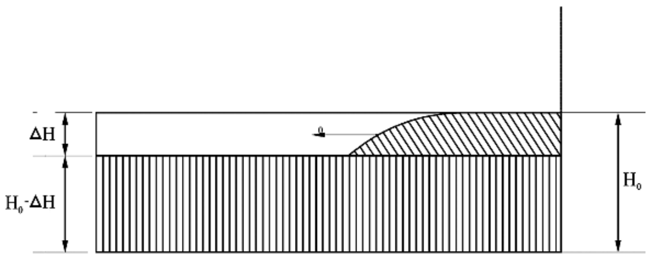
(**a**) Pump is delivering a constant flow rate through a pipe with length L
(**b**) Pump stops suddenly or the valve is closed rapidly. A pressure wave with subnormal pressure of (-ΔH) would start to move toward the discharge reservoir
(**Fig. 9.3** continued)



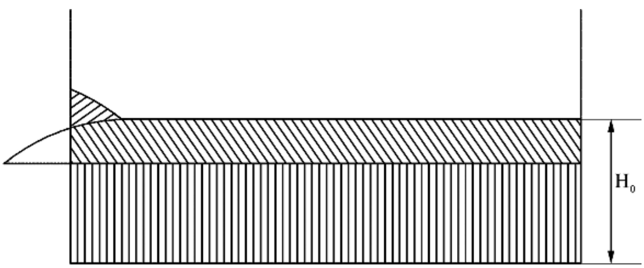
(**c**) The sub-normal wave reaches the reservoir



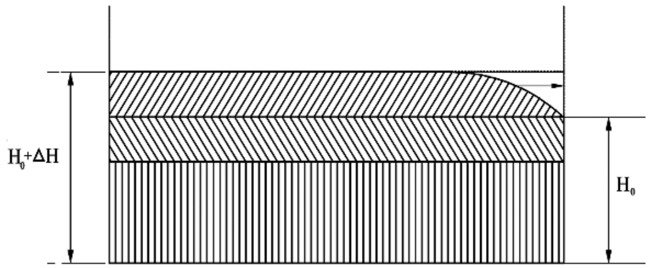
(**d**) The sub-normal pressure wave is now a wave with high pressure and moves back from the reservoir



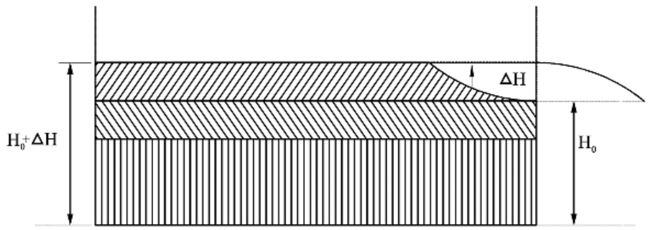
(**e**) The high pressure wave is moving toward the pump



(**f**) The high pressure wave reaches the closed valve and returns with a pressure equal to $H_0+\Delta H$
(**Fig. 9.3** continued)



(**g**) The high pressure wave moves toward the reservoir



(**h**) The high pressure wave reaches the reservoir and again returns back toward the pump with a subnormal pressure. This cycle repeats from step (b)

Fig. 9.3 A schematic diagram of pressure waves in the pipeline after a sudden pump shutdown (or a valve shutdown)[3]

to zero would decrease the static pressure of the liquid in the delivery pipe to $(H_0 - \Delta H)$, see Fig. 9.3b, where

$$\Delta H = \frac{a\Delta V}{g} \tag{9.6}$$

c. The pressure wave with a pressure of ΔH less than the normal pressure would travel, with a velocity of a, toward the discharge reservoir. The static pressure in the discharge reservoir is equal to H_0 which is constant, since the height of water in the reservoir has not been changed. The pressure wave is confronted with a liquid with higher pressure as H_0, see Fig. 9.3c.
d. When the pressure wave reaches the reservoir, ΔH becomes zero, Fig. 9.3d.
e. The pressure wave returns toward the pump, Fig. 9.3e.
f. Once the pressure wave reaches the closed non-return valve, its speed becomes zero and its pressure rises as high as ΔH, Fig. 9.3f.
g. The pressure wave with a pressure of $(H_0 + \Delta H)$ travels toward the discharge reservoir again, Fig. 9.3g.
h. By the time this wave reaches the reservoir, due to the fix pressure of the tank, i.e. H_0, its pressure reduces to H_0 and returns back to the pump, Fig. 9.3h.

This cycle, steps “a” to “h,” will repeat until the wave energy is totally dissipated. To avoid any damages arising from the pressure fluctuation in this period, proper types of control devices should be installed on the delivery pipe.

In general, the hydraulic transient conditions which would occur after the shutdown of the driving motor depend on three factors:

1. the inertia of the rotating parts of motor and pump
2. characteristic curves of the pump
3. characteristic curves of the delivery pipe.

In Fig. 9.4, the wave propagation diagrams in the pipeline after a sudden stop of a pump, considering the pressure losses in the pipe, are shown. The difference between Fig. 9.3 and the latter figure is the slope of the hydraulic line in Fig. 9.4, which is an indication of the energy loss in the system.

9.4 Water Column Separation

Water column separation is one of the consequences of water hammer which occurs during the subnormal pressure period in the delivery pipe. The problem can be expressed as follows.

When the pump stops suddenly, the subnormal pressure waves created at the pump location travel with a high speed toward the end of the pipe. If the profile of the pipeline and the hydraulic conditions of the flow are such that at some point the pressure in the pipe becomes lower than the vapor pressure of the liquid (water), water is evaporated and two bulks of liquid would be separated at the middle. This phenomenon is called the "water column separation."

If this phenomenon occurs, in the next period when the high pressure waves move toward the pump, pressure would increase at the same location causing the vapor to be condensed rapidly. Due to rapid volume reduction in this location, two columns of water would collide with each other. This would create a very high pressure rise in the pipe and it could even result in serious damages to the pump station, pipeline, or other installed equipments.

The water column separation usually occurs at the highest points of the systems, close to the hydraulic line. An example is shown in Fig. 9.5.

9.5 Water Hammer and Important Parameters

9.5.1 Pumping Systems with High or Low Pressures

The water hammer phenomenon has more consequences in the pumping systems working at low pressures than in the systems with higher pressures. The reason is that in both systems the flow velocities in the pipes are in the same order of magnitude. Since the pressure rise due to the water hammer is proportional to the velocity change in the pipes, the pressure rise in both high and low pressure systems are in the

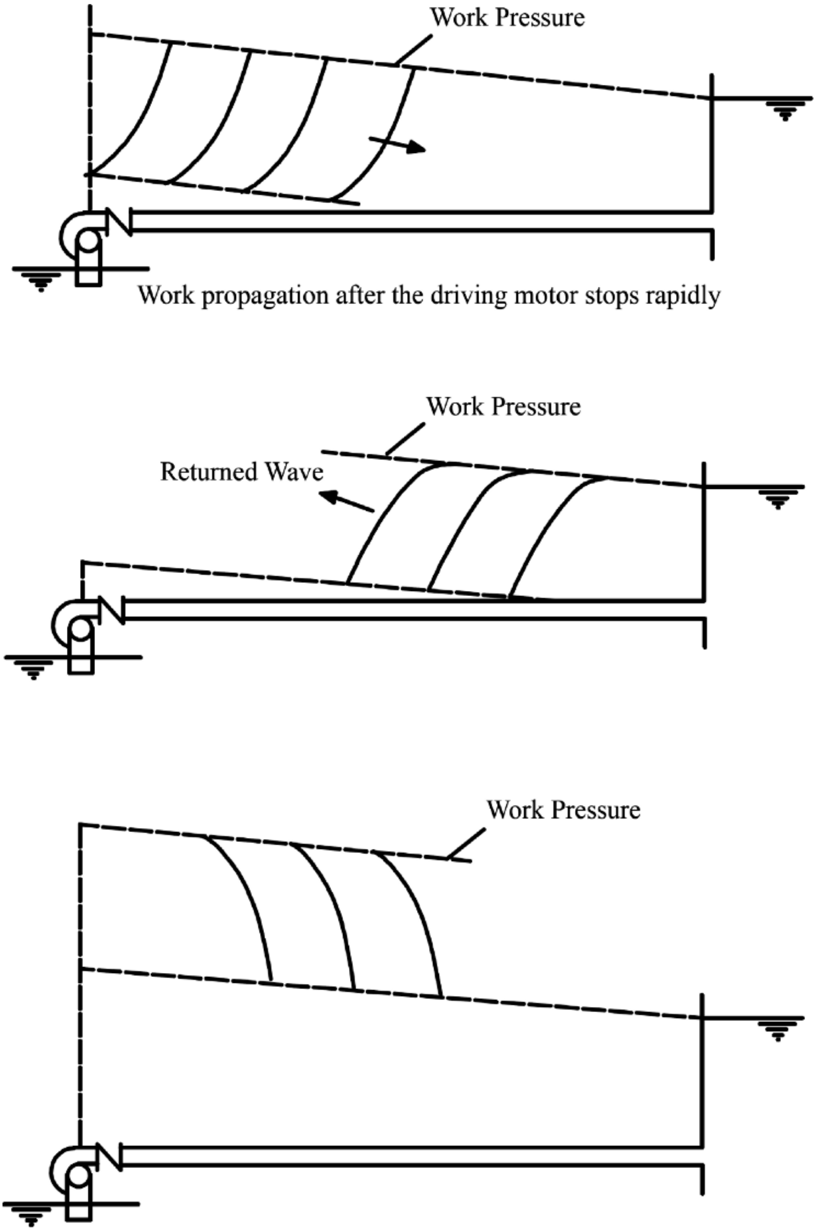


Fig. 9.4 Wave propagation after the driving motor stops rapidly in a pipe with friction loss

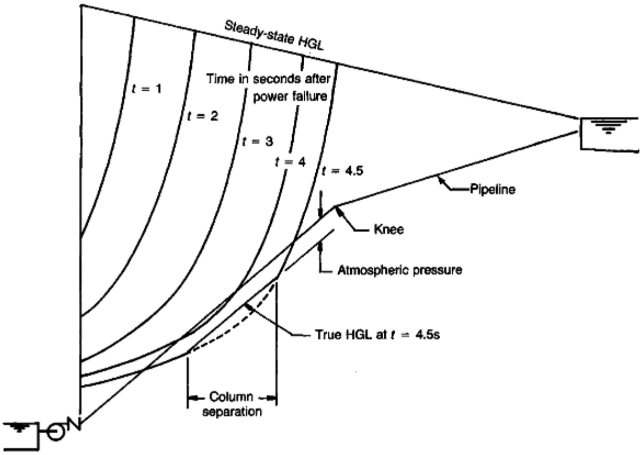


Fig. 9.5 Profile of a discharge pipe and the subnormal pressure curves at different times after the pump has stopped [4]

same order of magnitude as well. However, in low pressure systems all equipments, pipe, and fittings are designed based on a low anticipated pressure. Therefore, the pressure rise due to the water hammer may cause more serious damages in these systems.

9.5.2 Layout of the Discharge Pipe

In a pumping system, the layout of the discharge pipe is designed based on many parameters such as technical and economical requirements, as well as the geological consideration. However, when considering many possible design schemes, one specific requirement is also very important: When the driving motor of a pump stops suddenly, the profile of the maximum pressure line along the pipeline would have a concave shape, as shown in Fig. 9.5. The water hammer effect would then be less severe in pipelines that are also concave shaped. Especially, none of the points on the pipeline should locate above the maximum pressure curve when plotted in the same system of coordinates. For this reason, when possible, the discharge pipes of the pumping systems are installed under the ground.

As mentioned before, the minimum and maximum pressure curves during a water hammer episode depend on many items such as the characteristic curve of the system. Therefore, the water hammer curve is not independent of the profile of the pipeline and once a profile for a pipe is designed, the pressure curves must be drawn and the critical points of it must be identified.

9.5.3 Pipe Diameter

The diameter of a pipeline is selected based on the required flow rate and velocity, as well as the economical considerations. It must be noted that by increasing the diameter, the flow velocity would decrease which in turn would reduce the effect of the pressure waves. This is one of the solutions to control the harmful effects of the water hammer on the pipes. Although this would increase the cost of the pipeline, in some cases the expenses are less than the expensive controlling devices that must be used for preventing water hammer.

9.5.4 Number of Pumps

The number of pumps that are connected to a discharge line in a parallel arrangement are determined based on the required flow rate, characteristics of the pumps, and the installation conditions. Nevertheless, the number of pumps connected to each discharge pipe, also affects the way water hammer occurs in the pipe.

By looking at the characteristic curve of several pumps that are connected in parallel, one can see that as the number of pumps is increased, the pressure rise supplied by each pump is decreased (see Sect. 8.3.1). This would also reduce the pressure impacts caused at the start up period.

When one of the pumps suddenly stops, or when one of the non-return valves fails to operate properly, the existence of several pumps in the line is more advantageous. One reason is as the number of pumps increases, the flow rate delivered by each of them reduces and the flow rate in the discharge line would be less affected.

But if all pumps stop to work simultaneously, like in a power failure scenario, the lesser the number of the pumps connected to the pipe, the smaller the pressure variations and other hydraulic transient effects. This is because to deliver a specific flow rate, by increasing the number of pumps, the size of each pump is smaller. This means the kinetic energy of the rotating parts would be also considerably smaller (less inertia to continue rotation). Whereas with less number of pumps, the pumps are bigger and thus their kinetic energies are larger. Therefore, to deliver a specific flow rate, the velocity changes and the effect of the water hammer would be minimal if there is only one pump connected to the discharge pipe.

9.5.5 Flywheel Effect

When a pump stops suddenly and if the kinetic energies of the rotating parts are relatively large, the flow rate changes more slowly in the discharge. For this reason, one of the methods to reduce the effects of the water hammer is to increase the "flywheel effect" or the moment of inertia of the rotating parts of the pump.

Usually, the driving motor provides about 90% of the total moment of inertia of the pump–motor assembly. To increase this effect, a flywheel can be installed on the axis of the motor. This method is very useful in small pump stations and would reduce the necessity to use much more expensive solutions compared to other pressure control systems. In Fig. 9.6, the effect of increasing the inertia on the pressure waves, by using a flywheel, is shown.

9.5.6 Specific Speed of the Pump

As mentioned before, the hydraulic transient conditions due to the water hammer depend on many parameters such as the characteristic curves of the pump and the way they change during the pressure wave propagation period. For this reason, pumps with different specific speeds would show different behaviors when flow returns to the pump.

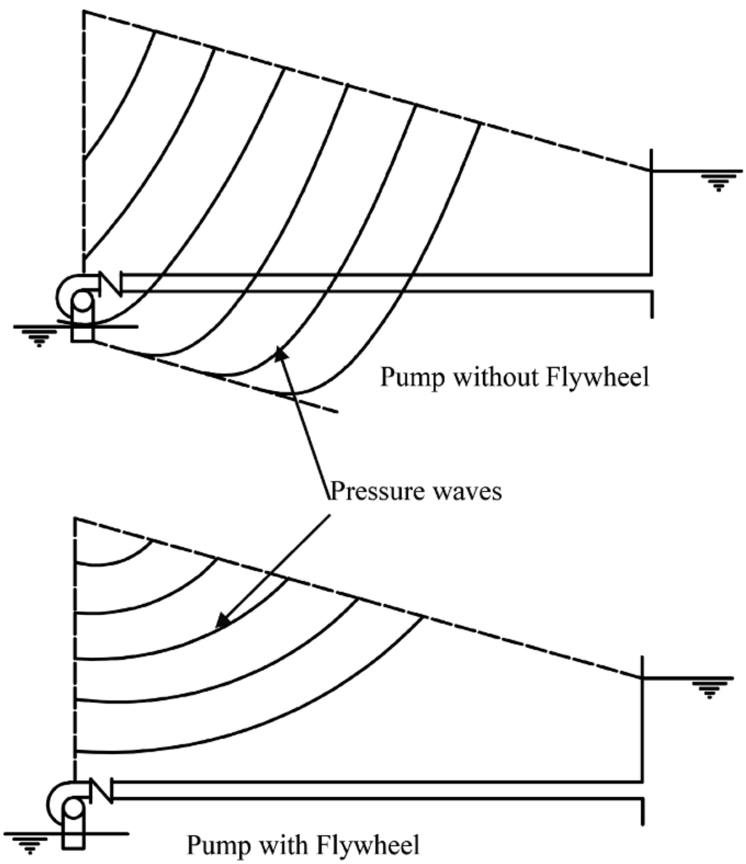


Fig. 9.6 Flywheel effect [1]

In general, for a unique discharge pipeline under the same initial conditions, the maximum pressure rise that is formed during the flow return toward the pump, see Fig. 9.3e, depends on two parameters:

1. The maximum flow rate that can flow through the pump in energy dissipation regime, when the pump works as a turbine, see Fig. 9.2.
2. The flow rate that passes the pump when it works as a turbine in "runaway speed."

When the flow rate moves in the opposite direction through the pump, the centrifugal pumps (low specific speed) would supply higher pressures than the axial flow pumps and the mixed flow pumps (this is only true if the water column separation does not occur in the line). Also during the sudden shutdown of the driving motor, the subnormal pressure due to the water hammer in the centrifugal pumps is slightly more than in the axial (high specific speed) or mixed flow pumps.

However, when in a pumping system with no non-return valve the water hammer occurs, the reverse rotational speed in the axial flow pumps is higher than the centrifugal pumps. For this reason the driving motors of the pumps with high specific speed must be protected against any damages during the water hammer period.

9.6 Water Hammer Controlling Methods

To prevent the severe pressure rise during a water hammer occurrence, the following methods can be used.

9.6.1 Design the Discharge Pipe Based on Lower Liquid Velocities

As mentioned before, since the pressure rise due to the water hammer is directly proportional to the velocity change in the pipe, by decreasing the flow velocity, the effect of the water hammer will be minimized.

9.6.2 Increasing the Moment of Inertia of the Pump

Adding a flywheel on the rotating axis of the driving motor would prevent the rotational speed to reduce sharply and therefore restrain the excess pressure decrease or increase. This method is usually economical for small pump stations and the discharge pipes for up to 3 km.

9.6.3 By-Pass Pipes

One of the simpler methods to prevent the damaging effects of the water hammer is to install a by-pass pipe with a non-return valve, as shown in Fig. 9.7.

Under normal conditions, the pressure supplied by the pump would keep the non-return valve closed. However, after the shutdown of the pump, pressure will be decreased in the discharge pipe and once it becomes less than the suction pressure, the non-return valve will open and the liquid would enter from the suction pipe to the discharge pipe thereby preventing more pressure reduction. This method may be used in systems in which the supplied head of the pump is not very high.

9.6.4 Surge Tanks

These tanks act as a reservoir to suppress the pressure waves and are installed on the discharge pipe. When the pressure in the pipe increases, liquid enters the tank and is stored there. During periods of subnormal pressure in the pipe, then, the liquid would flow back to the pipe, preventing rapid velocity changes.

These tanks are designed to work in one-way or two-way arrangement. In two-way designs, as mentioned before, the tank is working at both high and low pressure periods. This design would be used in the piping systems in which the height of the hydraulic line is almost the same as the pipeline itself (Fig. 9.8).

In one-way surge tanks, liquid would only enter the discharge pipe from the tank during the subnormal pressure period. A non-return valve would prevent the liquid from flowing into the surge tank during the high pressure period (Fig. 9.9). These tanks are used in systems in which the height of the hydraulic line of the pipe at some locations is higher than the pipeline itself. The one-way surge tanks are very safe devices to prevent the water column separation in the pipe. To keep the surge tank filled with liquid, other auxiliary equipments such as floating valves must be used.

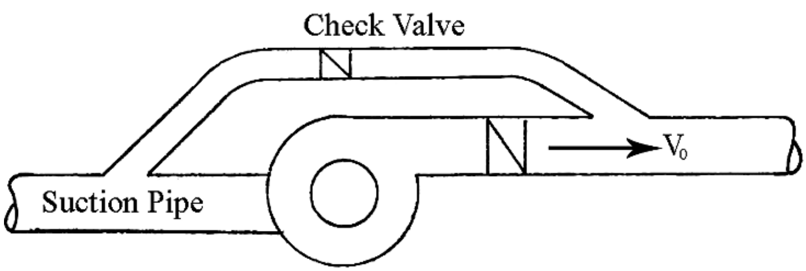


Fig. 9.7 By-pass pipe with a non-return valve

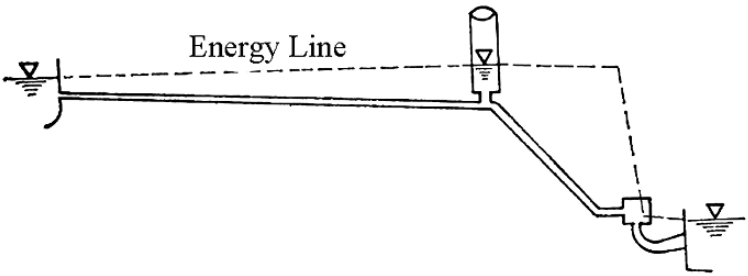


Fig. 9.8 Application of a two-way surge tank

9.6.5 Air Chambers

Air chambers are basically a type of high pressure surge tanks which can be built in smaller sizes. In these tanks, the pressurized air locates on the top of liquid, as shown in Fig. 9.10.

The design of the pressurized tanks must be done carefully. The size of the chamber must be large enough to compensate the liquid in the subnormal pressure periods without allowing air to enter into the system. The volume of the air must be chosen such that during filling period of the pipes, its pressure does not change significantly. If the volume of the air is small, the pressure fluctuations are high and this would not help preventing water hammer.

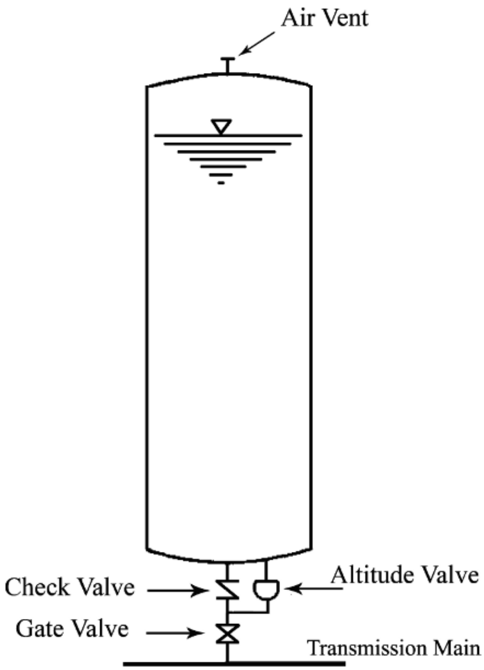


Fig. 9.9 Schematic of a one-way surge tank [5]

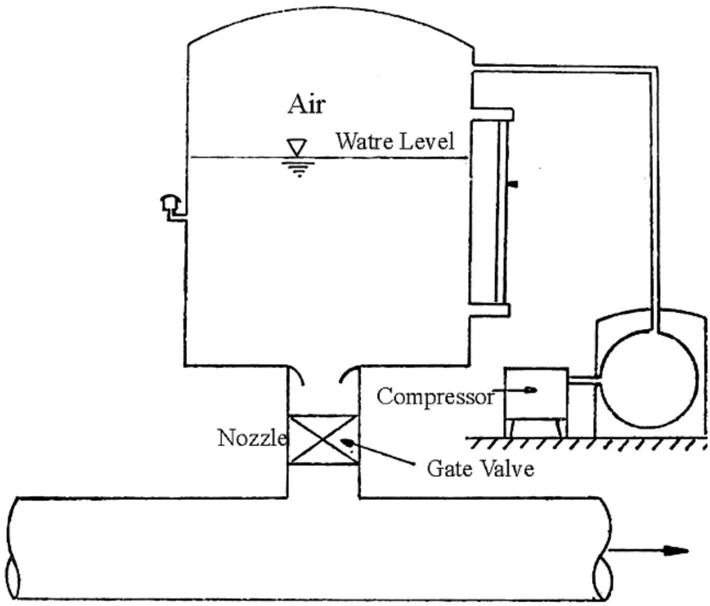


Fig. 9.10 Schematic of an air chamber with corresponding control equipment [4]

The large volume of the air would also increase the cost of the equipment. The pressure change in the air depends on the initial volume of the air, the discharged water, and the thermodynamic process in the tank. The pressure changes usually are calculated from a poly-tropic relations, i.e. $PV^n = cte$, where n is between 1.2 and 1.4.

To enhance the effect of the pressurized tank as a pressure damper, other devices which make artificial pressure loss may be used at the opening of the tank. Experience has shown that if the energy loss of the liquid at the time of entering the tank is more than the pressure loss when water flows back to the pipe, the pressure fluctuations would dissipate faster following several waves' travels.

One of the methods to increase the pressure loss in return period is to put a converging nozzle at the entrance of the tank, see Fig. 9.10. Another method is to install a by-pass pipe with a non-return valve or an orifice in the same place. When liquid flows from the smaller diameter pipe into the tank, its pressure loss will increase. By adjusting the valve on the pipe, the pressure loss in the by-pass pipe can be kept in a desirable level. In Fig. 9.11, such arrangement is shown.

9.6.6 Non-return Valves

The discharge pipes of the pumps are normally equipped with non-return valves. The main application of these valves is to prevent the flow running toward the pump

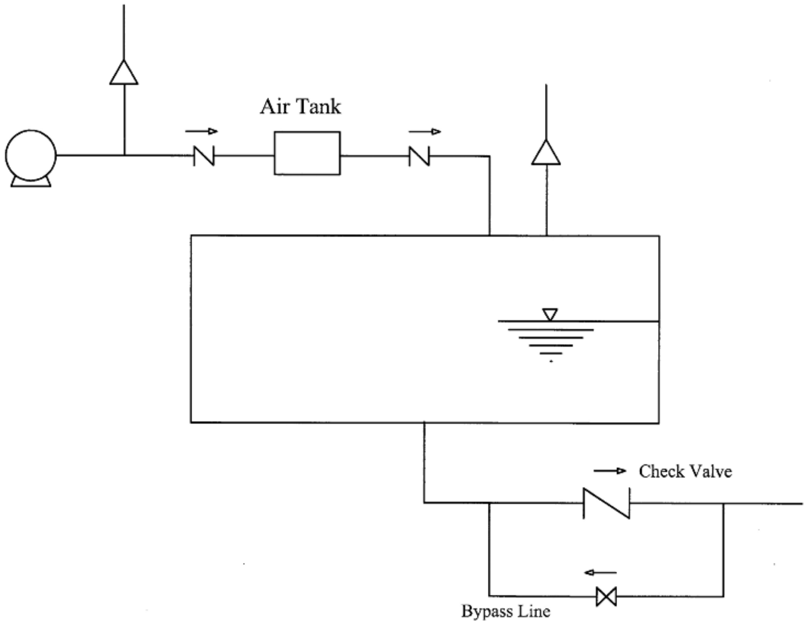
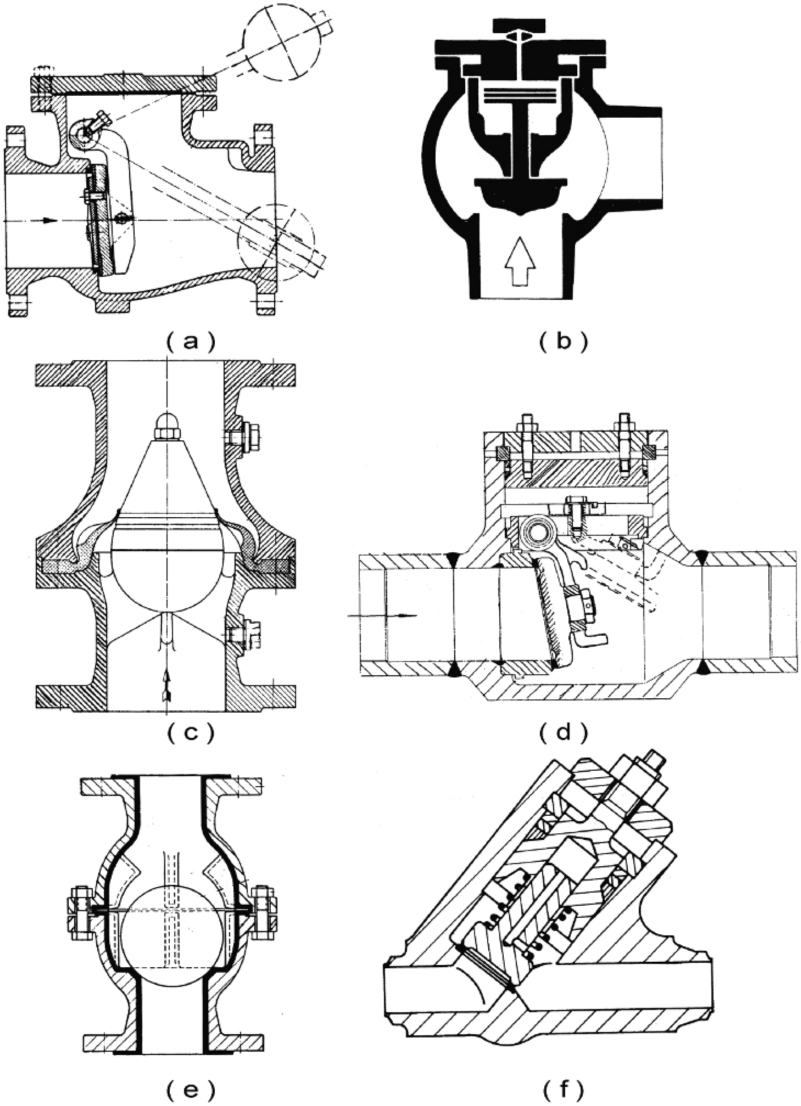


Fig. 9.11 Air chamber with by-pass pipe

when it stops, thereby reducing the adverse effects. During normal working conditions of the pump, the supply flow would keep the non-return valve open. Upon sudden stop of the pump, the flow rate would reduce rapidly until it reaches zero and would then flow back to the pump. Once the liquid flow is reversed, the disk of the valve is sharply closed, causing an intense impact on the valve seat. This would create more pressure waves. Neglecting the pressure losses, this pressure rise is the same as the subnormal pressure produced when the flow returns back to the pump.

The ordinary non-return valves usually close completely, few moments after the liquid starts to flow in the opposite direction (the short delay is due to the friction in the hinges, inertia of the disc, or malfunctioning of the valve itself). The sudden closure of the valve would cause a very intense impact on the seat that may cause more damages. To avoid this situation, other types of non-return valves can be used, such as those which do not close immediately or the valves that are closed gradually. The former valves would close before the reverse flow starts and therefore prevents any harsh impact. To close the disk in these valves a special spring or a weight is used. In the second type of valves, in the last few steps of the closing action, the valve seat moves slowly and in a controlled way, thereby avoiding any hard impact.

In Figs. (9.12) and (9.13), few samples of non-return valves are shown. The non-return valves with diaphragm are the best choice in terms of operation during a water hammer occurrence.



(a): Non-return valve with lever and weight
(b): Slow closing non-return valve
(c): Non-return valve with diaphragm
(d): Non-return valve with disc
(e): Non-return valve, ball type
(f): V shape non-return valve with spring

Fig. 9.12 Samples of non-return valves [1]

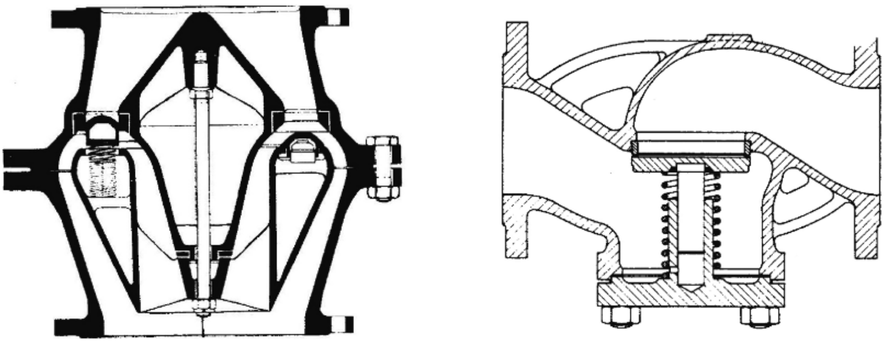


Fig. 9.13 Samples of non-return valves that close rapidly [1]

9.6.7 Pressure Control Valves

These valves are designed to open at very high pressures and are installed at the critical points of the piping system. During the pressure rise period, the valve would release liquid to the outside. This would reduce the pressure in the line and prevents any possible damages.

9.6.8 Vacuum Valves

These valves are installed on those points of the piping system in which there is a possibility of liquid evaporation due to subnormal pressures. When the pressure reduces beyond certain level in the pipe, these valves close and let the atmospheric air to enter the system.

9.6.9 Special Brakes

These devices are installed on the axis of the pump and would prevent the pumps and motor to rotate in the opposite direction. Along with this device, a by-pass pipe would discharge some of the liquid from the discharge pipe until the effect of the pressure wave is dissipated. In this way pressure rise due to the water hammer will be avoided. The usage of these brakes is not recommended on the small pumps, because the sudden stop of the pump axis could create some mechanical problems.

In Table (9.1), a guideline is presented that shows different methods for preventing water hammer effects. It is obvious that choosing the best method depends on the hydraulic and physical characteristics of the system and any system must be investigated separately and thoroughly for providing the best solution.

For more information about water hammer, see also [6, 7, 8, 9, 10, 11, 12, 13, 14].

Table 9.1 Guideline for selecting the method for preventing water hammer. The parameters in the table are all defined in the text

Method	Criteria for selection	Application
Increasing the moment of inertia of the pump (flywheel)	$\frac{WR^2N^2}{\gamma aLH_0^2} > 0.01$	For short pipelines. WR^2 is the moment of inertia of all rotating parts of the pump and γ is the specific weight of the liquid.
By-pass pipe with non-return valve	$\frac{aV_0}{gH_0} > 1$	When the supplied head of the pump is much less than $\frac{aV_0}{g}$.
Two-way surge tank	Low initial head, H_0	When the height of the hydraulic line with respect to the pipeline is not high.
One-way surge tank	$\frac{aV_0}{gh} > 1$	h is the equivalent height of the pressure of the surge tank. When the pipeline profile has a positive slope.
Safety valves	$\frac{aV_0}{gh} < 1 \& \frac{2L}{a} > 5\,s$	When the pipeline profile is downward and the possibility of the water column separation exists.
Air chamber	$\frac{aV_0}{gh} < 1$	For almost all cases.

References

1. Wijdieks J., “Water Hammer”, Delft Hydraulics Laboratory, June 1983.
2. Parmakian J., “Water Hammer Analysis”, Dover Publications, 1963.
3. “Planning of Centrifugal Pumping Plants”, Pump Division, Sulzer Brothers Limited, Winterthur Switzerland.
4. Watters G. Z., “Analysis and Control of Unsteady Flow in Pipelines”, 2nd ed., Butterworths, Stoneham, 1984.
5. Sanks R. L., “Pumping Station Design”, Butterworth- Heinemann, 1998.
6. Wijdieks J., “Practice and Criteria of Pump Sump Model Investigation”, Delft Hydraulic Laboratory, 1985.
7. Popescu M., “Applied Hydraulic Transients”, Taylor & Francis Publications, 2003.
8. Smith P., Zappe, R., W., “Valve Selection Handbook”, 5th ed., Elsevier, 2004.
9. Streeter V. L., “Fluid Transient in Systems”; Wylie, E. B., Prentice Hall, 1993.
10. Rich G. R., “Hydraulic Transients”, Dover, New York, 1963.
11. Skousen Philip L., “Valve Handbook”, McGraw-Hill, 1998.
12. Tullis J. Paul, “Hydraulics of Pipelines: Pumps, Valves, Cavitation, Transients”, John Wiley & Sons, 1989.
13. Sharp B. B., “Water Hammer Problems and Solutions”, Edward Arnold Publishers Ltd, London, 1981.
14. Neshan H., “Water Hammer”, PUMPIRAN Co., Tehran, 1986.

Chapter 10
Pump Stations

In a pump station, in addition to the pumps and driving motors, there are other equipments such as piping, electrical, control, and safety systems. In small pump stations, these auxiliary equipments may be designed very simple for economical reasons. But in larger pump stations, the technical complexity and requirements for design of auxiliary systems would be increased.

In designing pump stations, the following general parameters must be considered:

1. Operational methods and technical issues
2. Initial investment for the installation
3. Operational costs.

Economical evaluation for the pump stations will be done based on the above parameters. A designer usually considers all technical and economical issues before selecting one or many alternative designs to start the project. At first, one or two alternatives are investigated in more details and then in the second step the detailed drawings and technical specifications of the systems and equipments are provided.

In the following sections, the major parameters to be considered during the pump station design as well as the hydraulic parameters that could affect the pump performances will be discussed in more details.

10.1 Operational Methods and Technical Issues

The safety and easy operation of a pumping system must always be considered by a designer. The geographical situation of the pump station and the possibility of hiring expert personnel to maintain and operate the station are also among important parameters that have to be considered when designing a station. For example, once the capacity of the pump station is determined, by using a larger number of smaller pumps to deliver the same flow rate, the size of the units is reduced; therefore, the maintenance and operation of each system is less complicated. However, this would increase the cost of the project.

The selection of manual, semi- or fully automatic systems for controlling the equipment must be done also having these parameters in mind. The ability of using spare parts that are standard productions of the Manufacturers would be very important during operation time. The geographical location of the pump station is one of the most important parameters. One cannot use the design of a specific pump station to build another pump station in a different location. An experienced designer usually starts a particular design considering the socio-economical conditions of the location of the pump station, the anticipated operational methods, and the selected equipments which are also dependent on the latter parameter.

In this section, the technical consideration for designing a pump station will be discussed in more details. It must be mentioned that most of the technical issues that are discussed in this section pertain to the water supply stations.

10.1.1 Type of Pump Station

The method of suction from the suction reservoir is an important parameter in selecting the type of pump station. Horizontal pumps are used more commonly in pump stations. These pumps are installed on a fixed foundation, with suction pipe connected to the suction reservoir. The reservoir can be located at a higher or lower level than the pumps, depending on the required NPSH (see Sect. 4.4). Though this is the simplest arrangement for a pump station, in many cases this solution is not technically or economically viable.

For example, when it is not possible to control the level of the liquid (water) in the suction reservoir (like seasonal changes of the water level in the natural reservoirs or dams), the submersible pump stations are the best choice. In these types of stations, the station would move as the water level goes up or down. In some cases, using vertical pumps is more practical than using horizontal pumps. In this case, depending on the condition of the site, the driving motor can be installed under the water (submersible pumps) or above the water. Submersible pumps can also be installed horizontally. These pumps can be installed on the river beds or irrigation canals by constructing suitable basins. In this way the constructional expenses of the station would be reduced significantly.

In general, the types of pumps selected for a particular station directly depend on the construction costs of the station. For this reason, it is always a good idea to compare many alternative designs before making the final decision.

10.1.2 Pumped Liquid

The type of liquid that is pumped is a major parameter in selecting the type of pump and technical specifications. The existence of suspended particles in the water, their sizes, and the corrosive properties of the water would determine the type of impeller and the material of different parts of the pump. The material of the pump for pumping the sea water would be different when it is used to pump industrial sewage water. The existence of the solid particles in water may even imply such limitations that the usage a turbopump would not be feasible any more. In some cases, provisions must be sought to filter the water before entering the pump. To select the proper filtering system, the maximum allowable particle size that can pass through the pump must be determined by the pump manufacturer. It should be mentioned that there are turbopumps that are specifically designed to carry liquid with large-sized suspended particles. However, these pumps are usually more expensive and less efficient. The costs for constructing the filtration system or sedimentation basins would affect the decision-making about the pump type and the percentage of allowable particles that can pass through it.

10.1.3 Head, Flow Rate, Efficiency, and NPSH

For any pump station, the required flow rate and total head is known, leaving the designer to decide about other parameters. The geometrical head is an important parameter in selecting the pump type and the pumping method. Normally, single stage centrifugal pumps can supply heads up to 90 m. To deliver liquids with higher pressure, the multi-sage pumps can be used. Naturally, this would require equipments with higher working pressure and higher costs for pipes and fittings.

In some cases the required head is so large that pumping must be done in multiple stages. In these conditions, multiple pump stations would be installed in series. This type of arrangement is usually seen in oil pipelines. A designer must consider multiple designs with respect to the number of pump stations and their locations. When pumping is done in a series of stations, in achieving synchronized operation of the pumps and pump stations is very important. Sometimes, to avoid any problems and to have more flexibility, several discharge reservoirs are constructed at different levels. This will increase the construction costs, as well as the required maintenance for the reservoirs. However, this solution may be feasible in specific cases, e.g. for irrigation purposes where the quality of the water and therefore the type of the water reservoir are not very important factors.

The required pumping head is obtained from adding the pressure losses in the pipe to the geometrical height between the discharge and the suction reservoirs. Determining the pressure losses in the pipeline is one of the important tasks in designing a pipeline. If the pipe diameter is chosen large, the piping costs will increase at the expense of lowering the pressure losses. This would reduce the required head and therefore the size of the pumps. Consequently, this would reduce the purchasing and installation costs of pumps and other auxiliary systems like electrical equipments. It is obvious that the required power for the electro-motors and annual electrical costs would decrease as well.

Another parameter in determining the pressure losses is the anticipated age of the pump station. Pressure loss depends on the roughness of the internal pipe walls and this parameter is not the same for new and old pipes. As the life of the pipeline is increased, the roughness of the walls is changed and the hydraulic characteristic of the pipe will change correspondingly. In some pipes with increase in the service life, the pipe diameter becomes smaller due to the solid deposits on the walls. This will increase the pressure loss. The result would be that following few years of service life, the flow rate of a pump station is decreased. In small pumping systems in which the length of the pipeline is short, the flow rate change may not be noticeable. However, for all these reasons, the design of the pump stations is usually performed based on old pipeline information.

The life of a pump station is usually between 30 and 40 years and, therefore, the pressure losses in the pipe should be calculated based on this expected service life. However, at the beginning of the work, the flow rate would be more than the required flow rate, because of this over estimation. The flow rate would, then, be corrected as the station continues to work, which means that the working point of all pumps will change as the work station continues to operate. This situation requires more attention and consideration during the design, since the efficiency of the pump will change over the time.

The required flow rate determines the number of the required pumps which could operate in parallel (pumps work in parallel has been discussed in Sect. 8.3.1). When one or more pumps are connected to the piping system, the efficiencies of the pumps are changed. Usually, the pump selection is a trial-and-error procedure. The important thing is that the efficiency of the pumps, when working alone or in parallel, should not change significantly during the life time of the station. This kind of analysis must be done during the design procedure because, as will be seen later, any change in the pump efficiency would change the operational cost.

Another important issue is to calculate the available NPSH in the suction pipe and compare it with the required NPSH of the pump in order to avoid cavitation during the life time of the station. After such analysis the foundation level and the methods through which pumps must be connected to the suction reservoir can be determined. As mentioned before, in some cases, the level of the water in the suction reservoir is not fixed and would change over time. For this reason the minimum expected water level in the reservoir must be used in design process. By moving the level of the pump station to a lower level than the suction reservoir, the safety of the station in terms of occurrence of cavitation in long term is guaranteed. However, this will increase the volume and cost of the construction works.

10.1.4 Architecture of the Station

As mentioned before, in many cases in order to reduce the geometrical suction head, the pump station is located at a level lower than the suction reservoir. The upper level of the station building, then, is used to install the control equipments. The dimension of the station must be such that it can accommodate all equipments and piping system, and still there should be enough space for easy access for the personnel to reach the equipments for every day operation and necessary maintenances, as well as removing the units if necessary. In addition, provisions should be made to allow the electrical equipments to be moved to an upper level in case of a flood or similar accidents. For this reason and also for installation and maintenance purposes, a suitable roof crane must be provided in the station. The capacity of such crane must be enough to lift the heaviest unit in the station.

As a general rule, all controlling equipments must be positioned in such locations that are easily visible by the operator during normal operation of the station as well as during start up or shutdown of the units.

Finally, in a pump station apart from the pumps, piping, and other fittings, necessary provisions must be considered for drainage, lighting, heating, cooling, and fire prevention units. Appropriate lighting of the system, good interior design, and easy access on the floor would definitely make a pleasant environment for the operators.

10.2 Initial Investment for Pump Station

One of the important parameters in making decisions about type and arrangement of the mechanical, electrical, and control systems, as well as the construction works in a pump station is the amount of the initial investment. Although the exact amount of the cost can be determined after completion of the design, at the beginning the designer must have a rough estimate of the available resources.

In modern pump stations, almost all operations are done automatically. This will increase the initial investment; however, the operational cost in terms of personnel costs will decrease. For this reason, two different designers may propose two designs for the same station (with the same hydraulic conditions) with two different initial costs. To avoid this confusion, the project should be defined considering some specific requirements in terms of type of the equipments and operational cost.

10.3 Operational Cost

The operational cost of a station is determined based on all other parameters. After considering the initial investment, personnel salaries, life of the station, depreciation of the equipments, and the energy consumption, a designer could determine the cost of pumping 1 m^3 of water from one specific location to the desired destination. This would determine how much the construction of the station is economically feasible or if it is viable at all. That is the reason behind considering different distinct designs in the preliminary design process, so the designer and the project manager could decide upon the best choice.

One important factor which is usually ignored is the efficiency of the pumps during operation. As mentioned before the efficiency of the pump would change according to flow rate demand or as the life of the station is increased. The following example shows the importance of this parameter in determining the operational cost.

The absorbed power for a pump for delivering 200 l/s with a head of water of 100 m at efficiencies of 70% and 80% would be 280 kW and 245 kW, respectively. The industrial pumps are usually designed to work for 2400 hours per year. This means that at one year the pump with lower efficiency would consume $(280 - 245) \times 2400 = 84000$ kWh more energy than the other pump. This number would be substantial during 40 years of the life of the station.

Selection of the type and method of regulating the flow rate is another important issue in determining the operational cost. To control the flow rate, there are different methods such as using in a valve on the discharge pipe, changing the rotational speed of the driving motor, trimming the diameter of the impeller, and installing an orifice on the discharge pipe. The pressure loss produced in the valve or orifice is essentially an energy waste. Whereas by changing the rotational speed of the driving motor, the flow rate adjustment can be done more efficiently. However, costs of these two methods are not the same.

At the end of this section, a partial list of the necessary decisions that must be made by the designers and documents that should be available as part of the design procedure is presented:

1. Selecting the type of the pump station (e.g., submerged or lower level station) and corresponding preliminary drawings.
2. Selecting the pump types (e.g., axial or centrifugal flow, horizontal or vertical, single stage or multi-stage),
3. Technical specifications of the piping system (working pressure, material, pipe diameters and thickness, and type of valves and other fittings).
4. General specification of the auxiliary mechanical equipments (surge tank, crane, vacuum suction pump, etc.).
5. General specification of the electrical equipment (type and power of the driving motor, electrical panels, and transformers).
6. General specification of the control and safety systems.
7. General specification of the heating, cooling, and fire protection systems.
8. The estimated cost of building the pump station (including all electro-mechanical, control, safety systems, as well as construction).
9. The estimation of the operational cost (cost of pumping 1 m^3 of water to the destination).
10. Specifying the winner alternatives, comparing them with the original requested specifications, helping the consulting engineer to make decision and starting the phase one of the project.

In the next sections, the important issues that must be considered in the phase one of any pump station project will be discussed in detail.

10.4 Determining the Pump's Specifications

Once the type of the pump is specified, i.e. axial, centrifugal, submersible, etc., it is time to determine the pump's specifications using the following information:

- Required flow rate and the range of its changes.
- Head of the pump (geometrical height between two reservoirs + pressure losses in the pipes and other fittings).
- Type of the liquid to be pumped.

Having the above information and using the characteristic curves of the available pumps, type and required number of pumps are determined. The working points of the pumps must be close to the maximum efficiency point. Since the flow rate of the pump would not remain constant and will change during the year for reasons such as changing of seasons, the "working range" of the pump must be examined very carefully. The method for connecting the pump, either in parallel or in series, depends on the desired range of changes in the flow rate and head. By connecting pumps in parallel during the failure of one pump, other pumps can deliver the flow rate partially. Usually, depending on the total number of pumps in the system, one or more additional pumps are also installed in the system. These pumps do not work under normal situation. However, in case of emergency they can easily be connected to the system.

As mentioned in Sect. 8.4.2, when pumps are connected in parallel, the slope of the H–Q characteristic curve of the pumps has a remarkable effect on the operational conditions. If the pressure losses in the pipes and local pressure drops are high (i.e., the slope of the system characteristic curve is steep), it is not recommended to use pumps with moderate to low slope characteristic curves (flat curves), because increasing the number of pumps would not increase the flow rate remarkably. In such cases, pumps with steeper characteristic curves have better performances (see Chap. 8). For this reason, during the design of the pump station and to determine the number of pumps working in parallel, the characteristic curve of the system must be obtained first. Then, the performance of this curve must be compared with the characteristic curves of the pumps. The result of such analysis would determine the optimum number of pumps as well as the range of their performances.

Sometimes many alternative designs must be investigated to find the best arrangement that is also economically feasible. Once the best arrangement is chosen, the designer will start to perform detailed analysis and would prepare the characteristic curves and the hydraulic information for the pump station, such as those shown in Table (10.1). With the help of such curves and information, the operating points of the pumps can be obtained and the operator(s) can determine the flow rate of the station when one or several pumps are working in the system. Of course in larger pump stations the control devices are installed to show the flow rate on each line, but still this information can be used as additional control measures. When the reading of the flow meter is not in the same order as the one obtained from the curves, this would be an indication of a problem in the pump or in the flow meter. Figure 10.1 and Table (10.1) show typical performance of a pump station with nine pumps working in parallel and three discharge pipes as shown. The curves and the table also show the working point of the pumps when they are alone or all together in the system.

One of the important parameters in pump stations is the required NPSH for the pumps. If the suction head is positive, normally there would be no problem in terms

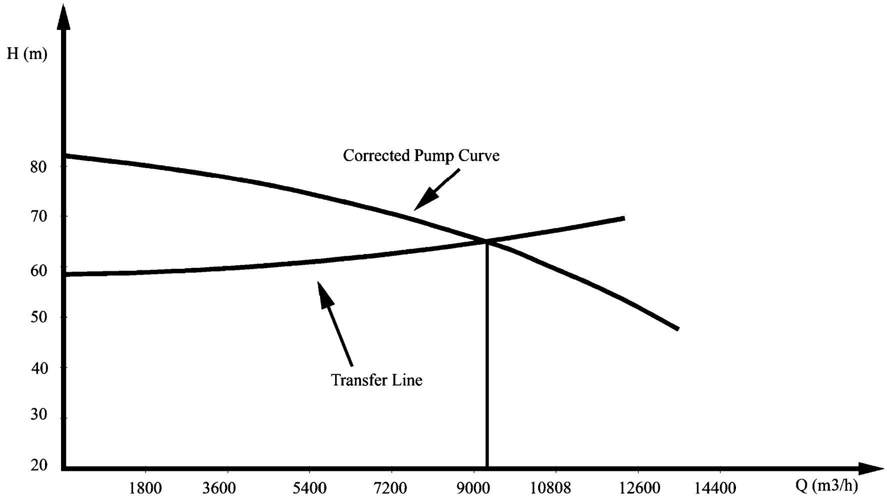


Fig. 10.1 The working point of a pump station, consisting of three delivery pipes and nine turbopumps

of cavitation. Otherwise, the available NPSH in the system must be obtained and compared with the one required by the pumps.

Another important parameter in the selection of a pump is the maximum value of torque that the pump shaft can support. Beyond this limit, either the pump shaft would break or the bearings will be damaged after a period of time. For this reason,

Table 10.1 Performance of the station shown in Fig. 10.1 [1]

Number of tries	Number of pumps	Number of transfer lines	Total flow of station (m^3/h)	The flow of each pump (m^3/h)	The head of each pump(m)	Total power (KW)	Efficiency
1	1	1	1174	1174	62.3	240	83.2
2	1	2	1185	1185	62	242	83
3	1	3	1200	1200	61	244	82.5
4	2	1	2267	1124	64	474	84
5	2	2	2359	1180	62	484	83
6	2	3	2395	1198	61	488	82.5
7	3	1	3133.3	1044	68	684	85
8	3	2	3482	1161	63	720	83.5
9	3	3	3566	1189	62	726	83
10	4	2	4502	1125.5	64.5	940	84.2
11	4	3	4681	1170	62.3	960	83.2
12	5	2	5438	1088	66	1160	84.5
13	5	3	5758	1152	63.5	1190	83.7
14	6	2	6266.6	1044	68	1368	85
15	6	3	6803	1134	64	1422	84
16	7	3	7744	1106	65	1631	84.5
17	8	3	8632	1079	66	1856	84.5
18	9	3	9400	1044	68	2052	85

on the characteristic curves of the pumps, especially large pumps, the maximum transmittable torque is specified (this value is denoted by the ratio of P/n, where P is the power and n is the rotational speed of the pump). In Fig. 10.2, the characteristic curves for a pump with limiting axis torque are shown (In this Figure that was taken from KSB catalogue, power is shown by N). Therefore, when selecting a pump with larger size, it should be considered that the working point of the pump does not need be close to this point.

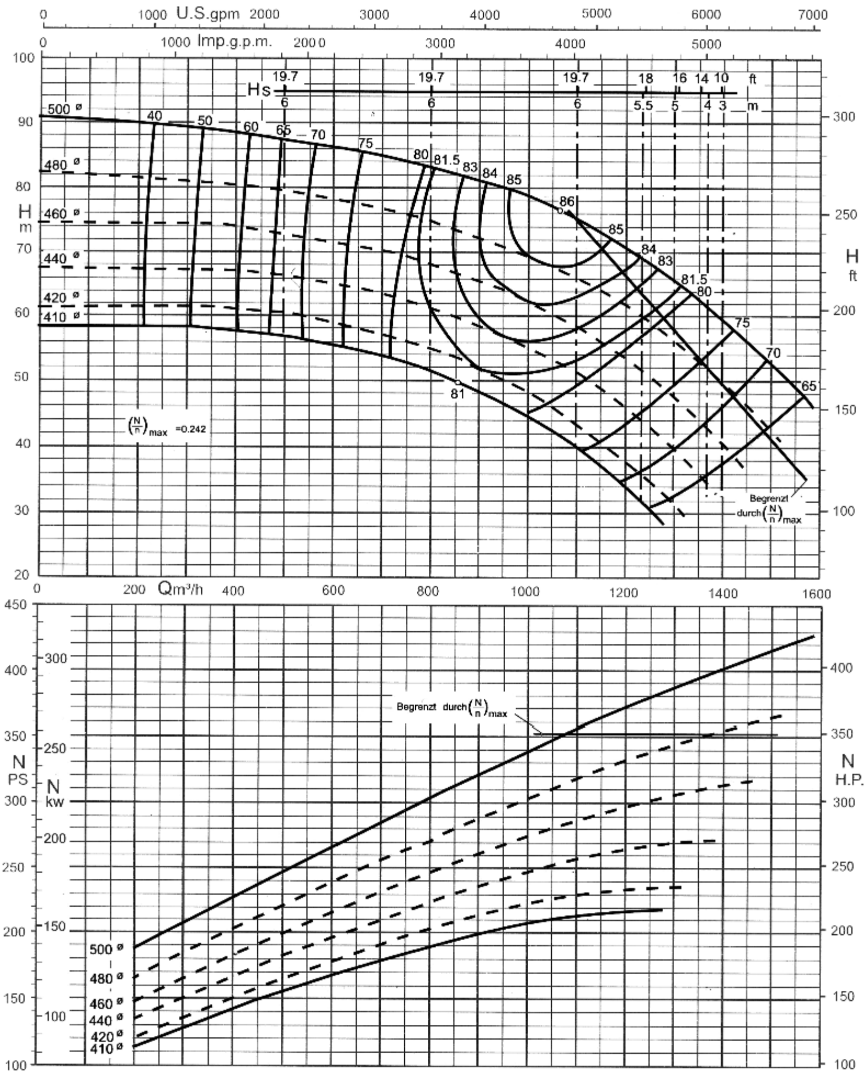


Fig. 10.2 The characteristic curve for a pump with limited transferred torque. In this Figure power is denoted by N (KSB Product Catalogue)

10.5 Driving Motor

One of the important tasks in designing a pump station is determining the type of the driving motors and their complete specifications. Along with the $H{-}Q$ and efficiency characteristic curves for each pump, the manufacturers usually present the $P{-}Q$ curve as well. Based on this curve, the required power for the turbopump can be determined.

Normally in pump stations, the electro motors with alternative current (usually squirrel cage induction motors) are used. However, the direct current motors, internal combustion motors, and even gas or steam turbines can be used as the driving motors for the turbopumps. The choice of the driving motor for a pump depends on the location of the pump station and local conditions. The initial cost of the motor, the operational and maintenance costs, spare parts, and availability of the expert personnel are among the other important parameters to be considered in selecting a driving motor.

Another important parameter is the rotational speed of the motor and the possibility to change it. If changing the rotational speed is possible, the flexibility of the pump station would be enhanced, mainly because the flow rate can be adjusted by changing the rotational speed (without requiring the usage of the flow control valves). This will increase the life time of the valves and at the same time will save energy by avoiding the unnecessary pressure losses in the valves. But DC motors are usually more expensive than the AC motors. There are also devices that can be used on AC motors to change the rotational speed which are used in pump stations. But the initial and maintenance costs of these devices must be reasonable.

Diesel machines are practically the only drivers that can be used as the driving motor when there is no electricity available. One of the advantages of these motors is their good response to the water hammer phenomenon. It is almost impossible that in a pump station all diesel motors stop working at the same time, whereas, all the electrical motors would stop at the same time in case of electrical failure in the pump station. Also, diesel motors are heavier than the electrical motors and therefore their flywheel effect is high preventing the harmful water-hammer consequences. In hydraulic, steam, and gas turbine power plants (and often in oil refinery plants), the generated power by turbines can be used to drive the turbopumps.

The power of the driving motor must be more than the required power of the pump for safety reasons. In small pumps, usually the maximum point of $P{-}Q$ curve is used to determine the power of the electro motor. In this way, under the worst conditions, the electro motor can still work without any problem. In centrifugal pumps, this point is located at the maximum flow rate point, whereas in axial flow pumps the maximum power is at flow rate equal to zero. However, in larger pumps, especially when the required power at the working point of the pump is much different with the maximum power on the maximum point of $P{-}Q$ curve, this is not a practical solution because it will increase the cost of the motor. In such cases the safety devices must be provided to shutdown the motor when the electrical current is larger than a specific value.

Table 10.2 Safety factors to calculate the power of the motor [2]

Required power for pump, P_{inP}	Safety factor
1.5 kW	1.25
1.5–4 kW	1.25
4–7.5 kW	1.20
7.5–40 kW	1.15
Over 40 kW	1.1

In order to find the power of the electromotor when the $P-Q$ curve is not available, the absorbed power for turbopump is calculated and is then multiplied by the safety factors shown in Table (10.2).

The nominal powers of the electro motors are usually standard. Therefore, the next power that is larger than the calculated power is selected. The next step would be to choose the insulation type and the casing of the motor. Another parameter is the surrounding temperature of the motor. By increasing this temperature, the heat transfer to the environment will decrease and eventually the efficiency of the motor would decline.

10.6 Design of the Suction Reservoir and Pipes

The suction reservoir is the first element of a piping system from which the liquid enters into the system. The design of this reservoir and correct installation of the suction pipes have great effects on the performance of the system.

The design of the suction reservoir and layout of the necessary equipments must be such that mud, sand, or other particles in the water cannot enter the suction pipe and eventually the pump. For this purpose, usually a stilling tank is installed before the main suction reservoir. This tank would let the water stand still long enough until all suspended particles settle in this reservoir. It is obvious that this tank must be cleaned occasionally to remove these substances. Therefore, provisions must be considered for the movements of trucks, cranes, and other equipments. The detailed design of such system requires more information and will not be discussed in this book. In this section, only the hydraulic parameters that are important in the design will be discussed.

The suction reservoir must be located such that the fluid flow toward pumps in all directions becomes uniform and laminar. The uniformity of the flow is especially very important for pumps with high specific speeds (axial flow pumps), because the performances of these pumps are very sensitive to the existence of turbulence and any distortion in the suction flow.

The suction of the liquid can be from natural wells, basins, open canals, pipes, or closed reservoirs. Pumps can be installed directly in the suction reservoir (vertical turbine pumps) or connected to the reservoir through suction pipes. The suction head can be positive (the suction reservoir at a level higher than the pump) or negative (the reservoir is lower than the pump). In all these conditions, if the suction pipe of the pump is directly connected to the reservoir, the suction is called “free suction.” If the suction pipe of the pump is connected to a collector pipe, the suction is called “pipe suction.”

10.6.1 Free Suction

The important issues in designing the reservoirs with free suction are discussed in the following sections.

10.6.1.1 The Depth of Suction Pipe

There should always be enough space between the end of the suction pipe and the bottom of the reservoir. This is to avoid any remaining sand or dirt that normally exists on the bottom to enter the pipe.

The exact distance, which also depends on the flow rate of the pumps, can be obtained from the curve that is presented in Fig. 10.4. To use this curve, Fig. 10.3 can be used as guidance.

The minimum distance between the water level in the reservoir and the end of the suction pipe, D_{min} = H–C is also important to avoid any vortex flow or air entering to the pipe, see Fig 10.3. This minimum distance can be obtained from the following relation [10]:

$$D_{min} = \frac{V_s^2}{2g} + 0.1 \tag{10.1}$$

where D_{min} is in meter and V_s the velocity in the suction pipe is in m/s. This distance can also be obtained from Fig. 10.4.

10.6.1.2 Layout of the Suction Pipe

The distance between the axis of the suction pipe and the reservoir walls depends on the flow rate and the pipe diameter (the minimum distance is shown in Fig 10.3). When several suction pipes are fed from the same reservoir, the distance of the pipes from each other must be respected. This distance is also shown in Fig. 10.4.

As mentioned before, proper equipments must be installed to prevent solid particles, mud or sand entering into the pump. Installation of proper screens or filters is a necessary step at the suction reservoir. Also, sedimentation must be done in this place by providing a slope to move the solid particles away from the suction point.

If there is more than one suction pipe, their positions must be such that they do not interfere with each other and the liquid is divided equally between them. By installing suitable partitions between the pipes, flow can be divided equally as shown in Fig. 10.5.

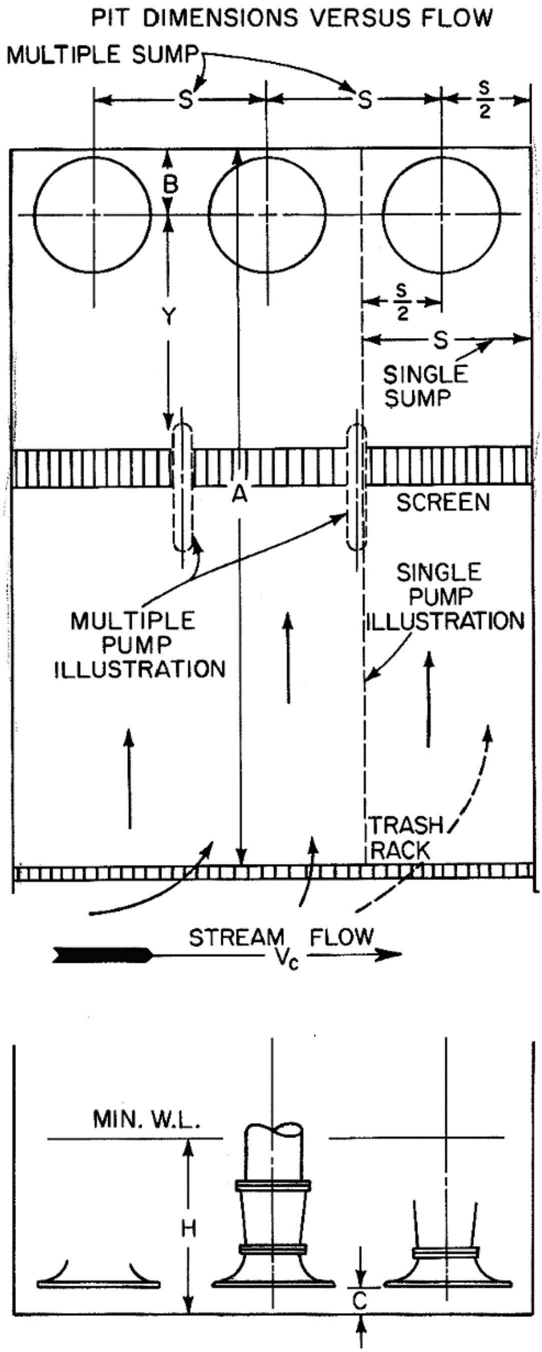


Fig. 10.3 Arrangement of a pump station and corresponding distances. All distances between suction pipe and the walls are defined in Fig. 10.4 [3]

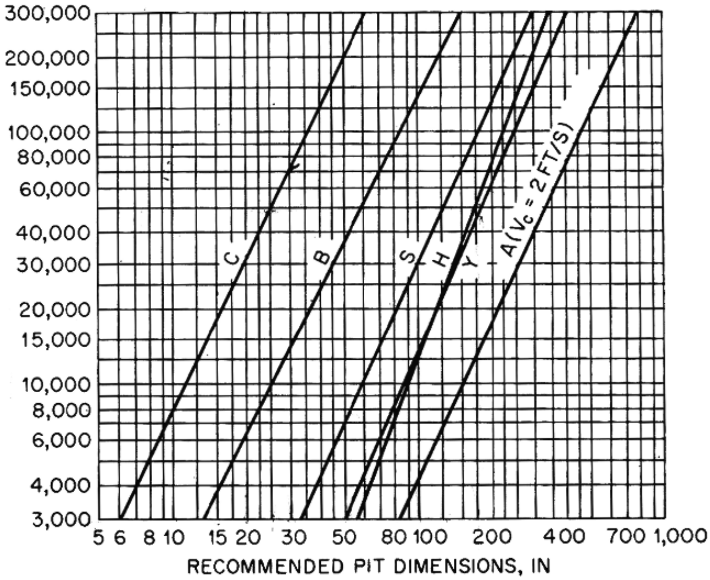


Fig. 10.4 Recommended distances for the suction reservoir. (English Unit System) [3]

To avoid any disturbance in the flow at the entrance to the suction pipe, the walls of the reservoir are designed with necessary curves or slopes. For example, when the water is taken from a river or a main canal, the connecting branch to the reservoir should be in the opposite direction of the flow in the main canal. In this way, the disturbances in the main flow would not affect the flow going to the suction reservoir.

As the final recommendation, the suction reservoir must be designed such that it is easily accessible and it is possible to control and inspect the station.

10.6.2 Pipe Suction

In some pump stations, the pump suction pipe is connected to a collector or a main pipe. In these cases the flow velocity in the collector is one of the important parameters. Usually, the velocity should not exceed 1 m/s, otherwise the flow entering the pump would not be uniform and laminar. When two or more pumps are connected to a single collector, as the flow rate decrease in the collector, the diameter must be changed in order to make the flow velocity constant. In this way the suction conditions for all pumps would be the same.

The best method to branch a suction pipe from the main pipe is to use a 30–45° connection. When there are several pumps connected to a main collector, if the

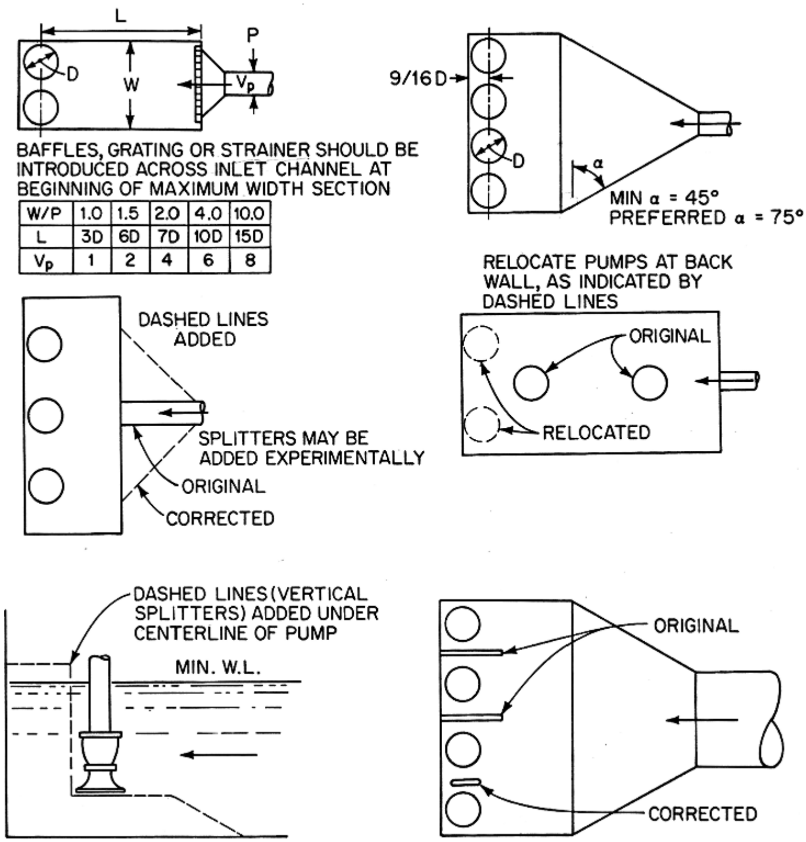


Fig. 10.5 Good arrangements for suction pipes in the suction reservoirs [3]

above angle is used and the flow velocity in the suction pipes of the pumps is kept less than 1.5 m/s, then there would be no limitation in terms of distances between the pipes. In Fig. 10.6, two methods of connecting suction pipes to the main collector are shown.

When suction is done vertically from a canal or a tunnel (under pressure), the suction pipes can be arranged such that the flow conditions in the main canal remains independent of the pump's performances. This situation is shown in Fig. 10.7. If the suction pipe diameter is more than 1/3 of the canal diameter, the distance between branches must be at least 12 times of the pipe diameters. This distance could decrease to 6 times of the suction pipe diameter, if the pipe diameter is less than 1/3 of the diameter of the main canal. Also, the pump entrance must be located at least at a distance twice the suction pipe diameter above the upper wall of the canal and the flow velocity must be kept under 2.5 m/s.

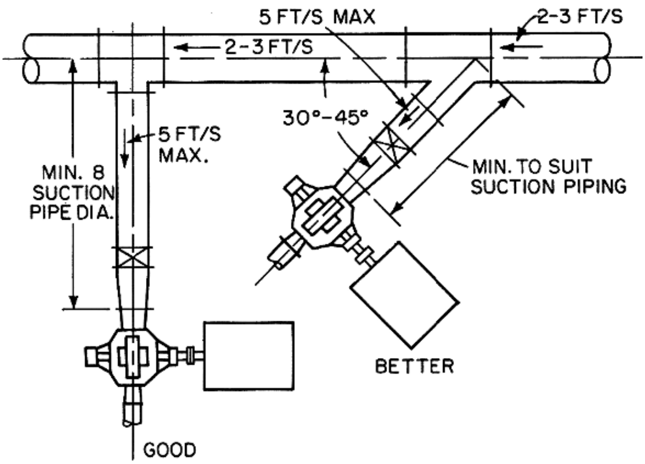


Fig. 10.6 Proper distances and suction velocity in a "pipe suction" scheme [4]

10.6.3 Suction Pipe of the Pump

The stream lines of the flow entering a turbopump must be parallel and the velocity must be uniform to guarantee a smooth operation and less possible damage to the impeller. To achieve this goal the following important parameters must be considered in designing the suction pipe of a pump:

- The straight section of the suction pipe before the pump flange must be at least three times of the pipe diameter and the pipe diameter must be at least equal to the diameter of the pump entrance. If the pipe diameter is larger, a reducer can be used before the pump to adjust the diameters. The converging piece must be from the non-symmetric type, with a flat top to avoid any air entrapment (Fig. 10.8).
- When the suction is done from a reservoir or a basin, the flow velocity in the suction pipe must be kept between 1 and 2 m/s. When the suction head is positive, this velocity can be increased to maximum of 3 m/s. When the suction is done from a main pipe or manifold, the flow velocity in the suction pipe must be kept below 1.5 m/s. If the branch from the main collector is at angle of 30°–45°, the distance between the pump and the main collector can be as small as desired.
- Under no circumstances air can enter the suction pipe. For this reason, at the

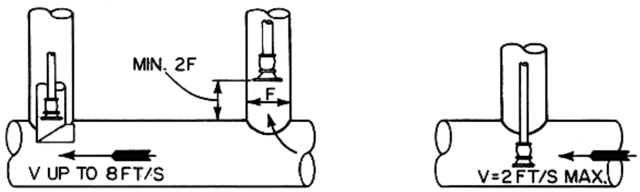


Fig. 10.7 Suitable methods to connect suction pipes to a pressurized tunnel [4]

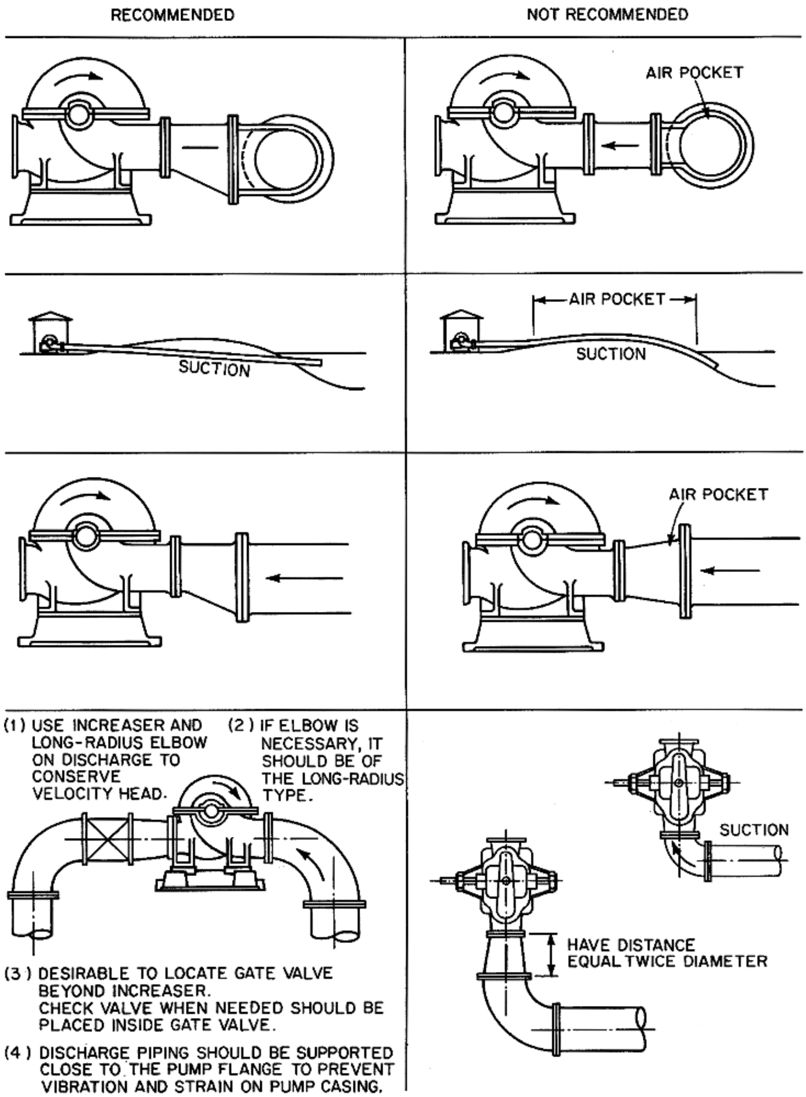


Fig. 10.8 Comparison of different methods to lay out the suction pipe for a pump [3]

suction side of the pump the usage of welded-fittings are better than the flanged-fittings. All the seams must be air proofed.

- The suction pipe must be designed straight and very simple as much as possible. All resistance to the flow must be minimized, otherwise the energy loss increases and the risk of cavitation is also increased.
- A good economical balance between the pump size, its speed, $NPSH_{req}$, pipe diameter and velocity in the suction pipe must be achieved. For example, if the reduction of the suction head by changing the suction reservoir would cost

too much, it is better to increase the pipe diameter between the reservoir and the pump instead and to design the pipe as straight as possible to maintain the $NPSH_{avail}$. But if the pump is to be used in a cooling tower system, where it is located above the reservoir, both the flow velocity and the pipe length can be increased. When the suction is done from a well (negative suction head), pump must be located as close as possible to the reservoir.

- The high points in the suction pipes should be avoided since there is always possibility of air or vapor to be trapped there. This happens, for example, when the suction pipe has a negative slope toward the pump or when the valve on the suction pipe is installed vertically as shown in Fig. 10.8 because in both cases there is a possibility of the air to remain at the higher elevation. Also, the reducers must be from the non-symmetric type such that the flat part of them is located at the top. Otherwise, proper devices must be installed to release the air from the suction pipe.
- The suction pipe must be designed such that the flow experiences least disturbances. For example, the usage of bends before the pump is not recommended. Of course it is possible to use the reducing bends before the pump, since the flow in these types of bends is more laminar. In any case, the large radius bends should be installed before the pumps.
- In pumps that are connected to each other in series, the suction pressure for the second pump and other pumps following it will be high. Therefore, all fittings and flanges must be from the welded type to handle the pressure. Also, no expansion joint must be used in these arrangements, since the hydraulic forces to the pump are high. Because of a high pressure in the suction pipes of the second pump and consequent pumps, the diameter of the suction pipe can be smaller.

10.6.4 Discharge Pipe

The flow conditions in the discharge pipe are less critical than those in the suction pipe. Nonetheless, some parameters must be considered:

- The flow velocity in the discharge pipe must be kept low so as to avoid unnecessary energy losses. This must be done considering a good balance between the initial cost of the projects and the required energy for the pump.
- The discharge pipe should endure higher pressures than the suction pipe to withstand the normal pressure of the pump as well as the pressure rise due to the water hammer (as much as possible) during the start up or shutdown or other emergency cases.
- The layout of the discharge pipe is very important. The discharge of the water from the pipe to the discharge reservoir can be free or the end of the pipe can be connected to the bottom of the tank. One of the good designs is to use a siphon shape pipe at the end of discharge line in order for the pump to supply

less pressure, compared to a discharge pipe with free end. The siphon pipes are specifically good for pumps with low manometric heads.

10.7 Equipments for Suction and Discharge Pipes

10.7.1 Suction Pipe

As mentioned before, the suction pipe must be designed very simple with the least number of fittings and other equipments. If the suction head is positive or the pump is equipped with the filling device in start up period, there could be no valves on it. But if the suction head is negative and there is no other equipment to fill up the pipe, there should be a non-return valve at the beginning of the pipe. This valve which is called the swing valve or foot valve would prevent the flow return to the reservoir during the shutdown period.

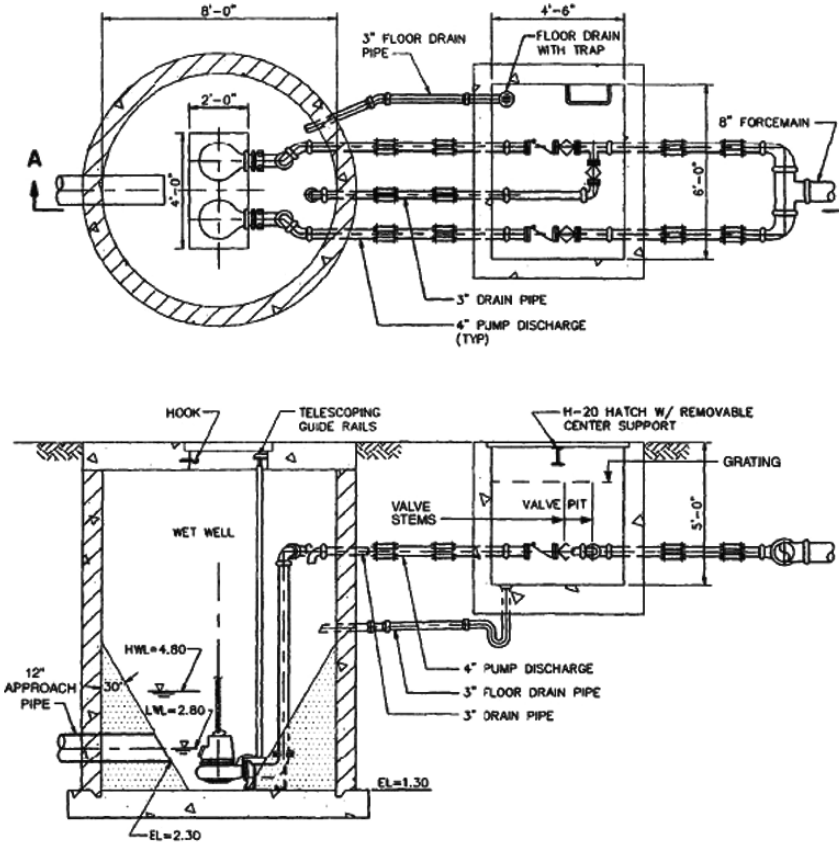


Fig. 10.9 A sample of general layout of a pump station and design parameters [4]

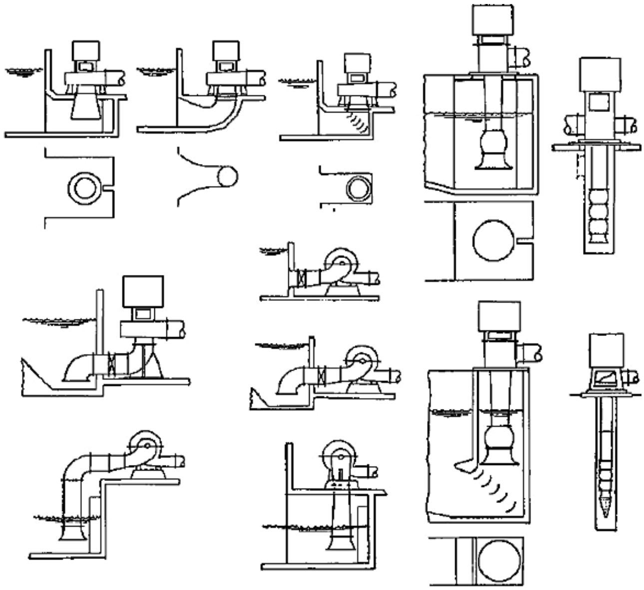


Fig. 10.10 Different methods of installing the suction pipes [4]

Other equipments could be a valve for shutting down the flow (this valve must be completely open during the normal work of the pump), a reducer, if necessary, and an expansion joint.

10.7.2 Discharge Pipe

The required equipments on the discharge pipe are (starting from the pump side) an expander (if necessary), a non-return valve, a shut off valve, and an expansion joint.

If necessary, a proper surge tanks or air valve can be installed in suitable distances, for the water hammer periods. The location of these valves must be selected carefully. When the surge tanks are used, they are usually located at the first possible point after the discharge valve.

In Fig. 10.9, different design layouts for pump stations are shown. In Fig. 10.10, methods of installing the suction pipes are shown.

For more information, see also [5, 6, 7, 8].

References

1. Nourbakhsh S. A., "Pump and Pumping", University of Tehran Press, 2004.
2. "Basic Principles for the design of Centrifugal Pump Installations", SIHI Group, 1980.
3. Karassik Igor J. et al, "Pump Handbook", McGraw-Hill, 1976.

4. Karassik Igor J. et al., “Pump Handbook”, McGraw-Hill, 2001.
5. Sanks R. L., “Pumping Station Design”, Butterworth-Heinemann, 1998.
6. Smith P., Zappe, R., W. “Valve Selection Handbook”, 5th ed., Elsevier, 2004.
7. Skousen Philip L. , “ Valve Handbook”, McGraw-Hill, 1998.
8. Tullis J. Paul, “Hydraulics of Pipelines: Pumps, Valves, Cavitation, Transients”, John Wiley & Sons, 1989.
9. “Basic Principles for the Design of Centrifugal Pump Installations”, SIHI Group, 1980.
10. “Centrifugal Pump Design”, Technical Appendix, KSB, Frankenthal, Germany.

Appendix
Turbopump's Elements

The constructional components of turbopumps are different with each other depending on the type of pump, its main application, and the physical properties of the liquid to be handled. For the purpose of studying constructional elements of turbopumps, these components are first divided into two groups: major and auxiliary components. Major components are also divided into two groups:

1. Rotating parts, including impeller and pump shaft
2. Stationary parts, including inducer, diffuser, volute casing, and casing.

The auxiliary parts include:

- Stuffing box
- Ball bearing
- Bushings
- Leaking rings
- Equipments for balancing the axial and radial forces.

In this section of the book, only the major constructional elements of the turbopumps are studied.

A.1 Impeller

The impeller of a turbopump, on which the blades are installed, is the main component of the pump that is responsible for transferring the energy from the pump to the liquid. The structure of impellers can be examined from different points of views:

- Whether the impeller is open or closed.
- Blade types and the relation between the impeller shape and the pump specific speed.
- Whether the impeller is single suction or double suction.

A.2 Open and Closed Impellers

A.2.1 Closed Impellers

In closed impellers, blades are located between two shrouds or sidewalls. These impellers are mostly used in centrifugal and mixed flow pumps (Fig. A.1). Except for some special purposes, multi-stage turbopumps also use closed impellers.

The application of closed impellers in turbopumps is usually limited to cases in which the pumped liquid is single phase and there are no gases or suspended solid particles in the liquid. Usually, pump manufacturers determine maximum allowable dissolved gas in the liquid for each particular impeller types and then present this information in their catalogs. It should be mentioned that by reducing the number of blades in closed impellers, they can also be used to handle liquids with suspended particles. The shape and number of blades in these impellers are determined based on the size and type of the solid particles. The blade number for handling these types of liquids can even be reduced to only one blade. These types of impellers are used widely in pumps working in food processing plant, e.g. in turbopumps used for pumping the water and beats in sugar processing or water and fruit in compote processing plants.

Another important usage of impellers with smaller number of blades is in pulp and paper industry and water treatment plants. In Fig. A.2 the cross sections of several closed impellers with less number of blades are shown.

A.2.2 Semi-open Impellers

In these impellers the front wall of the impeller (when looking from the flow entrance side) is eliminated and blades are attached only to one surface (Fig. A.3). These impellers are used in centrifugal and mixed flow pumps that carry multiphase liquids. The liquids handled by these impellers can have higher percentages of gases and solid particles than those carried by closed-type impellers.

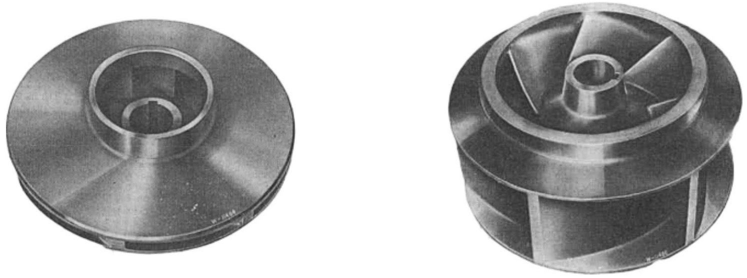


Fig. A.1 Closed impellers [1]

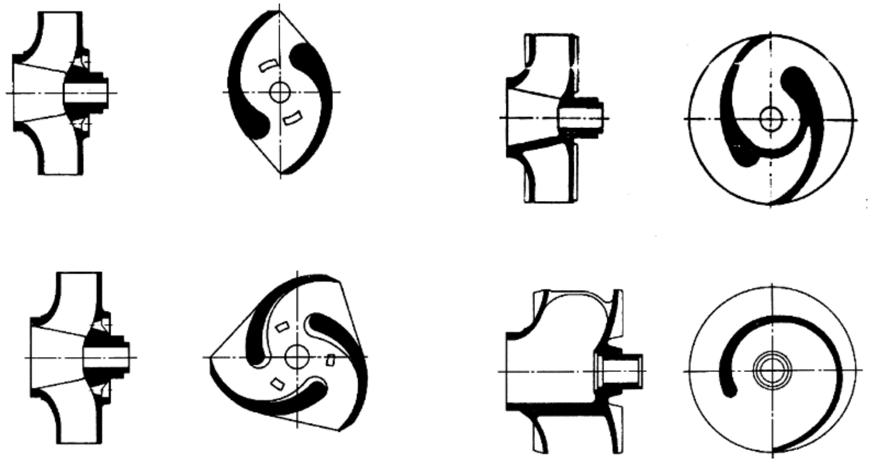


Fig. A.2 Profiles of different closed impellers [4]

For two-phase flows consisting of liquid-gas, the number of the blades can be larger. However, if the flow also contains solid particles, the number of blades must be reduced based on the solid particle sizes.

Semi-open impellers are widely used in special pumps used in sewage systems. In Fig. A.4, cross sections of several semi-open impellers are shown. Semi-open impellers are also used in the mixed flow pumps pumping single-phase flows when the flow rate is high. In this type of application the manometric head is usually low and pumps are installed vertically.

A.2.3 Open Impellers

Fully open impellers are normally used in vertical turbopumps. This type of impellers are also called "propellers" and are especially suitable for delivering very large flow rates in vertical pumps. These impellers are also used in ships and boats for driving mechanism. In Fig. A.5 an open vertical impeller is shown.

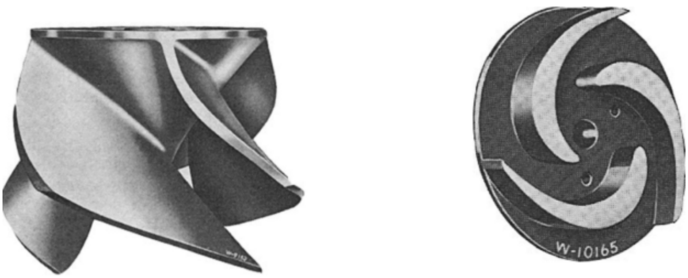


Fig. A.3 Semi-open impellers [1]

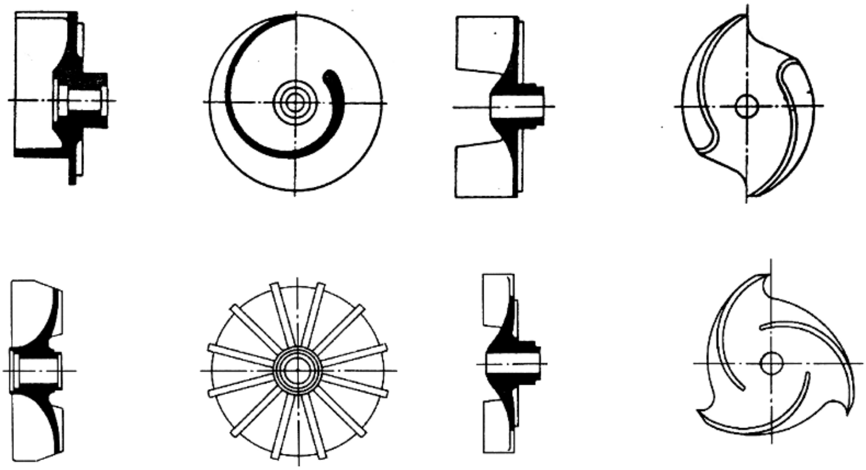
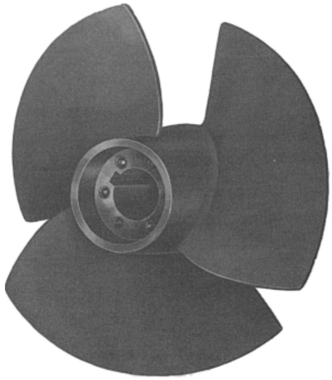


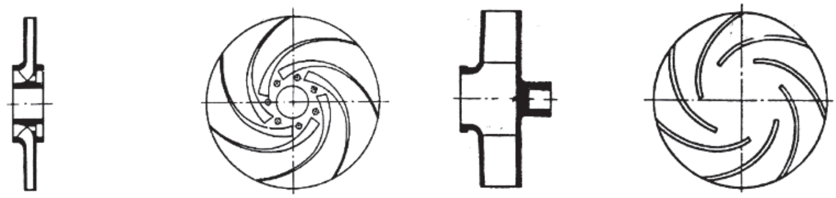
Fig. A.4 Cross sections of several types of semi-open impellers [4]

Fig. A.5 An open impeller [1]



A.3 Blade Types and the Relation Between the Blade Shape and Specific Speed

Pump impellers can be classified based on the specific speed or the direction of the flow with respect to the pump shaft. As the specific speed of an impeller increases, the flow exiting the pump changes its the direction from 90° to 180°. Therefore, the profile of the impeller and the pump are also changed. This type of pump classification indeed defines the limit of the application of a specific pump for delivering specific flow rates and manometric heads. This means that by decreasing the specific speed, a pump is more suitable for delivering lower flow rates in higher manometric heads, whereas, by increasing the specific speed, the delivered flow rate increases and the manometric head decreases. For this reason, impellers can be classified into three different categories.



(a) (b)

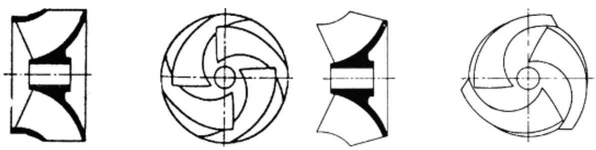
Fig. A.6 Two types of blade profile for centrifugal pumps [4]

A.3.1 Centrifugal Impellers (Low Specific Speed)

In these impellers, liquid enters the impeller parallel to the rotating shaft and leaves it perpendicularly. The outlet blade angle β_2' in centrifugal impellers is usually smaller or equal to 90°. The blade profile in these impellers consists of either one or two curvatures. In the first type of blades which are usually used in very low specific speeds, the entrance edge is parallel to the pump shaft, Figure A.6b, whereas, in the second type, the entrance edge of the blades are not parallel to the shaft. In the impellers with two curvatures, the first curvature locates in the entrance area of the pump, see Fig. A.6a and second curvature extends towards inside the impeller.

A.3.2 Semi-centrifugal (Mixed Flow) Impellers (Medium Specific Speeds)

In these impellers, liquid enters the pump parallel to the shaft and leaves it with an angle greater than 90° (with respect to the shaft). The blades' profiles used in these impellers are from two-curvature type with an external lades angel, β_2', smaller than 90°. The difference in this type with the centrifugal impellers is that the blade angles at the exit do not remain constant across the blade width. In Fig. A.7, two types of mixed flow impellers are shown. As one can see, in the impeller type (a) the exit edge of the blade is parallel to the shaft. But because of the specific shape of the



(a) (b)

Fig. A.7 Two types of mixed low impellers [5]

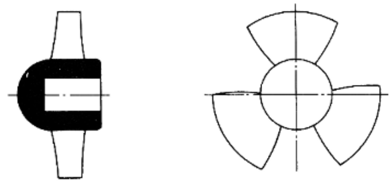


Fig. A.8 An axial impeller [4]

blade and higher specific speed, the impeller is considered as a semi-centrifugal impeller. In these types of impellers, the external blade angle, β_2', does not change very much.

A.3.3 Axial Flow Impellers (High Specific Speed)

In the axial impellers, the flow at the entrance and exit of the pump is parallel to the pump shaft. These impellers are very sensitive to small variation in the inlet flow and for this purpose the range of the flow rates in these impellers are very limited. To solve this problem, the blade angles can be changed automatically by a servomotor which is activated by flow rate changes. In Fig. A.8, the profile of an axial impeller is shown.

A.4 Suction Method

With respect to the method of suction, impellers are divided into two types: single-suction and double-suction impellers. In impellers with one suction side, liquid enters the impeller from one side (Fig. A.9a). These impellers have more applications because of simplicity in manufacturing and are mostly used in multi-stage pumps.

Impellers with two suction sides are actually two symmetrical impellers with one-side suction which are connected in tandem. Both impellers are installed inside one casing (Fig. A.9b). Two-suction sides are connected to one-suction pipe and liquid is distributed to both suction sides simultaneously. One of the important advantages of these impellers is that the flow rate of the pump, at the same manometric

Fig. A.9 Single-side (**a**) and double-side; (**b**) impellers [4]

(a) **(b)**

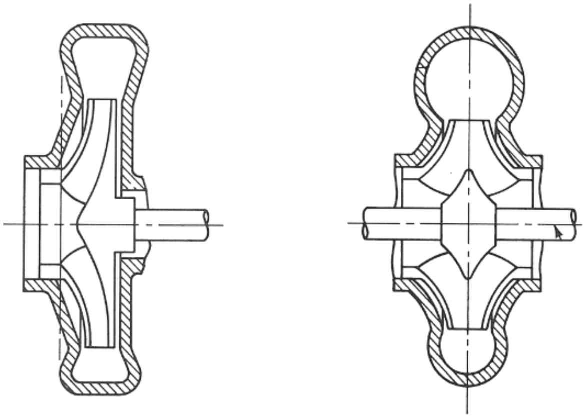


Fig. A.10 Two types of impellers:(**a**) hanging impeller; (**b**) double-sided impeller (no hanging) [1]

head, would be twice with respect to a single impeller. This is like arranging pumps in parallel which could also affect the $NPSH_{req}$ of the pump. Also, by entering the flow from two opposite sides of the impeller, the axial thrust is balanced more efficiently.

It should be mentioned that the shape of the inducer and of the suction pipe in pumps with two suction sides is different from single-suction pumps and their manufacturing is also more complicated.

At the end of this section, it must be added that the method of installing the impellers on the shaft could also be different in different pumps. In impellers that are called "hanging impellers," the pump shaft only extends to the back shroud of the impellers and, as a result, the cross-sectional area available for the flow entering the pump is a complete circle (Fig. A.10a). Whereas, in other types of pumps, the shaft extends all the way through the impeller and occupies part of the flow entrance area (Fig. A.10b). For this reason, hanging impellers are mostly used in pumps which are used for handling the liquid with solid particles, to open the flow area.

A.5 Suction Pipe

In order for the liquid flow to enter the impeller very smoothly and uniformly, usually a suction pipe is used before the pump. The shape of this pipe depends on the pump type and the installation method. However, the most important parameter in designing a suction pipe is to select the cross section of the pipe such that it delivers a uniform flow to the pump. In this section, several types of suction pipes are shown.

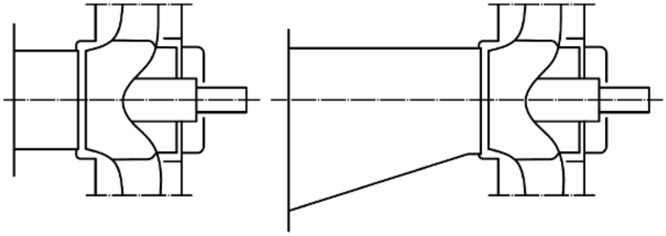


Fig. A.11 Two types of suction pipes, straight and reducing [3].

A.5.1 Straight Pipes: With Constant or Reducing Cross Section

These types of suction pipes are usually used in horizontal or vertical single-stage centrifugal pumps, mixed flow pumps, vertical, submersible pumps and also in small multi-stage horizontal pumps. The suction pipe must be from non-symmetric type (with respect to the axis of the pipe) such that the straight part of it locates on the top of the installation. In this way, there would be no danger of air entrapment in the suction pipe (Fig. A.11).

A.5.2 Bends: Uniform or Reducing Type

If necessary, bends can be installed before horizontal or vertical centrifugal pumps. However, they are not suitable for pumps with high specific speeds, because they would reduce the manometric head as well as the efficiency of the pump. In any case, the curvature of the suction bends must be large enough and they should not be installed immediately before the pumps. If required, one can use reducing bends in which the flow velocity increases gradually and therefore the stream lines remain uniform (Fig. A.12).

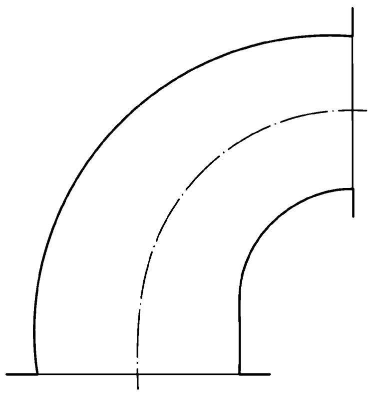


Fig. A.12 Suction bend [3]

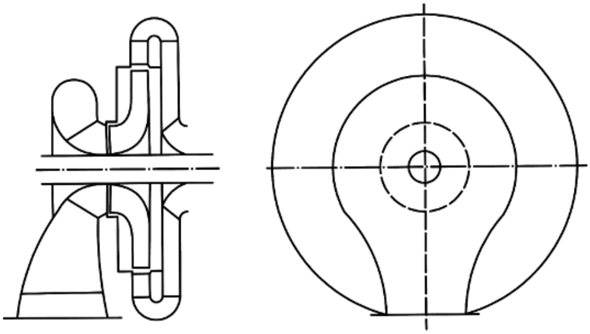


Fig. A.13 Concentric suction chambers [3]

A.5.3 Concentric Suction Chambers

These suction chambers are basically used in multi-stage pumps as well as in single-stage pumps with double-suction impellers (Fig. A.13). To avoid the flow circulation inside this chamber, blades can be installed close to the pump impeller.

In general, the design of these suction chambers must be done such that the flow separation does not occur close to the walls, vanes, or the flow velocity does not increase excessively, otherwise the risk of cavitations may increase.

A.5.4 Bell-Type Suction Pipe

These suction components are installed in vertical mixed flow pumps with high capacity and axial pumps. To select the inlet diameter of the bell pipe, one can use the information presented in Table (A.1) with parameters defined in Fig. A.14.

A.5.5 Volute Suction Element

The important applications of volute suction elements are in single-stage and multi-stage pumps with single- or double-suction impellers (Fig. A.15).

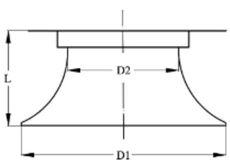


Fig. A.14 Bell-type shape suction pipe. For appropriate dimensions, refer to Table (A.1) [3]

Table A.1 Required dimensions for selecting a bell type suction pipe, see Fig. A.14 [6]

Nominal diameter	D_1	D_2	L
75	73	110	80
100	98	150	90
125	123	190	98
150	147	230	106
200	198	300	125
250	247.6	375	145
300	296.8	450	170
350	346.0	530	190
400	395.6	600	210
450	444.8	680	240
500	494.0	750	260
600	594.8	900	310
700	695.0	1050	340
800	796.0	1200	400
900	895	1350	440
1000	1001	1350	440
1100	1102	1400	450
1200	1202	1500	470
1300	1302	1600	490
1400	1404	1750	510
1500	1504	1850	530
1600	1604	2000	560
1700	1706	2100	580
1800	1806	2250	620
1900	1908	2350	640
2000	2010	2500	690

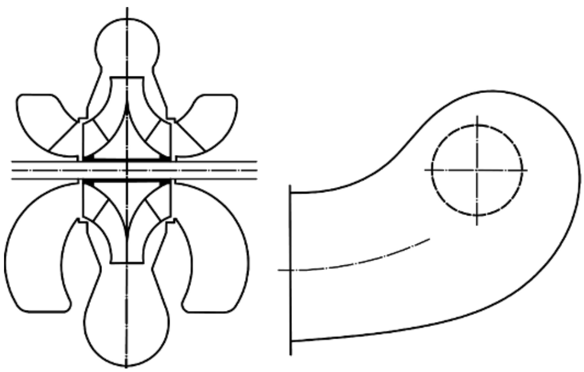


Fig. A.15 Volute suction [3]

A.6 Diffuser

As was shown before, the energy that is transferred from the pump to the liquid is the sum of two energies: kinetic (proportional to the velocity) and potential energy. Since in the process of transporting a liquid, pressure losses increase by increasing the flow velocity, it is necessary that some portion of the kinetic energy of the flow in the pump is transformed to potential energy, before the liquid leaves the pump. For this reason, pumps in which most of the energy is transformed to potential energy have better performances.

To transform kinetic energy to the potential energy after the liquid leaves the impeller, a diffuser is used. The diffuser is a stationary part of the pump that has no vanes. However, its cross section increases as the liquid goes through it, resulting in decrease in flow velocity and therefore an increase in manometric pressure.

There are three different types of diffusers as shown in Fig. A.16.

1. Diffusers without vanes with parallel walls, Fig. A.16a.
2. Diffusers without vanes with diverging walls, Fig. A.16b.
3. Diffusers with vanes, Fig. A.16c.

The type of diffuser and its application in a pump depend on many parameters. Usually, in small single-stage centrifugal pumps a diffuser is not used and this is because attaining a high efficiency is not very important. In these pumps the velocity reduction takes place in the volute casing or in the discharge pipe (expanding pipe). As the size and capacity of the pump are increased and achieving a high efficiency becomes more important, different types of diffusers with parallel walls, diverging walls, or eventually diffusers with vanes must be used.

It should be mentioned that in pumps equipped with diffusers with vanes, the flexibility with regard to the flow rate variation decreases. These pumps, which have better efficiency in their design points (compared to the pumps without vaned diffusers) perform poorly under other conditions (lower or higher flow rates).

In multi-stage centrifugal pumps (horizontal or vertical), diffusers with return canal steer the leaving flow from one stage to the next stage and in this way correct the flow path with respect to the next impeller (Fig. A.17). For this reason diffusers are a main component in the multi-stage pumps. In Fig. A.18, the cross section of a single-stage centrifugal pump with diffuser is shown. In vertical mixed flow and

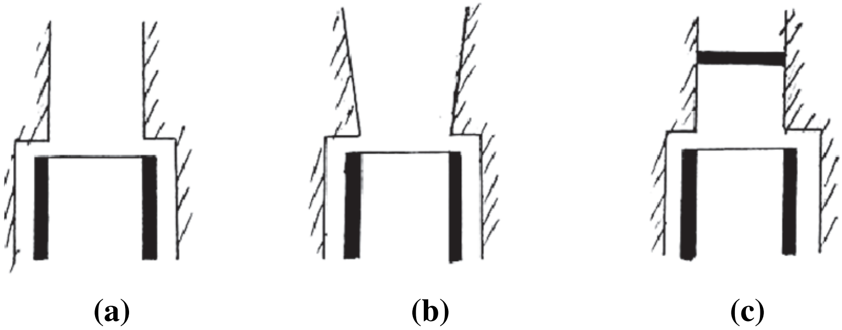


Fig. A.16 Three types of diffusers. (**a**) Diffuser with parallel walls; (**b**) Diffuser with diverging walls; (**c**) Diffuser with vane

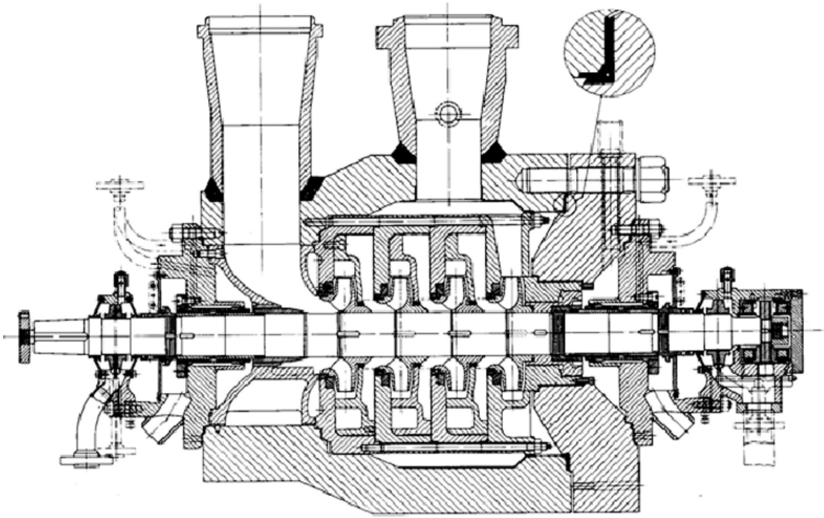


Fig. A.17 Diffuser with return canal in a multi-stage pump (Worthington Pump, Inc.) [2]

axial pumps, the role of diffuser is very important and its design is performed with extreme care, because these pumps deliver higher flow rates with low manometric head; therefore, the pressure losses must be kept minimal.

One of the important roles of the diffuser is to eliminate the flow circulations that are usually formed around the pump shaft after flow leaves the impeller and corrects the flow pattern around the shaft. This cannot be done without a diffuser. In Figs. A.19 and A.20, a vertical mixed flow pump and vertical axial flow pump with diffuser are shown, respectively.

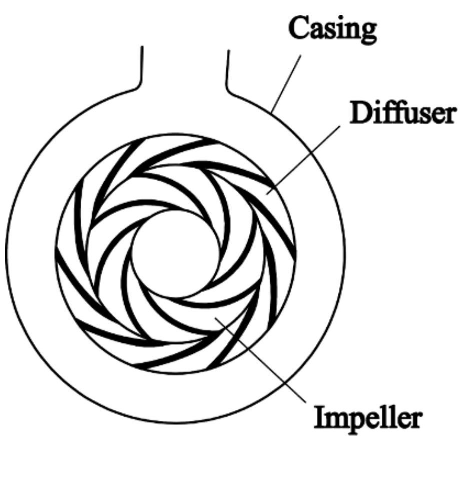


Fig. A.18 Diffuser with vanes in a centrifugal pump [2]

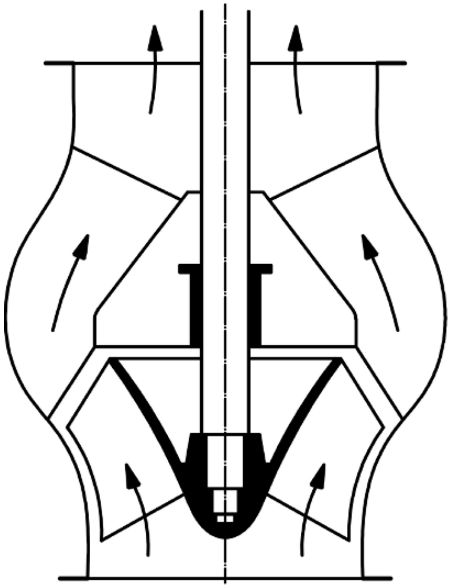


Fig. A.19 Vertical mixed flow pump with diffuser [3]

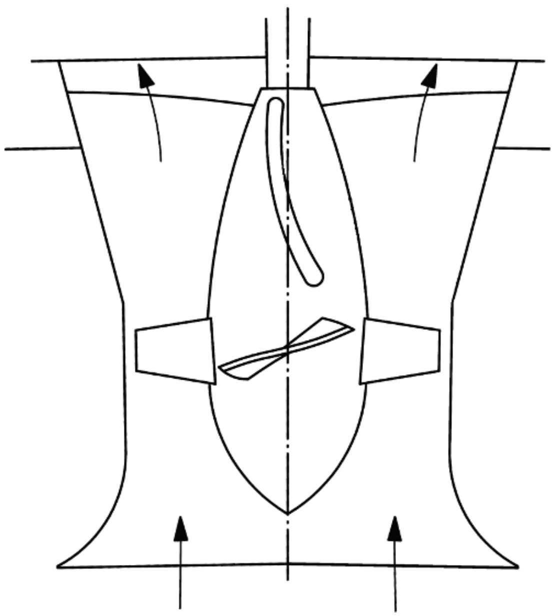


Fig. A.20 Vertical axial flow pump with diffuser [3]

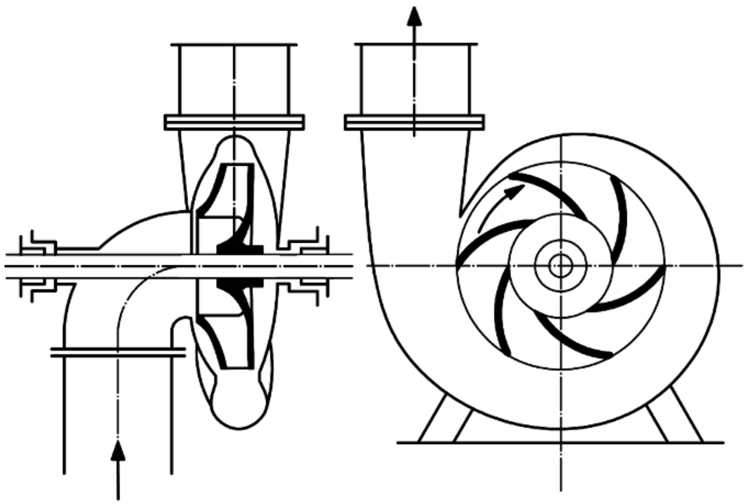


Fig. A.21 Volute casing in single-stage centrifugal pump [3]

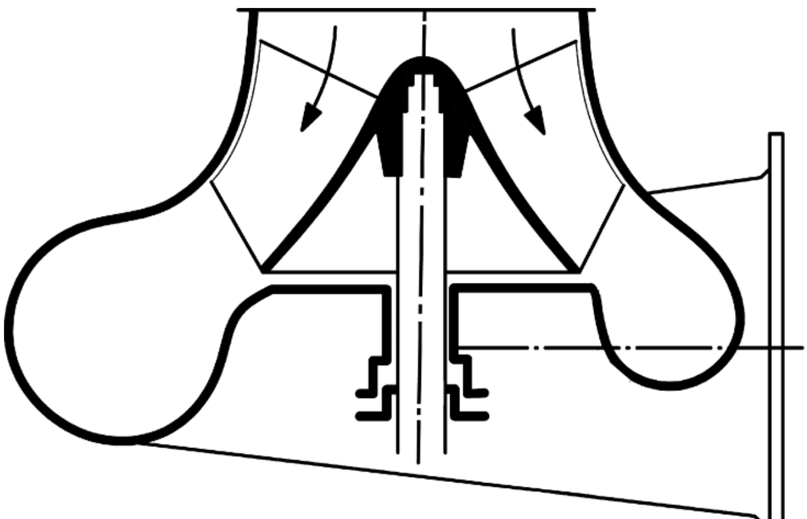


Fig. A.22 Volute casing in a mixed flow pump [3]

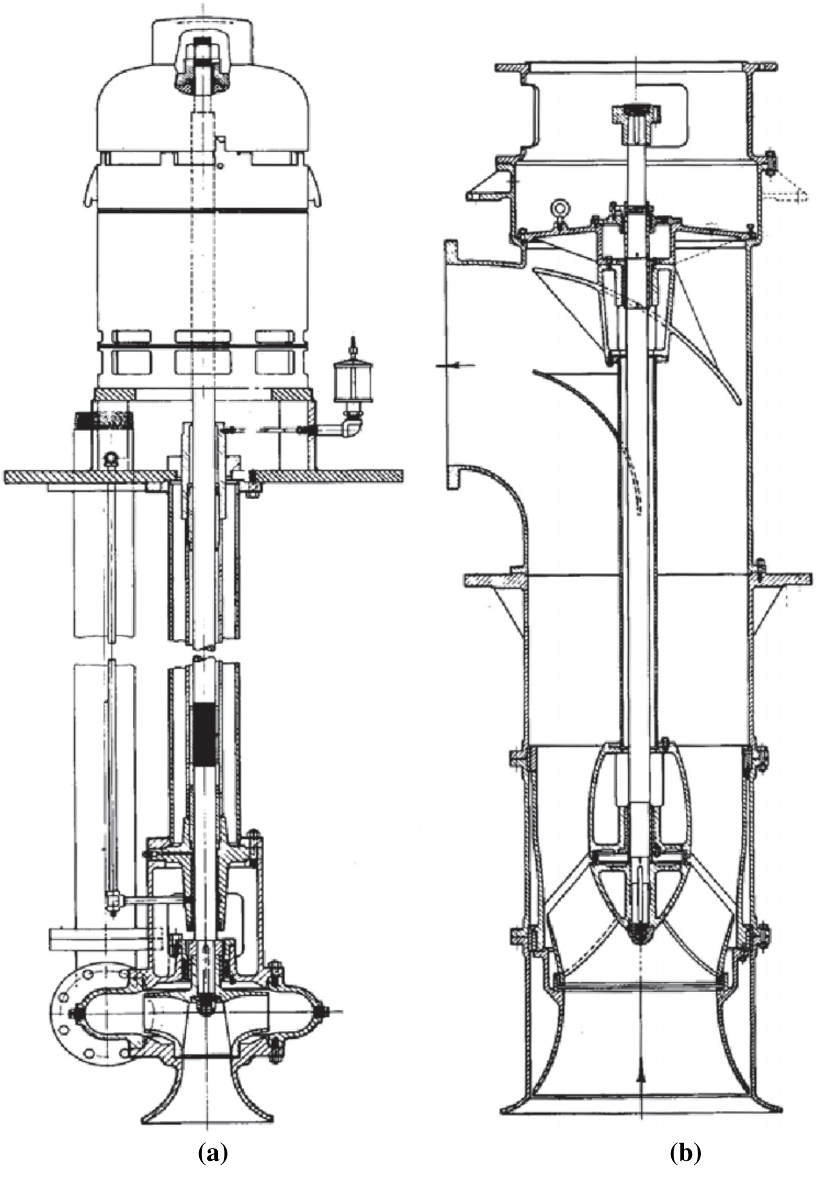


Fig. A.23 (a) Vertical wet-pit or non-clogging pump [1]. (b) Vertically modified propeller pump with removable bowl and shafting assembly (Worthington Pump, Inc.) [2]

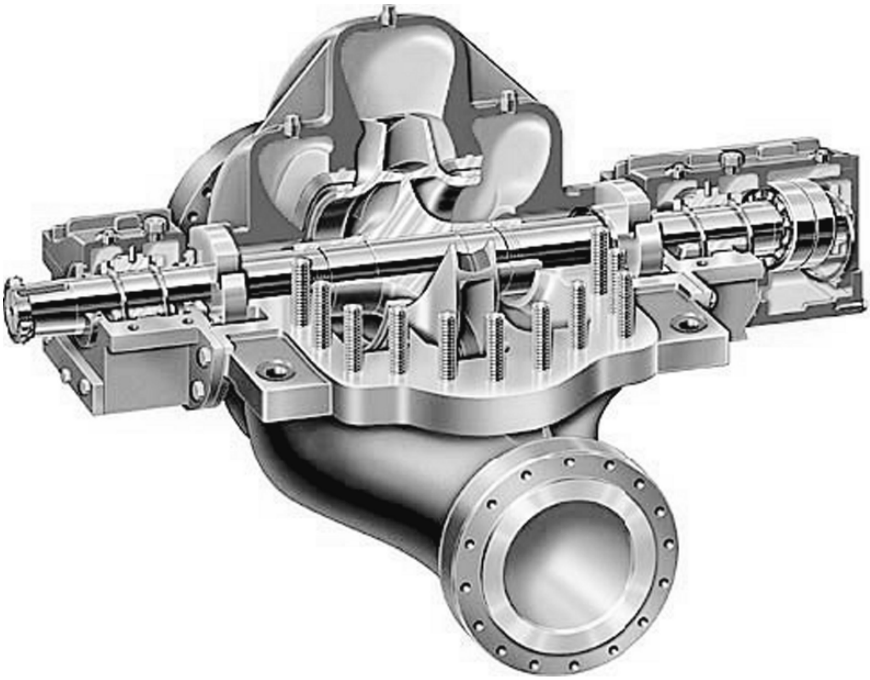


Fig. A.24 Casing of a multi-stage pump (www.flowserve.com)

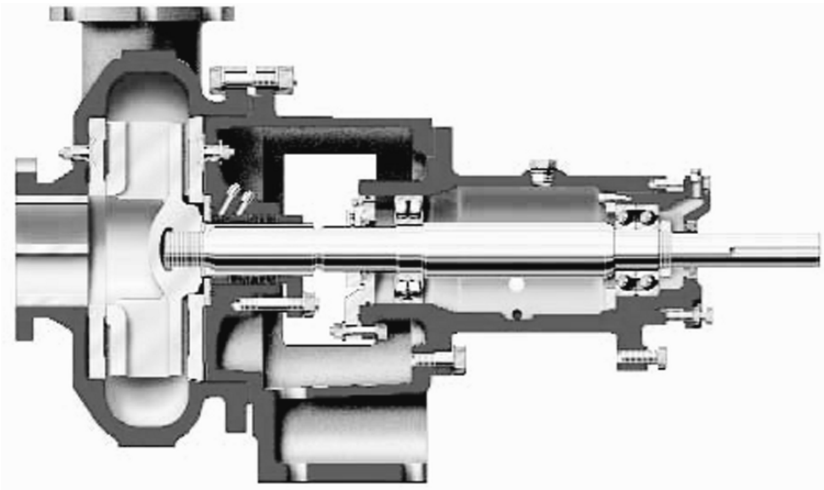


Fig. A.25 Casing of a single-stage pump (www.flowserve.com)

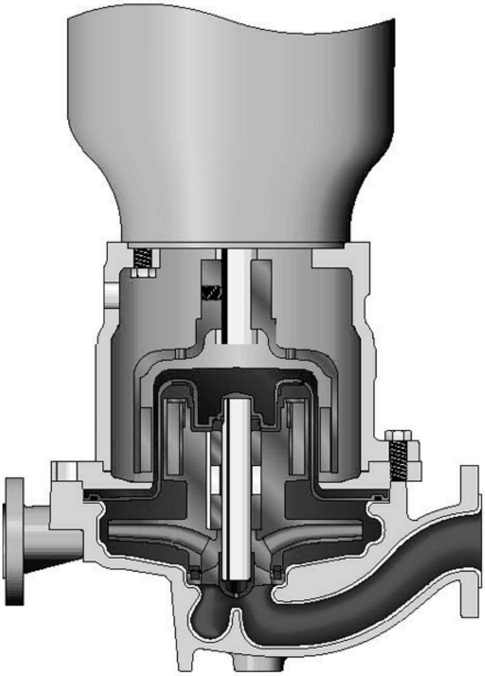


Fig. A.26 Vertical chemical pump (www.ksb-ltd.com)

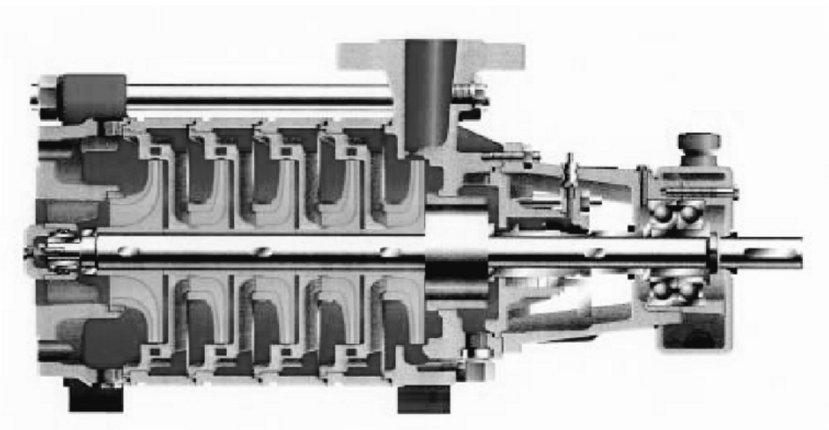


Fig. A.27 Horizontal multi-stage centrifugal pump (www.ksb-inc.com)

In designing a vaned diffuser, it is important that the cross sections of the flow passages change very gradually. Also, the number of vanes should not be equal to the number of impeller's blades. Otherwise, the vibration will increase. Usually, the number of vanes in the diffuser is one more than that in the impeller's blades.

A.7 Volute Casing

To collect the liquid after leaving the impeller or diffuser in horizontal centrifugal or mixed flow pumps, volute casing are used (Fig. A.21 and A.22). These volutes also form the casing for the whole pump assembly.

In vertical pumps, depending on the type of the pump and its design, the volute casing can be eliminated and only diffusers are used (Fig. A.23a). In axial pumps, on the other hand, it is not possible to use volute casing and only cylindrical casings are used (Fig. A.22b).

The reason for using the word ‘volute" for this part of the pump is that in the volute casing, the cross-sectional area of the flow gradually increases from the volute mouth, 0°, to the pump exit, 360°, creating a volute shape. This increase in flow cross section is done in the same direction as the rotational speed of the impeller.

It must be mentioned that depending on the pump application and the value of the hydraulic radial thrust in the volute, the shape of the volute may be different (see Section 5.3.2).

In Figs. A.21– A.23, a centrifugal pump, mixed flow, and vertical pumps with casing and vertical pump with diffuser are shown, respectively.

Volute casing of a pump must be manufactured such that assembling and dismantling of different parts of the pump can be done easily. Also, by reducing the number of parts in a pump, its production becomes economically more viable. Pump casings, depending on the type of the pump, number of stages, configuration of the suction, and delivery pipes, may have different shapes. In Figs. A.24–A.27 different types of casing are shown.

In multi-stage pumps, in order to balance the axial thrust, different methods are used for installing the impellers, inlet pipes, and outlet pipes. This will also change the general appearance of the multi-stage pumps.

References

1. Karassik Igor J. et al., "Pump Handbook", McGraw-Hill, 2001.
2. Karassik Igor J. et al., "Pump Handbook", McGraw-Hill, 1976.
3. Lazarkiewicz Stephen, "Impeller Pumps", Pergamon Press, 1965.
4. "Centrifugal Pump", Lexicon, KSB, Frankenthal, Germany, 1980.
5. "Centrifugal Pump Design", Technical Appendix, KSB, Frankenthal, Germany.
6. "Kubota Pump Handbook", Vol. 1, Technical Manual, Kubota Ltd., 1972.

Index