HANDBOOK OF HYDRAULIC FLUID TECHNOLOGY

MECHANICAL ENGINEERING

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edited by

GEORGE E. TOTTEN

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Preface

One of the most frustrating practices of my career has been the search for information on hydraulic fluids, which includes information on fluid chemistry; physical properties; maintenance practices; and fluid, system, and component design. Although some information on petroleum oil hydraulic fluids can be found, there is much less information on fire-resistant, biodegradable, and other types of fluids. Unfortunately, with few exceptions, fluid coverage in hydraulic texts is typically limited to a single-chapter overview intended to cover all fluids. Therefore, it is often necessary to perform a literature search or a time-consuming manual search of my files. Some time ago it occurred to me that others must be encountering the same problem. There seemed to be a vital need for an extensive reference text on hydraulic fluids that would provide information in sufficient depth and breadth to be of use to the fluid formulator, hydraulic system designer, plant maintenance engineer, and others who serve the industry.

Currently, there are no books dedicated to hydraulic fluid chemistry. Most hydraulic fluid treatment is found in handbooks, which primarily focus on hydraulic system hardware, installation, and troubleshooting. Most of these books fit into one of two categories. One type of book deals with hydraulic equipment, with a single, simplified overview chapter covering all hydraulic fluids but with a focus on petroleum-derived fluids. The second type of book provides fluid coverage with minimal, if any, discussion of engineering properties of importance in a hydraulic system.

The purpose of the *Handbook of Hydraulic Fluid Technology* is to provide a comprehensive and rigorous overview of hydraulic fluid technology. The objective is not only to discuss fluid chemistry and physical properties in detail, but also to integrate both classic and current fundamental lubrication concepts with respect to various classes of hydraulic fluids. A further objective is to integrate fluid dynamics with respect to their operation in a hydraulic system in order to enable the reader to obtain a broader understanding of the total system. Hydraulic fluids are an important and vital component of the hydraulic system.

The 23 chapters of this book are grouped into three main parts: hardware, fluid properties and testing, and fluids.

HARDWARE

Chapter 1 provides the reader with an overview of basic hydraulic concepts, a description of the components, and an introduction to hydraulic system operation. In Chapter 2, the rolling element bearings and their lubrication are discussed. An extremely important facet of any well-designed hydraulic system is fluid filtration. Chapter 3 not only provides a detailed discussion of fluid filtration and particle contamination and quantification, but also discusses fluid filterability.

An understanding of the physical properties of a fluid is necessary to understand the performance of a hydraulic fluid as a fluid power medium. Chapter 4 features a thorough overview of the physical properties, and their evaluation and impact on hydraulic system operation, which includes: viscosity, viscosity-temperature and viscosity-pressure behavior, gas solubility, foaming, air entrainment, air release, and fluid compressibility and modulus.

FLUID PROPERTIES AND TESTING

Viscosity is the most important physical property exhibited by a hydraulic fluid. Chapter 5 presents an in-depth discussion of hydraulic fluid viscosity and classification. The hydraulic fluid must not only perform as a power transmission medium but also lubricate the system. Chapter 6 provides a thorough review of the fundamental concepts involved in lubricating a hydraulic system. In many applications, fluid fire resistance is one of the primary selection criteria. An overview of historically important fire-resistance testing procedures is provided in Chapter 7, with a discussion of currently changing testing protocol required for industry, national, and insurance company approvals. Ecological compatibility properties exhibited by a hydraulic fluid is currently one of the most intensive research areas of hydraulic fluid technology. An overview of the current testing requirements and strategies is given in Chapter 8.

One of the most inexpensive but least understood components of the hydraulic system is hydraulic seals. Chapter 9 provides a review of mechanical and elastomeric seal technology and seal compatibility testing. An often overlooked but vitally important area is adequate testing and evaluation of hydraulic fluid performance in a hydraulic system. There currently is no consensus on the best tests to perform and what they reveal. Chapter 10 reviews the state-of-the-art of bench and pump testing of hydraulic fluids. Vibrational analysis not only is an important plant maintenance tool but is also one of the most important diagnostic techniques for evaluating and troubleshooting the operational characteristics of a hydraulic system. Chapter 11 provides an introductory overview of the use of vibrational analysis in fluid maintenance. No hydraulic system operates trouble-free forever. When problems occur, it is important to be able to identify both the problem and its cause. Chapter 12 provides a thorough discussion of hydraulic system failure analysis strategies.

FLUIDS

Although water hydraulics do not constitute a major fluid power application, they are coming under increasing scrutiny as ecocompatible alternatives to conventional hydraulic fluids. Chapter 13 offers an overview of this increasingly important technology.

The largest volume fluid power medium is petroleum oil. In Chapter 14, the reader is provided with a thorough overview of oil chemistry, properties, fluid maintenance, and change-out procedures. Chapter 15 reviews additive technology for petroleum oil hydraulic fluids.

There are various types of synthetic hydraulic fluids. Chapter 16 describes the more important synthetic fluids, with a focus on aerospace applications.

Chapters 17 to 20 describe fire-resistant hydraulic fluids. Emulsions, water-glycols, polyol esters, and phosphate esters are discussed individually and in depth in Chapters 17, 18, 19, and 20, respectively. This discussion includes fluid chemistry, physical properties, additive technology, maintenance, and hydraulic system conversion.

Vegetable oils are well-known lubricants that have been examined repeatedly over the years. Currently, there is an intensive effort to increase the utilization of various types of vegetable oils as an ecologically sound alternative to mineral oil hydraulic fluids. Chapter 21 provides a review of vegetable oil chemistry, recovery, and properties. The applicability of these fluids as hydraulic fluid basestocks is examined in detail.

Chapter 22 discusses electrorheological fluids, which are becoming increasingly interesting for use in specialized hydraulic applications.

In Chapter 23, various standardized fluid maintenance procedures are discussed and a summary of equivalent international testing standards is provided.

The preparation of a text of this scope was a tremendous task. I am deeply indebted to many colleagues for their assistance, without whom this text would not have been possible. Special thanks go to Dr. Stephen Lainer (University of Aachen), Prof. Atsushi Yamaguchi (Yokohama National University), Prof. Toshi Kazama (Muroran Institute of Technology), K. Mizuno (Kayaba Industrial Ltd.), and Jürgen Reichel (formerly with DMT, Essen, Germany).

Special thanks also goes to my wife, Alice, for her unending patience, and to Susan Meeker, who assisted in organizing and editing much of this material; to Glenn Webster, Roland J. Bishop, Jr., and Yinghua Sun, without whose help this text would never have been completed; and to Union Carbide Corporation for its support.

George E. Totten

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1

Basic Hydraulic Pump and Circuit Design

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1 INTRODUCTION

Hydraulics, according to Webster, is defined as "operated, moved, or effected by means of water." In the 17th century, it was discovered that a fluid under pressure could be used to transmit power [1]. Blaise Pascal (1623–1662) observed that if a fluid in a closed container was subjected to a compressive force, the resulting pressure was transmitted throughout the system undiminished and equal in all directions [1].

Hydraulics is by far the simplest method to transmit energy to do work. It is considerably more precise in controlling energy and exhibits a broader adjustability range than either electrical or mechanical methods. To design and apply hydraulics efficiently, a clear understanding of energy, work, and power is necessary.

In this chapter, fundamentals of hydraulic pump operation and circuit design will be provided. This will include the following:

- Hydraulic principles
- Hydraulic system components
- Hydraulic pumps and motors
- System design considerations

2 DISCUSSION

2.1 Hydraulic Principles

Work is done when something is moved. Work is directly proportional to the amount of force applied over a given distance according to the following relation,

$$\text{Work (ft-lbs)} = \text{Distance (ft)} \times \text{Force (lbs)} \tag{1.1}$$

Power is defined as the rate of doing work and has the units of foot-pounds per second. A more common unit of measure is "horsepower (hp)." Horsepower is defined as the amount of weight in pounds that a horse could lift 1 ft in 1 s (Fig. 1.1) [1]. By experiment, it was found that the average horse could lift 550 lbs. 1 ft in 1 s; consequently,

$$1 \text{ hp} = \frac{550 \text{ ft-lbs}}{\text{s}} \tag{1.2}$$

Energy is the ability to do work. It may appear in various forms, such as mechanical, electrical, chemical, nuclear, acoustic, radiant, and thermal. In physics, the Law of Conservation of Mass and Energy states that neither mass nor energy can be created or destroyed, only converted from one to the other.

In a hydraulic system, energy input is called a "prime mover." Electric motors and internal combustion engines are examples of prime movers. Prime movers and hydraulic pumps do not create energy; they simply convert it to a form that can be utilized by a hydraulic system.

The pump is the heart of the hydraulic system. When the system performs improperly, the pump is usually the first component to be investigated. Many times, the pump is described in terms of its pressure limitations. However, the hydraulic pump is a flow generator, moving a volume of fluid from a low-pressure region to a higher-pressure region in a specific amount of time depending on the rotation speed.

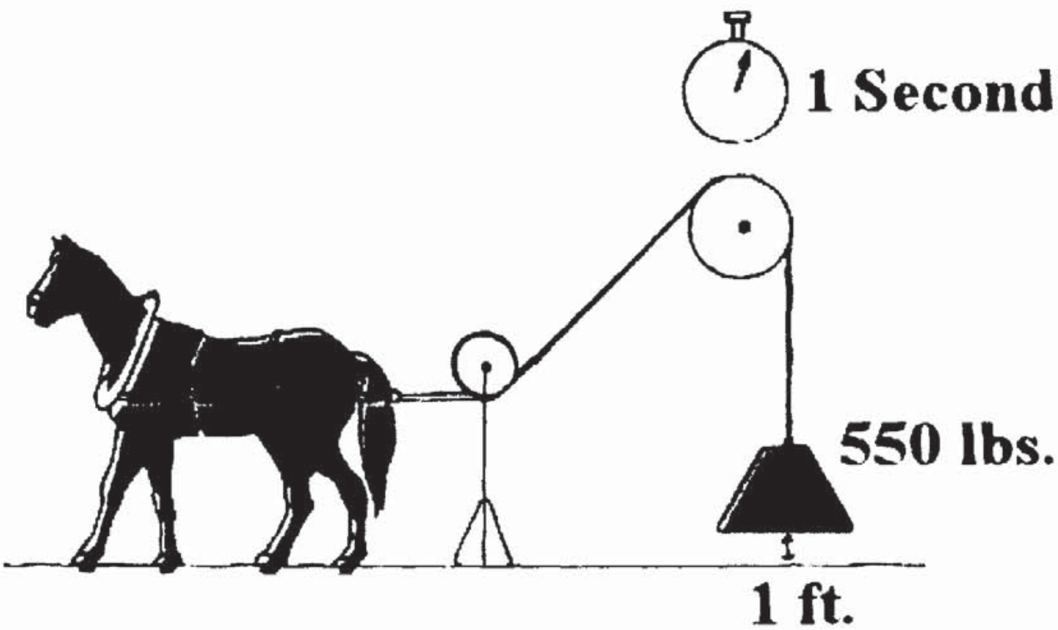


Figure 1.1 Illustration of the horsepower concept.

Therefore, the pump is properly described in terms of its displacement or the output flow rate expected from it.

All pumps used in a hydraulic system are of the positive displacement type. This means that there is an intentional flow path from the inlet to the outlet. Therefore, the pump will move fluid from the inlet or suction port to the outlet port at any pressure. However, if that pressure is beyond the pressure capability of the pump, failure will occur. The pressure which exists at the outlet port of the pump is a function of the load on the system. Therefore, hydraulic system designers will always place a pressure-limiting component (i.e., a relief valve) at the outlet port of a pump to prevent catastrophic failures from overpressurization.

Most pumps in hydraulic systems fall into one of three categories: vane pumps, gear pumps, or piston pumps (Sec. 2.2). The action of the hydraulic pump consists of moving or transferring fluid from the reservoir, where it is maintained at a low pressure and, consequently, a low-energy state [2]. From the reservoir, the pump moves the fluid to the hydraulic system where the pressure is much higher, and the fluid is at a much higher-energy state because of the work that must be done by the hydraulic system. The amount of energy or work imparted to the hydraulic system through the pump is a function of the amount of volume moved and the pressure at the discharge port of the pump:

$$\text{Work} \propto \text{Pressure} \times \text{Flow} \tag{1.3}$$

From an engineering standpoint, it is common to relate energy to force times distance:

$$\text{Work} \propto \text{Force} \times \text{Distance} \tag{1.4}$$

However, hydraulic pressure is force divided by area, and volume is area times distance:

$$\text{Pressure} = \frac{\text{Force}}{\text{Area}} \tag{1.5}$$

$$\text{Volume} = \text{Area} \times \text{Distance} \tag{1.6}$$

From these relationships, Eq. (1.7) shows that pressure times volume is equivalent to force times distance.

$$P\ (\text{lbs/in.}^2) \times V\ (\text{in.}^3) = F\ (\text{lbs}) \times D\ (\text{in.}) \tag{1.7}$$

The hydraulic pump is actually a "three-connection" component. One connection is at the discharge (outlet) port, the second is at the suction (inlet) port, and the third connection is to a motor or engine (Fig. 1.2) [1]. From this standpoint, the pump is a transformer. It takes the fluid in the reservoir, using the energy from the motor or engine, and transforms the fluid from a low-pressure level to a higher-pressure level. In fact, the hydraulic fluid is actually a main component of the hydraulic system, and as we will see throughout this book, has a major influence in the operation of the system. Hydraulic pumps are commonly driven at speeds from 1200 rpm to 3600 rpm or higher and maximum pressures may vary from <1000 psi (pounds per square inch) to >6000 psi. Tables 1.1 and 1.2 [1] show typical pressure and speed limitations for various types of pumps and motors.

In addition to pressure, there is also a temperature limitation imposed by the hydraulic fluid. This is caused by the decrease in viscosity as the fluid temperature

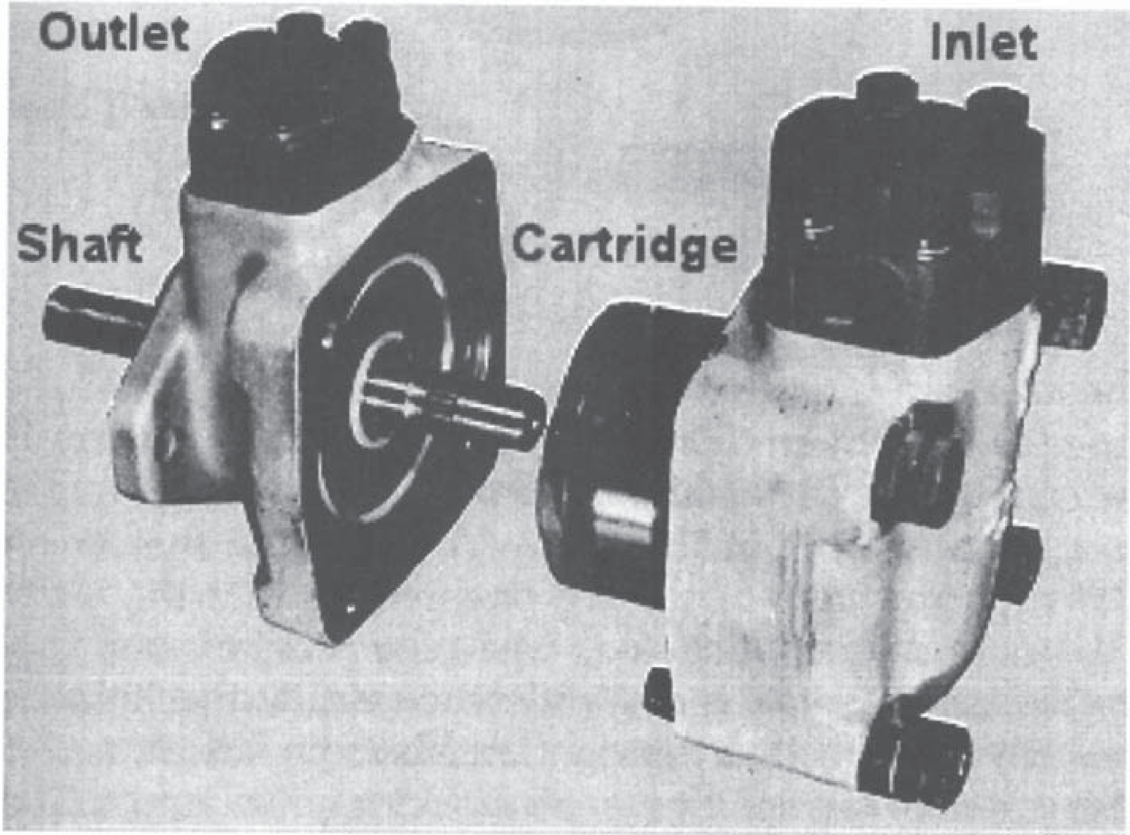


Figure 1.2 Illustration of pump connections.

increases. Generally, pump manufacturers set upper and lower viscosity limits on the hydraulic fluids used in their pumps. The upper viscosity limit determines the minimum temperature for pump start-up to prevent cavitation. Cavitation occurs when there is insufficient fluid flow into the pump inlet. During cavitation, the fluid will release any dissolved gases or volatile liquids. The gaseous bubbles produced will then travel into the high-pressure region of the pump where they will collapse under high pressure and may cause severe damage to the pump. Also, overheating of the pump bearings could result because of insufficient cooling as a result of inadequate flow.

Table 1.1 Typical Pump Performance Parameters for One Manufacturer

Hydraulic Pump Type	Flow	Maximum Pressure (psi)	Maximum Speed (rpm)	Total Efficiency (%)
External Gear	Fixed	3,600	500–5,000	85–90
Internal Gear	Fixed	3,000	900–1,800	90
Vane	Fixed	2,500	900–3,000	86
Vane	Variable	1,000–2,300	750–2,000	85
Radial Piston	Fixed	10,000	1,000–3,400	90
Axial Piston (bent axis)	Fixed	5,100–6,500	950–3,200	92
Axial Piston (bent axis)	Variable	5,800	500–4,100	92
Axial Piston (swash plate)	Variable	4,600–6,500	500–4,300	91

Table 1.2 Typical Motor Performance Parameters for One Manufacturer

Hydraulic Motor Type	Flow	Maximum Pressure (psi)	Maximum Speed (rpm)	Total Efficiency (%)
Gear	Fixed	3600	500–3000	85
Radial Piston	Fixed	6100–6500	1–500	91–92
Radial Piston	Variable	6100	1–500	92
Axial Piston (bent axis)	Fixed	5800–6500	50–6000	92
Axial Piston (bent axis)	Variable	6500	50–8000	92
Axial Piston (swash plate)	Variable	4600–5800	6–4900	91

The lower viscosity limit will establish the upper temperature limit of the fluid. If the upper temperature limit is exceeded, the viscosity will be insufficient to bear the high operating loads in the pump and, thus, lubrication failure will result in shortened pump life and/or catastrophic pump failure. Table 1.3 shows the effect of viscosity index on the pump operating temperatures for one major pump manufacturer [1].

Viscosity index is a measure of the viscosity change or resistance to flow of a liquid as the temperature is changed. A higher viscosity index produces a smaller viscosity change with temperature than a fluid having a lower viscosity index (see Chapter 4).

2.1.1 Torque and Pressure

The input parameters to the hydraulic pump from the prime mover are speed and torque. The input torque to the pump is proportional to the pressure differential between the inlet port and the discharge port:

$$\text{Torque}_{\text{input}} \propto (\text{Pressure}_{\text{outlet}} - \text{Pressure}_{\text{inlet}}) \times \text{Displacement} \tag{1.8}$$

The torque required to drive a positive displacement pump at constant pressure is [3–7]

$$T_a = T_t + T_v + T_f + T_c \tag{1.9}$$

where T_a is the actual torque required, T_t is the theoretical torque due to pressure differential and physical dimensions of the pump, T_v is the torque resulting from viscous shearing of the fluid, T_f is the torque resulting from internal friction, and T_c is the constant friction torque independent of both pressure and speed.

Table 1.3 A Pump Manufacturer's Viscosity Index and Temperature Guidelines

Temperature (°F)	Viscosity Index (50)	Viscosity Index (95)	Viscosity Index (150)
Minimum	18	5	0
Optimum	85–130	80–135	75–140
Maximum	155	160	175

Substituting the operational and dimensional parameters (using appropriate units) into Eq. (1.9) produces the following expression:

$$T_a = (P_1 - P_2)D_p + C_vD_p\mu N + C_f(P_1 - P_2)D_p + T_c \tag{1.10}$$

where P_1 is the pressure at the discharge port, P_2 is the pressure at the inlet port, D_p is the pump displacement, C_v is the viscous shear coefficient, μ is the fluid viscosity, N is the rotation speed of the pump shaft, C_f is the mechanical friction coefficient, and T_c is the breakaway torque.

Actual values of the parameters such as viscous shear coefficient, mechanical friction coefficient, and breakaway torque are determined experimentally. From Eq. (1.10), the required torque to drive a pump is primarily a function of the pressure drop across the pump ($P_1 - P_2$) and the displacement (D_p) of the pump.

2.1.2 Rotational Speed and Flow

The output flow of a hydraulic pump is described by

$$Q_a = Q_t - Q_l - Q_r \tag{1.11}$$

where Q_a is the delivery or actual flow rate of the pump, Q_t is the theoretical flow, Q_l is the leakage flow, and Q_r is the losses resulting from cavitation and aeration (usually neglected).

Substituting the operational (speed, flow, and pressure) and dimensional (pump displacement) parameters into Eq. (1.11) yields

$$Q_a = ND_p - \frac{C_sD_p(P_1 - P_2)}{\mu} \tag{1.12}$$

where Q_a is the delivery or actual flow rate of the pump and C_s is the leakage flow (slip) coefficient. From Eq. (1.12) it is apparent that pump delivery (Q_a) is primarily a function of the rotational speed (N) of the pump, and the losses are a function of the hydraulic load pressure ($P_1 - P_2$).

The leakage flow or slip that takes place in a positive displacement pump is caused by the flow through the small clearance spaces between the various internal parts of the pump in relative motion. These small leakage paths are often referred to as capillary passages. Most of these passages are characterized by two flat parallel plates with leakage flow occurring through the clearance space between the flat plates (Fig. 1.3) [1]. Therefore, the fundamental relationships for flow between flat plates are normally applied to the leakage flow in hydraulic pumps. However, to apply Eq. (1.12), it is necessary to obtain the value of the slip flow coefficient through experimental results [3,6].

2.1.3 Horsepower (Mechanical and Hydraulic)

Of interest to the designers and users of hydraulic systems is the power (Hp_{input}) required to drive the pump at the pressure developed by the load and the output power ($\text{Hp}_{\text{output}}$), which will be generated by the pump. The expression (using appropriate units) which describes the "mechanical" input horsepower required to drive the pump is

$$\text{Hp}_{\text{input}} = \frac{T_aN}{5252} \tag{1.13}$$

where Hp_{input} is the required input power (hp), T_a is the actual torque required (lb-

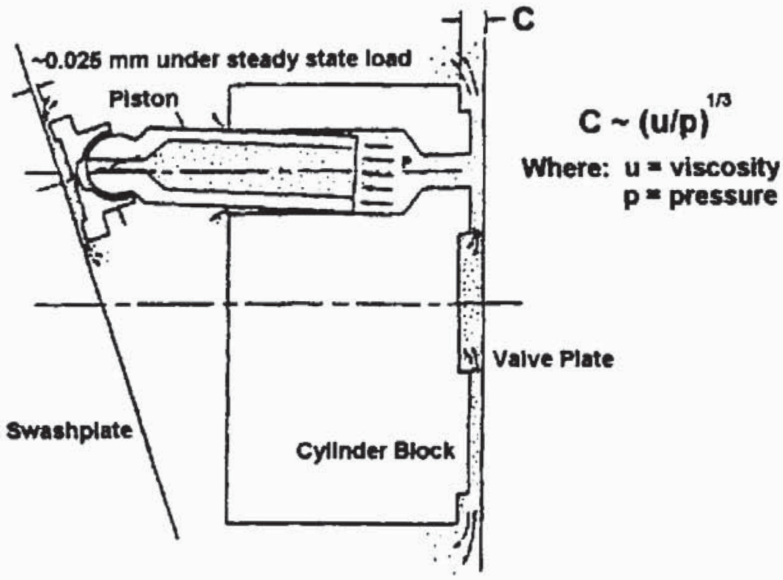


Figure 1.3 Illustration of the leakage path between two flat plates in a piston pump.

ft), and N is the rotational speed of the pump shaft (rpm). The "hydraulic" output horsepower of a hydraulic pump is described by the expression

$$Hp_{output} = \frac{PQ}{1714} \tag{1.14}$$

where Hp_{output} is the power output of the pump (hp), P is the pressure at the pump discharge port (psi), and Q is the delivery or actual flow output (gpm).

However, in the real world, $Hp_{input} > Hp_{output}$. This is always true because no pump or motor is 100% efficient. As will be shown in the next subsection, this is mainly due to internal mechanical friction and fluid leakage within the pump.

2.1.4 Pumping Efficiency

Hydraulic pumping efficiency or total efficiency (E_t) is a combination of two kinds of efficiencies: volumetric (E_v) and mechanical (E_m). Volumetric efficiency is given by

$$E_v = \frac{\text{Actual flow output}}{\text{Theoretical flow output}} \tag{1.15}$$

The second kind of efficiency is called the torque efficiency or the mechanical efficiency (E_m). The mechanical efficiency is described by

$$E_m = \frac{\text{Theoretical torque input}}{\text{Actual torque input}} \tag{1.16}$$

The overall or total efficiency (E_t) is defined by

$$E_t = E_v E_m \tag{1.17}$$

The total efficiency is also related to power consumption by

$$E_t = \frac{\text{Power output}}{\text{Power input}} \tag{1.18}$$

Substituting Eqs. (1.13) and (1.14) into Eq. (1.18) produces the following expression for the overall efficiency of a hydraulic pump:

$$E_t = \frac{0.326T_a N}{PQ} \tag{1.19}$$

Theoretical pump delivery is also determined from the dimensions of the pump. However, if the dimensions of the pump are not known, they are determined by pump testing. The overall efficiency of a pump may also be measured by testing. However, it is difficult to measure mechanical efficiency. This is because internal friction plays a major role and there is no easy way of measuring this parameter within a pump. Rearrangement of Eq. (1.17) gives the mechanical efficiency as the overall efficiency divided by the volumetric efficiency:

$$E_m = \frac{E_t}{E_v} \tag{1.20}$$

2.1.5 Hydraulic System Design

When designing a hydraulic system, the designer must first consider the load to be moved or controlled. Then, the size of the actuator is determined. The "actuator" is a component of a hydraulic system which causes work to be done, such as a hydraulic cylinder or motor. The actuator must be large enough to handle the load at a pressure within its design capability. Once the actuator size is determined, the speed at which the load must move will establish the flow rate of the system (gpm; gallons per minute):

For hydraulic cylinders,

$$\text{gpm} = \frac{AV}{231} \tag{1.21}$$

where A is the area (in.^{3}) and V is the velocity (in./min)
For hydraulic motors,

$$\text{gpm} = \frac{(D)(\text{rpm})}{231} \tag{1.22}$$

where D is the displacement (in.^{3}/rev.)

For example, calculate the hydraulic cylinder bore diameter (D) and flow (Q) required to lift a 10,000 lb load at a velocity of 120 in./min with a hydraulic load pressure not exceeding 3000 psi. Using the fundamental hydraulic expression,

$$\text{Force} = \text{Pressure} \times \text{Area} \tag{1.23}$$

Force is equal to 10,000 lbs and pressure is 3000 psi. The area of the head end of the cylinder is calculated by rearranging Eq. (1.23) and solving for the area:

$$\begin{aligned} \text{Area} &= \frac{\text{Force}}{\text{Pressure}} = \frac{10{,}000}{3{,}000} \\ &= 3.333 \text{ in.}^2 \\ &= \frac{\pi D^2}{4} \\ D &= 2.06 \text{ in.} \end{aligned} \tag{1.24}$$

Therefore, a double-acting single-rod cylinder with a cylinder bore of 2.25 or 2.5 in. may be used, depending on the requirements of the head side of the cylinder (Fig. 1.4) [1]. However, on the rod side, the area of the piston is reduced by the area of the rod. Because the effective area on the rod side of the cylinder is less than that on the head side, the cylinder would not be able to lift as large a load when retracting, because of pressure intensification at the rod end causing the system relief valve to open. For this reason, all single-rod cylinders exert greater force at the rod end when extending than retracting. On the other hand, a double-rod cylinder of equal rod diameters would exert an equal force in both directions (extending and retracting).

Once the size of the cylinder is selected, the designer must consider the speed requirements of the system. The speed at which the load must be moved is dependent on the pump flow rate. Of course, the velocity of the cylinder rod and the speed of the load must be the same. By using the equation

$$Q = \frac{VA}{231} \tag{1.25}$$

where Q is the flow rate into the cylinder (gpm), V is the velocity of the cylinder rod (in./min), and A is the cylinder area (in.^{2}).

The flow rate (Q) needed to produce a specific velocity (V) can now be calculated by combining Eqs. (1.24) and (1.25); using a 2.5-in.-diameter cylinder as an example,

$$\begin{aligned} Q &= \frac{\pi D^2 V}{924} \\ &= \frac{(3.14)(2.5)^2(120)}{924} \\ &= 2.6 \text{ gpm} \end{aligned} \tag{1.26}$$

Once the pressure needed to support the load and flow to produce the specified load velocity is determined, the pump selection process and system design may begin.

There are several factors in pump selection and hydraulic system design that have not been addressed. For example, the service life required by the system must be decided along with the contamination level that must not be exceeded in the system. The piping sizes and the inlet conditions to the pump must be considered. The fluid to be used in the hydraulic system is an obvious consideration. These and other factors will be addressed in later sections of this chapter.

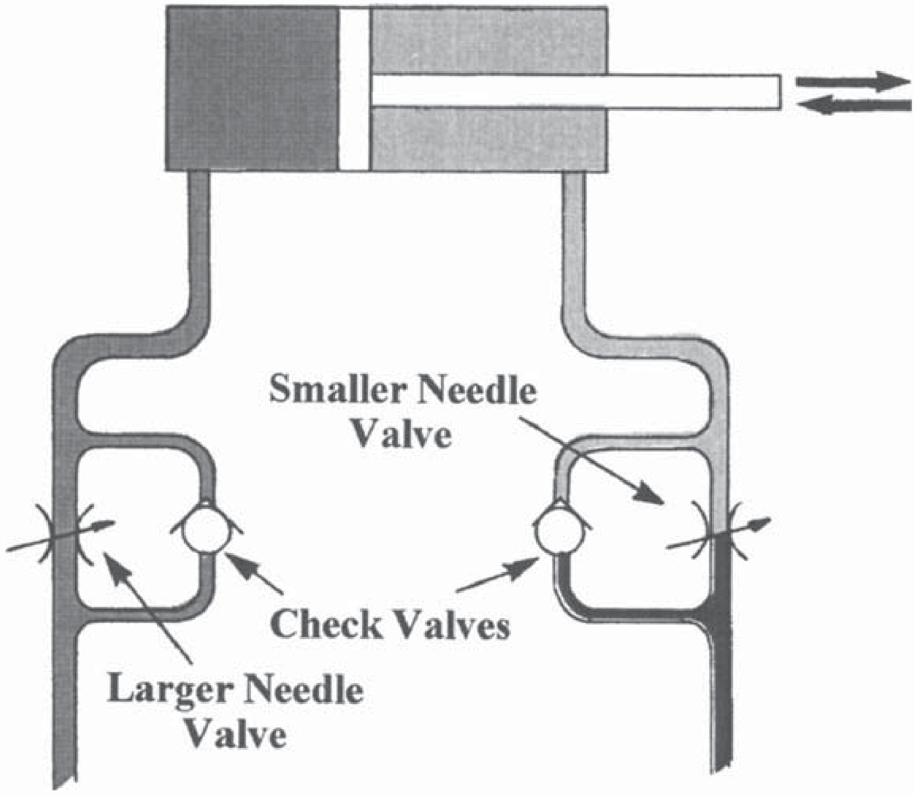


Figure 1.4 Illustration of a single-rod double-acting cylinder.

2.2 Hydraulic Pumps and Motors

There are three types of pumps used predominately in hydraulic systems: vane pumps, gear pumps, and piston pumps (Fig. 1.5) [1].

Although there are many design parameters that differ between a hydraulic pump and a hydraulic motor, the general description is fundamentally the same, but their uses are quite different. A pump is used to convert mechanical energy into hydraulic energy. The mechanical input is accomplished by using an electric motor or a gasoline or diesel engine. Hydraulic flow from the output of the pump is used to power a hydraulic circuit. On the other hand, a hydraulic motor is used to convert hydraulic energy back into mechanical energy. This is accomplished by connecting the output shaft of the hydraulic motor to a mechanical actuator, such as a gear box, pulley, or flywheel.

2.2.1 Vane Pumps

A typical design for the vane pump is shown in Fig. 1.6 [8]. The vane pump relies upon sliding vanes riding on a cam ring to increase and decrease the volume of the pumping chambers within the pump (Fig. 1.7). The sides of the vanes and rotor are sealed by side bushings. Although there are high-pressure vane pumps (>2500 psi), this type of pump is usually thought of as a low pressure pump (<2500 psi), see Table 1.1.

There are two vane pump designs. One is a balanced design, whereas the other is unbalanced. In the balanced design, there are opposing pairs of internal inlet and outlet ports which distribute the thrust force evenly around the shaft (Fig. 1.8) [8]. All modern vane pumps are of the balanced design. The vane pump is considered

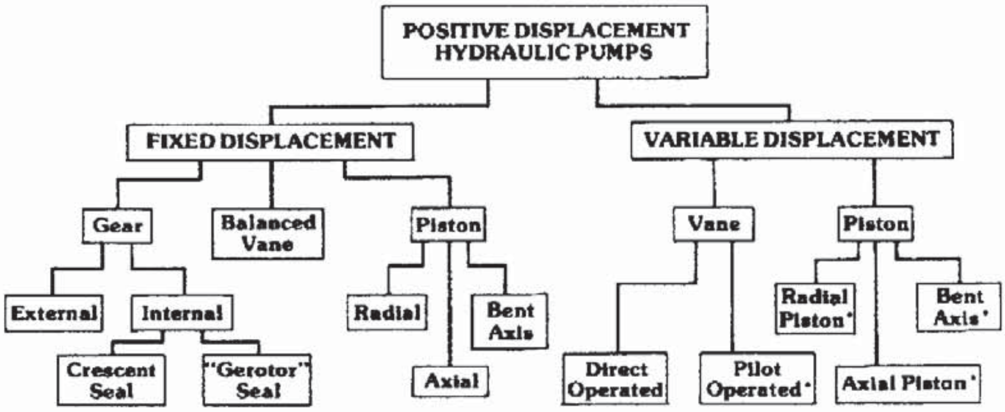


Figure 1.5 The family of hydraulic pumps.

one of the simplest of all positive displacement pumps and can be designed to produce variable displacement; that is, the output flow can be changed to suit the needs of the hydraulic system (Fig. 1.9) [8]. Fluid leakage in vane pumps occurs between the high- and low-pressure sides of the vanes and across the side bushings which results in decreased volumetric efficiency and, hence, reduced flow output. The unbalanced design suffers from shortened bearing life because of the unbalanced thrust force within the pump.

2.2.2 Gear Pumps

It is generally agreed that the gear pump is the most robust and rugged type of fluid power pump. Although there are many gear-type pumps, three are used predominately for hydraulic service. One is the external gear pump (Fig. 1.10) [8] and the other two are internal gear pumps of the "crescent" seal and "gerotor" seal type (Figs. 1.11 and 1.12, respectively) [8].

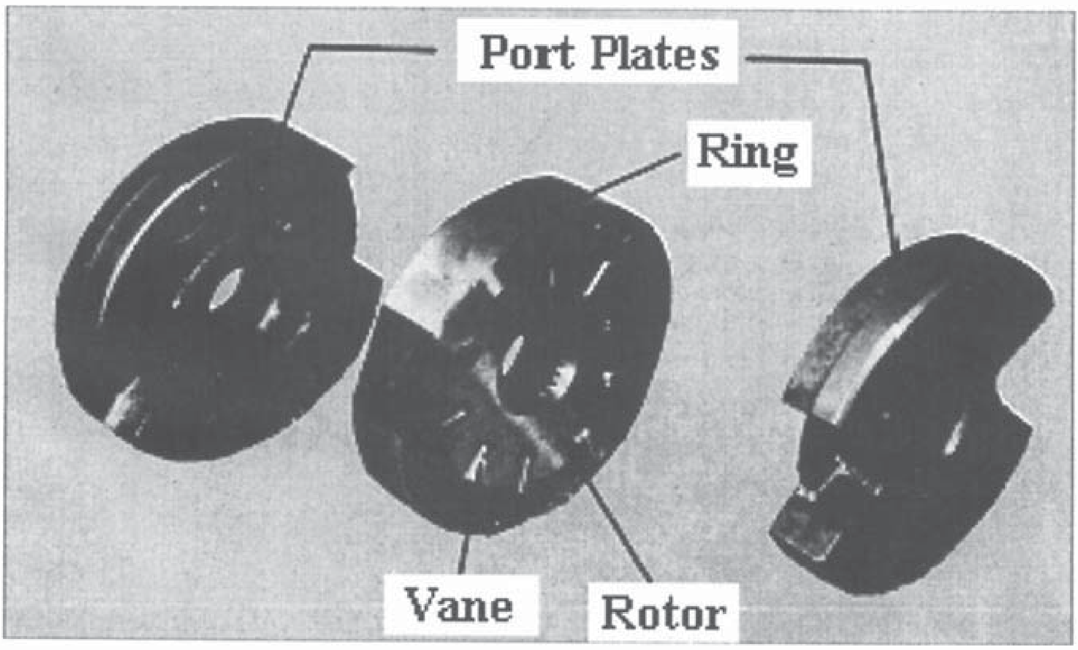


Figure 1.6 Illustration of a typical vane pump.

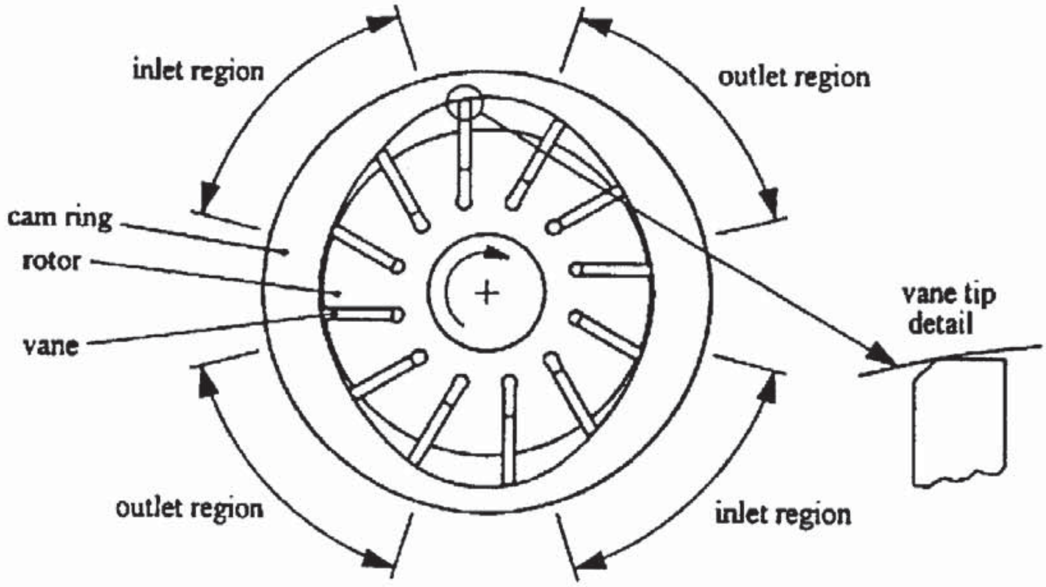


Figure 1.7 Illustration of cam ring and vanes in a vane pump.

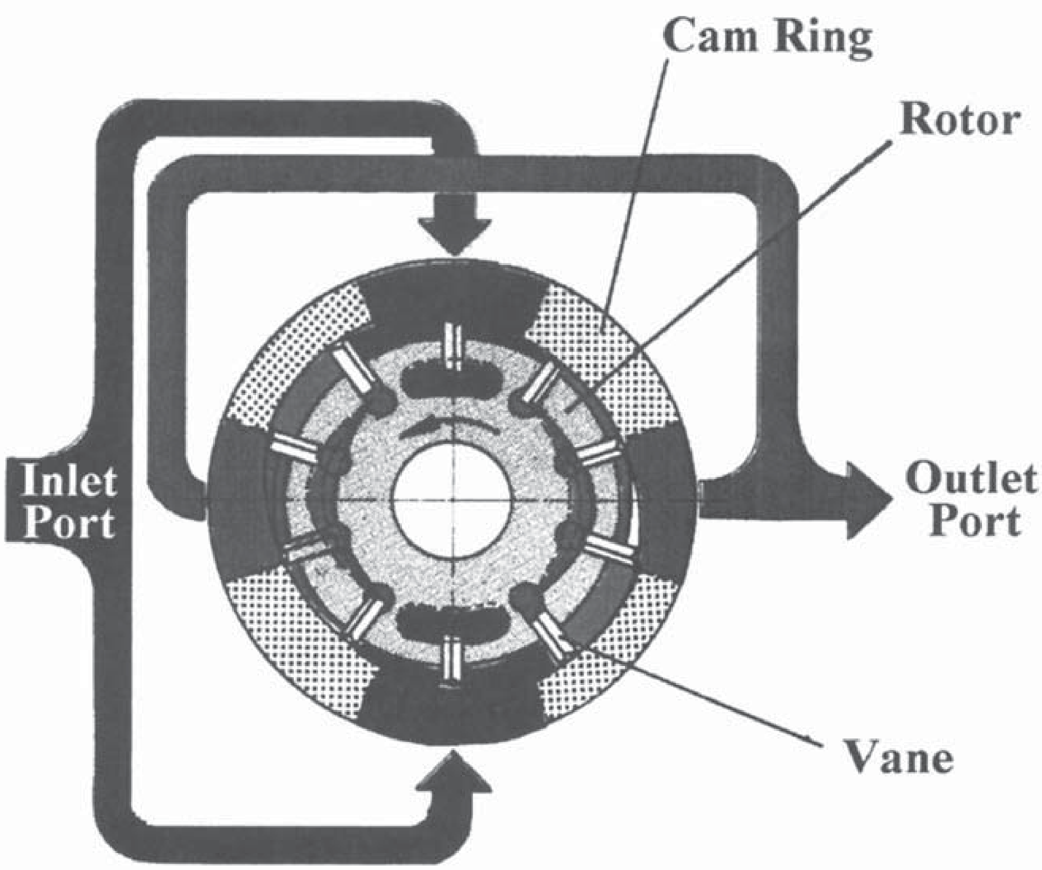


Figure 1.8 Illustration of a balanced-design vane pump.

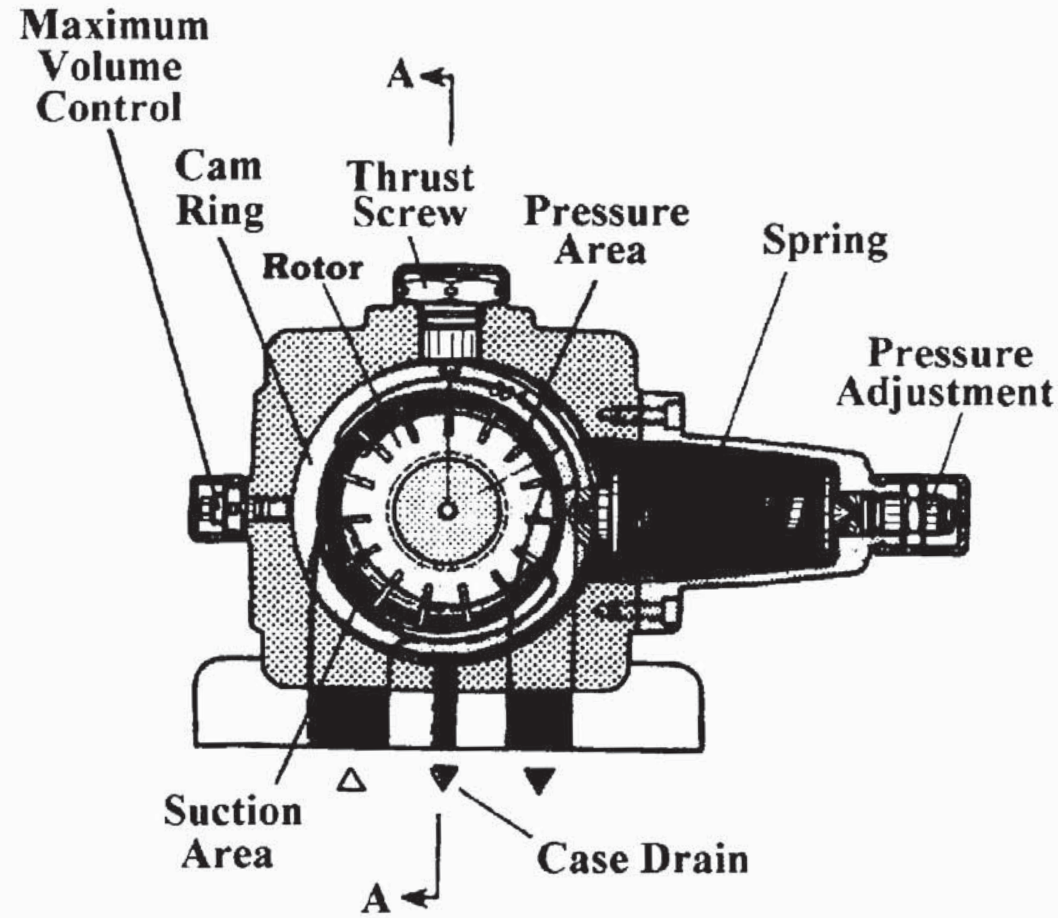


Figure 1.9 Illustration of a variable-displacement vane pump.

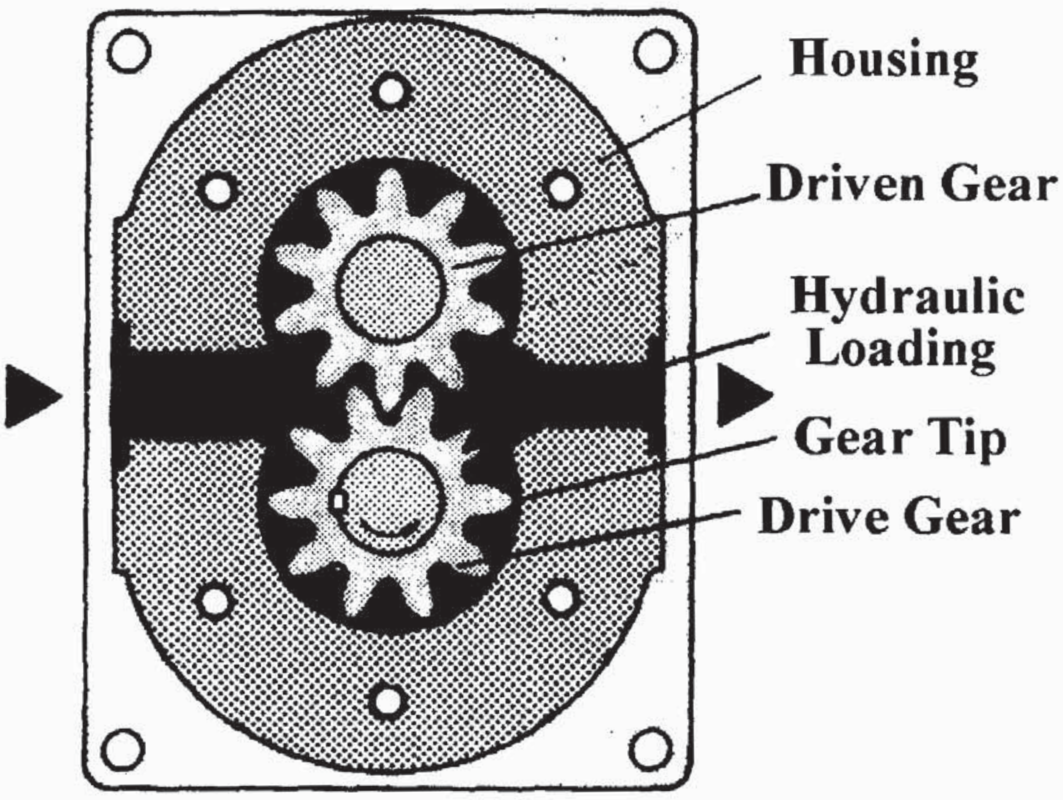


Figure 1.10 Illustration of an external gear pump.

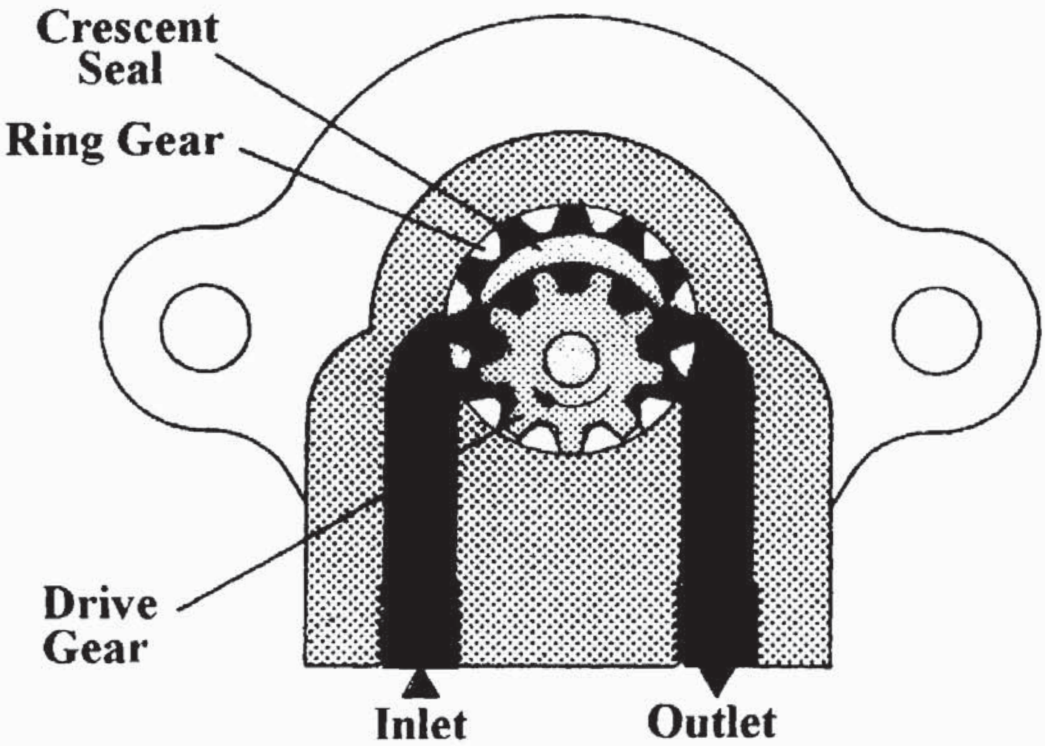


Figure 1.11 Illustration of an internal (crescent) gear pump.

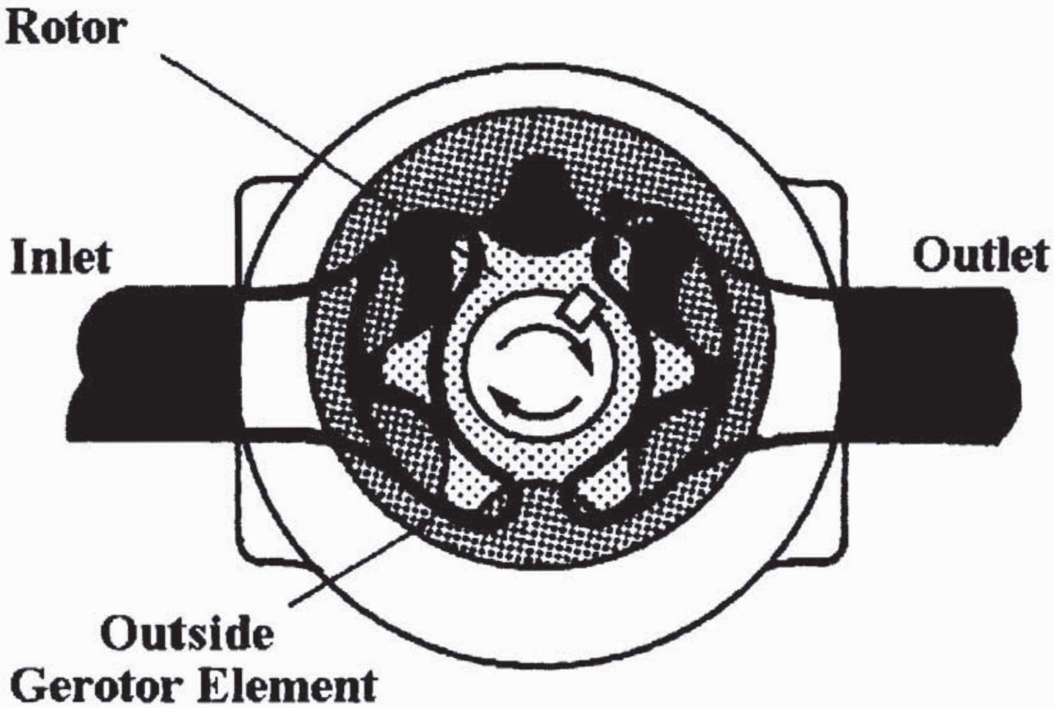


Figure 1.12 Illustration of an internal (gerotor) gear pump.

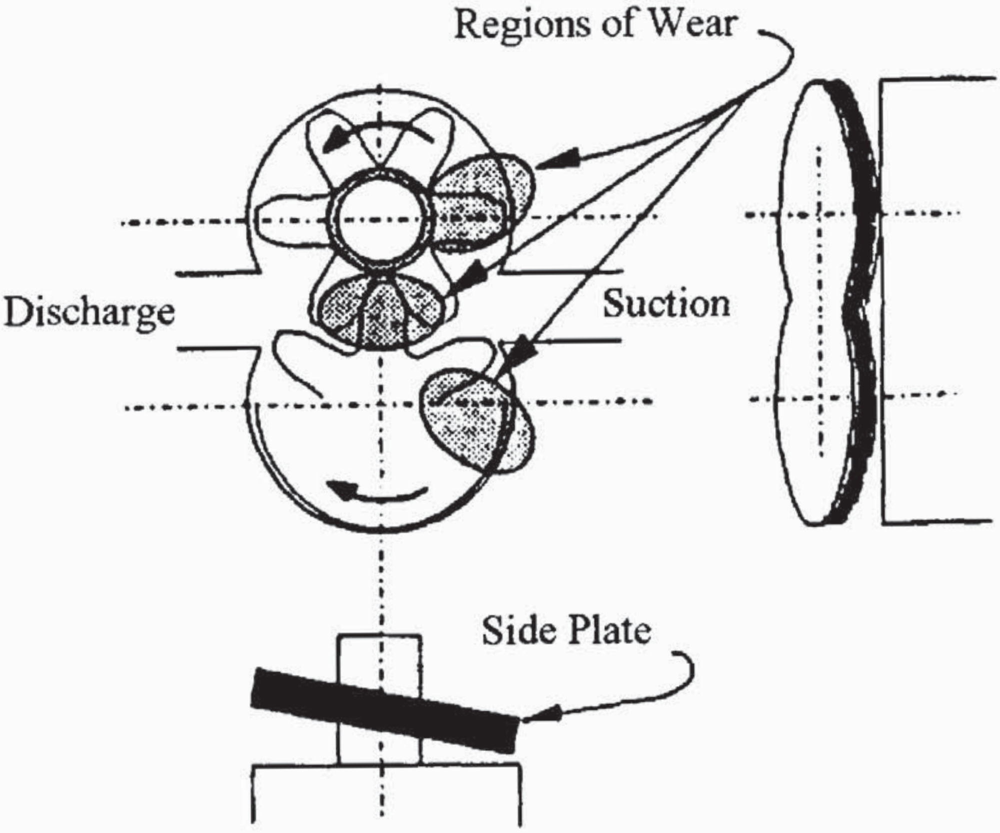


Figure 1.13 Illustration of the wear and leakage areas of the external gear pump.

The external gear pump is the most prevalent (Fig. 1.10) [8]. Note that there are two gears: a drive and a driven gear. The number of teeth, the pitch circle diameter, and the width of the gears are the dominant parameters which control the displacement. The gears are enclosed by the housing and a side plate. Fluid leakage in this type of pump occurs between the tips of the gears and across the side plate (Fig. 1.13) [8].

The crescent seal internal gear pump consists of a small internal gear and a larger ring gear (Fig. 1.11) [8]. The small internal gear is driven by the prime mover. The internal gear meshes with the ring gear and turns it in the same direction. The sealing of the high-pressure chamber from the pump's inlet is achieved by a crescent seal between the upper teeth of the internal small gear and the upper teeth of the ring gear. In the gerotor gear pump, the inner gerotor has one less tooth than the outer element (Fig. 1.12) [8]. The internal gear is driven by the prime mover and, in turn, drives the outer element in the same direction. There is no satisfactory gear pump design in which the displacement can be varied.

2.2.3 Piston Pumps

The piston pump is operated at the highest pressure of all of the pumps normally found in hydraulic applications. The piston pump is manufactured in the axial, bent-axis, and radial configurations. In addition, there are both fixed- and variable-displacement bent-axis configurations (Figs. 1.14 and 1.15, respectively) [8]. The axial design configuration predominates in hydraulic systems and will be the basis of the discussions here. A typical example of an axial fixed-displacement piston pump is

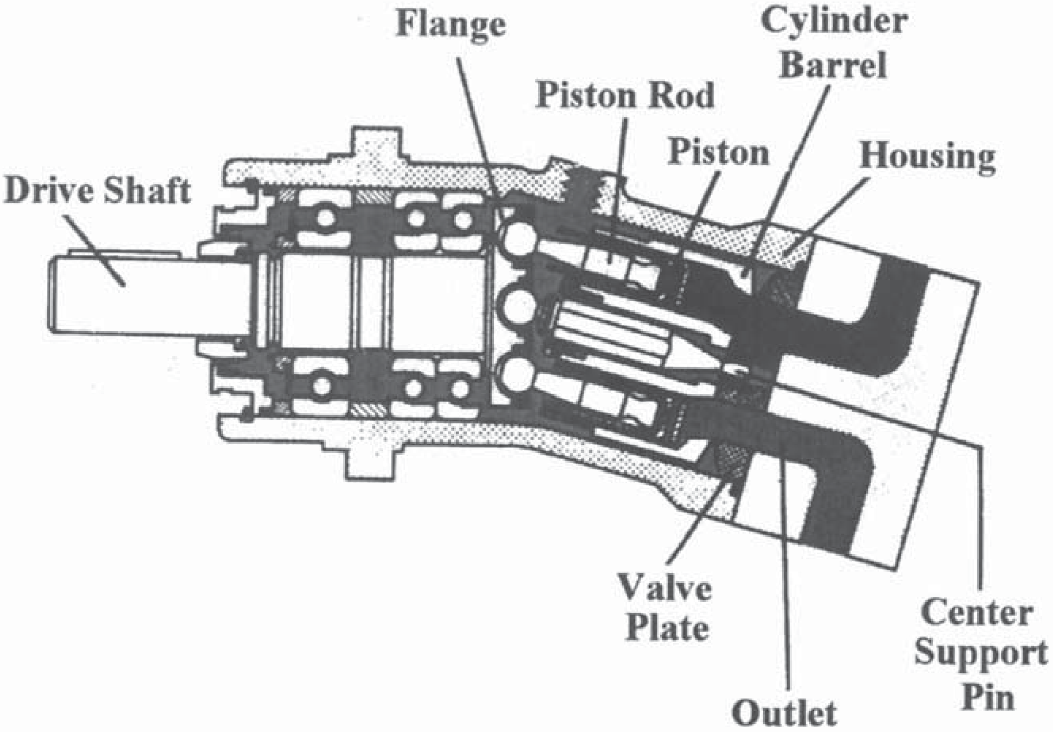


Figure 1.14 Illustration of a bent-axis fixed-displacement piston pump.

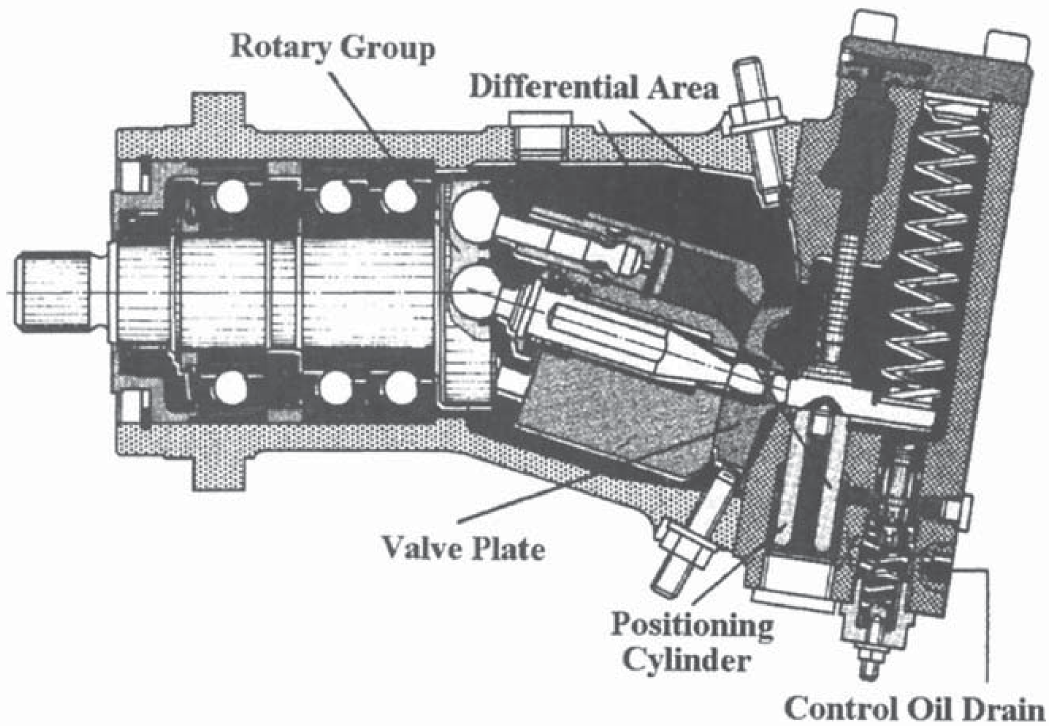


Figure 1.15 Illustration of a bent-axis variable-displacement piston pump.

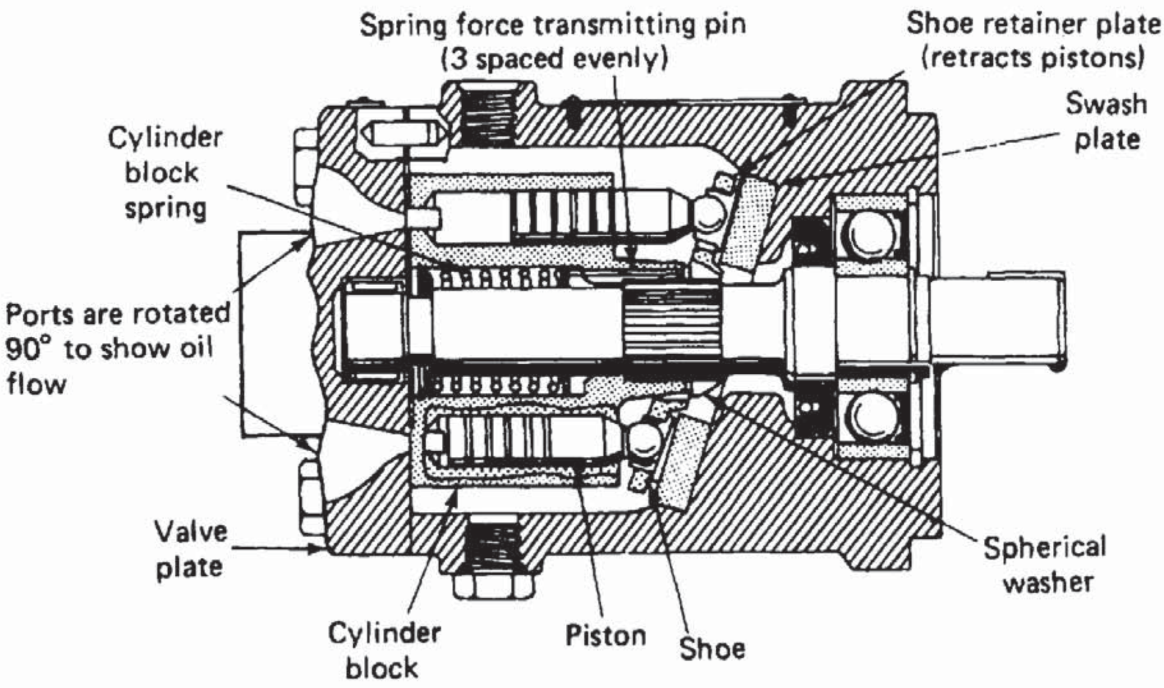


Figure 1.16 Illustration of an axial fixed-displacement piston pump.

shown in Fig. 1.16 [2] and a typical example of an axial variable-displacement piston pump is illustrated in Fig. 1.17 [2]. Variable-displacement piston pumps lend themselves to the incorporation of various valve mechanisms that will alter the performance of this pump; for example, it can be a pressure-compensated pump in one configuration, where the valve mechanism will alter the displacement of the pump to limit the outlet pressure to some preselected value. A pressure-compensated piston pump is illustrated in Fig. 1.17 [2].

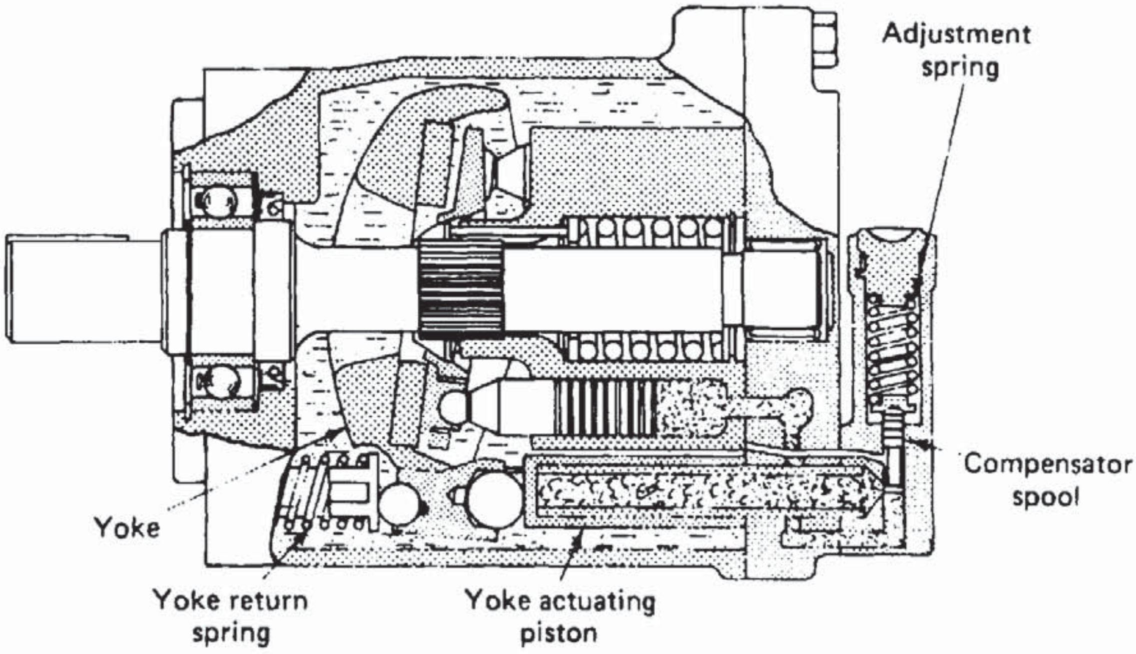


Figure 1.17 Illustration of an axial variable-displacement piston pump.

As seen in Figs. (1.16) and (1.17), the major components of a piston pump are the pistons, the piston or cylinder block, valve plate, piston shoes, swash plate, and the drive shaft. In operation, the shaft drives the piston block, which, in turn, rotates the pistons. The pistons are held against the swash plate by springs and a retainer plate. For the piston pump to produce a flow, the swash plate must be at some angle relative to the centerline of the shaft, which is also the axial center of rotation of the pistons. The pistons ride on the surface of the swash plate and the angle will force the pistons to move in and out of the piston or cylinder block. The greater the swash-plate angle, the larger the piston stroke and the greater the displacement of the pump. The dependence of pump displacement (V) on the swash-plate angle α is shown by the following equation and Fig. 1.18 [9]:

$$V = \frac{\pi d_k^2}{4} D_k(\tan \alpha) \tag{1.27}$$

where V is the pump displacement (in.^{3}), d_k is the piston bore diameter (in.), D_k is the piston bore circle diameter (in.), and α is the swash-plate angle (in degrees).

For an axial piston pump bent-axis design, the pump displacement (V) is described by the bent-axis angle α according to the following equation and Fig. 1.19 [9]:

$$V = \frac{\pi d_k^2}{4} (2r_h z)(\sin \alpha) \tag{1.28}$$

where r_h is the piston bore circle radius and z is the number of pistons.

The third kind of piston pump is the radial design (Fig. 1.20) [8]. In general, the radial piston pump has the highest continuous-pressure capability than any other type of pump (Table 1.1). Figure 1.21 [8] shows the basic configuration of a three-piston pump. The pistons are positioned radially to an eccentric drive shaft. Each

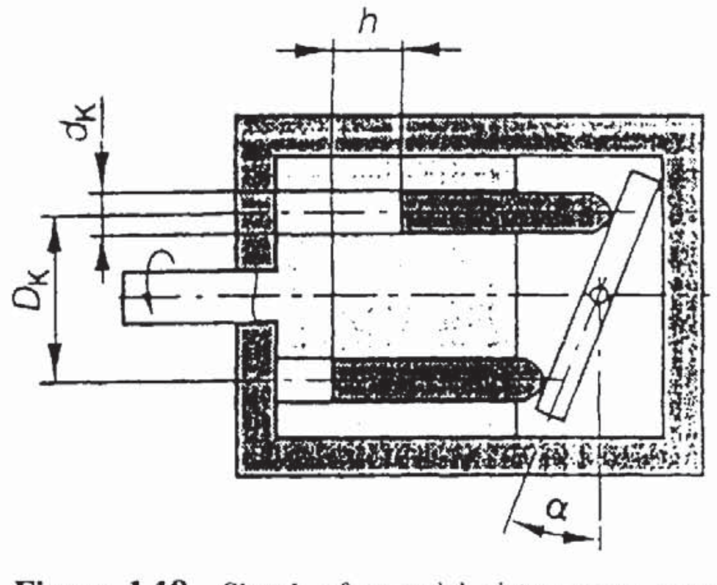


Figure 1.18 Sketch of an axial piston pump swash-plate design showing displacement angle α.

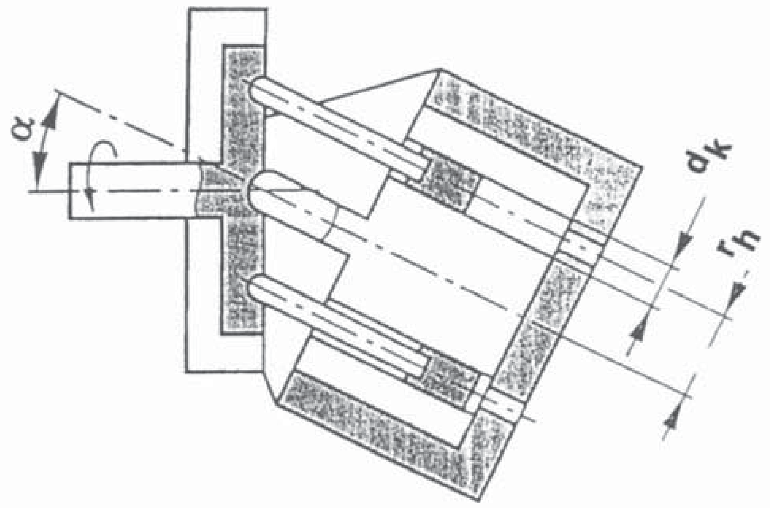


Figure 1.19 Sketch of an axial piston pump bent-axis design showing displacement angle α.

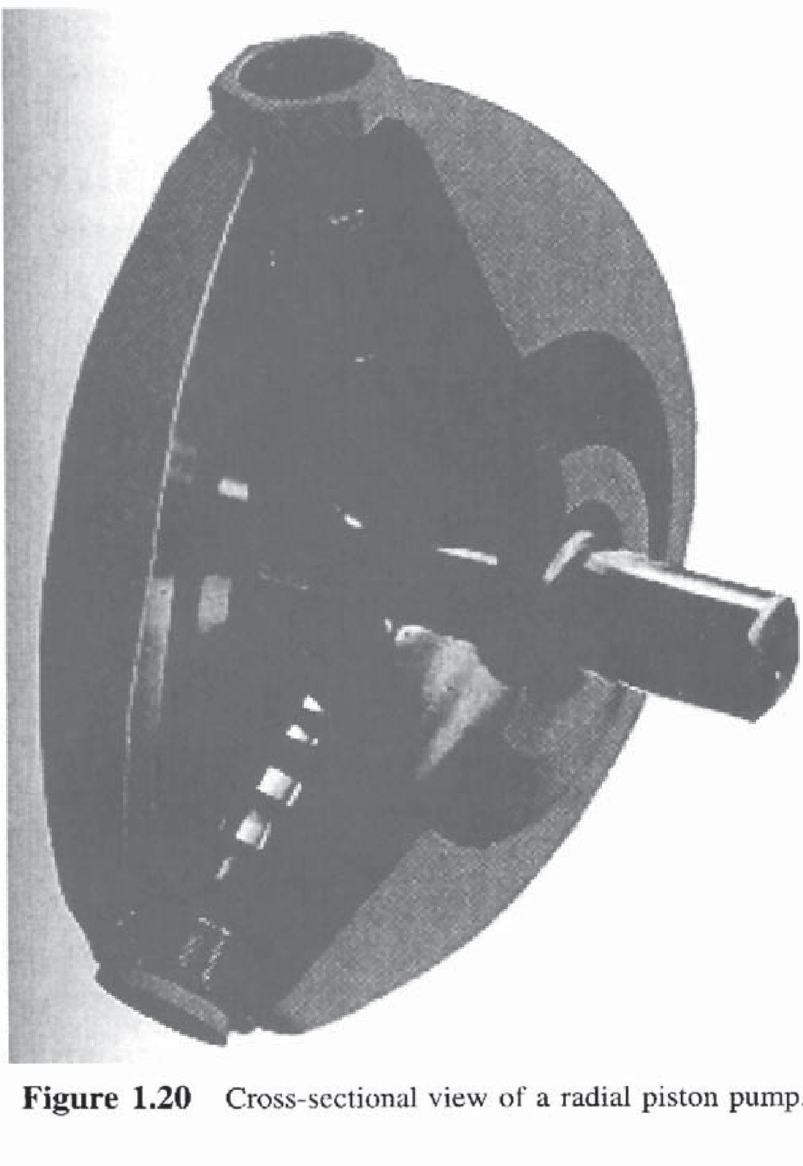


Figure 1.20 Cross-sectional view of a radial piston pump.

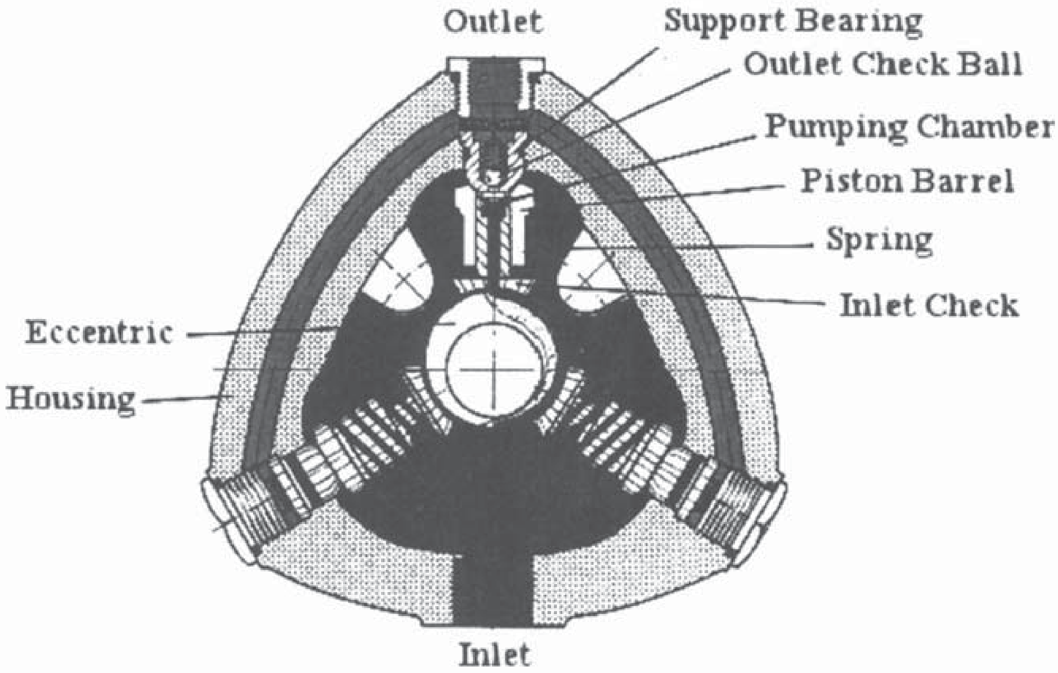


Figure 1.21 Illustration of the interior of a radial piston pump.

hollow piston consists of an inlet check valve, a spring, a piston barrel, a pumping chamber, an outlet check ball, and a support bearing.

As the drive shaft is rotated, the spring holds the base of the piston in contact with the eccentric cam shaft. The downward motion of the piston causes the volume to increase in the pumping chamber. This creates a reduced pressure that allows the inlet check valve to open, allowing oil to enter the pumping chamber. The oil enters the chamber by way of a groove machined into the cam-shaft circumference. Further rotation of the cam shaft causes the piston to move back into the cylinder barrel. The rapid rise in chamber pressure closes the inlet check valve. When the rising pressure equals the system pressure, the outlet check valve opens, allowing flow to exit the piston into the pressure port of the pump. The resulting flow is the sum of all the piston displacements. The number of pistons that a radial pump can have is only limited by the spatial restrictions imposed by the size of the pistons, housing, and cam shaft.

For a radial piston pump, the pump displacement (V) is defined by the piston bore (d_k) and the cam thrust (e) according to the following equation and Fig. 1.22 [9]:

$$V = \frac{d_k^2 \pi}{4} 2ez \tag{1.29}$$

where e, the cam thrust, is measured in inches.

Typical applications for radial pumps include cylinder jacking, crimping, and holding pressure on hydraulic presses. However, it should be noted that for extremely high-pressure applications, the displacements of radial pumps are usually not larger than 0.5 in.^{3}/rev.; for example, at 1800 rpm, a 0.5-in.^{3}/rev. displacement pump will only deliver ~3.9 gpm. Assuming an efficiency of 93% at a load pressure of 10,000 psi, the pump would require a 24-hp electric motor [Eqs. (1.14) and (1.18)].

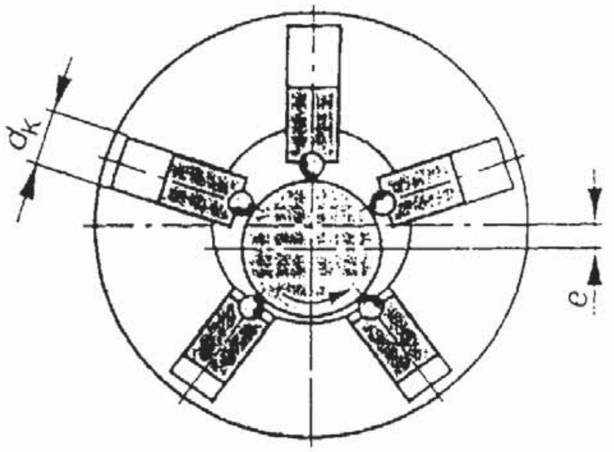


Figure 1.22 Sketch of a radial piston pump showing the parameters used to calculate displacement.

2.2.4 Sizing and Selection of Pumps

As was mentioned earlier, the sizing of the system pump actually begins with the load. The specification for a hydraulic system only deals with the movement of a load; therefore, the size of the pump must be calculated from this information. The pump is a flow generator which develops flow; pressure is a result of the pressure losses within the system and the pressure necessary to maintain the motion of the load. Hence, the first parameter in sizing a pump is to determine the required flow. Then, as was shown earlier, the pressure capability of the pump must be considered. The pressure necessary to deal with the load is determined by sizing the cylinder. Then, it is only necessary to add the system losses to the pressure to arrive at the pressure capability of the pump.

Once the size and pressure capability of the pump are known, the type of pump and the manufacturer must be selected. There are basically three types of pumps normally used. The main criterion that will play heavily in the selection of the pump is the type of control used. In an open-center system, which is not extremely high pressure, any of the three pump types can be used. In this case, price and personal opinions of the designer will prevail. However, if the system is to be a closed-center one with pressure or load compensation, the usual selection is the variable-displacement piston pump.

Open-Loop Circuit

The open-loop circuit is by far the most popular design. An example of an open-loop circuit is shown in Fig. 1.23 [8]. In this figure, an electric motor powers a variable-displacement pump which draws hydraulic fluid from the reservoir and pushes the fluid through a directional control valve. The fluid from the control valve can be directed to either side of a reversible hydraulic motor and then is sent back to the reservoir. Pumps used in open-loop applications can only pump fluid in one direction. In contrast to the reversible hydraulic motor, the pump's ports are not the same size—the inlet port is always larger than the outlet port. The advantage of an open-loop design is that, if necessary, a single pump can be used to operate several different actuator functions simultaneously. The main disadvantage is its large res-

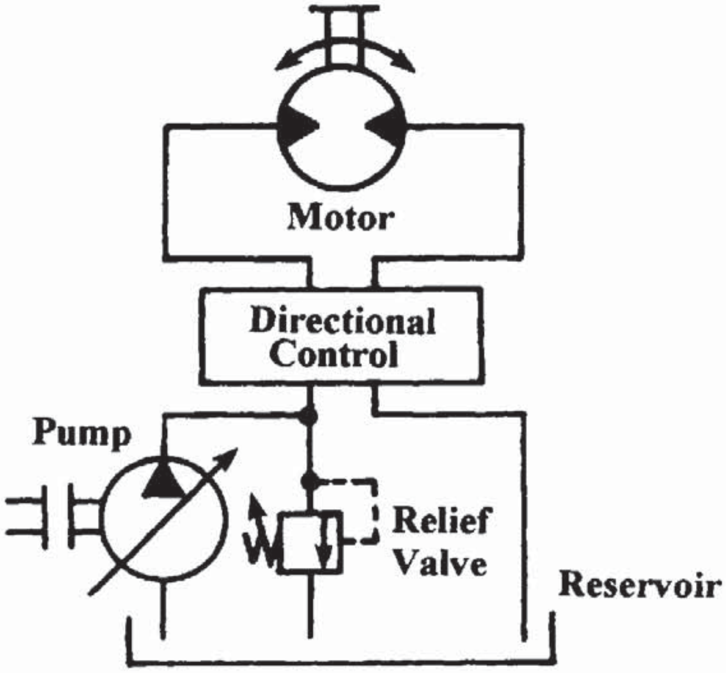


Figure 1.23 Illustration of an open-loop circuit.

ervoir size. Generally, the reservoir is sized to hold at least three times the volume of fluid that can be supplied by the pump in 1 min.

Closed-Loop Circuit

In contrast to the open-loop design, the closed circuit eliminates the need for a large storage reservoir. Figure 1.24 [8] shows an illustration of a closed-loop circuit. In this design, a reversible pump is used to drive a reversible hydraulic motor. The closed-loop design is always used in conjunction with a smaller "supercharge" circuit. The supercharge circuit consists of a small fixed-displacement pump (usually about 15% of the displacement of the main pump), a small fluid reservoir, filters, and a heat exchanger.

The supercharge circuit always works on the low-pressure side of the main loop. Its function is to pump freshly filtered fluid into the closed loop through check valves while bleeding-off a percentage of the hot fluid through a bleed valve. This hot fluid is then cooled by a heat exchanger and stored in a small reservoir before returning to the main system. The pressure in the supercharge circuit is limited to 100–300 psi by the supercharge relief valve. The pressure setting of this valve is determined by the requirements of the pump/motor combination and the operating conditions of the system. The cross-port relief valves on the motor are there only to protect the actuator from load-induced pressure spikes. They are not intended to function like those found in open-loop designs, which would cause severe overheating of the circuit due to insufficient fluid supply, inherent in a closed loop system, to carry away this extra heat. In closed-loop circuits, pressure, flow, and directional motor control are all achieved by controlling the variable-displacement pump.

The advantages of a closed-loop circuit is that high-horsepower systems are compact and efficient and require less hydraulic fluid storage. The high efficiency of this circuit is the result of the pump control being designed to supply only the fluid flow required by the actuator to operate at the load-induced pressure. The pump is

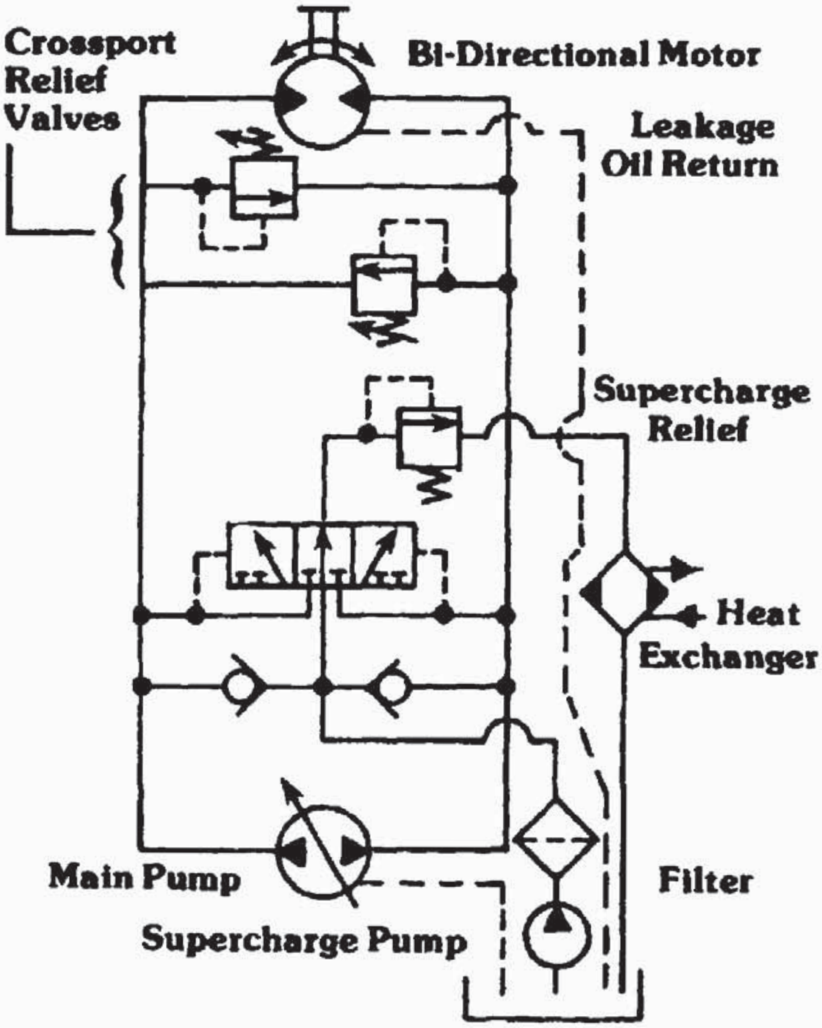


Figure 1.24 Illustration of a closed-loop circuit.

the heart of the system and controls the direction, acceleration, speed, and torque of the hydraulic motor, thus eliminating the need for pressure and flow control components.

A major disadvantage of a closed-loop circuit is that a single pump can only operate a single output function or actuator. In addition, this type of hydraulic circuit is generally used only with motor actuators.

Half-Closed-Loop Circuit

Figure 1.25 [8] is an illustration of a "half-closed"-loop circuit. This circuit is similar to the closed circuit except that it can be used with cylinder actuators having differential areas. As can be seen from the figure, during cylinder extension, the pump must generate a larger flow from its left-hand port than is being returned to its right-hand port from the cylinder. The extra fluid needed by the pump is supplied by its left-hand inlet check valve, which is an integral part of the pump.

When the pump control strokes the pump over the center, the flow from the pump is reversed and the cylinder begins to retract. During retraction, the differential area of the cylinder piston causes a larger flow than needed at the inlet of the pump. This excess flow is directed to the reservoir through the "unloading valve." The

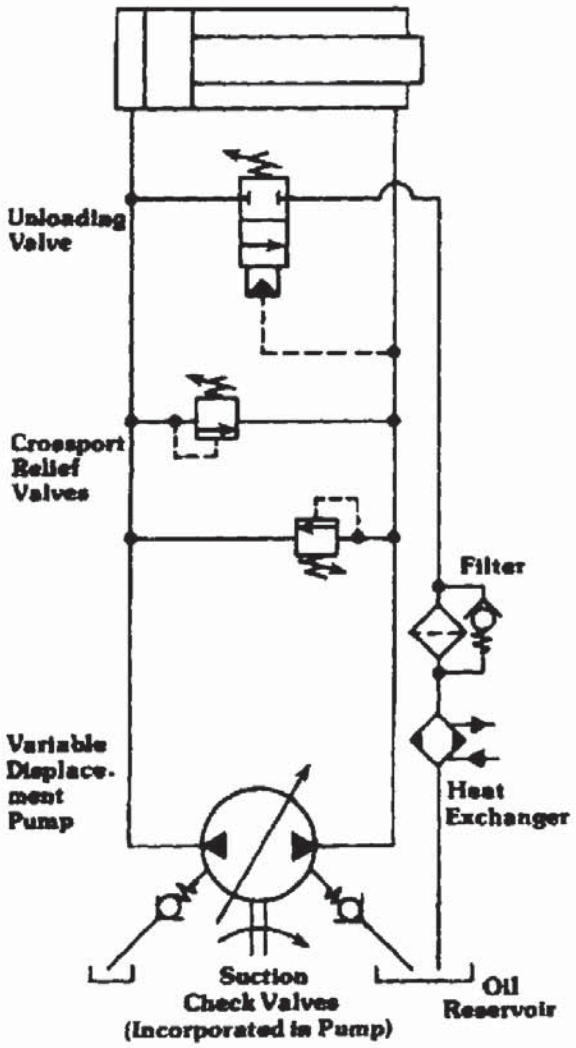


Figure 1.25 Illustration of a half-closed-loop circuit.

unloaded fluid is filtered and cooled prior to its return to the reservoir. In this way, a portion of the closed-loop fluid is filtered and cooled in an open-loop circuit each time the cylinder is cycled.

2.2.5 Contamination Considerations

Every hydraulic system will have particulate contamination entrained in the circulating fluid. These contaminant particles can enter the clearance space of every component, but more specifically, the hydraulic pump [10]. Contaminants will enter the system from reservoir breathers, seals, and so forth as well as from the wear of internal surfaces.

There are three ways of addressing the contamination problems in a hydraulic system. This can be accomplished by using return-line and reservoir breather filters. One way is to prevent particles from entering the system. Because the particles that enter the hydraulic system will cause wear on the internal surfaces of the components, the elimination of these particles will also reduce the production of wear debris. The second approach is to filter the particles that become entrained in the circulating fluid, and the third approach is to select components that effectively resist the contaminant attack.

In selecting the pump, none of these approaches are completely adequate. Due to the development of contaminant-sensitivity test procedures at the Fluid Power Research Center, formerly located at Oklahoma State University, it is now possible to evaluate the efficiency of seals [11,12] and breathers in preventing the entrance of particulate contamination as well as the removal efficiency of hydraulic filters [13,14]. In addition, contaminant sensitivity test procedures are available to evaluate the resistance or tolerance of a hydraulic component, such as a pump, to entrained particulate contaminants [15–17]. Therefore, knowing the ability of the seals and breathers to prevent the entrance of contamination and the effectiveness of the filter in removing that contaminant which does enter the system, a reasonable selection of the pump to produce the desired service life can be made.

Table 1.4 shows the level of cleanliness that can be achieved as a function of degree of filtration as it applies to various hydraulic components. This subject is covered in much greater detail in Chapter 3.

Preparation of Pipes and Fittings

When installing pipes and fittings on a hydraulic system, it is imperative that they be as clean as possible. The following steps are recommended to prepare metal pipes and fittings prior to installation [18]:

1. Ream inside and outside edges of pipe or tubing and clean with a wire brush to loosen and remove any particles.
2. Sandblast short pieces of pipe and tubing to remove any rust and scale. In the case of longer pieces or short pieces having complex shapes, they first should be cleaned of all grease and oil in a degreasing solvent and then pickled in a suitable solution until all rust and scale is removed.
3. After pickling, rinse all parts thoroughly in cold running water and then immerse parts in a tank containing neutralizing solution at the proper temperature and length of time as recommended by the manufacturer.
4. Rinse parts in hot water and place into another tank containing an antirust solution. If parts are not to be immediately installed, they should be left to air-dry with antirust solution remaining on them. If pipes are dry and will be stored, they should be capped to prevent dirt from entering. Before using any pickled part, it should be thoroughly flushed with a suitable degreasing solvent.
5. Cover all openings into the hydraulic system to prevent dirt and any foreign matter from entering the system.
6. Inspect all threaded fittings and remove any burrs or metal slivers.

Table 1.4 Filtration and Cleanliness Guidelines for Various Hydraulic Components

Filtration (μm)	Cleanliness (class)	Hydraulic Application
1–5	0–1	Servo valves
10	2–4	Piston pumps and motors, flow controls, relief valves
20–25	4–5	Gear and vane pumps
40	6	Infrequent duty cycle and noncritical components

7. Before filling or adding hydraulic fluid to the reservoir, make sure that the fluid is as specified and that it is clean.
8. When adding hydraulic fluid to the reservoir, use a fluid filtration cart to prefilter the fluid as it enters the reservoir. Never add fluid directly from the storage container or drum without filtering.

2.2.6 Pump Performance Characteristics

The performance of the pump is extremely important in the overall success of any hydraulic system. A pump exhibits mechanical-type losses as well as volumetric losses [19]. Mechanical losses are the result of the motion of the working element within the pump because of friction. As shown previously [Eq. (1.10)], the theoretical torque required to drive the pump is equal to the product of the pump displacement and the pressure differential across the pump. Obviously, the actual torque is greater than this theoretical value in order to make up for the mechanical losses within the pump. As stated earlier, the mechanical efficiency is the ratio of the theoretical torque to the actual torque [Eq. (1.16)].

The effectiveness of a pump in converting the mechanical input energy into output hydraulic energy must be measured by tests. Theoretically, at low pressures, the output flow of a positive displacement pump is equal to the product of the displacement and the shaft rotational speed as shown by

$$\text{Flow (gpm)} = \frac{\text{Displacement (in.}^3\text{/rev.)} \times \text{Speed (rpm)}}{231} \tag{1.30}$$

As the differential pressure across the pump increases, the flow through the clearances or leakage paths within the pump will increase to create slip flow, which will be subtracted from the output flow. Therefore, the volumetric efficiency is the ratio of the actual flow to the theoretical flow; this parameter reflects the magnitude of the volumetric losses [Eq. (1.15)].

Overall efficiency is equal to the volumetric efficiency multiplied by the mechanical efficiency [Eq. (1.17)]. Figure 1.26 illustrates the relationship among volumetric, mechanical, and overall efficiency of a hydraulic pump. To properly select a hydraulic pump for a given application, efficiency information is extremely important. These data must be acquired by the pump manufacturer by testing and are normally reported in a graph resembling that shown in Fig. 1.27. The upper curves in this figure show that at a given load pressure, the volumetric efficiency increases with speed. This is because all fluids have a property known as viscosity, and at greater and greater speeds, there is insufficient time available for the fluid to leak across to case or slip past clearances in the pump. Also, the lower curves indicate that at a given load pressure, the input power requirement increases as speed increases. This is the result of the increased flow output of the pump as rotational speed increases [Eqs. (1.12) and (1.13)]. It should be noted that these data are usually run at one temperature using one fluid. No viscosity or density effects are taken into consideration in these data. The cost penalty paid for poor efficiency can be significant. Therefore, care must be taken to evaluate the efficiency characteristics of the pump during the selection process.

The mechanical strength of the pump to survive the pressure duty cycle expected in the application is reflected by the proof- and burst-pressure information.

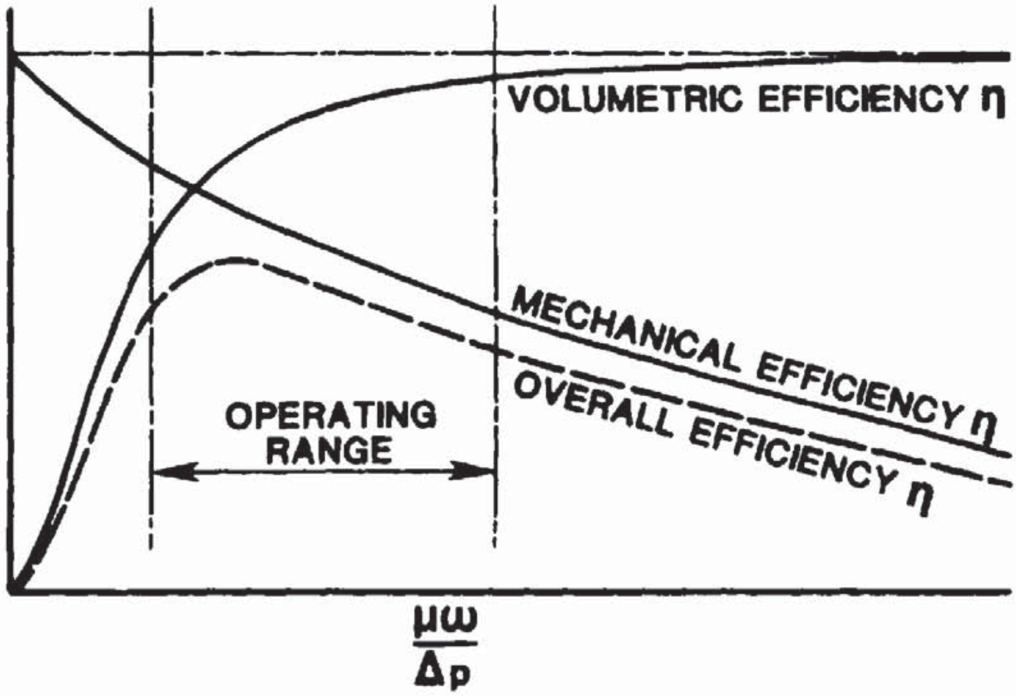


Figure 1.26 Typical hydraulic pump efficiency curves.

This information is obtained through failure tests conducted by the pump manufacturer. The proof pressure is normally 1.5 the rated pressure and the burst pressure is 2.5 times the rated pressure. The ability of the pump to operate for a sufficient time at the actual pressure duty cycle is determined by conducting endurance tests at conditions agreed upon by the industry.

2.2.7 Pump Inlet Condition

In theory, a positive displacement pump will produce flow in direct proportion to the shaft speed [Eq. (1.12)]. However, if the fluid cannot be supplied to the pumping chambers of the pump, this relationship will not hold and the pump is said to cavitate. The flow versus shaft speed for a typical hydraulic pump will be linear up to the point that fluid can no longer enter the pumping chambers of the pump as these chambers are opened and closed because of shaft rotation. When this occurs, the chambers will only partially fill and the outlet flow will reduce. Under these conditions, the pump will be starved for fluid. The speed that this starvation will occur depends on the viscosity and the density of the hydraulic fluid as well as the physical configuration of the pump inlet and the connecting lines. This phenomenon is illustrated in Fig. 1.28.

In considering the starvation of a positive displacement pump, there is normally very little that can be done with the configuration of the pump inlet by the system designer. Also, the fluid being used in the system is generally selected for reasons other than the pump inlet conditions, such as for high-temperature operation, fire resistance, or biodegradability. Therefore, it is necessary to size the inlet piping and position the pump relative to the reservoir such that the inlet pressure to the pump is positive. The pressure at the inlet to the pump is normally called the Net Positive Suction Head (NPSH) and may be calculated in terms of absolute pressure [20]. The

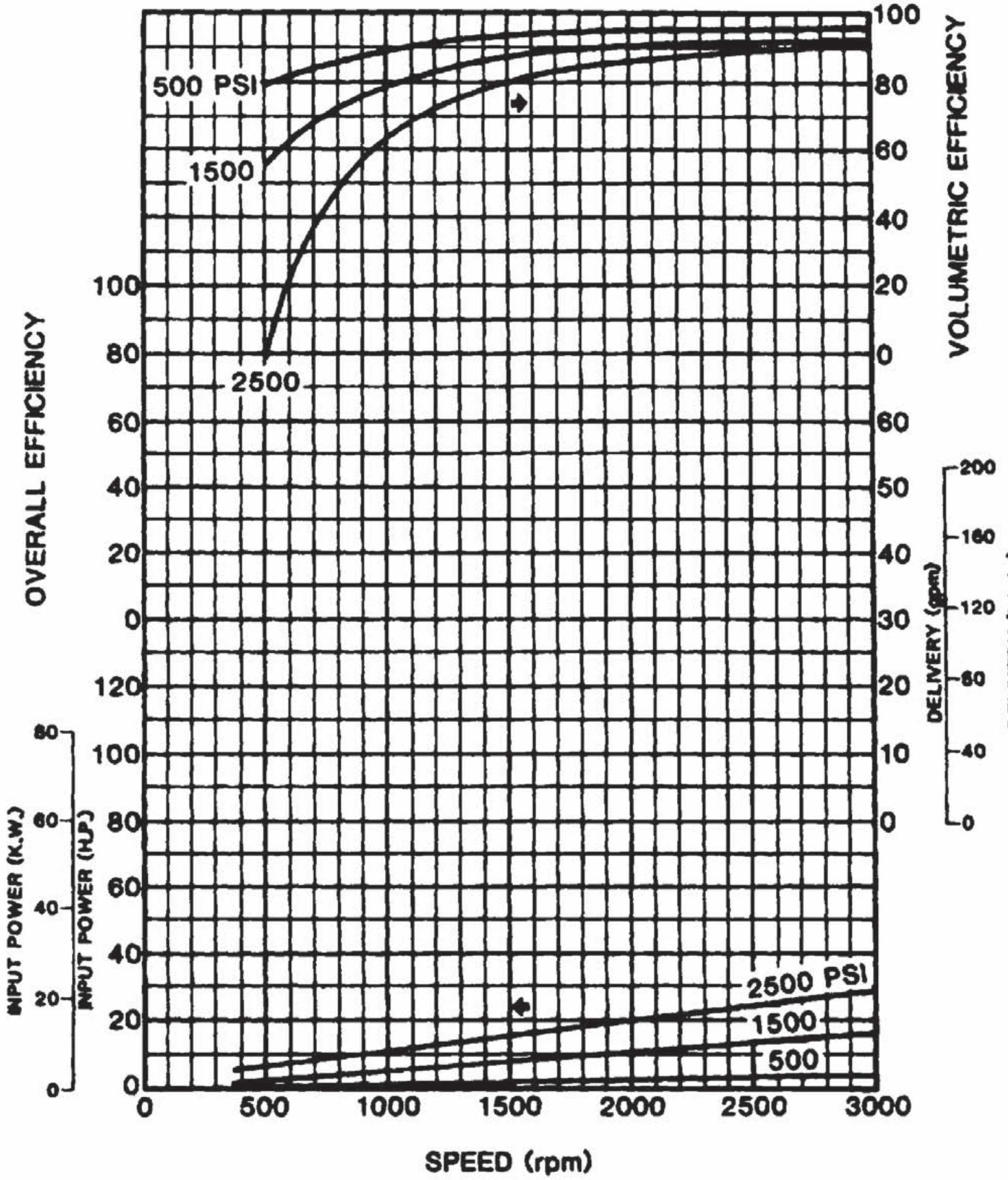


Figure 1.27 Typical performance data reported by a pump manufacturer.

entire system from the fluid level in the reservoir to the inlet port of the pump must be taken into account when determining the NPSH (Fig. 1.29) [20]. The primary factors in determining the NPSH are as follows:

- The atmospheric head or the atmospheric pressure at the particular location (H_A).

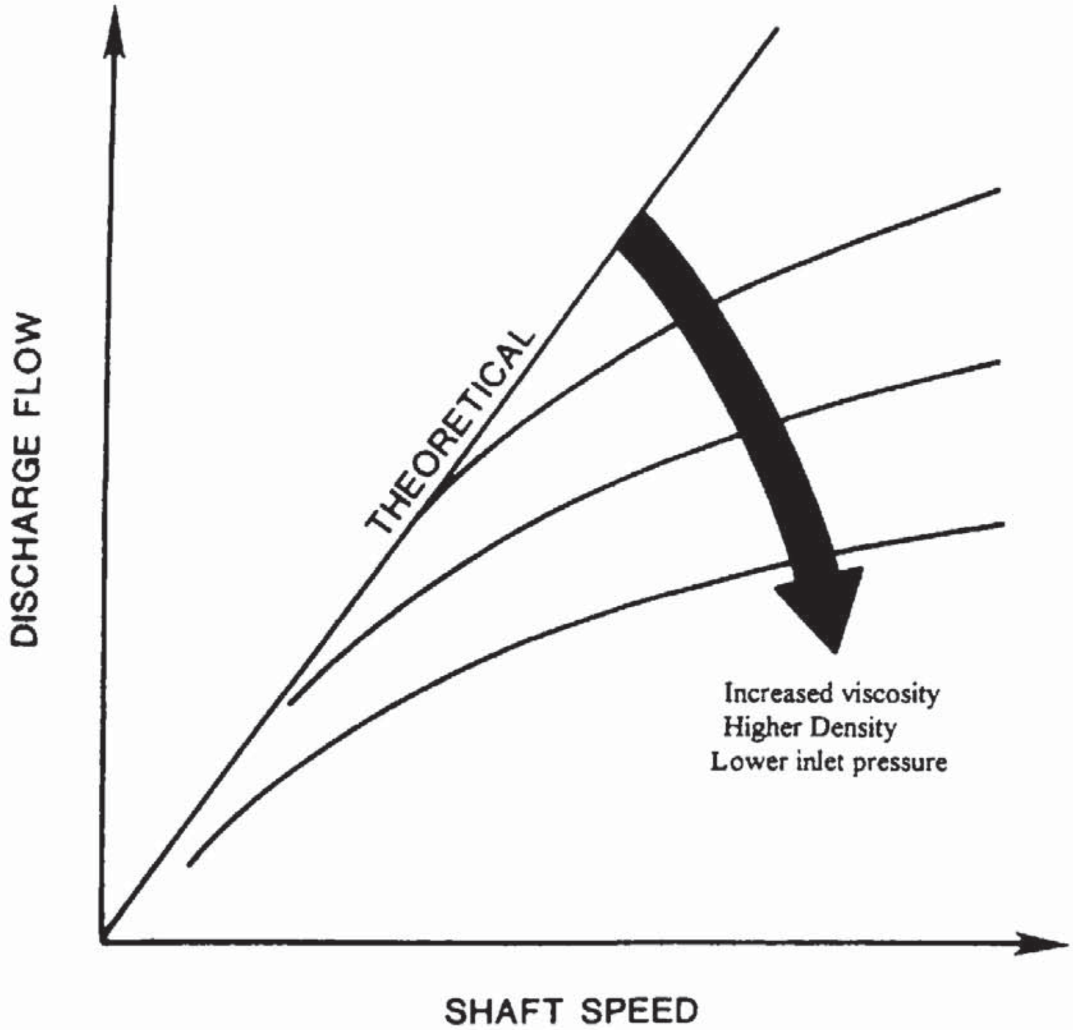


Figure 1.28 Discharge flow of a positive displacement pump as a function of shaft speed, fluid viscosity, and density.

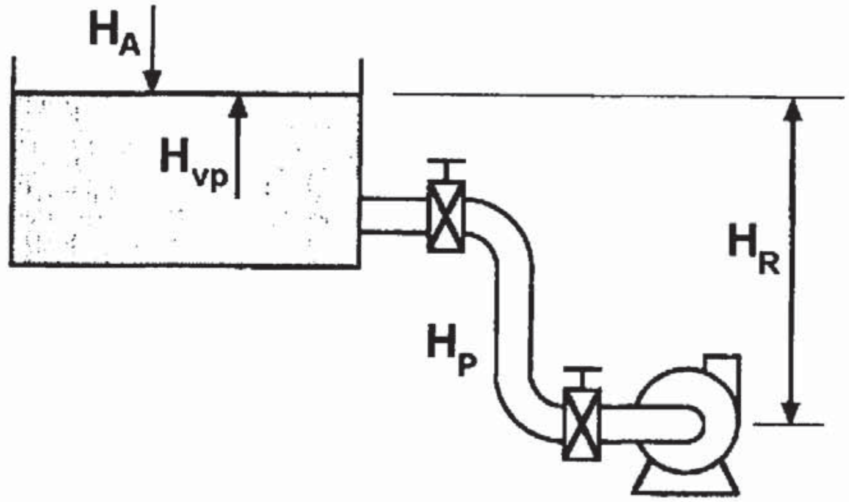


Figure 1.29 Illustration of the NPSH parameters.

- The friction head or the pressure needed to overcome the losses due to friction when the fluid is flowing through the pipe, fittings, valves, and area changes (H_P). Table 1.5 [21] shows data for various hose and pipe diameters.
- Static inlet head or the vertical distance from the centerline of the pump inlet to the free surface level in the reservoir (H_R).
- The vapor pressure of the fluid at the fluid temperature (H_{VP}).

The above parameters can be used to calculate the NPSH according to the equation,

$$\text{NPSH} = H_A + H_R - H_P - H_{VP} \tag{1.31}$$

It is interesting to note that the term H_{VP}, although negligible for mineral-oil-based hydraulic fluids, can be the most dominant term where volatile fluids are concerned, such as with water-based fire-resistant hydraulic fluids. Figure 1.30 shows a plot of water vapor pressure as a function of temperature [22].

There are equations for each of the factors involved in establishing the NPSH. However, numerous graphs and nomographs have reduced the burden of calculating the NPSH for a given pump and inlet condition. One such graph is given in Fig. 1.31. With this graph, one can determine the approximate NPSH at the pump inlet from the flow and speed of the pump. As an example, a pump having a displacement of 0.05 gpm running at 1800 rpm would, according to the nomograph, require a NPSH of ≈24 ft of oil pressure, or 0.35 × 24 = 8.4 psi (for mineral oil), to prevent cavitation. This graph provides information concerning the minimum NPSH and is fairly accurate for viscosities below about 200 SUS and specific gravities of about 0.9. Care should be taken when dealing with a fluid with a high specific gravity, such as many of the synthetic fluids (polyol esters, phosphate esters, etc.) and water-

Table 1.5 Pressure Drop ΔP (psi/ft) and Flow Rates (gpm) for Various Hose (H) and Pipe (P) Diameters at Typical Flow Velocities

Hose/pipe inner diameter (in.)	ΔP (gpm) 2 ft/s	ΔP (gpm) 4 ft/s	ΔP (gpm) 10 ft/s	ΔP (gpm) 15 ft/s	ΔP (gpm) 20 ft/s
0.500 H	0.176 (1.22)	0.352 (2.45)	0.880 (6.12)	1.32 (9.18)	2.92 (12.2)
0.750 H	0.0782 (2.75)	0.156 (5.51)	0.391 (13.8)	1.06 (20.7)	1.76 (27.5)
0.875 H	0.0575 (3.75)	0.115 (7.50)	0.432 (18.7)	0.878 (28.1)	1.45 (37.5)
1.00 H	0.0440 (4.90)	0.0880 (9.79)	0.365 (24.5)	0.743 (36.7)	1.23 (49.0)
2.00 H	0.0110 (19.6)	0.0220 (39.2)	0.154 (97.9)	0.312 (147)	0.517 (196)
3.00 H	0.00489 (44.1)	0.0186 (88.1)	0.0925 (220)	0.188 (330)	0.311 (441)
4.00 H	0.00275 (78.3)	0.0130 (157)	0.0646 (392)	0.131 (588)	0.217 (783)
0.493 P	0.181 (1.19)	0.362 (2.38)	0.905 (5.95)	1.36 (8.92)	2.97 (11.9)
0.742 P	0.0799 (2.70)	0.160 (5.39)	0.400 (13.5)	1.08 (20.2)	1.78 (27.0)
0.884 P	0.0563 (3.83)	0.113 (7.65)	0.426 (19.1)	0.867 (28.7)	1.43 (38.3)
1.049 P	0.0400 (5.39)	0.0800 (10.8)	0.344 (26.9)	0.700 (40.4)	1.16 (53.9)
2.067 P	0.0103 (20.9)	0.0206 (41.8)	0.147 (105)	0.300 (157)	0.496 (209)
3.068 P	0.00468 (46.1)	0.0181 (92.2)	0.0900 (230)	0.183 (346)	0.303 (461)
4.026 P	0.00272 (79.4)	0.0129 (159)	0.0641 (397)	0.130 (595)	0.215 (794)

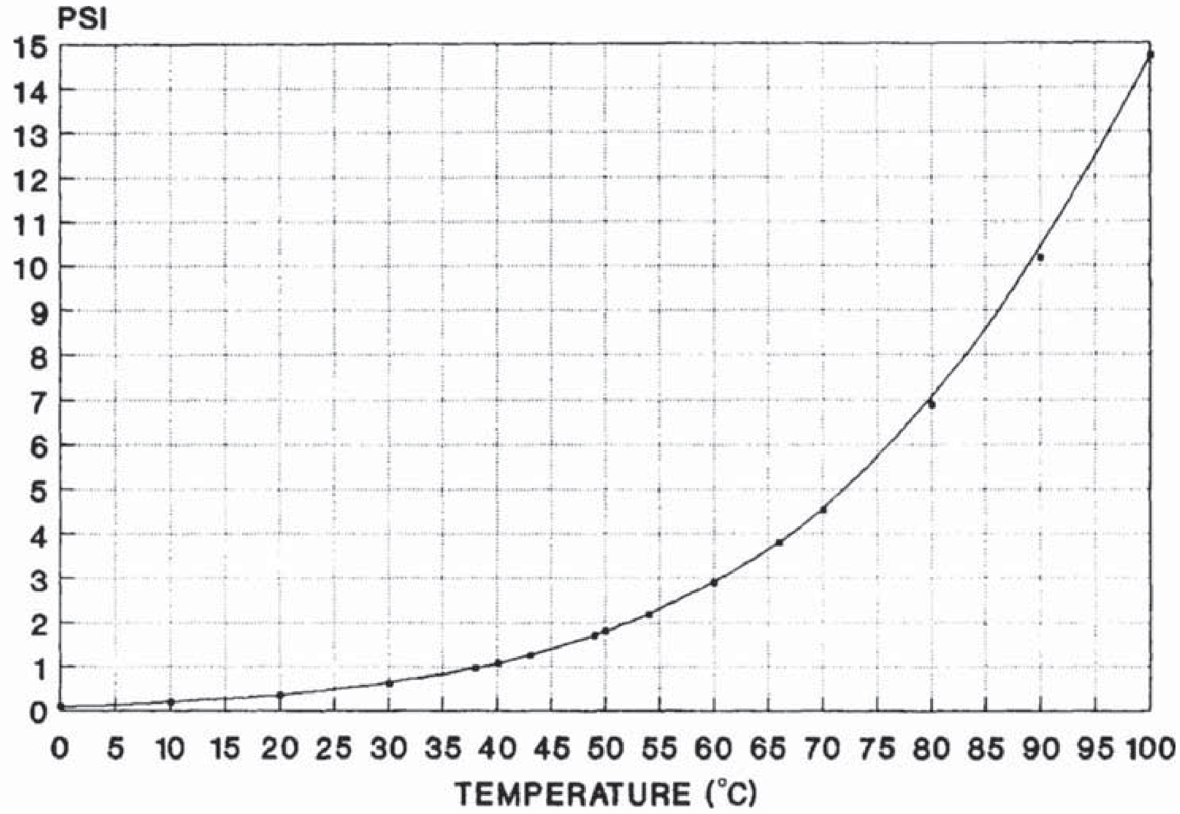


Figure 1.30 Water vapor pressure curve.

based fluids. In addition, complicated or exceptionally long inlet lines should be considered as special cases when determining the NPSH.

Friction head or the losses resulting from the pipe friction acting upon the fluid flowing through the pipe can be calculated using Darcy's equation:

$$\Delta P = \lambda \frac{L\rho Q^2}{2DA^2} \tag{1.32}$$

where ΔP is the pressure loss because of friction, λ is the friction factor, L is the length of pipe, ρ is the density of the fluid, Q is the flow through the pipe, D is the pipe diameter, and A is the cross-sectional area of the pipe.

The friction factor, λ, can be obtained from a modified Moody diagram, as shown in Fig. 1.32 [2]. It should be noted that the solid line on the left-hand side of the graph is for fully developed laminar flow, and the solid line on the right-hand side is for fully developed turbulent flow. The dashed lines show the changes that occur when the laminar flow is not fully developed and at very high Reynolds numbers. Equation (1.32) can be rewritten in terms of head loss as follows:

$$h = \frac{\Delta P}{\rho G} \tag{1.33}$$

where h is the head loss (ft) and G is the gravitational constant (ft/s^2).

The frictional losses can also be found using nomographs. The nomograph shown in Fig. 1.33 [2] can be used to find the pressure loss in a pipe because of friction under conditions of laminar flow. In the example in the figure, the fluid velocity is 2.0 m/s and the fluid viscosity is 30 cP. A straight line is drawn between

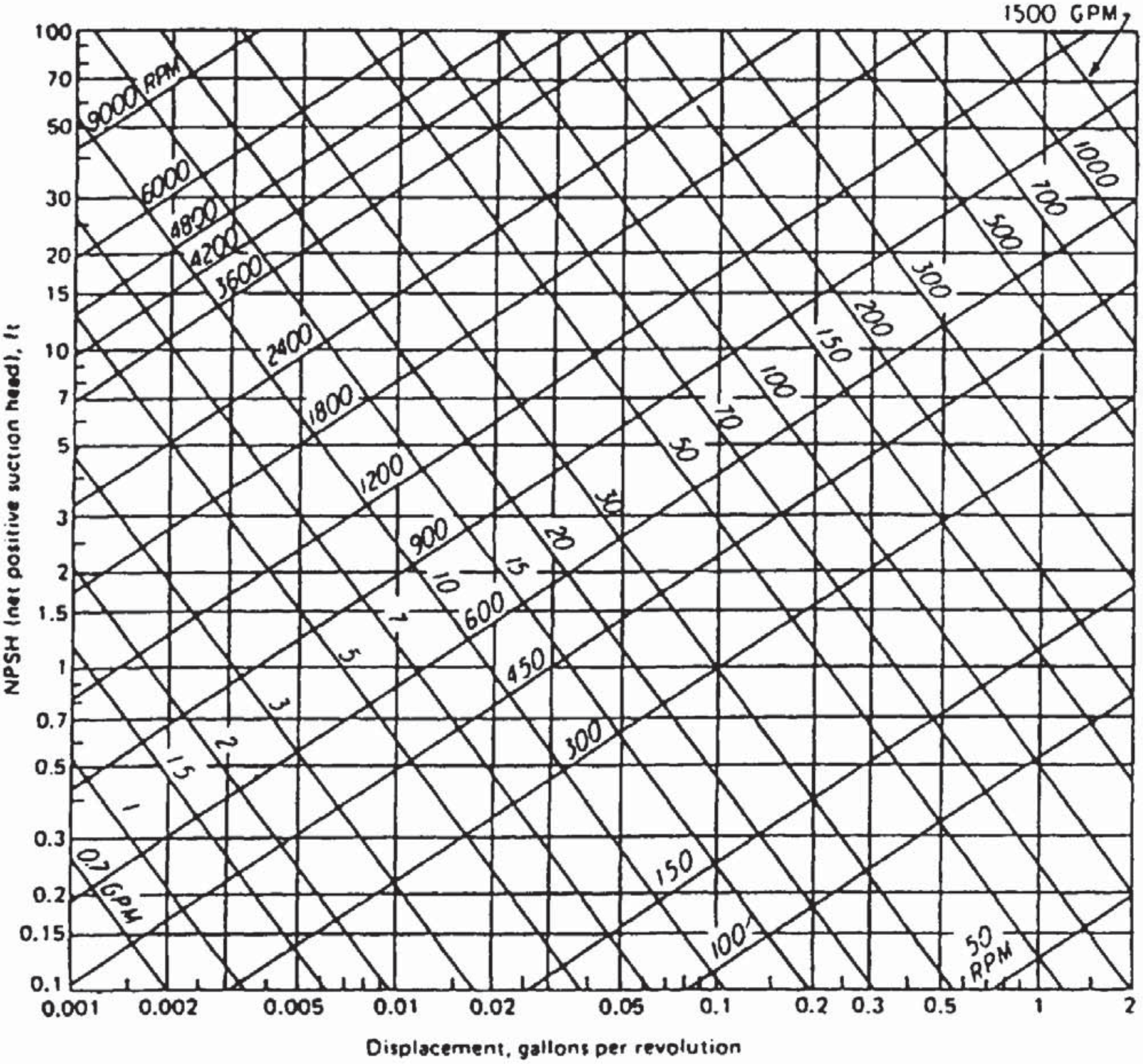


Figure 1.31 Minimum pump head pressure estimation nomograph.

these two points. The pipe diameter is 20 mm. By drawing a straight line between the pipe diameter of 20 mm through the intersection of the first line drawn with the turning line, one will find the pressure loss per foot of this pipe at these flow conditions to be approximately 0.06 bar/m. Then, multiply this number by the total length of the pipe to find the total pressure loss. The nomograph shown in Fig. 1.34 [2], for turbulent flow, is used in exactly the same way as that shown in Fig. 1.33 for laminar flow.

2.3 Hydraulic System Components

There are probably as many different hydraulic systems and component designs as there are designers. However, a fundamental hydraulic system consists of the following components and circuits in addition to the pumping component.

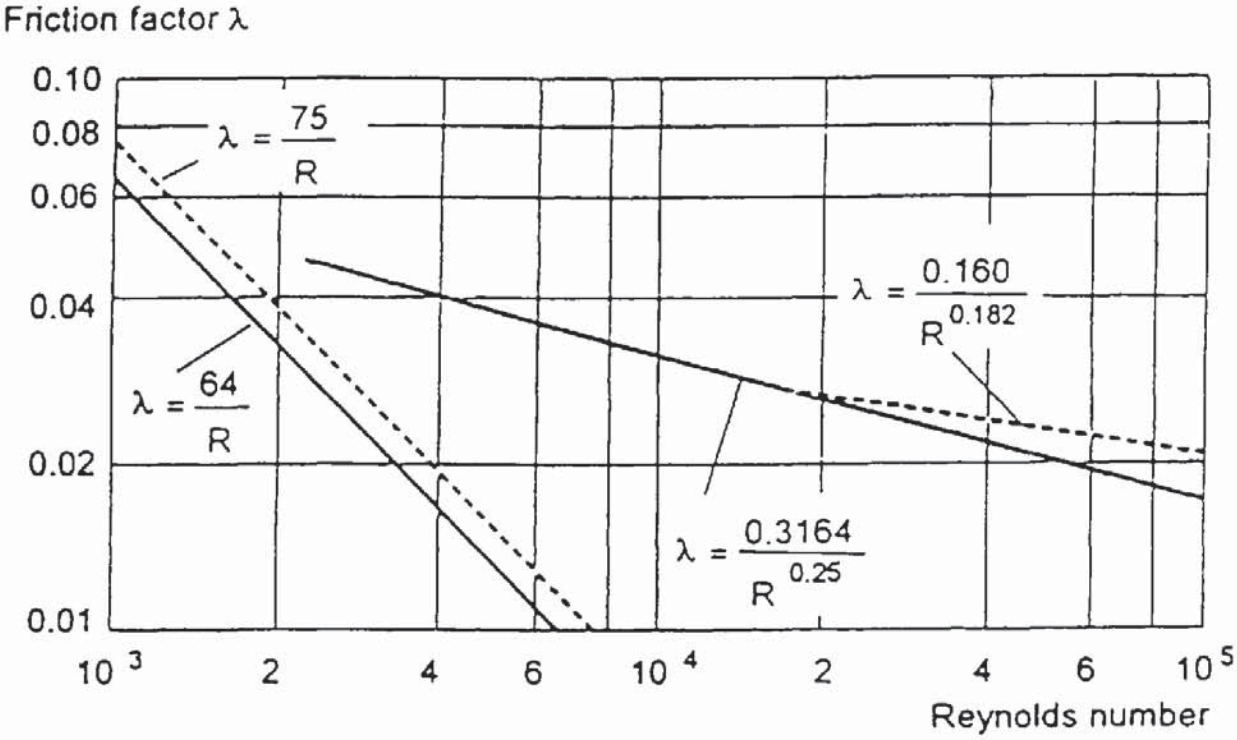


Figure 1.32 Friction factor versus Reynolds number (Moody diagram).

- Flow control
- Pressure control
- Rotary and linear actuators
- Accumulators
- Piping and hose
- Reservoir

Each of these components have a unique mission in the operation of a hydraulic system.

2.3.1 Flow Control

Flow control in a hydraulic system is commonly used to control the rod velocity of linear actuators or the rotary speed of hydraulic motors. There are three ways to accomplish flow control. One is to vary the speed of a fixed-displacement pump; another is to regulate the displacement of a variable-displacement pump. The third way is with the use of flow control valves. Flow control valves may vary from a simple orifice to restrict the flow to a complex pressure-compensated flow control valve and to flow dividers.

Uncompensated Flow Control Needle Valves

The simplest uncompensated flow control is the fixed-area orifice. Normally, these orifices are used in conjunction with a check valve so that the fluid must pass through the orifice in one direction, but in the reverse direction the fluid may pass through the check, thus bypassing the orifice. Another design incorporates a variable-area orifice so that the effective area of the orifice can be increased or decreased (usually

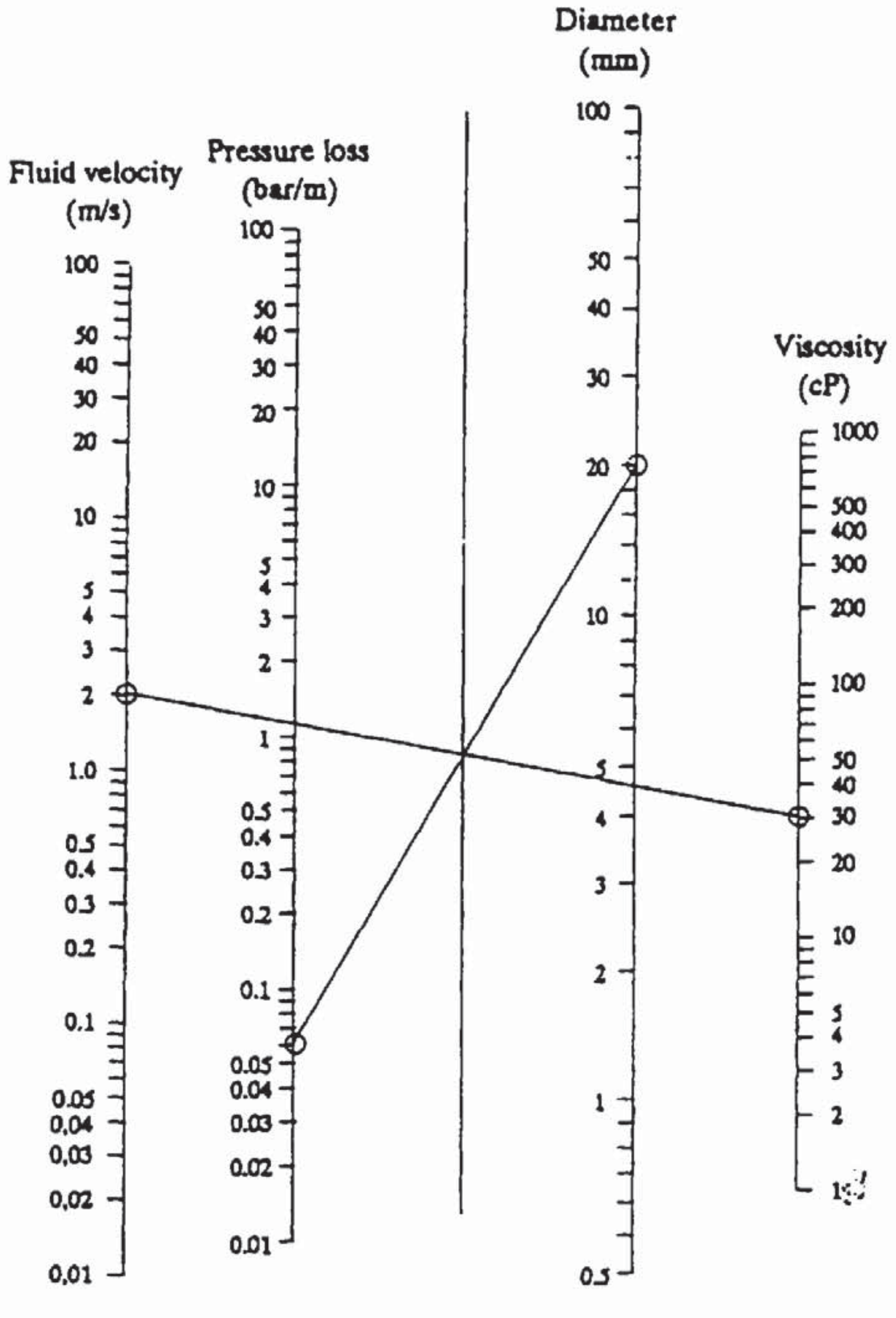


Figure 1.33 Pressure loss per unit pipe length for laminar flow.

manually). One style of the variable-area orifice with a reverse-flow check valve is shown in Fig. 1.35 [2]. These uncompensated flow control valves are used where exact flow control is not critical.

Flow through an orifice is proportional to the pressure drop across the orifice. Therefore, if the pressure differential increases, flow will also increase. To avoid this, a compensated flow control valve must be utilized.

Compensated Flow Control Needle Valves

A very simple compensated flow control valve is shown in Fig. 1.36 [2]. In this valve, the force opposed by the spring is a function of the pressure drop across the fixed orifice, not the pressure drop across the entire valve. As the pressure differential

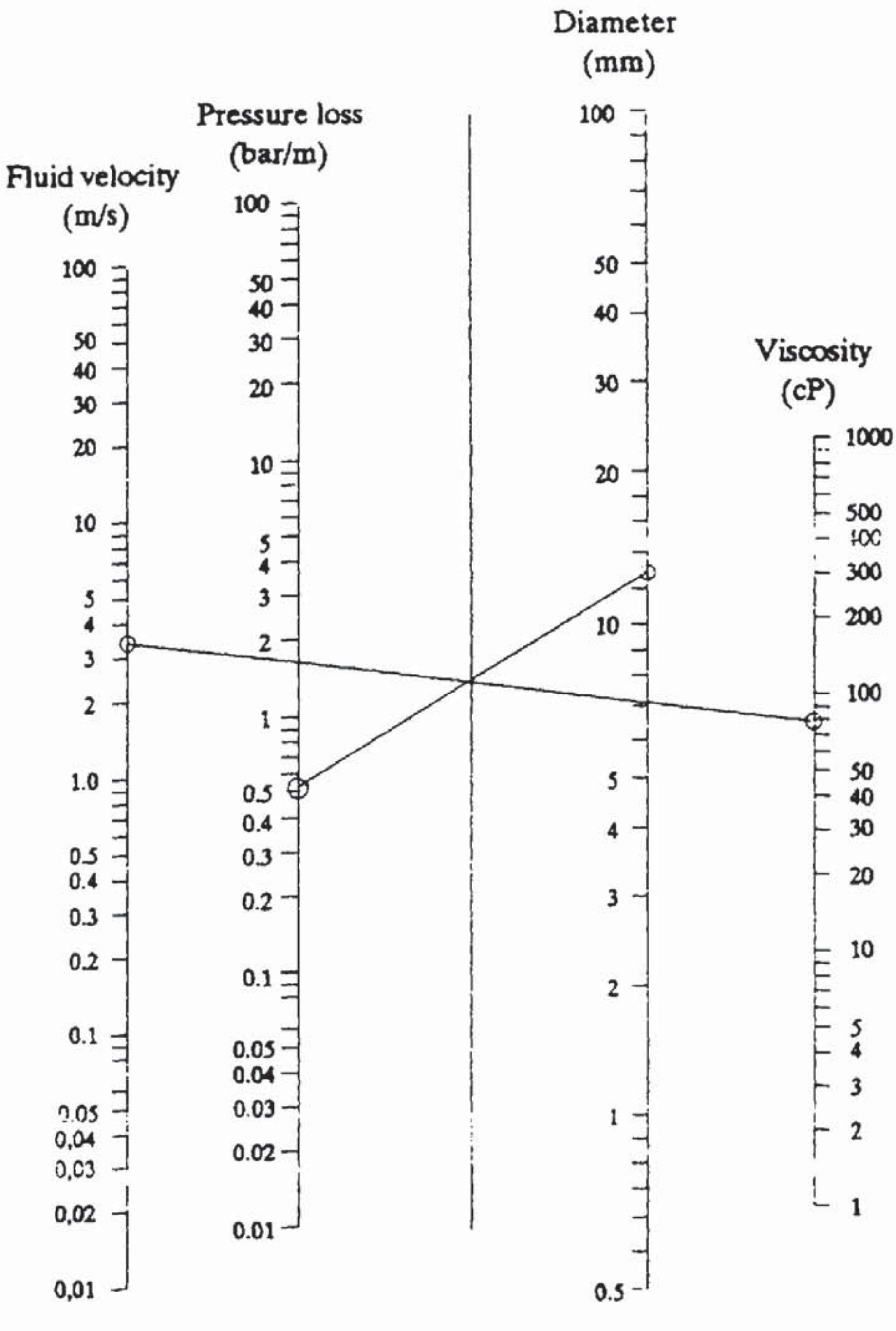


Figure 1.34 Pressure loss per unit pipe length for turbulent flow.

across the valve from the inlet to the outlet increases, flow will also attempt to increase. However, any increase in flow will be accompanied by a resulting increase in the pressure drop across the fixed orifice. When this pressure differential becomes larger than the spring preload, the valve spool will shift and the outlet port will be restricted. There are compensated flow control valves that are much more complex than the one shown; however, most operate the same because the pressure drop across the control orifice is held constant by utilizing a secondary variable orifice.

The following equations can be used to calculate the flow rate through a needle valve, or a series of valves, at a given system pressure. Refer to Fig. 1.37 [21] for the orifice coefficient (CD) values and circuit definitions.

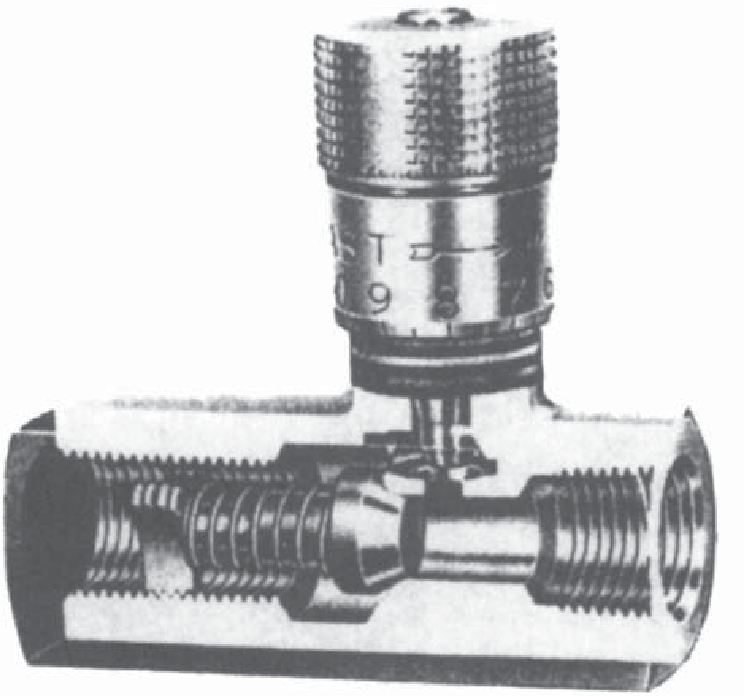


Figure 1.35 Illustration of an uncompensated flow control needle valve.

Parallel circuits:

$$Q = 29.81 \sqrt{\frac{\Delta P}{SG}} \,(CD_1 D_1^2 + CD_2 D_2^2 + \cdots) \tag{1.34}$$

where Q is the flow rate (gpm), ΔP is the pressure drop (psig), SG is the specific gravity of fluid, CD is the orifice coefficient, and D is the orifice diameter (in.)

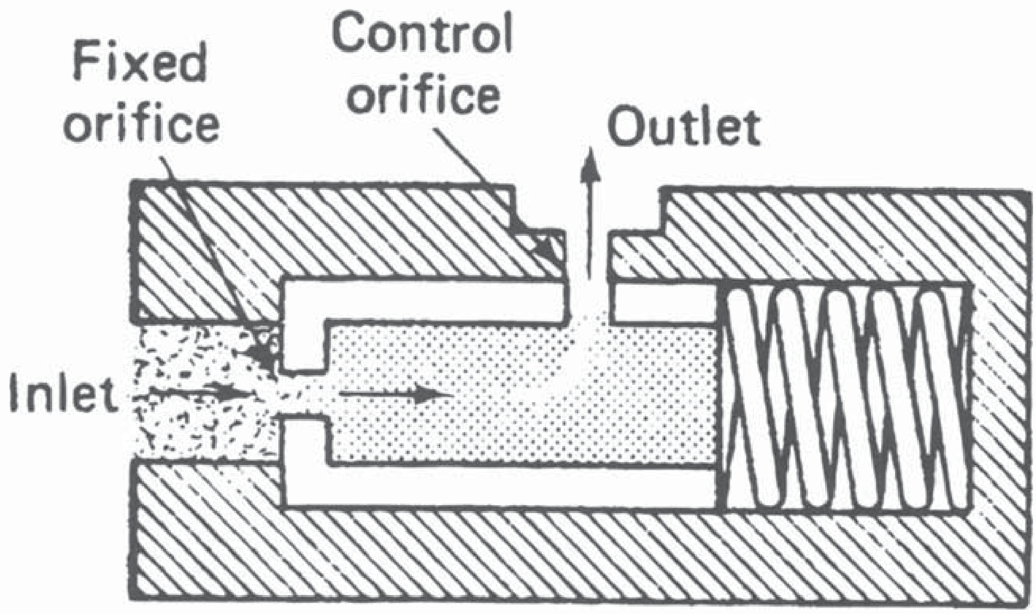


Figure 1.36 Illustration of a compensated flow control needle valve.

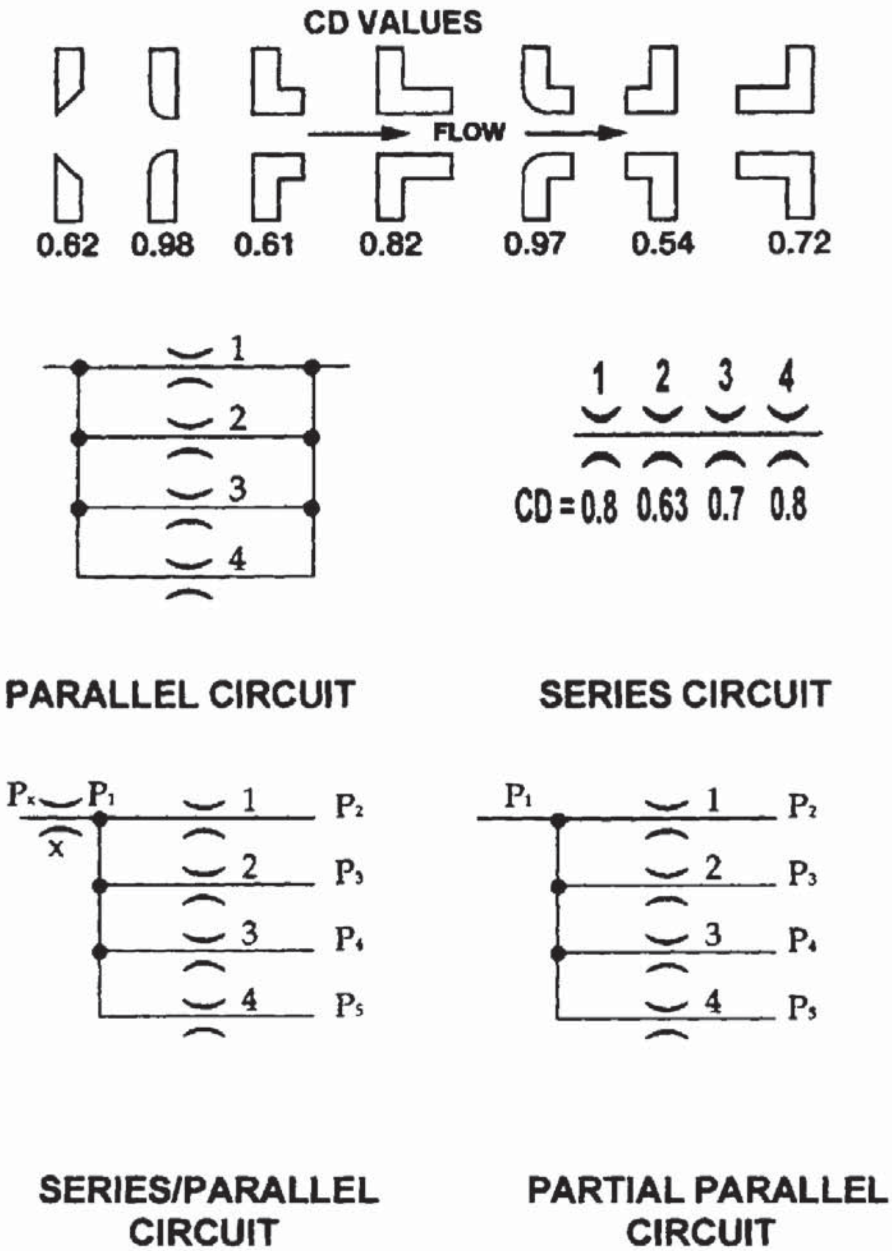


Figure 1.37 CD values and circuit definitions for orifice calculations.

Series circuits:

$$Q = \sqrt{\frac{\Delta P(29.81)^2}{SG[(1/CD_1D_1^2)^2 + (1/CD_2D_2^2)^2 + \cdots]}} \tag{1.35}$$

Series/parallel circuits:

$$Q_x = 29.81CD_xD_x^2\sqrt{\frac{\Delta P}{SG}} \tag{1.36}$$

Partial parallel circuits:

$$Q = \frac{29.81}{\sqrt{SG}}(CD_1D_1^2\sqrt{P_1 - P_2} + CD_2D_2^2\sqrt{P_1 - P_3} + \cdots) \tag{1.37}$$

Flow Dividers

Flow dividers are a form of flow control valves. There are at least two types of flow dividers. One is called a priority flow divider; the other is a proportional flow divider. The priority type of flow control provides flow to a critical circuit at the expense of other circuits in the system. For example, many of the earth-moving machines are equipped with power steering. From a safety standpoint, the steering system is a very critical function.

Figure 1.38 [2] illustrates a priority flow divider. In operation, the flow will enter the priority flow divider from the right-hand end, as shown [2]. When the flow reaches a value such that the pressure drop across the fixed orifice produces a force larger than that provided by the spring, the spool will move to the left. This action will begin to close the priority outlet port and open the secondary outlet. When the flow is below the designed priority flow, the spool will be all the way to the right, the secondary will be closed, and the priority will be wide open. The proportional-type flow divider follows the same principle as the priority flow divider, except that two orifices are used and the spool is normally spring-loaded to a particular flow-split ratio.

2.3.2 Pressure Control

The primary pressure control valves are relief valves. There are direct-acting relief valves and pilot-operated relief valves. In addition, the pressure-reducing and the counterbalance valves fall under the pressure control category. There are a great many more valves that would fall into the pressure control category which will not be discussed here.

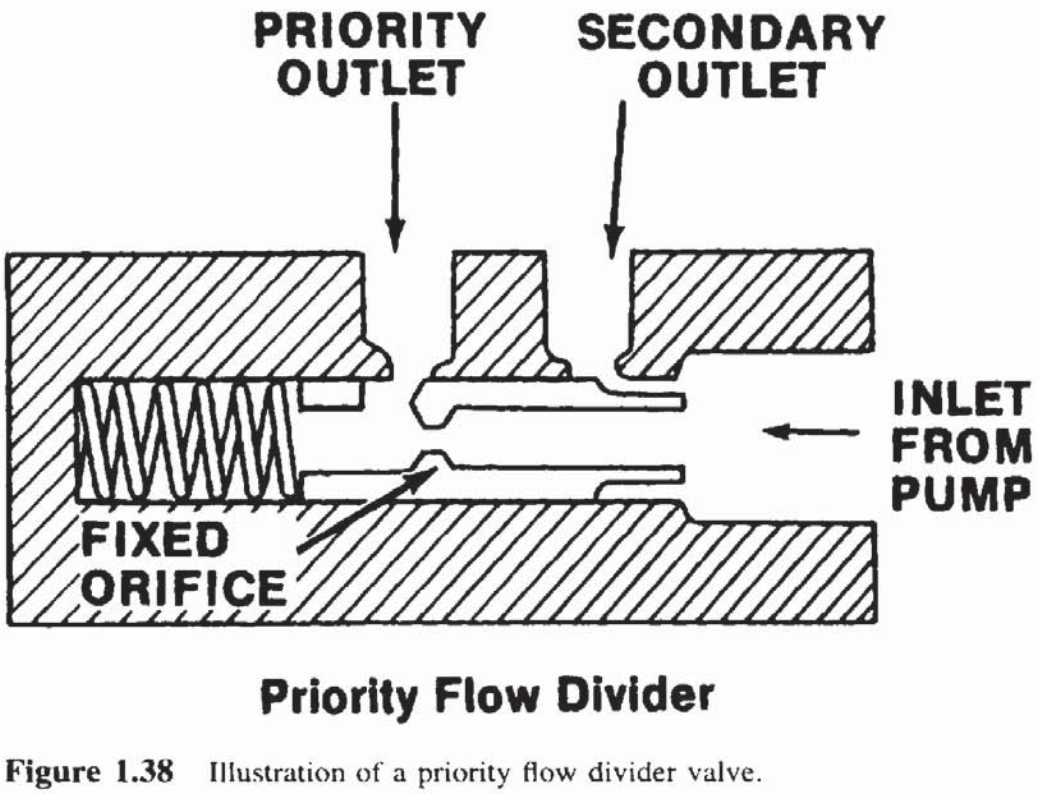


Figure 1.38 Illustration of a priority flow divider valve.

Pressure Relief Valves

There are two major kinds of pressure relief valves. One is described as a direct-acting valve and the other is pilot operated. The direct-acting relief valve is shown in Fig. 1.39 [2]. The model shown is actually adjustable, but not all direct-acting relief valves are externally adjustable. In operation, the flow enters from the bottom of the valve shown in Fig. 1.39. When the inlet pressure reaches the value such that the pressure times the exposed area of the ball is greater than the spring setting, the valve will begin to pass hydraulic fluid. Note that the spring must be compressed in order for the seat (ball) to move and provide greater flow area. Therefore, the pressure will increase as the flow through the valve increases. The pressure at which the valve first begins to open is called the cracking pressure; the pressure at rated flow is termed the full-flow pressure. In the case of the direct-acting relief valve, the difference between the cracking pressure and the full-flow pressure could be large. This difference is called the override pressure.

The pilot-operated relief valve is shown in Fig. 1.40 [2]. The pilot-operated pressure relief valve increases pressure sensitivity and reduces the pressure override normally found in relief valves using only the direct-acting force of the system

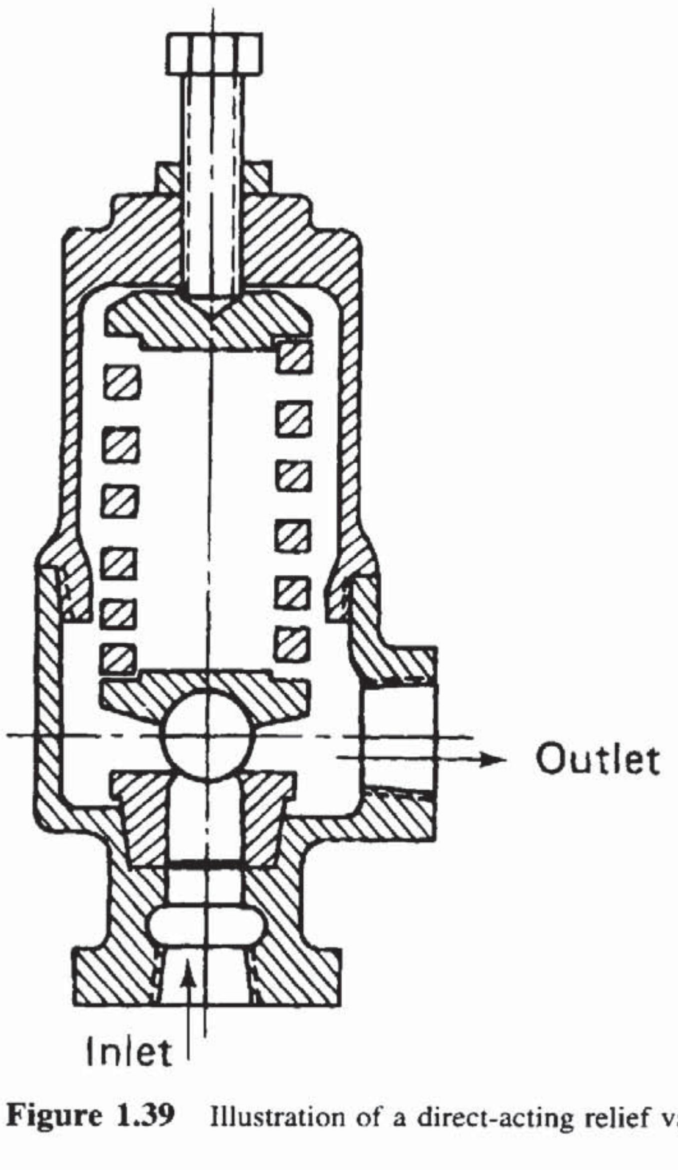


Figure 1.39 Illustration of a direct-acting relief valve.

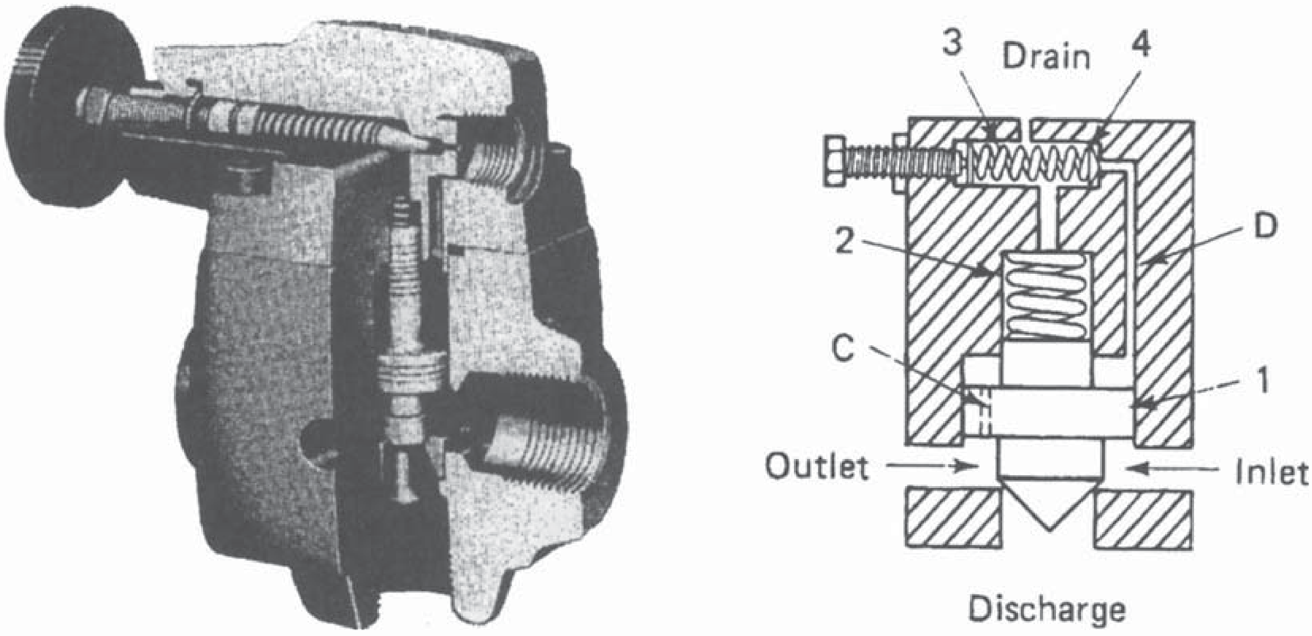


Figure 1.40 Illustration of a pilot-operated relief valve.

pressure against a spring element. In operation, the fluid pressure acts upon both sides of piston (1) because of the small orifice (C) through the piston and the piston is held in the closed position by the light-bias spring (2). When the pressure increases sufficiently to move the pilot poppet (4) from its seat, the fluid behind the piston will be directed to the low-pressure area, such as the return line. The resulting pressure imbalance on the piston will cause it to move in the direction of the lower pressure, compressing the spring and opening the discharge port. This action will effectively prevent any additional increase in pressure. The setting of the pilot-operated relief valve is adjusted by the preload of the poppet spring (3).

Pressure-Reducing Valves

Pressure-reducing valves are used to supply fluid to branch circuits at a pressure lower than that of the main system. Their main purpose is to step the pressure down to the requirements of the branch circuit by restricting the flow when the branch reaches some preset limit. The pressure-reducing valve is illustrated in Fig. 1.41 [2]. In operation, a pressure-reducing valve permits fluid to pass freely from port C to port D until the pressure at port D becomes high enough to overcome the force of the spring (2). At this point, the spool will move, obstructing flow to port D and thus regulating the downstream pressure. The direction of flow is irrelevant with a pressure-reducing valve, as the spool will close when the pressure at port D reaches the set value. If free reverse flow is required, a check valve must be used.

Counterbalance Valves

The normal use of counterbalance valves is to maintain back pressure on a vertically mounted cylinder to hold vertical loads such as encountered in hydraulic presses. A typical circuit using a counterbalance valve is shown in Fig. 1.42 [2]. Counterbalance valves can be operated by either a direct pilot or a remote pilot. As shown in Fig. 1.42, when a direct pilot is utilized, the pressure on the rod side of the cylinder must reach the valve setting before it will open and permit flow. When the valve is op-

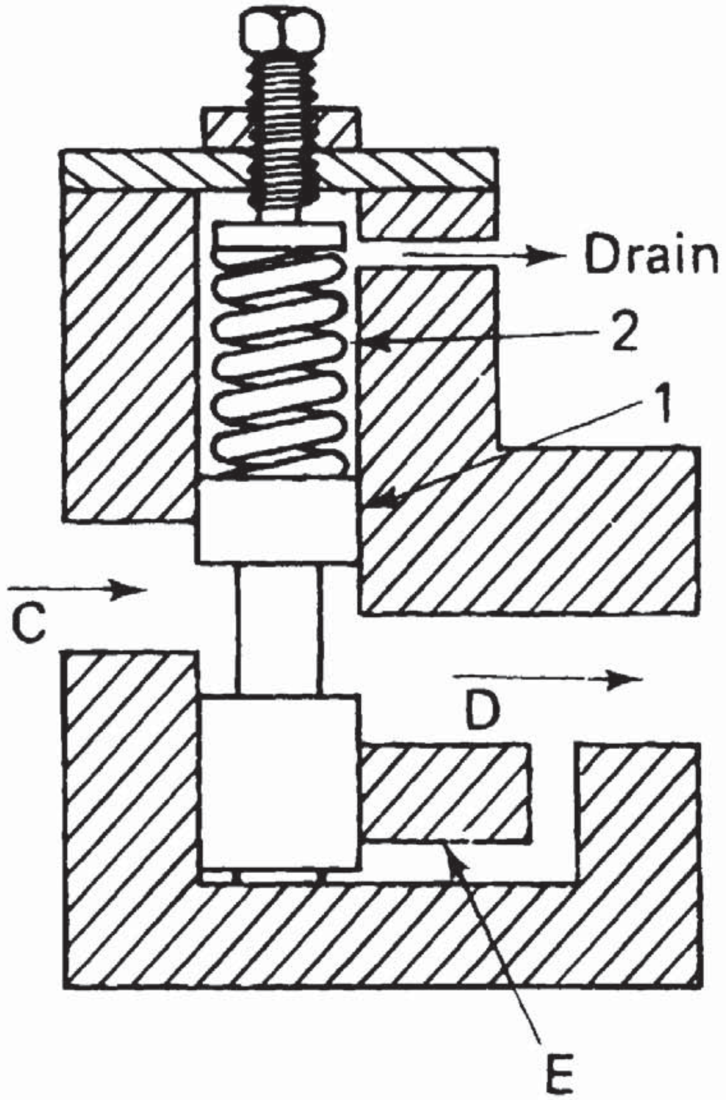


Figure 1.41 Illustration of a pressure-reducing valve.

erated by a remote pilot, this pilot line can be connected to the pump outlet. In this case, the valve will open when the inlet pressure to the cylinder reaches some value. There will be very little rod side pressure in this arrangement. Reverse flow will not pass through the counterbalance valve, as shown in Fig. 1.42. Therefore, a bypass check valve must be included to permit the cylinder to be raised.

2.3.3 Check Valves

Check valves are normally used to control the direction of fluid flow. However, their operation is similar to that of a direct-operated relief valve. Figure 1.43 [23] shows a simple check valve and a cross-sectional illustration of the parts. The valve consists of a seat, a poppet, and a spring. The valve remains closed against flow until the pressure at its inlet creates sufficient force to overcome the spring force. Once the poppet leaves its seat, hydraulic fluid is permitted to flow around and through the poppet to the valve outlet port. For this reason, a simple check valve can only allow flow in one direction. Like direct-operated relief valves, simple check valves have a cracking pressure. By changing the spring, cracking pressures between 5 and 75 psi can be obtained. For special applications, a "no-spring" version is also available.

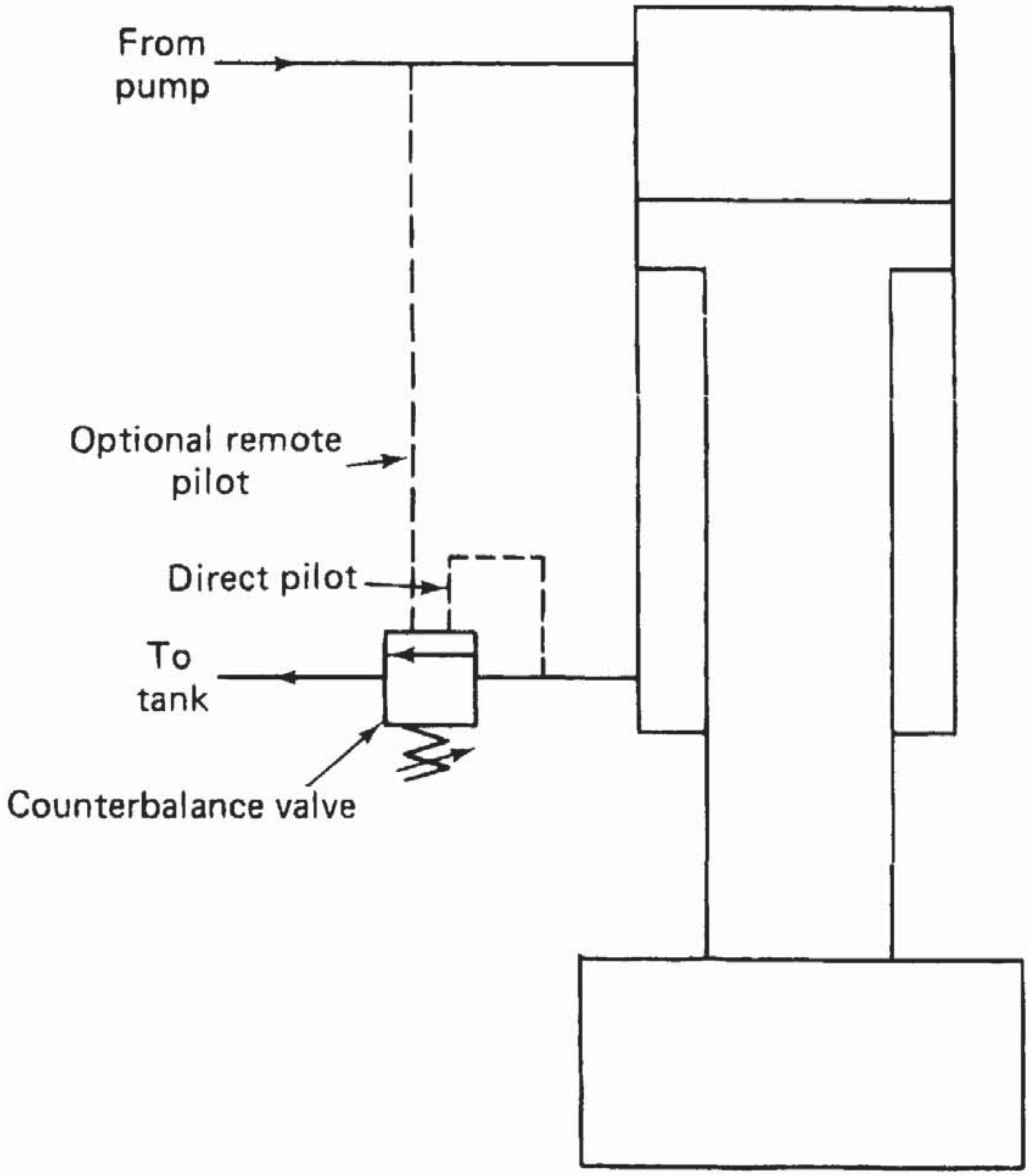


Figure 1.42 A typical counterbalance-valve hydraulic circuit.

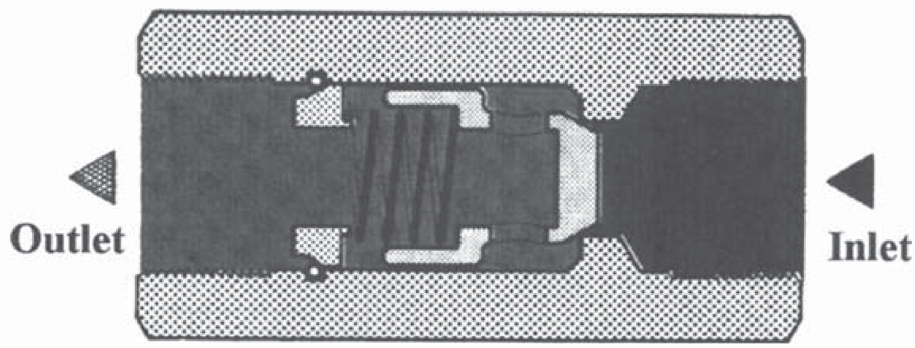


Figure 1.43 Illustration of a simple check valve.

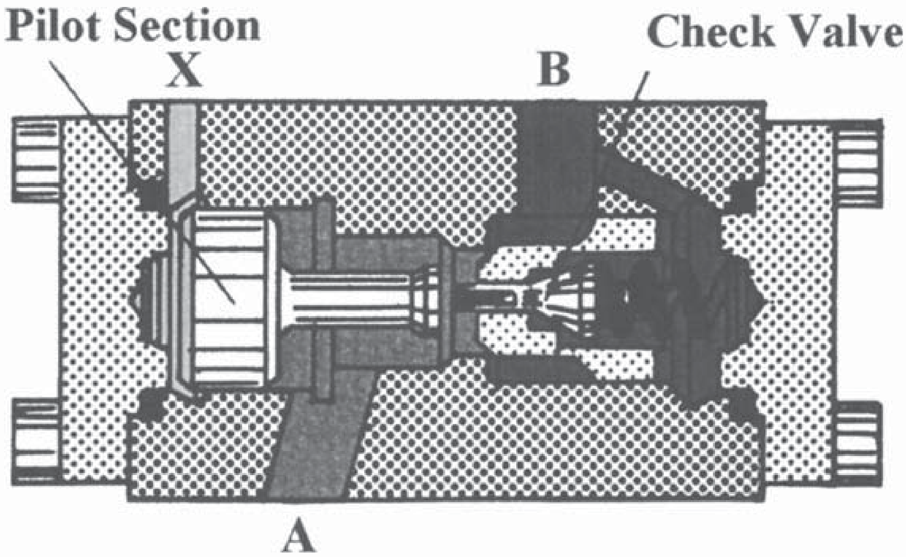


Figure 1.44 Illustration of a pilot-operated check valve.

For load holding and in decompression-type hydraulic press circuits, a pilot-operated check valve is used. It performs the same function as the simple check valve described above. However, in contrast to the simple check valve, a pilot-operated check valve can be piloted "open" when a reverse flow is required. Figure 1.44 [23] illustrates the components of a pilot-operated check valve. The valve has two distinct sections, the check-valve section and the pilot section. The check-valve section allows free fluid flow from port A to port B while preventing reverse flow from B to A without leakage. However, if a pilot pressure signal is supplied to port X, then a force is applied to the pilot piston, which forces the piston rod against the check-valve poppet. This force then unseats the poppet, allowing free flow of fluid from port B to port A.

2.3.4 Directional Control Valves

In typical hydraulic systems, there may be rotary or linear actuators present. These actuators normally have two ports. If oil is pumped into one of the ports while the other is connected to tank, the actuator will move in one direction. In order to reverse its direction of motion, the pump and tank connections must be reversed. The sliding-spool-type directional control valve has been found to be the best way to accomplish this change.

Figure 1.45 [24] shows an illustration of a sliding-spool-type directional control valve. The valve has a cylindrical shaft called a spool, which slides in a machined bore in the valve housing. The housing has ports to which the hydraulic pump, return line to tank, and lines for the actuator are connected. The number of ports designates the valve type. For example, a valve with four ports is referred to as a "4-way" valve; a valve with three port connections would be called a "3-way" valve. Furthermore, spool valves can be classified as "2-position" or "3-position" valves.

A 2-position valve can only be shifted fully left or fully right. A common use for such a valve would be in a cylinder application which only requires the cylinder to extend or retract to its fullest positions. Another application would be in hydraulic motors which only run in forward or reverse directions.

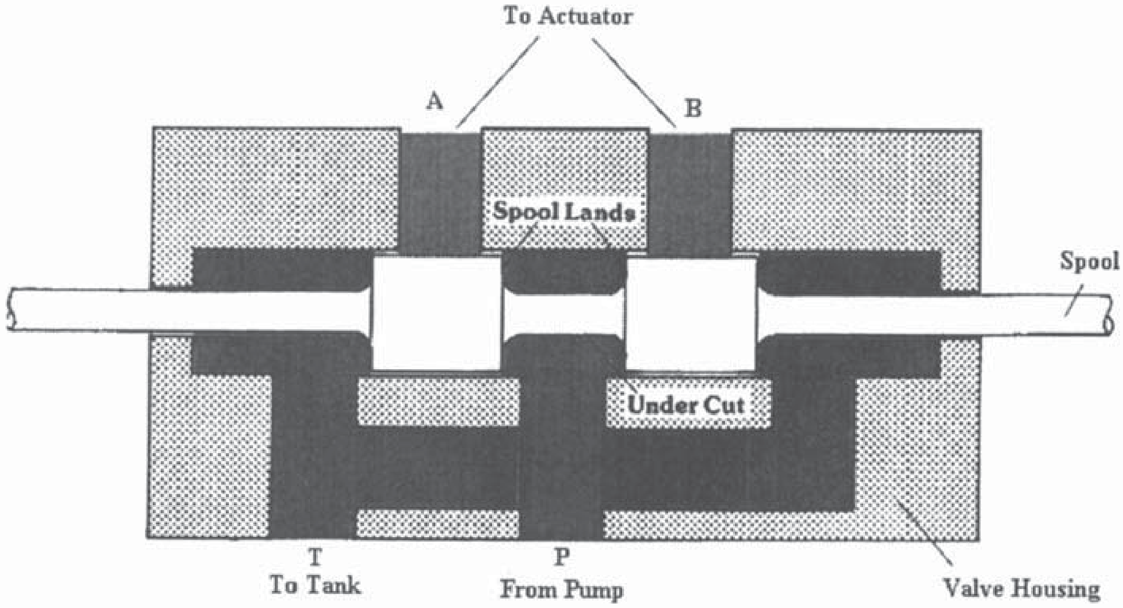


Figure 1.45 Illustration of a spool-type directional control valve.

A "3-position" valve is similar in operation to a "2-position" valve except that it can be stopped in a third or "neutral" position between ports A and B. While in the centered or neutral position, flow may or may not be possible, depending on the spool design of the center position. Figure 1.46 [24] shows some common "3-position" spool designs.

Most valve manufacturers test their valves for flow capacity and develop charts that plot valve flow rate versus pressure drop (ΔP). From these plots, a flow factor

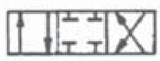
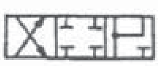
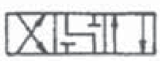
TYPICAL FLOW PATHS AVAILABLE WITH 3 POSITION VALVE SPOOLS			
	Closed Center		Restricted Open Center
	Open Center		Regenerative End Closed Center
	Tandem Center		B Blocked P & A→T
	Float Center		P & B→T A Blocked
	Regenerative Center		P & B Blocked A→T
	Restricted Float Center		P & A Blocked B→T

Figure 1.46 Typical flow paths available for 3-position spool valves.

denoted CV can be determined for each valve. The CV factor then can be used to calculate the flow characteristics of the valve at other conditions; for example, a valve with a CV = 1 will flow 1 gpm at a 1-psig pressure drop using a 1.0 specific gravity (SG) fluid. Figure 1.47 [21] lists the circuit definitions used in the following equations. The CV factor is calculated by

$$\mathrm{CV} = \frac{Q\sqrt{\mathrm{SG}}}{\sqrt{\Delta P}} \tag{1.38}$$

Parallel circuits:

$$Q = (\mathrm{CV}_1 + \mathrm{CV}_2 + \cdots)\sqrt{\frac{\Delta P}{\mathrm{SG}}} \tag{1.39}$$

Series circuits:

$$Q = \sqrt{\frac{\Delta P}{\mathrm{SG}(1/\mathrm{CV}_1^2 + 1/\mathrm{CV}_2^2 + \cdots)}} \tag{1.40}$$

Series/parallel circuits:

$$Q_x = \mathrm{CV}_x \frac{\sqrt{P_x - P_1}}{\sqrt{\mathrm{SG}}} \tag{1.41}$$

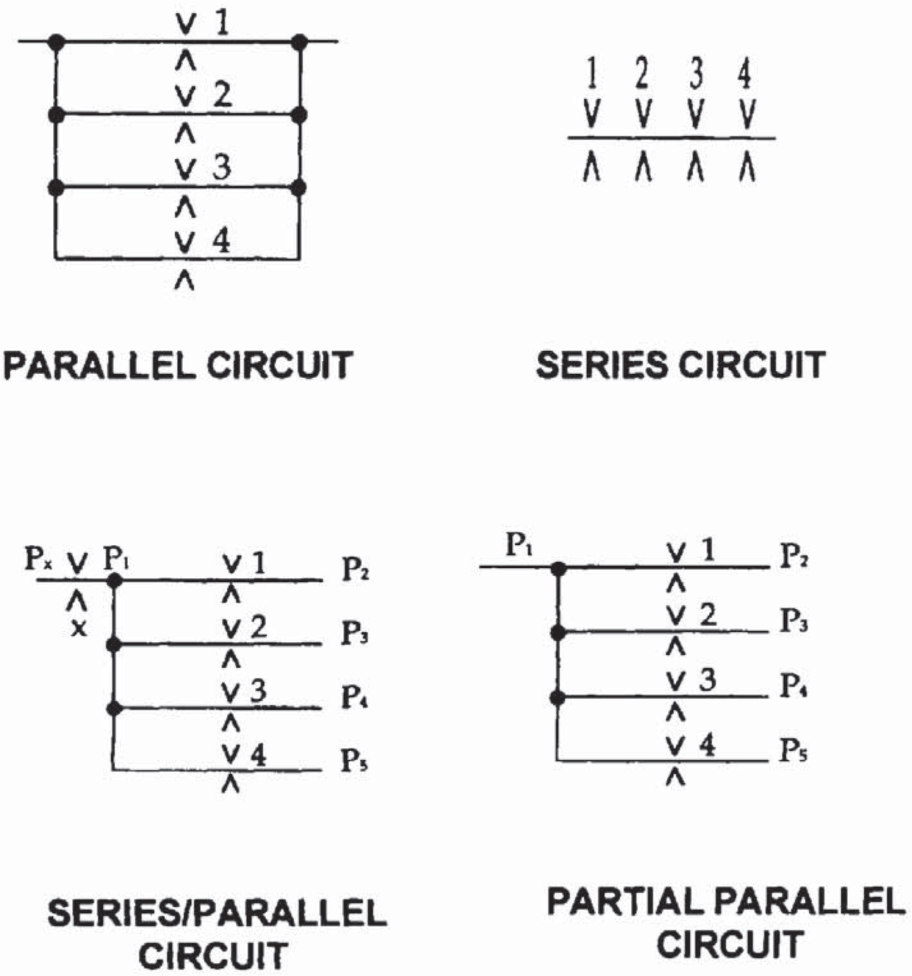


Figure 1.47 Circuit definitions for valve flow CV factor calculations.

Partial parallel circuits:

$$Q = \frac{CV_2\sqrt{P_1 - P_2} + CV_3\sqrt{P_1 - P_3} + \cdots}{\sqrt{SG}} \tag{1.42}$$

2.3.5 Rotary and Linear Actuators

Rotary motors and linear cylinders are used to convert the energy in the hydraulic circuit to either rotary torque and speed or linear force and velocity. Rotary actuators or motors can be gear, vane, or piston design and will operate very similar to a pump except that flow and pressure are inputs, and torque and rotation are outputs. These are normally referred to as continuous-rotation actuators. Another type of rotary actuator is the limited-rotation design and is sometimes called a rotary cylinder. In this design, the output shaft is limited, usually to less than 360° of rotation. By far, the most prevalent actuator found in hydraulic systems is the linear actuator or cylinder.

Cylinders are either single acting or double acting. Hydraulic cylinders are normally constructed of a barrel, piston assembly, piston rod, end caps, ports, and seals, as shown in Fig. 1.48 [7]. The piston provides the effective area against which the fluid pressure is applied and supports the piston end of the rod. The opposite end of the rod is attached to the load. The cylinder bore, end caps, ports, and seals maintain a fluid-tight chamber into which the fluid energy is connected. Whether the rod will extend or retract in a double-acting cylinder depends on which port fluid is directed. In a single-acting cylinder, there is only one port which when adequately pressurized will extend the rod. The single-acting cylinder depends on external forces such as weight and gravity to retract the rod.

Hydraulic cylinders are normally sized to accommodate the load requirements (Table 1.6) [21]; for example, if the load requirements are such that the cylinder must move a load of 20,000 lbs at a speed of 20 ft/min in the extend direction, this

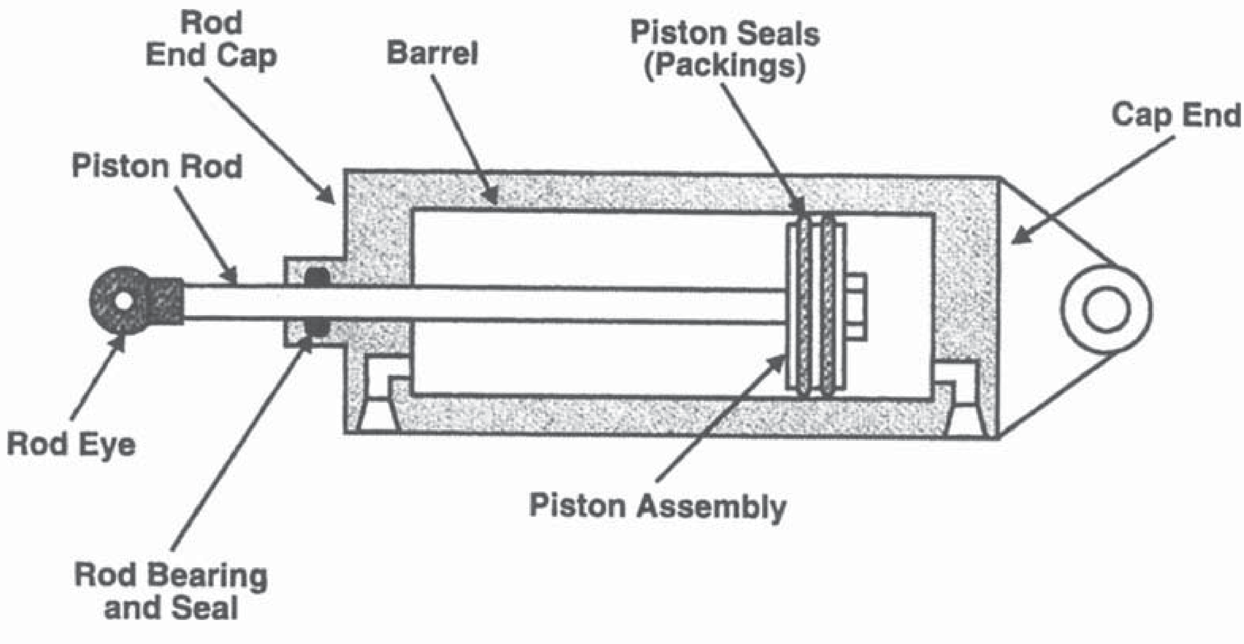


Figure 1.48 Components of a typical double-acting cylinder.

Table 1.6 Cylinder Size, Load, and Pressure Data

Bore diam. (in.) rod (2.5 in.)	Mode	Eff. area (in.^{2})	d (lb) at 1000 (psi)	Load (lb) at 2000 (psi)	Load (lb) at 3000 (psi)	Load (lb) at 5000 (psi)
4	Pull	7.66	7,700	15,000	23,000	38,000
5	Pull	14.7	15,000	29,000	44,000	74,000
6	Pull	23.4	23,000	47,000	70,000	120,000
6	Push	28.3	28,000	56,000	85,000	140,000

information will determine the size of the cylinder, the necessary fluid pressure, and the input flow rate, as was shown in (Section 2.1.5).

2.3.6 Accumulators

The purpose of an accumulator in a hydraulic system is to store or provide fluid at a pressure to minimize short-duration pressure spikes or to reach a short-duration high-flow demand. Most accumulators used in hydraulic systems are the spring-loaded or the gas-charged type. The spring-loaded accumulator simply uses the spring force to load a piston. When the fluid pressure increases to a point above the preload force of the spring, fluid will enter the accumulator to be stored until the pressure reduces. The gas-charged accumulator can be either a piston type or a bladder type, as shown in Figs. 1.49 and 1.50, respectively [25]. In the gas-charged accumulator, an inert gas such as dried nitrogen is used as a precharge medium. In operation, this type of accumulator contains the relatively incompressible hydraulic fluid and the more readily compressible gas. When the hydraulic pressure exceeds the precharge pressure exerted by the gas, the gas will compress, allowing hydraulic fluid to enter the accumulator.

2.3.7 Components of a Hydraulic Circuit Diagram

The proper planning of any hydraulic system should start with a properly drawn hydraulic circuit, using ISO-1219-approved graphic symbols. Figures 1.46 [24], 1.51 [18], and 1.52 [18] show the most common symbols found in circuit diagrams. An example of a simple hydraulic circuit used by hydraulic fluid manufacturers is ASTM D-2882-83 "Standard Method for Indicating the Wear Characteristics of Petroleum and Non-Petroleum Hydraulic Fluids in a Constant Volume Vane Pump." This pump test is currently the only one that has ASTM status. Figure 1.53 is an illustration of the hydraulic circuit diagram for this test. The electric motor (1) supplies mechanical energy to a Vickers V-104 vane pump (2), which, in turn, converts mechanical energy into hydraulic energy. The pump outlet pressure is monitored by a pressure gauge (4). The relief valve (5) is adjusted to induce a load pressure of 2000 psi as measured by the pressure gauge (4). The outlet of the relief valve is at low pressure <20 psi. The fluid then passes through a filter (6), then through a flow meter (7), which measures the flow rate to be ~5–6 gpm (8 gpm at no load). The fluid then passes through a heat exchanger (8) and then into the reservoir (9). After the reservoir, the fluid passes through a 60-mesh filter (usually inside the reservoir at the outlet to the pump). A thermoregulator valve (13) is used to maintain a constant reservoir tem-

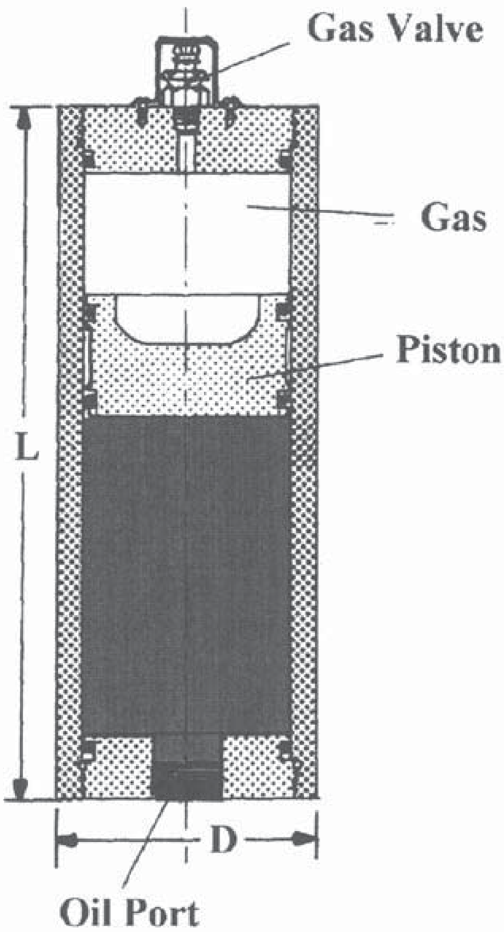


Figure 1.49 Illustration of a piston-type gas-charged accumulator.

perature (14) (set at 80°C for oil or 65°C for water-based fluids). Based on the known parameters of speed (1200 rpm), pressure (2000 psi), and flow (8 gpm, theoretical), other parameters, such as power, torque, and heat, can be readily calculated for this circuit as follows:

$$\text{Power}_{\text{input}} = \frac{(\text{gpm})(\text{psi})}{1714E_t} = \frac{(8.0)(2000)}{(1714)(1.0)} = 9.3 \text{ hp} \tag{1.43}$$

$$\text{Torque}_{\text{input}} = \frac{(\text{hp})(63{,}025)}{(\text{rpm})E_t} = \frac{(9.3)(63{,}025)}{(1200)(1.0)} = 490 \text{ in.-lbs} \tag{1.44}$$

$$\frac{\text{BTU}}{\text{h}} = 1.5(\text{gpm})(\text{psi}) = (1.5)(8.0)(2000) = 24{,}000 \tag{1.45}$$

It should be noted that these calculations have assumed the pump to be 100% efficient ($E_t = 1.0$). In the real world, this is never the case. Typically, vane pumps have $E_t < 0.9$. The effect of this would be to increase the power requirement of the pump to deliver the desired flow rate at a given load-induced pressure. However, from actual experience with this ASTM test, the Vickers V-104 pump delivers only ~5–6 gpm at 2000 psi. This is because the test requires running the pump at a 2000-psi-load pressure, which is 1000 psi higher than the designed maximum pressure for this pump. Therefore, the volumetric efficiency E_v = ~6/8 = ~0.75; assum-

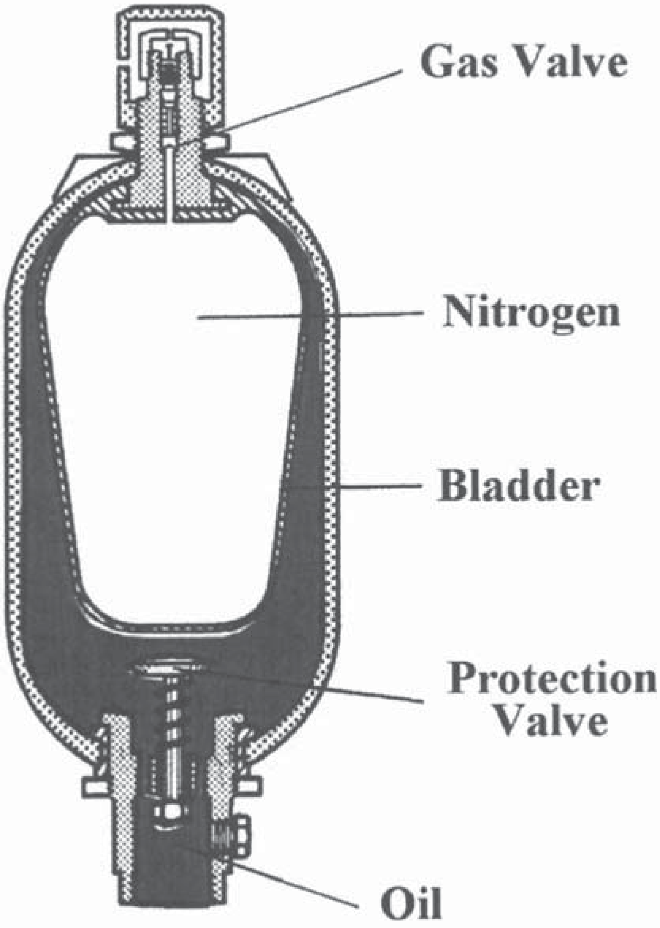


Figure 1.50 Illustration of a bladder-type gas-charged accumulator.

ing the mechanical efficiency $E_m = 0.9$, we have the total efficiency $E_t = (0.75)(0.9) = 0.68$ as a more reasonable value for the total efficiency of this pump at 2000 psi.

2.4 Basic Hydraulic System Design

2.4.1 Pipe and Hose Sizing

Whereas it is necessary to connect the various components in a hydraulic system with some kind of piping, such piping will produce flow resistance and therefore cause parasitic losses in the hydraulic system. To avoid as much loss as possible, the piping or hose must be sized properly. The internal diameter of the hose is extremely important because the fluid velocity at any given flow rate will depend on that diameter. In fact, the fluid velocity will equal the flow rate divided by the internal area of the pipe as follows:

$$V = \frac{0.3208Q}{A} \tag{1.46}$$

where V is the velocity (ft/s), Q is the flow rate (gpm), and A is the internal pipe area (in.^{2}).

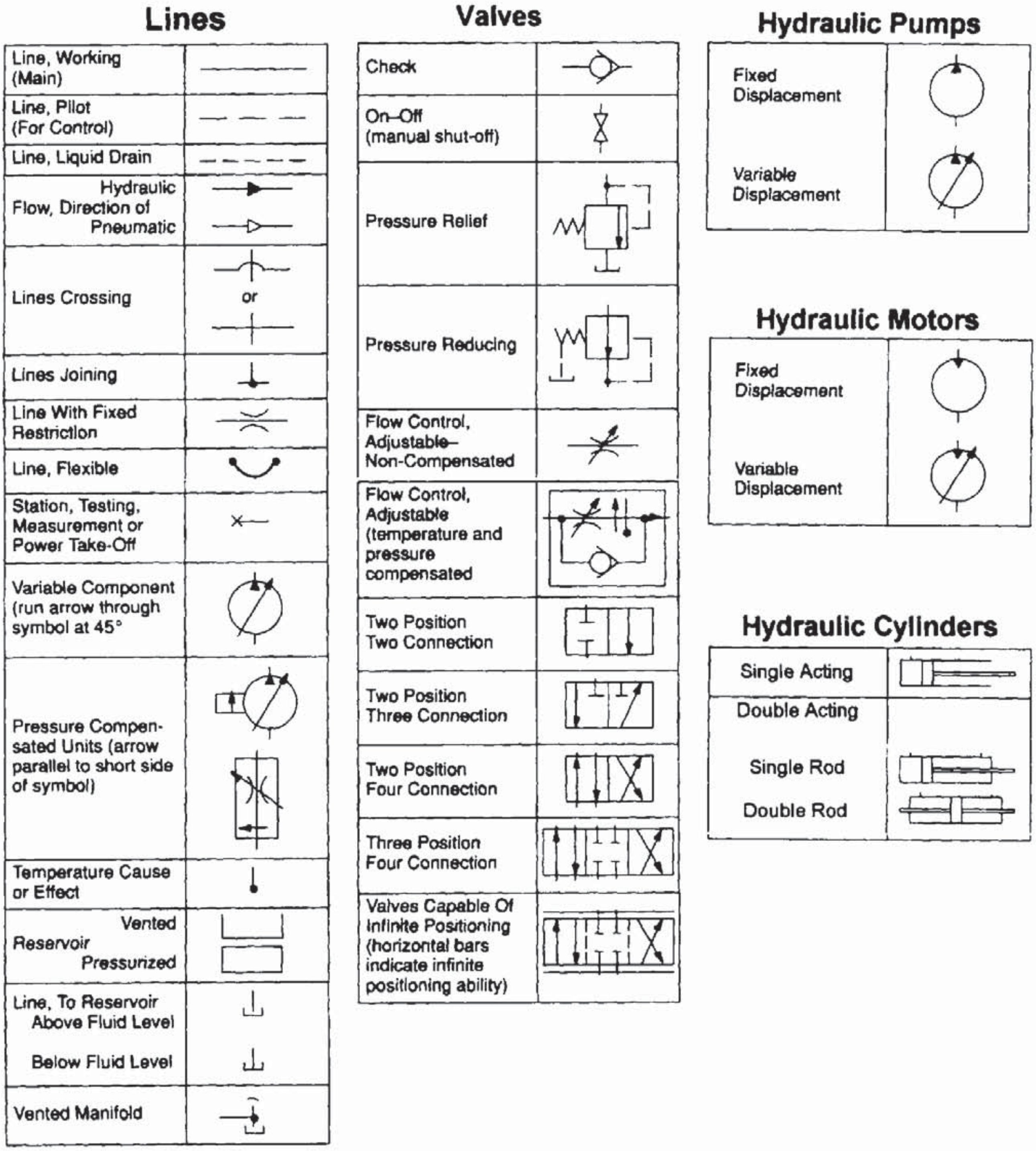


Figure 1.51 Basic hydraulic symbols—1.

The fluid velocities recommended for hydraulic systems are given in Table 1.7. Pressure-drop calculations for the piping or hose can be made using Eq. (1.32). Although most texts refer to the flow regime present in the pipe when making such calculations, the Moody diagram will take the flow regime into consideration, as it relies on the Reynolds number (*N*,) which depends upon fluid velocity, fluid viscosity, and the inside diameter of the pipe as shown by the equation,

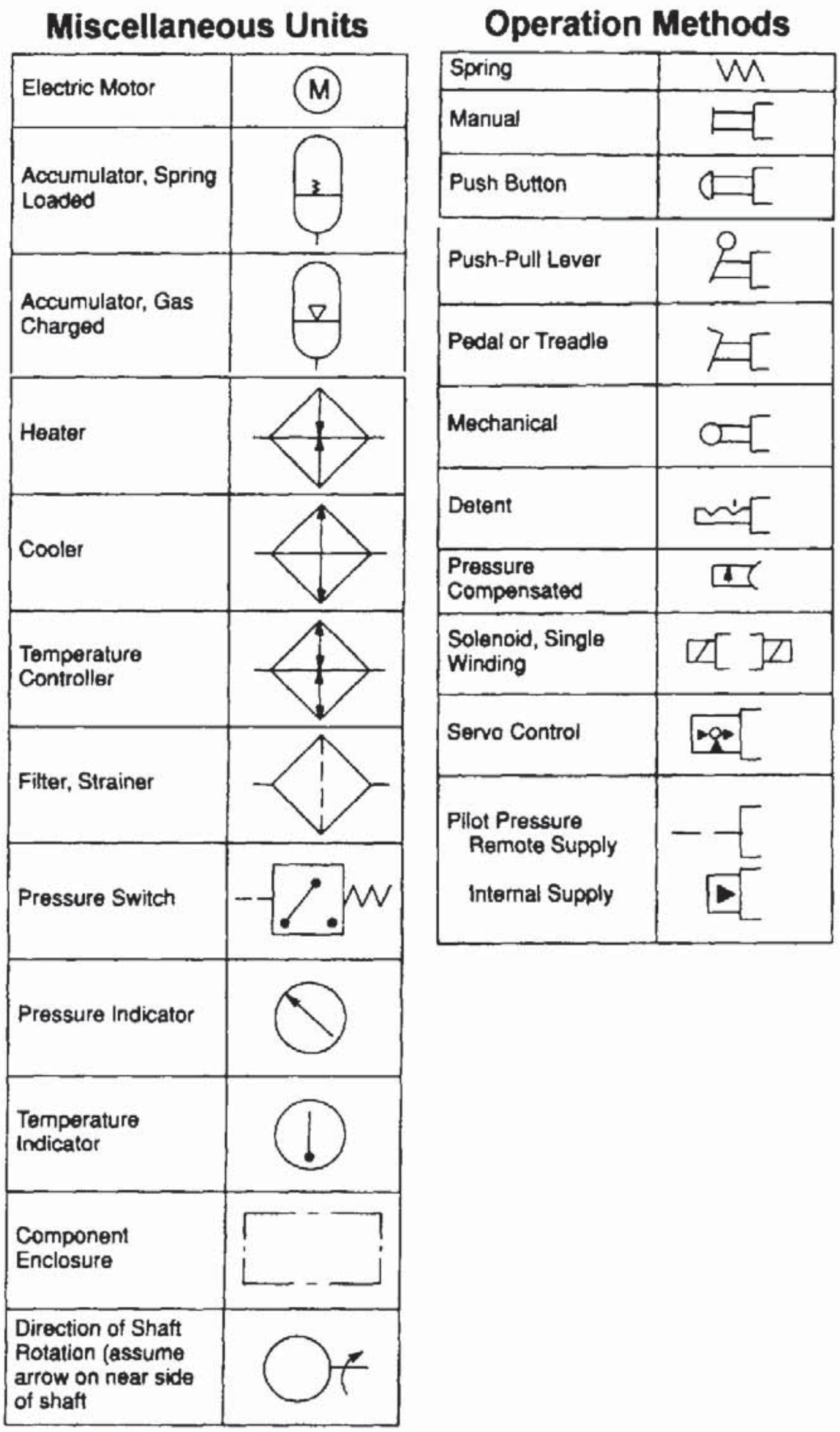


Figure 1.52 Basic hydraulic symbols—2.

$$N_r = \frac{3162Q}{\mu d} \tag{1.47}$$

where N_r is the Reynolds number, μ is the viscosity (cSt), and d is the pipe inner diameter (in.).

Fittings and valves must be handled somewhat differently than straight runs of pipe. The easiest way to calculate the losses resulting from fittings and valves is to use the equivalent-length method to estimate the effect by treating it as if it were an

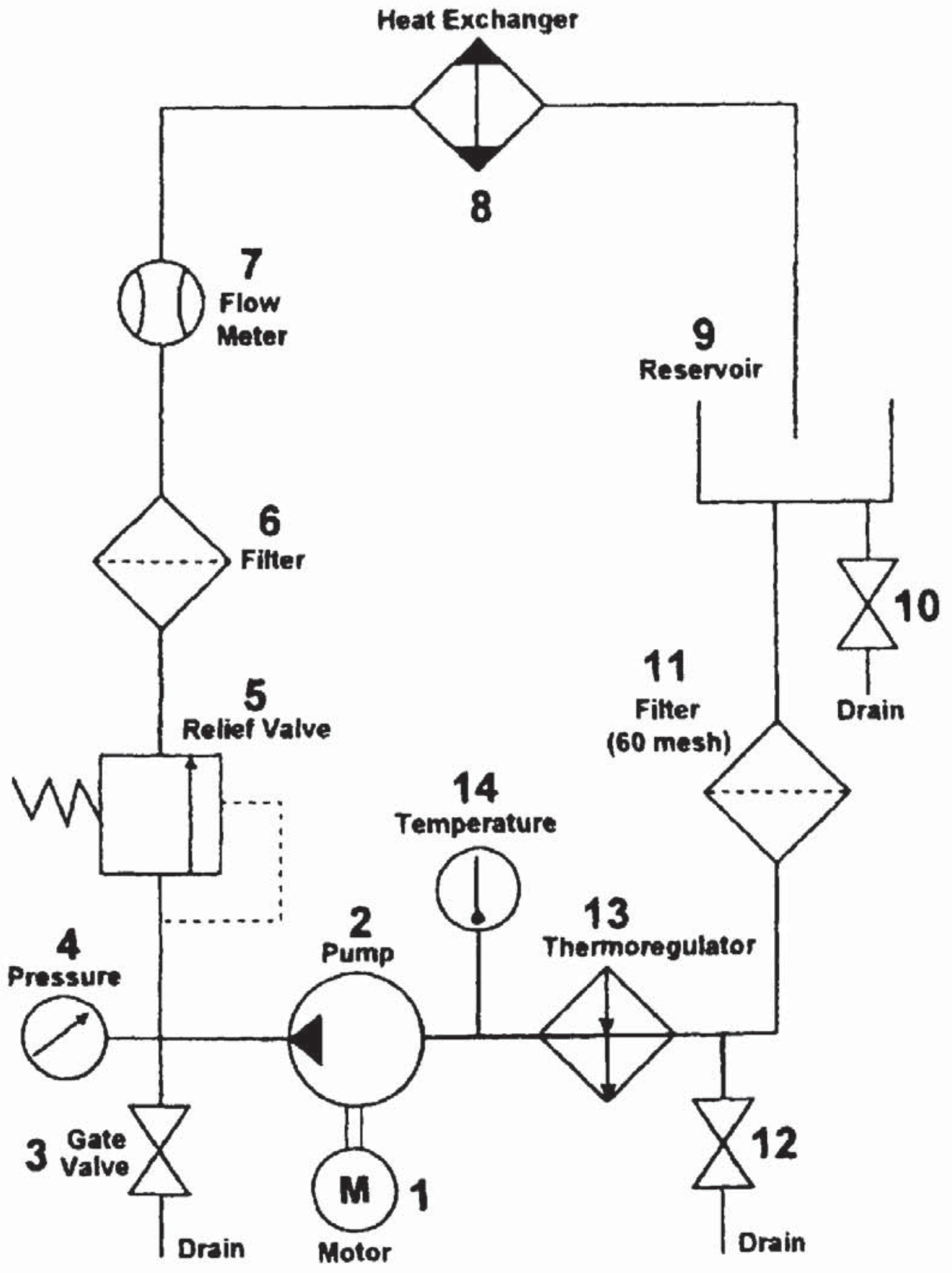


Figure 1.53 ASTM D-2882 pump test hydraulic circuit diagram.

additional length of pipe. Table 1.8 lists some common devices and their equivalent-length values, which are given as the length-to-diameter (L_e/D) ratios so that they can be used directly in the modification of the Darcy equation as follows:

$$h_r = \lambda \frac{L_e}{D} \frac{v^2}{2g} \tag{1.48}$$

where h_f is the equivalent length, λ is the friction factor, L_e/D is the equivalent-length values, v is the fluid velocity, and g is the gravitational constant.

The analytical methods presented here to calculate pressure losses in hydraulic piping and fittings is accurate but can be very time-consuming. A method that is less

Table 1.7 Recommended Hydraulic Circuit Flow Velocities

Suction line		Pressure line		Return line
Viscosity (SUS)	Velocity (ft/s)	Pressure (psi)	Velocity (ft/s)	velocity (ft/s)
700	2.0	365	8.2–10.0	5.5–15.0
465	2.5	725	11.5–13.0	5.5–15.0
230	4.0	1450	14.5–16.5	5.5–15.0
140	4.3	2900	16.5–20.0	5.5–15.0
140–700	4.3–2.0	<2900	20.0	5.5–15.0

accurate but provides a reasonable estimate of pressure losses in hydraulic systems involves the use of tables available from pipe manufacturers and in various handbooks concerning fluid flow (Table 1.5).

2.4.2 Reservoir Design

A typical design for an industrial reservoir is shown in Fig. 1.54 [7]. Several features can be seen in this figure. The overall dimensions should enclose a sufficient volume of oil to permit air bubbles and foam to escape during the resident time of the fluid in the reservoir. The depth must be adequate to assure that during peak pump demands, the oil level will not drop below the pump inlet level. The pump should be mounted below the reservoir so that a positive head pressure is available at all times. This is very critical when water-based hydraulic fluids are used, as these fluids can have a higher specific gravity as well as a much higher vapor pressure than mineral-oil-based fluids (Sec. 2.2.7). The reservoir should be sized to afford adequate fluid cooling. Baffles are provided to prevent channeling of the fluid from the return line to the inlet line. The bottom of the return line is usually cut at a 45° angle to assist in the redirection of the fluid away from the inlet. A cleanout plate is provide to promote cleaning and inspection. Sight gauges are normally used to monitor the fluid level. A breather system with a filter is provided to admit clean air and to maintain atmospheric pressure as fluid is pumped into and out of the reservoir. With water-based hydraulic fluids, a pressurized reservoir is recommended. Special breather caps

Table 1.8 Equivalent Length Values

Device	Equivalent length (L_e/D)
Check valve	150
90° Standard elbow	30
45° Standard elbow	16
Close return bend	50
Standard tee-run	20
Standard tee-branch	60

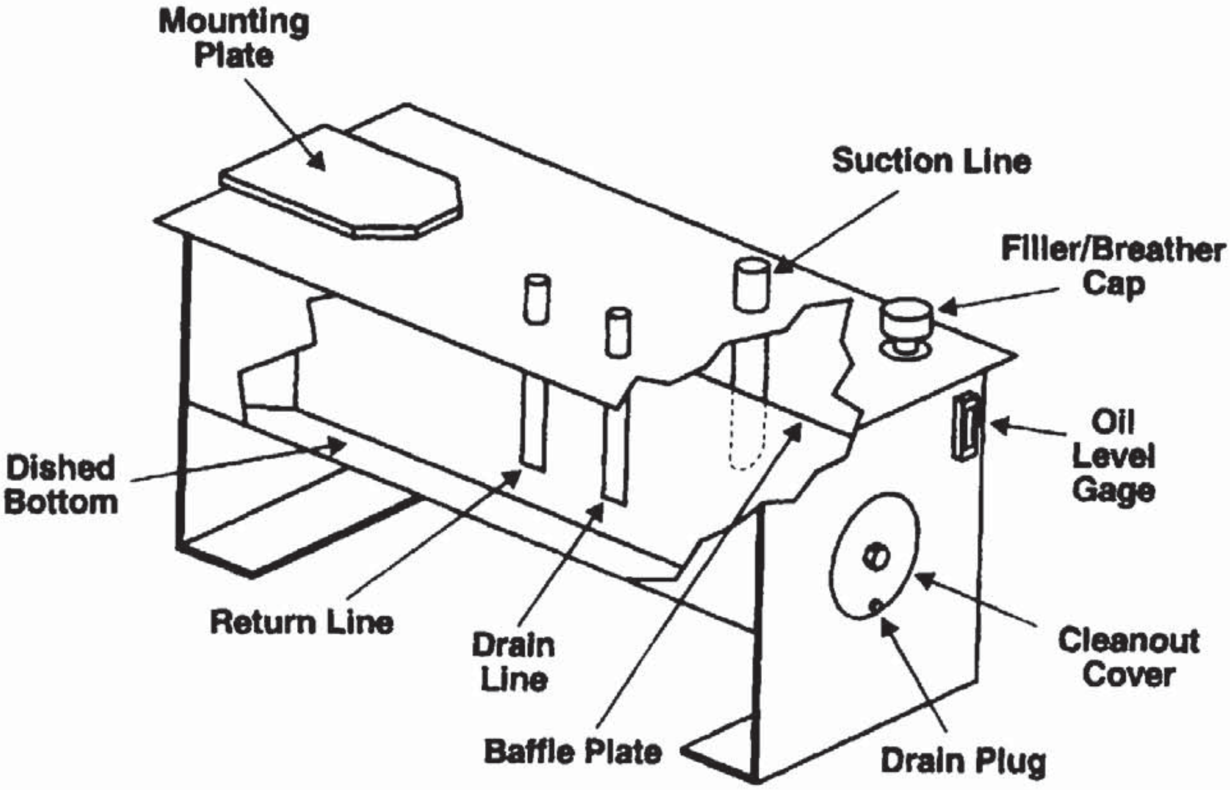


Figure 1.54 A typical design for an industrial reservoir.

can be purchased to vent between 1 and 15 psig. If one of these are used, make sure that it has a vacuum brake to vent at approximately −0.5 psig. (**Note:** Not all pressure caps have a vacuum brake.) This is important so that when the reservoir is cooling down, no appreciable vacuum develops in the reservoir tank. This feature will minimize pump cavitation upon start-up and also prevent a possible tank implosion.

2.4.3 Natural Frequency and Time Response

When designing any hydraulic system, especially when heavy masses are moved quickly, there is one very important design factor that needs to be considered. That factor is known as the "natural frequency" ($\bar{\omega}_0$) of the system. Knowledge of this frequency is important because it determines how fast one can accelerate a given load and, thus, its maximum achievable velocity.

From the physical laws of motion, the natural frequency of a hydraulic system can be found by taking the square root of the effective spring constant divided by the effective moving mass:

$$\omega_0 = \sqrt{\frac{C}{M}} \tag{1.49}$$

where $\bar{\omega}_0$ is the natural frequency, C is the effective spring constant, and M is the effective moving mass. This is a simple statement; however, determination of the effective spring constant and effective moving mass is not so simple. The effective spring constant not only includes the compressibility of the trapped hydraulic fluid

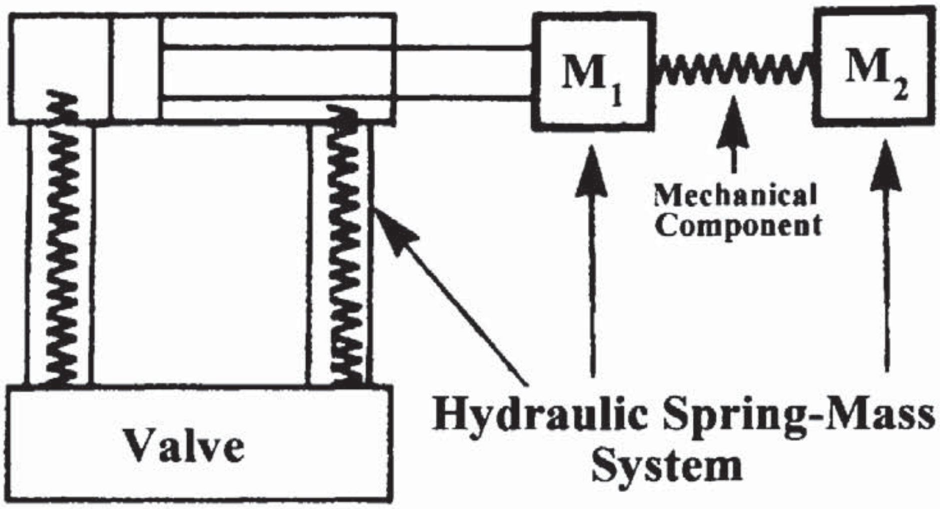


Figure 1.55 Illustration of the hydraulic spring-mass system.

between the valves and the actuators but also the movement of any hoses or piping, as well as structural vibrations. The effective mass of the system is the combination of all the moving loads, including the mass of the trapped fluid between the valves and actuators, as illustrated in Fig. 1.55 [26].

In the simplified case of a linear cylinder in a closed circuit (Fig. 1.56) [26], the natural frequency can be calculated using the following expression:

$$\omega_0 = \sqrt{\frac{A_b^2\beta}{V_1M} + \frac{A_e^2\beta}{V_2M}} \tag{1.50}$$

or

$$F_0 = \frac{\omega_0}{2\pi}$$

where $\bar{\omega}_0$ is the natural frequency (rad/s), A_b is the cylinder blind end area (in.^{2}), A_e is the cylinder extending end area (in.^{2}), β is the bulk modulus of fluid, V_1 is the cylinder blind end volume (in.^{3}), V_2 is the cylinder extending end volume (in.^{3}), M is the effective moving mass (lbs-s^2/ft, slugs), and F_0 is the natural frequency (Hz). There are computer programs available which can be used to determine the frequency response of a hydraulic system by using the impulse method.

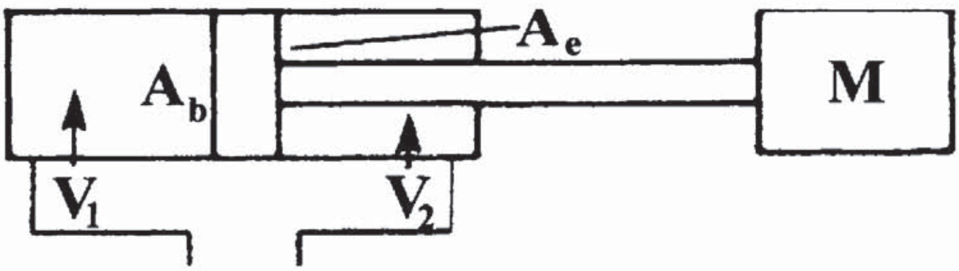


Figure 1.56 Illustration of the natural frequency parameters.

The time response of a hydraulic system is the synergistic result of the response times of all of the components used in the system [27]. Therefore, most component manufacturers will provide information relative to the responsiveness of their components. Unfortunately, the information derived from the component manufacturers is not consistent. The ability to understand and utilize the response information obtained from component manufacturers using a second-order system depends on the definition of several aspects of the response subject as follows:

- Delay time: the time required for the output to reach 50% of the steady output
- Rise time: the time required for the output to rise from 10% to 90% of the final output value
- Maximum overshoot: the time at which the maximum overshoot occurs
- Settling time: the time for the system to reach and stay within a stated plus-and-minus tolerance band around the steady-state output

A graph illustrating these parameters is presented in Fig. 1.57. Control technology can be used to evaluate the response of a complete hydraulic system if all of the component information is given in consistent and correct terms.

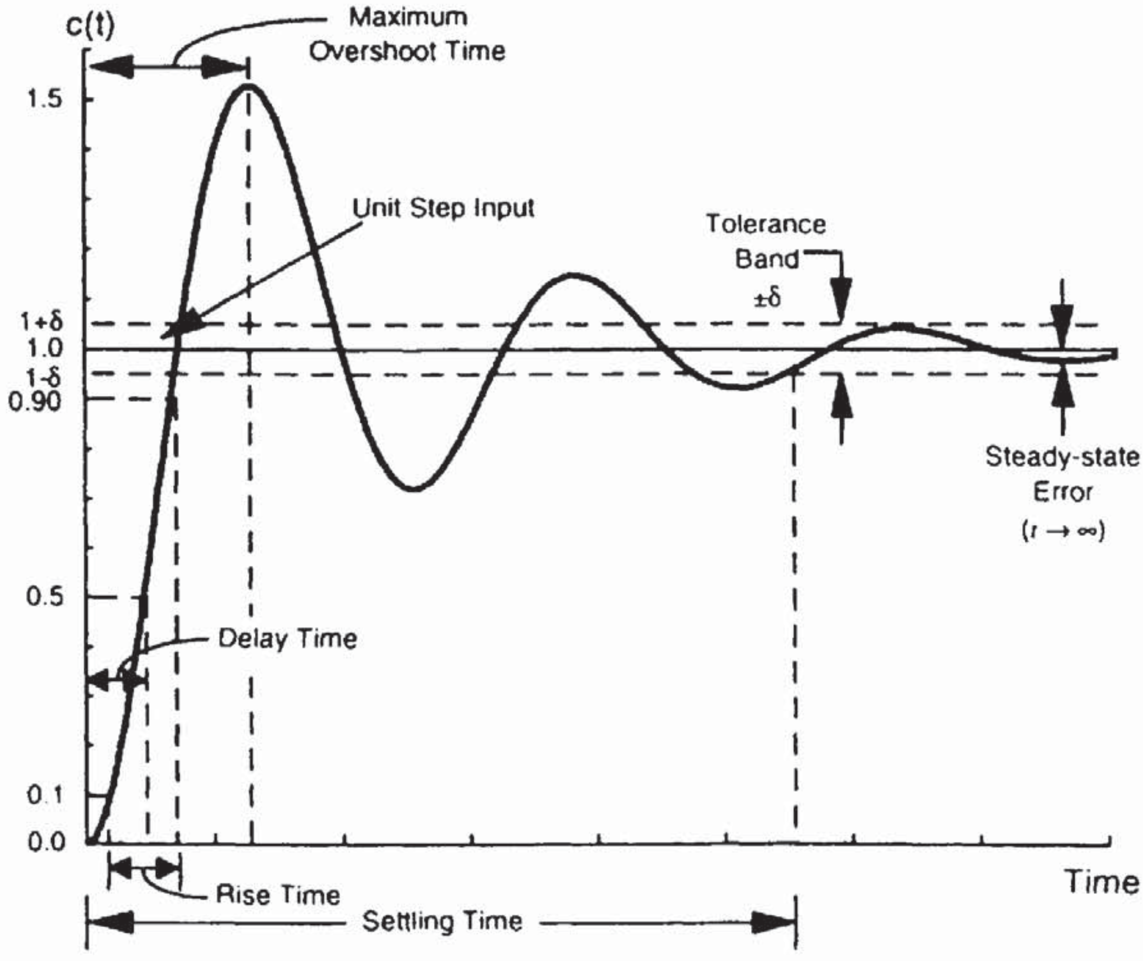


Figure 1.57 Step response of a second-order system.

Calculation of Natural Frequency, Acceleration, Maximum Velocity, Acceleration Pressure, and Flow Rate

For economic reasons, it is often desirable to operate a hydraulic system as fast as possible. This is especially true on automated assembly lines, where hydraulics are used to move parts.

As an example of a simple calculation, consider the following application, where one needs to determine the maximum speed and shortest cycle time to perform a repetitive task. By way of a single-rod hydraulic cylinder (1.5-in. bore, 1-in. rod), a proportional directional control valve is used to accelerate a 1000-lb load (M) to a constant velocity over a distance of 30 in. in 1 s and then decelerate the load to a stop. The load is then retracted in the same manner to start the cycle over again (Fig. 1.58) [26]. To solve this problem, the natural frequency must first be calculated so that the time to accelerate the load can be determined. Then, the maximum velocity, acceleration pressures, and required flow rates can be calculated for both the extending and retracting modes. The following information is given:

$w = 1000$ lbs (load)
$T_s = 1.0$ s (stroke time)
$A_b = 1.76$ in.^{2} (1.5 in. cylinder bore)
$A_e = 0.98$ in.^{2} (1.5-in.^{2} bore area − 1-in. rod area)
$S = 30$ in. (stroke distance)
$\beta = 200{,}000$ lb/in.^{2} (bulk modulus of oil)
$L_1 = 46.50$ in. (cylinder blind-end pipe length)
$L_2 = 38.75$ in. (cylinder rod-end pipe length)
$D = 0.62$ in. (pipe inner diameter)
Pipe size = 0.75-in. outer diameter × 0.065-in. wall

The first step is to calculate the pipe trapped volumes between the control valve and the cylinder blind-end inlet (V_3) and the rod-end inlet (V_4) in Fig. 1.58 [26] from the following equation:

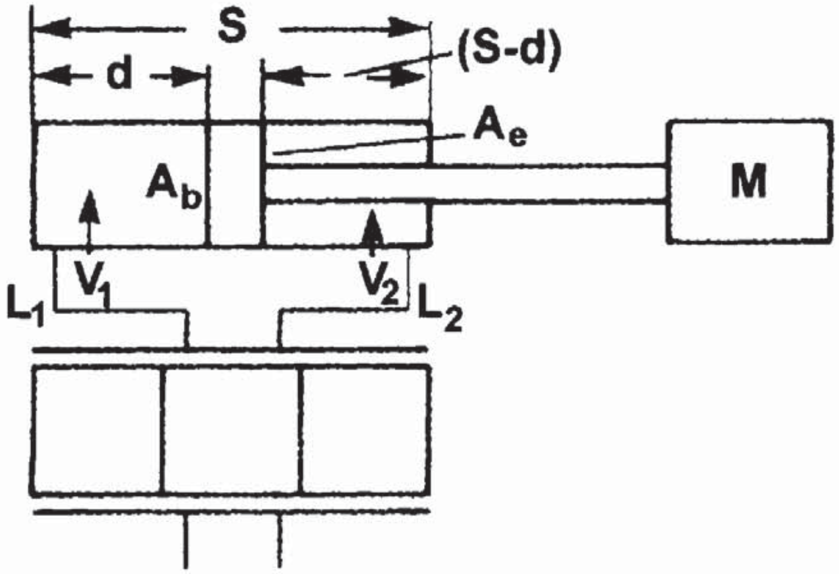


Figure 1.58 Illustration of a typical natural frequency calculation.

$$
\begin{aligned}
V_3 &= \frac{\pi D^2}{4} L_t \\
&= \frac{\pi(0.62)^2}{4}(46.5) \\
&= 14.04 \text{ in.}^3 \\
V_4 &= (0.30)(38.75) \\
&= 11.7 \text{ in.}^3
\end{aligned}
\tag{1.51}
$$

Next, calculate the dimension d, using the above values along with the given parameters, using the following expression:

$$
\begin{aligned}
d &= \left(\frac{A_r S + V_4}{\sqrt{A_r^3}} - \frac{V_3}{\sqrt{A_b^3}}\right)\left(\frac{1}{\sqrt{A_r}} + \frac{1}{\sqrt{A_b}}\right)^{-1} \\
&= \left(\frac{(0.98)(30) + 11.7}{\sqrt{(0.98)^3}} - \frac{14.04}{\sqrt{(1.76)^3}}\right)\left(\frac{1}{\sqrt{0.98}} + \frac{1}{\sqrt{1.76}}\right)^{-1} \\
&= 20.6 \text{ in.}
\end{aligned}
\tag{1.52}
$$

Next, calculate the total trapped volume between the valve and cylinder blind end (V_1) and cylinder rod end (V_2), using the following relations:

$$
\begin{aligned}
V_1 &= V_3 + A_b d \\
&= 14.04 + 1.76(20.6) \\
&= 50.3 \text{ in.}^3 \\
V_2 &= V_4 + A_r(S - d) \\
&= 11.7 + 0.98(30 - 20.6) \\
&= 20.9 \text{ in.}^3
\end{aligned}
\tag{1.53}
$$

Convert the load to units of mass as follows:

$$
M = \frac{W}{g} = \frac{1000}{386} = 2.59 \left(\frac{\text{lbs} - \text{s}^2}{\text{in.}}\right) \tag{1.54}
$$

where M is the effective moving mass (lbs $-$ s^2/in., slugs), w is the load force (lbs), and g is the gravitational constant (in./s^2). Then, substitute the known quantities into Eq. (1.50) to obtain the natural frequency ($\bar{\omega}_0$) of this system:

$$
\begin{aligned}
\bar{\omega}_0 &= \sqrt{\frac{(1.76)^2(200{,}000)}{(50.3)(2.59)} + \frac{(0.98)^2(200{,}000)}{(20.9)(2.59)}} \\
&= 91.1 \text{ rad/s}
\end{aligned}
\tag{1.55}
$$

In calculating $\bar{\omega}_0$, we have not taken into consideration other factors that contribute to the spring constant of the system, namely hoses and other mechanical components. However, it has been shown over the years that a good approximation to determine the usable acceleration is to divide the calculated natural frequency by 3 [26]. This simplification avoids a much more complex mathematical analysis, which would have required variables which are difficult, if not impossible, to define. Therefore, the usable frequency ($\bar{\omega}$) can be estimated as

$$\omega = \frac{\omega_0}{3} = \frac{91.1}{3} = 30.4 \ \frac{\text{rad}}{\text{s}}$$

or

$$F = \frac{\omega}{2\pi} = \frac{30.4}{2\pi} = 4.8 \text{ Hz} \tag{1.56}$$

The acceleration time (T) or the time for one complete oscillation can now be calculated:

$$T = \frac{1}{\omega} = \frac{1}{30.4} = 0.033 \text{ s} \tag{1.57}$$

However, it has been determined that this period is too short for acceleration to stabilize using proportional valves. Generally, for stable acceleration, the time allowed must be a minimum of four to six times the time period for one oscillation [26]. Therefore, the acceleration stabilizing time (T_b) is calculated to be

$$T_b = 6T = 6(0.033) = 0.20 \text{ s} \tag{1.58}$$

From the stroke distance (S), the acceleration time (T_b) and the stroke time (T_s), the maximum velocity (V_{max}), acceleration (A_{max}), and the acceleration force (F_a) can be easily calculated from the following expressions:

$$V_{max} = \frac{S}{T_s - T_b} = \frac{30}{1.0 - 0.2} = 37.5 \ \frac{\text{in.}}{\text{s}} \tag{1.59}$$

$$A_{max} = \frac{V_{max}}{T_b} = \frac{37.5}{0.20} = 188 \ \frac{\text{in.}}{\text{s}^2} \tag{1.60}$$

$$F_a = MA_{max} = \frac{w}{g} A_{max} = \left(\frac{1000}{386}\right) 188 = 487 \text{ lbs} \tag{1.61}$$

Before we can calculate the acceleration pressure at the blind end (P_b) and rod end (P_r) of the cylinder, the frictional force that the load imposes on the system needs to be determined. For this calculation, it is assumed that the coefficient of friction (μ) equals 0.58; we can then determine the force due to friction (F_μ) and the total force (F_t) as follows:

$$F_\mu = \mu w = (0.58)(1000) = 580 \text{ lbs} \tag{1.62}$$

$$F_t = F_\mu + F_a = 580 + 487 = 1067 \text{ lbs} \tag{1.63}$$

$$P_b = \frac{F_t}{A_b} = \frac{1067}{1.76} = 606 \text{ psi}$$

$$P_r = \frac{F_t}{A_r} = \frac{1067}{0.98} = 1089 \text{ psi} \tag{1.64}$$

One should note that for a single-rod cylinder, the rod-end pressure is always greater than the blind end, but only with double-rod cylinders having equal rod diameters will the pressure be the same at both ends.

Finally, the flow rate required at the blind end (Q_b) and rod end (Q_r) may be calculated as follows:

$$Q_h = \frac{V_{max}A_h(60)}{231} = \frac{(37.5)(1.76)(60)}{231} = 17.1 \text{ gpm}$$

$$Q_r = \frac{V_{max}A_r(60)}{231} = \frac{(37.5)(0.98)(60)}{231} = 9.6 \text{ gpm} \tag{1.65}$$

2.5 Hydraulic Fluid Considerations

2.5.1 Foaming

Most hydraulic fluids have an antifoaming agent as an additive. These additives have caused discussions among hydraulic system designers and users. Most of the additives used to control the foaming tendencies of hydraulic fluids accomplish this task by increasing the surface tension of the fluid. When the surface tension increases, the size of air or vapor bubbles which will coexist in the fluid become smaller and are therefore less likely to rise to the surface and cause a foaming situation. However, when the air is allowed to remain in the fluid, the compressibility of the fluid increases or, stated in another way, the bulk modulus of the fluid decreases. The suspension of air or vapor in the circulating fluid of a hydraulic system is a fault of the system; that is, a well-designed system will not permit air or vapor to become entrained in the fluid. Some expert designers of hydraulic systems have said that they would rather not have an antifoam agent present. Without the addition of the antifoaming agent, a system that is poorly designed will be readily apparent and can be fixed. Details on foaming, air entrainment, and air release are provided in Chapter 4.

2.5.2 Bulk Modulus

The bulk modulus of a fluid is a term used to describe the compressibility of the fluid. In fact, the bulk modulus is inversely proportional to the compressibility. The purpose of a hydraulic system is to raise the potential energy of the system by increasing the pressure of the fluid. This potential energy can then be converted into kinetic energy that will do useful work. However, a fluid with a low bulk modulus will be very compressible and the energy necessary to raise the pressure must also be sufficient to compress the fluid. Most hydraulic fluids have a very high bulk modulus in the pristine condition. However, when air is present, the effective bulk modulus will be low and the system fluid will need to absorb the heat generated when the compression takes place. Calculation procedures for bulk modulus and fluid compressibility is described in more detail in Chapter 4.

Fluid Compressibility and Cylinder Lunge

Fluid compressibility has a great effect on cylinder performance—especially when the fluid type is changed, such as changing from a mineral-oil- to a water-based or synthetic fluid. Hydraulic cylinders are especially sensitive to changes in bulk modulus. In critical operations, it is often necessary to extend the cylinder smoothly and at a very constant velocity. If the load changes, the compressibility of the hydraulic fluid will have a negative influence on the constant velocity. Also, any change in the volume (ΔV) of the fluid under compression will translate into a change in cylinder stroke (ΔS) defined as "lunge." The following expressions can be used to calculate "lunge" (ΔS) and the resultant velocity change (Δv):

$$\Delta S = \frac{(V_p + AS)\Delta L}{A^2\beta} \tag{1.66}$$

$$\Delta\nu = \frac{(\Delta S)(60)}{\Delta\tau} \tag{1.67}$$

where ΔS is the lunge (in.), V_p is the volume in the pipe (in.^{3}), A is the effective piston area (in.^{2}), S is the Stroke (in.), ΔL is the load change (lbs), β is the bulk modulus of the fluid, $\Delta\nu$ is the velocity change (in./min), and $\Delta\tau$ is the load change time (s).

We will now apply these equations to the meter-in (Fig. 1.59) [28] and the meter-out (Fig. 1.60) [28] circuits under the following conditions:

$A_1 = 4.9$ in.^{2} (blind-end area)
$A_2 = 2.5$ in.^{2} (rod-end area)
$S = 24$ in. (stroke)
$L_1 = 3000$ lbs (full load)
$L_2 = 1000$ lbs (reduced load)
$\Delta L = 2000$ lbs (load change, $L_1 - L_2$)

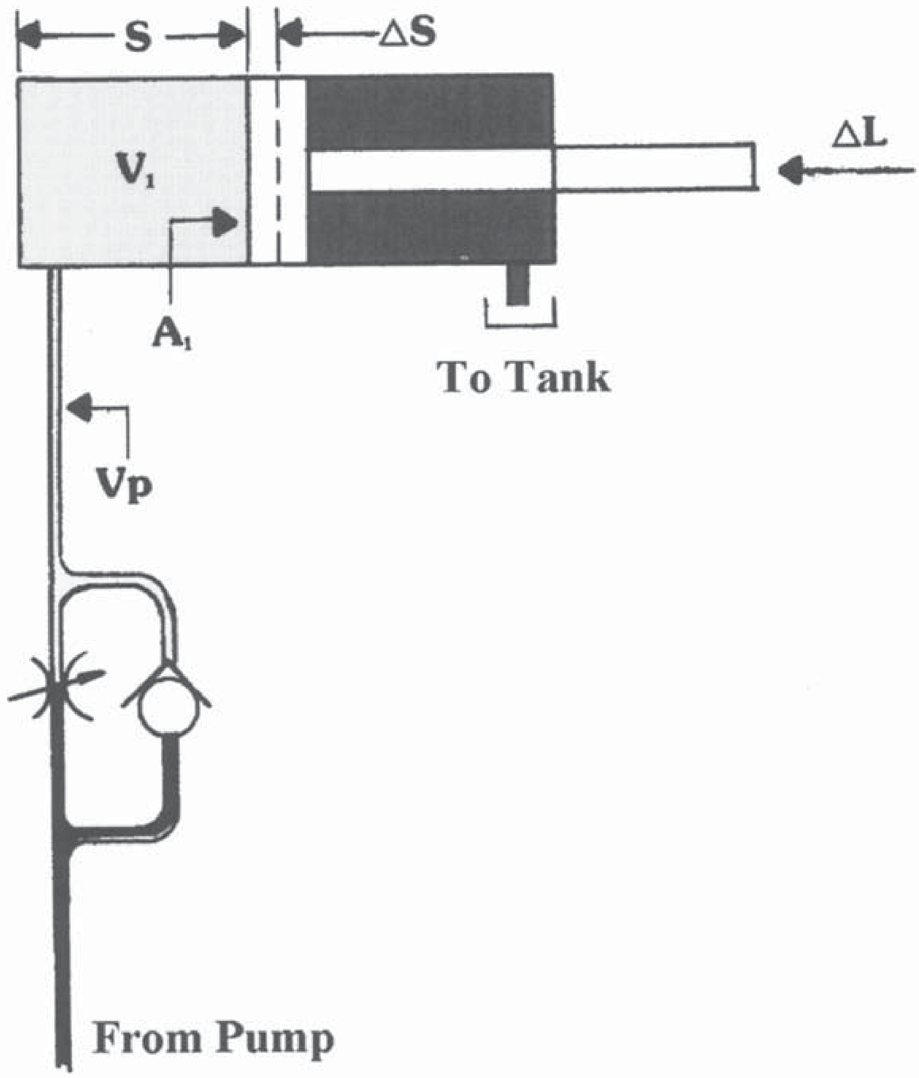


Figure 1.59 Illustration of a cylinder meter-in circuit.

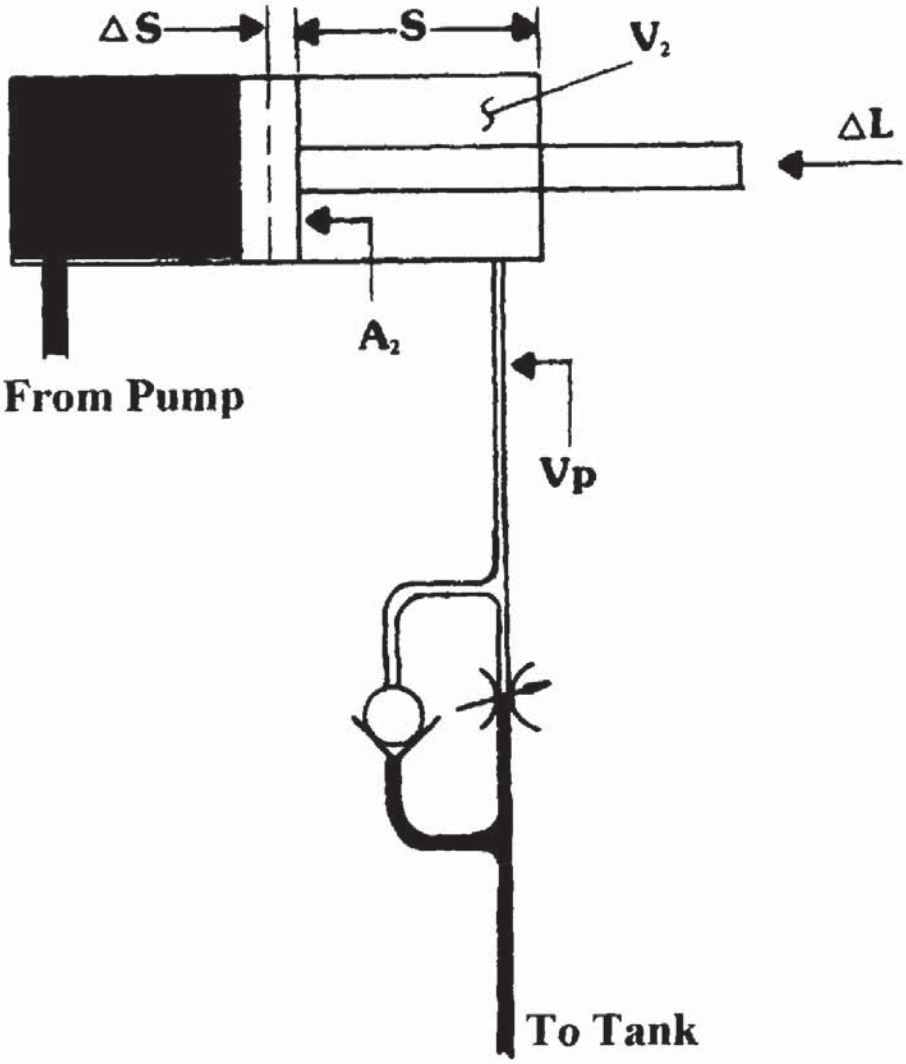


Figure 1.60 Illustration of a cylinder meter-out circuit.

$\Delta\tau = 1$ s (load change time)
$V_p = 36$ in.^{3} (oil line volume)
$\beta = 200{,}000$ lb/in.^{2} (bulk modulus of oil)

For the meter-in mode (Fig. 1.59) using Eqs. (1.66) and (1.67), we calculate

$$\Delta S = \frac{[36 + (4.9)(24)](2000)}{(4.9)^2(200{,}000)} = 0.064 \text{ in.} \tag{1.68}$$

$$\Delta \nu = \frac{(0.064)(60)}{1} = 3.8 \frac{\text{in.}}{\text{min}} \tag{1.69}$$

For the meter-out mode (Fig. 1.60) using Eqs. (1.66) and (1.67), we calculate

$$\Delta S = \frac{[36 + (2.5)(24)](2000)}{(2.5)^2(200{,}000)} = 0.154 \text{ in.} \tag{1.70}$$

$$\Delta \nu = \frac{(0.154)(60)}{1} = 9.2 \frac{\text{in.}}{\text{min}} \tag{1.71}$$

As can be seen from the above examples, the degree of "lunge" is directly proportional to the load change and inversely proportional to the bulk modulus of the fluid. Also, the cylinder lunge is greater in the meter-out mode than in the meter-in mode. This is due to pressure intensification in the rod end of the cylinder as discussed earlier in this chapter (Section 2.1.5).

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