FLUID POWER CIRCUITS and CONTROLS

Fundamentals and Applications

John S. Cundiff

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Fundamentals and Applications

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Preface

This book was written as text for a one-semester course in fluid power. It is expected that the course will be taught to senior-status engineering students. In most engineering curricula, fluid power is an elective course. Students interested in machine design (particularly those with a controls focus) make room in their course of study for a single technical elective in fluid power. This elective course must give them the basic foundation for the subject and give as much design experience as possible. Students need to feel confident that they can actually go out and "do something" with fluid power.

As they begin their study of fluid power, it is important that students quickly learn to think of the collection of components (pump, valves, actuators) as a system. I hope I have chosen a presentation of the material that encourages this learning. Each concept, and the components available to implement that concept, is illustrated with a circuit diagram of an application at the time the concept is introduced. When each component is discussed, it is immediately placed in a circuit and some analysis of circuit performance done. This approach allows the students to immediately apply what they have learned and encourages them to think about how the component operating characteristics interact with the rest of the circuit.

Chapter 1 gives a brief introduction to the fluid power industry and then develops the basic concept for power delivery with fluids. Elementary circuits are analyzed to present the fact that fluid power generally has a lower energy efficiency than other power delivery methods. This disadvantage must be offset by one or more of the several significant advantages of fluid power to justify its selection over a mechanical or electrical option. Chapter 2 reviews basic concepts learned in fluid mechanics and discusses the key properties of the fluids.

The two key variables in a fluid power system are pressure and flow. Chapter 3 discusses the various methods used to control pressure in a circuit, and Chapter 4 discusses the creation and control of flow.

Chapter 5 deals with rotary actuators and, as might be expected, most of the chapter is on motors. Having learned the characteristics of pumps (Chapter 4) and motors (Chapter 5), it is logical to follow with a discussion of hydrostatic transmissions in Chapter 6. Chapter 7 presents an analysis of linear actuators and completes the presentation of the key elements of a fluid power system.

Chapter 8 deals with temperature and contamination control. The requirement to maintain a lubricating film, and the ability of the oil to seal clearances, is a function of viscosity, which is a function of temperature. Contamination due to chemical reactions in the oil is also a function of temperature; thus it is appropriate to discuss temperature and contamination control in the same chapter.

Characteristics of auxiliary components (hoses, tubing, fittings, reservoir) are covered in Chapter 9. The appendix to this chapter has some handbook data. Instructors will want to supplement these data by providing handbooks and other reference material. My goal was to write a textbook, not a handbook.

Chapter 10 is the single chapter on pneumatics. It does not duplicate the discussion of components that are similar for liquid and gas but focuses on the difference in power transmission using the two fluids.

Servo valves are covered in Chapter 11, and this discussion is followed by a discussion of proportional valves in Chapter 12. These two chapters do not require a course in automatic controls as a prerequisite. Students who have had a controls course do get a more complete understanding of the material in Chapters 11 and 12.

I have taught my fluid power course here at Virginia Tech for 10+ years. It has always been taught with two lectures and one laboratory per week. The laboratory requires a sizeable commitment of resources, and I continue to fight for these resources, because I believe the laboratory experience is vital.

Some discussion of my organization of the laboratories is needed for those who review this text. Laboratories 1–7 cover the basics, and Laboratory 8 is a hydrostatic transmission design problem. These laboratories were written by me and are available to those who request them.

Laboratories 9–12 are taken from the Amatrol training manuals (copies loaned to the students) and cover the electrical control of fluid power circuits (characteristics of certain transducers, ladder diagrams, different types of relays, etc.). The Amatrol material is commercially available, as is similar material supplied by other trainer manufacturers, and it was judged to be inappropriate to try to duplicate it in this text.

Laboratories 13 and 14 are demonstrations of the use of servo valves for position and angular velocity control, respectively. Students who have had an automatic control course are particularly pleased to see the principles they have learned being demonstrated.

The basic outline of the text is mine, but much of the detail was contributions by reviewers. I am greatly indebted to the long list of individuals who gave much-needed, and much-appreciated, help. Please note the list of acknowledgements.

A number of reviews were obtained. Any remaining errors are mine. Thank you in advance to all those who take the time to send me the corrections.

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About the Author

John Cundiff received his PhD in Biological and Agricultural Engineering from North Carolina State University in 1972. His early research was on tobacco transplant mechanization, thus beginning a continuing interest in the use of hydraulics on mobile machines. This interest was expanded to industrial applications of hydraulics and pneumatics in 1987 when he began teaching a senior-level engineering course, "Fluid Power Systems and Circuits." This textbook captures 15 years of experience teaching this course.

Dr. Cundiff did research at the University of Georgia for the first eight years of his academic career and has taught at Virginia Tech in the Biological Systems Engineering Department since 1980. In 2000, he was elected a Fellow in the American Society of Agricultural Engineers.

Dr. Cundiff has two married children and three grandchildren. His hobbies are snow skiing and working on his 1964 Ford truck.

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Dedication

To Layne (son) and Angela (daughter), who have brought much joy into my life. I appreciate so very much your bringing two new people into my life, Michelle (daughter-in-law) and Patrick (son-in-law), and now, together, you have brought grandchildren, Nicholas and Sarah Beth from Layne and Michelle, and Rachael from Patrick and Angela. I am blessed beyond measure.

Contents

1

Brief Overview of Fluid Power

1.1 Introduction

In 1906, oil began to replace water as the pressurized fluid in hydraulic systems, and the modern era of fluid power began. Electrical control of fluid power began to enter the commercial sector with the development of servo valves in World War II. Today, the fluid power industry is a multibillion dollar industry.

Most mobile machines in the extraction industries (mining, logging, farming, and fishing) have fluid power circuits, thus fluid power is a major factor in the collection of raw materials for the international economy. Fluid power is an important part of most vehicles in the transportation industry. In manufacturing, the application of fluid power has continuously increased the productivity of workers and thus has had direct impact on the standard of living. Today, fluid power is a part of every product we use and service we enjoy.

1.2 Concept of Fluid Power

This text leads the reader through several levels of complexity beginning with simple circuits with a simple function and concluding with an introduction to the use of servo valves to control heavy loads moving at high speed. As we proceed along this journey, the reader will be periodically reminded that the fundamental concept of fluid power is quite simple. Fluid power technology is the conversion of mechanical energy to fluid energy, delivery of this energy to a utilization point, and then its conversion back to mechanical energy. A fluid power circuit has all three features: conversion from mechanical energy to fluid energy, delivery, and conversion from fluid energy back to mechanical energy. An electrical circuit also has all three features, but often the designer focuses only for the final conversion step—electrical-to-mechanical. Generation of the electrical energy and its delivery are external to the design problem.

1.2.1 Basic Circuits

Many people have an intuitive understanding of a basic cylinder circuit and a basic motor circuit. The block diagram shown in Fig. 1.1 gives the concept for a motor circuit. Two parameters, torque (*T*) and shaft speed (*N*), are converted to two different parameters, pressure (*P*) and flow (*Q*), using a pump. The two new parameters, *P* and *Q*, are converted back to *T* and *N* using a motor. The principal reason for converting to fluid power is the convenience in transferring energy to a new location. The pressurized fluid, defined by the *P* and *Q* parameters, easily flows around corners and along irregular pathways before reaching the point where it is converted back to *T* and *N*.

Fluid power is used on many agricultural machines because of the need to transfer power to a remote location. Suppose a conveyor must be driven on the opposite side of a machine from the prime mover. (On mobile machines, the prime mover is typically an internal combustion engine, and on stationary machines, it typically is an electric motor.) Power could be mechanically transmitted using a right-angle gearbox, shafts, bearings, roller chains, or belts. Using fluid power, the task is accomplished with a hydraulic pump mounted at the prime mover, two hydraulic hoses, and a hydraulic motor at the conveyor. Often, machine weight is reduced, and reliability increased, by using fluid power. In addition, overload protection is provided by simply installing a relief valve.

1.2.1.1 Brief Review of Mechanics

Power is defined as the rate of doing work or work per unit time. Work is defined as *force times distance.* Suppose a force acts through a moment arm to

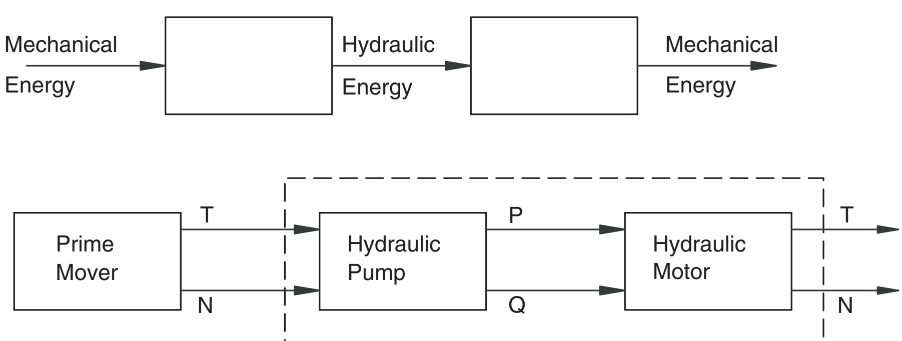


FIGURE 1.1
Concept of fluid power illustrated with basic motor circuit.

produce a torque. If the shaft is rotating at a given speed (N), the distance traveled in one minute is

$$x = 2\pi rN \tag{1.1}$$

where x = distance traveled (in)
r = moment arm (in)
N = rotational speed (rpm)

Work done in a one-minute interval is

$$\begin{aligned}\text{Work} &= \text{Force} \times \text{Distance}\\ &= F \times 2\pi rN\\ &= 2\pi TN\end{aligned} \tag{1.2}$$

where $T = \mathrm{F} \times \mathrm{r}$

Since power is the rate of doing work, the work done in one minute, Eq. (1.2), is

$$\mathcal{P} = \text{Work}/t = 2\pi TN/1 \tag{1.3}$$

where $\mathcal{P}$ = power (lb_f-in)/min.

One horsepower is 33,000 lb_f-ft/min, therefore,

$$\text{hp} = \frac{(\mathcal{P}/12)}{33,000}$$

If torque is expressed in lb_f-in and N is shaft speed in rpm, then power in hp is given by

$$\begin{aligned}\text{hp} &= \frac{2\pi(T/12)N}{33000}\\ &= \frac{TN}{63025}\end{aligned} \tag{1.4}$$

Mechanical power is proportional to the product of T and N. Is hydraulic power proportional to the product of P and Q?

1.2.1.2 Basic Concept of Hydraulic Cylinder

Suppose a flow of fluid is delivered to a hydraulic cylinder, causing it to extend. The cylinder has a cross-sectional area A and delivers a force F while moving a distance x (Fig. 1.2). The distance moved is related to the fluid volume delivered to the cylinder.

$$x = V/A \tag{1.5}$$

where x = distance (in)
V = volume (in^3)
A = area (in^2)

The force is related to the pressure developed at the cap end.

$$F = PA \tag{1.6}$$

where F = force (lb_f)
P = pressure (lb_f/in^2)
A = area (in^2)

(Throughout this text, pressure in lb_f/in^2 is expressed as psi.)
Work done is given by

$$\begin{aligned} \text{Work} &= Fx \\ &= (PA)\left(\frac{V}{A}\right) = PV \end{aligned} \tag{1.7}$$

Power is work per unit time,

$$\text{Power} = PV/t \tag{1.8}$$

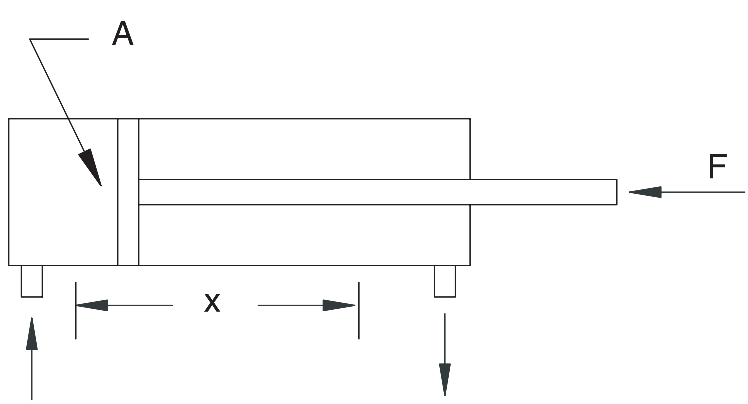


FIGURE 1.2
Hydraulic cylinder being extended a distance x against a force F.

Flow is defined as volume per unit time, $Q = V/t$; therefore,

$$\text{Power} = PQ \tag{1.9}$$

Mechanical power is the product of T and N and hydraulic power is the product of P and Q.

The relationship in Eq. (1.9) is the fundamental concept for our study of fluid power. The units used for pressure are typically lb_f/in^2, or psi, and the units for flow are gal/min, or GPM. To obtain hydraulic power with units of lb_f-ft/min, the following conversions are needed.

$$\mathcal{P}_{hyd} = PQ(231/12) \tag{1.10}$$

where $\mathcal{P}_{hyd}$ = hydraulic power (lb_f-ft/min)
P = pressure (psi)
Q = flow (GPM)

To obtain hydraulic horsepower,

$$\begin{aligned}\mathcal{P}_{hyd} &= \frac{PQ(231/12)}{33000} \\ &= \frac{PQ}{1714}\end{aligned} \tag{1.11}$$

It is useful to memorize this formula, as it will give a quick frame of reference when beginning a design. For example, if a pump delivers 5 GPM at 2,000 psi pressure drop, how much power is required?

$$\mathcal{P}_{hyd} = \frac{2000(5)}{1714} = 5.8\text{hp}$$

1.2.1.3 Basic Concept of a Hydraulic Motor

A flow of fluid is delivered to a hydraulic motor having displacement V_m. (The displacement of a hydraulic motor is the volume of fluid required to produce one revolution. Typical units are in^3/rev.) When a flow Q is delivered to this motor, it rotates at N rpm.

$$N = Q/V_m \tag{1.12}$$

where N = rotational speed (rpm)
Q = flow (in^3/min)
V_m = displacement (in^3/rev)

The derivation of an expression for torque produced by a hydraulic motor is straightforward but really not intuitive. We begin with the definition for mechanical horsepower.

$$\mathcal{P}_{mech} = \frac{2\pi TN}{33000} \tag{1.13}$$

where $\mathcal{P}_{mech}$ = mechanical power (hp)
T = torque (lb_f-ft)
N = rotational speed (rpm)

Substituting for N from Eq. (1.12),

$$\mathcal{P}_{mech} = \frac{2\pi T(Q/V_m)}{33000} \tag{1.14}$$

If the units for flow are GPM, and the units for torque are lb_f-in, then the expression becomes

$$\begin{aligned}\mathcal{P}_{mech} &= \frac{2\pi T/12(231Q/V_m)}{33000}\\ &= \frac{2\pi TQ/V_m}{1714}\end{aligned} \tag{1.15}$$

It has already been shown [Eq. (1.11)] that hydraulic horsepower is proportional to the product of pressure drop and flow. Equating Eqs. (1.11) and (1.15),

$$\text{hp} = \frac{\Delta PQ}{1714} = \frac{2\pi TQ/V_m}{1714}$$

Solving for torque, we find that

$$T = \frac{\Delta PV_m}{2\pi} \tag{1.16}$$

where T = torque (lb_f-in)
ΔP = pressure drop across motor (psi)
V_m = displacement (in^3/rev)

The relationship in Eq. (1.16) should be committed to memory. Torque that is delivered by a hydraulic motor is a function of the pressure drop across the motor and the displacement.

The derivation of Eq. (1.16) assumes that all the hydraulic power delivered to the motor is converted to mechanical power. This conversion is not 100% efficient; there are losses. Analysis of the losses is presented in Chapter 5. The formula in Eq. (1.16) is widely used and is a very useful approximation. It is important to remember that the actual torque available at the motor shaft is always less than that computed with Eq. (1.16).

1.2.2 Basic Circuit Analysis

The fluid power circuit shown in Fig. 1.3 has four components. The functions of these components are described below.

1. *Pump.* The pump develops a flow of fluid through the circuit. The pump shown in Fig. 1.3 is a fixed-displacement pump, which means that it delivers a fixed volume of fluid each revolution.
2. *Relief valve.* The relief valve protects the circuit. If the pressure rises high enough to offset the spring force keeping the valve closed, the valve opens, and flow returns to the reservoir, thus limiting the maximum circuit pressure.
3. *Directional control valve (DCV).* The directional control valve directs the flow of fluid based on its position. The valve in Fig. 1.3 is a three-position valve. In the center position, flow passes through the valve back to the reservoir. In the bottom position, flow is delivered to the cap end of the cylinder, causing it to extend. Simultaneously, fluid from the rod end flows to the reservoir. To retract the cylinder, the DCV is shifted to the top position, which reverses flow to the cylinder.
4. *Cylinder.* Another name for a cylinder is a *linear actuator.* The cylinder converts hydraulic energy into a force acting over some distance, known as the *stroke*.

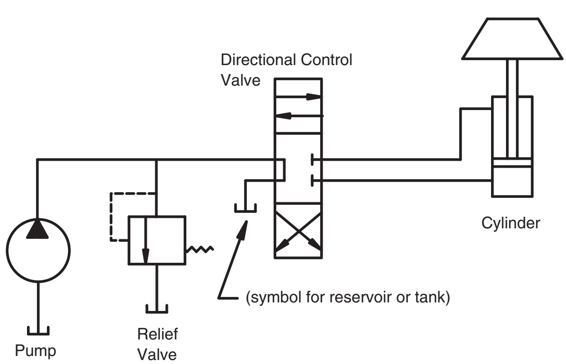


FIGURE 1.3
Basic circuit to extend cylinder.

The functional objective of the circuit shown in Fig. 1.3 is to lift a 5,000 lb_f weight. Suppose the cap-end area is 7.07 in^2, what pressure must the pump build to lift the weight?

$$\begin{aligned} P &= F/A \\ &= 5000/7.07 = 707 \text{ psi} \end{aligned}$$

This solution neglects an important aspect of circuit analysis. (Circuit analysis is basically the determination of flow and pressure at various points around the circuit.) Some pressure drop occurs as fluid flows through a section of hose, fitting, valve, or actuator. These individual pressure drops must be summed to calculate the total pressure required to achieve the functional objective.

In addition to the pressure drops around the circuit, the cylinder friction force must also be considered. When a cylinder extends, the cap-end seals slide along the inside of the cylinder, and the rod-end seals slide along the rod. The resulting friction force for the cylinder shown in Fig. 1.3 was measured and found to be 87 lb_f. This friction force opposes the motion of the cylinder, thus it increases the pressure to lift the load.

It is helpful to work a specific example. Pressure drops shown in Fig. 1.4 are defined as follows:

$\Delta P_{line\ 1}$ = Pressure drop between pump and DCV

ΔP_{DCV} = Pressure drop across DCV

$\Delta P_{line\ 2}$ = Pressure drop between DCV and cylinder cap end

$\Delta P_{line\ 3}$ = Pressure drop between cylinder rod end and DCV

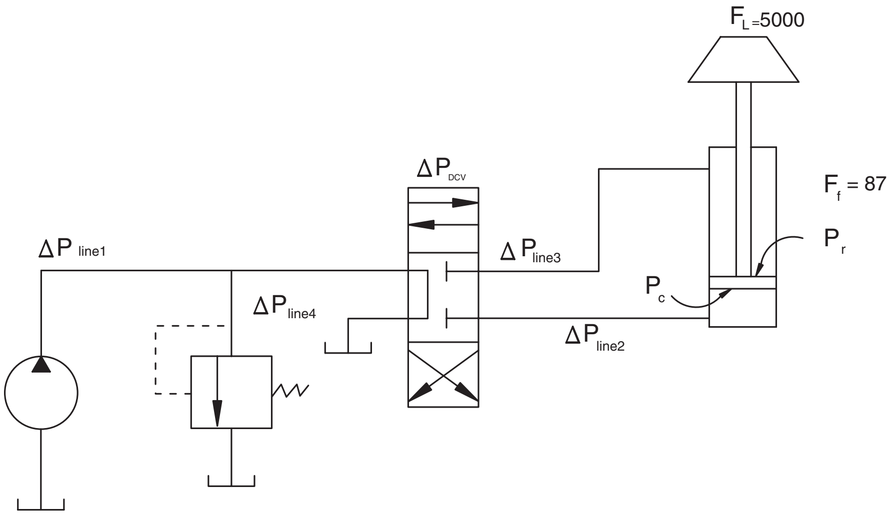
$\Delta P_{line\ 4}$ = Pressure drop between DCV and reservoir

The cylinder shown in Fig. 1.4 has a 3-in. bore and 1.25-in. rod. The cap-end area is then

$$A_c = \frac{\pi(3^2)}{4} = 7.07 \text{ in}^2$$

The rod-end area is

$$A_r = \frac{\pi(3^2)}{4} - \frac{\pi(1.25^2)}{4} = 5.84 \text{ in}^2$$



ΔP_{line1}	45 psi	ΔP_{line3}	20 psi
ΔP_{DCV}	35 psi	ΔP_{line4}	10 psi
ΔP_{line2}	15 psi		

FIGURE 1.4
Pressure drops around a circuit measured during cylinder extension.

During the extension, the total pressure at the rod-end port of the cylinder is

$$\begin{aligned} P_r &= \Delta P_{line3} + \Delta P_{DCV} + \Delta P_{line4} \\ &= 20 + 35 + 10 \\ &= 65 \text{ psi} \end{aligned}$$

Summing forces on the cylinder gives

$$P_c A_c = P_r A_r + F_f + F_L$$

or

$$\begin{aligned} P_c &= (P_r A_r + F_f + F_L)/A_c \\ &= [65(5.84) + 87 + 5000]/7.07 \\ &= 773 \text{ psi} \end{aligned}$$

The total pressure that must be developed at the pump is

$$\begin{aligned} P &= P_c + \Delta P_{line2} + \Delta P_{DCV} + \Delta P_{line1} \\ &= 773 + 15 + 35 + 45 \\ &= 868 \text{ psi} \end{aligned}$$

The actual pressure that the pump must develop is 868 psi, or 23% higher than the 707 psi calculated by ignoring the pressure drops and the cylinder friction force.

1.2.3 Efficiency

We have just learned that the actual pressure required to accomplish the functional objective of the circuit in Fig. 1.3 is 23% higher than the pressure required for the mechanical work done. It is intuitive that some energy has been "lost," meaning that some of the input mechanical energy was not delivered as output mechanical energy.

In Fig. 1.1, mechanical energy at one location is delivered to a second location. If this transfer could be done with a gearbox, typical efficiencies would be

Single reduction gearbox	98–99%
Double reduction gearbox	96–97%
Triple reduction gearbox	95%

Typical efficiency for a hydraulic pump to convert mechanical energy to hydraulic energy is 85%. A typical motor efficiency in converting hydraulic energy back to mechanical efficiency is 85%. Overall efficiency for the circuit in Fig. 1.1, not considering pressure drops, is then

$$0.85 \times 0.85 = 0.72$$

This result means that only 72% of the input mechanical energy is delivered as output mechanical energy.

Efficiency is an issue in almost all circuit designs. The input mechanical energy that is not delivered as output mechanical energy is converted to heat. Oil temperature increases in a hydraulic circuit until the rate of heat loss equals the rate of heat generation.

The operating temperature of a hydraulic circuit should not exceed 140°F. High temperatures reduce the clearance between moving parts and reduce oil viscosity. Both of these factors reduce lubrication; thus, the life of pumps and motors is shortened when operated at high temperatures.

The need for temperature control is mentioned to cause the reader to begin to think about the several factors that interrelate in the design of a fluid power circuit. The interaction of these factors should become clear as we proceed.

1.3 Summary

A design engineer, faced with the task of delivering mechanical energy to accomplish some functional objective, must consider the advantages and disadvantages for the several options available: mechanical, electrical, or fluid (hydraulic or pneumatic). Key advantages of fluid power are:

1. High power density (high power output per unit mass of system)
2. Control (speed of actuators easily controlled)
3. Not damaged when overloaded (relief valve opens to protect system)

The key disadvantage is the inefficiency. *A fluid power option should not be used unless the advantages offset the inefficiency.*

The basic concept of fluid power is simple; mechanical energy is converted to fluid energy, which is then converted back to mechanical energy. In the case of a pump-motor circuit, torque and rpm are converted to pressure and flow by the pump, and the motor converts the pressure and flow back into torque and rpm.

1. Pressure is required to obtain torque from a motor or force from a cylinder.
2. Flow is required to generate rotary motion with a motor or linear motion with a cylinder.

Accounting for the pressure drops around a circuit is a key factor in circuit analysis. Anytime a pressure drop occurs and no mechanical work is delivered, fluid energy is converted to heat energy. Efficiency is increased when these pressure drops are minimized.

The analyses in this chapter were done to illustrate the key concepts and were, by design, simplistic. More in-depth analysis will be done in subsequent chapters. The reader is reminded of the "can't see the forest for the trees" analogy. As we examine the individual "trees" in subsequent chapters, use the basic concept of fluid power as a framework and fit the details into place as you learn them.

Problems

1.1 The pressure drop across a pump is 1500 psi, and the pump output flow is 15 GPM. Assuming the pump is 100% efficient, what input power (hp) is required to drive this pump?

1.2 A simple circuit is shown in Fig. 1.5. During no-load extension, the pressure measurements were

$$P_1 = 150 \text{ psi}$$

$$P_2 = 120 \text{ psi}$$

The cylinder bore is 1.5 in., and the rod diameter is 0.625 in. Find the friction force (lb_f) for this cylinder. The friction force is defined as the force required to overcome the friction due to the piston seals sliding along the inside of the cylinder and the rod seals sliding on the rod.

1.3 Sometimes pump flow rate is unknown, and a flowmeter is not available to insert in the line and measure flow. An unloaded cylinder can be used to get an approximate flow reading. This procedure is illustrated as follows.

The cylinder in Fig. 1.5 has an 8-in. stroke. Total time for extension was 2.4 s. Find the flow rate (GPM) to the cylinder.

1.4 A hydraulic motor has a displacement of 0.915 in^3/rev, and the pressure drop across the motor is 1740 psi.

a. How much torque (lb_f-in) is this motor delivering?

b. If the speed is 820 rpm, what output power (hp) is it delivering?

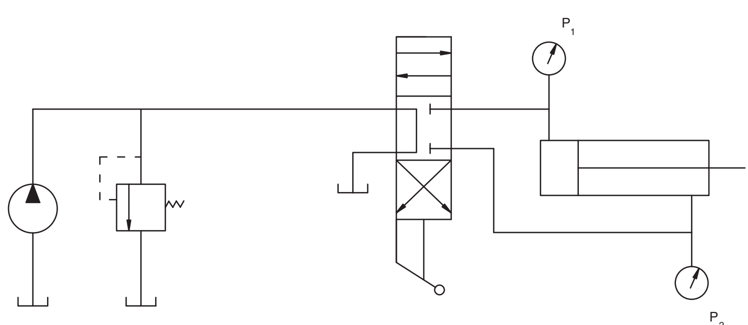


FIGURE 1.5
Circuit for Problem 1.2.

1.5 The circuit shown in Fig. 1.6 has the following pressure drops:

ΔP_{line1} = Pressure drop from pump to DCV
= 35 psi

ΔP_{DCV} = Pressure drop across DCV (We assume this is the same for flow in both directions through the valve.)
= 30 psi

ΔP_{line2} = Pressure drop between DCV and motor
= 5 psi

ΔP_m = Pressure drop across motor

ΔP_{line3} = Pressure drop between motor and DCV
= 10 psi

ΔP_{line4} = Pressure drop between DCV and reservoir
= 15 psi

The relief valve is connected immediately downstream from the pump outlet so that it "sees" the maximum pressure developed in the circuit. The motor has a displacement of 2.3 in^3/rev and is delivering 823 lb_f-in torque.

a. What is the pressure developed at the relief valve?

b. It is good practice to set the relief valve cracking pressure at 500 psi above the maximum pressure needed to achieve the functional objective of the circuit. This setting ensures that flow will not "leak" across the relief valve when the maximum pressure is developed. What cracking pressure should be set for this circuit?

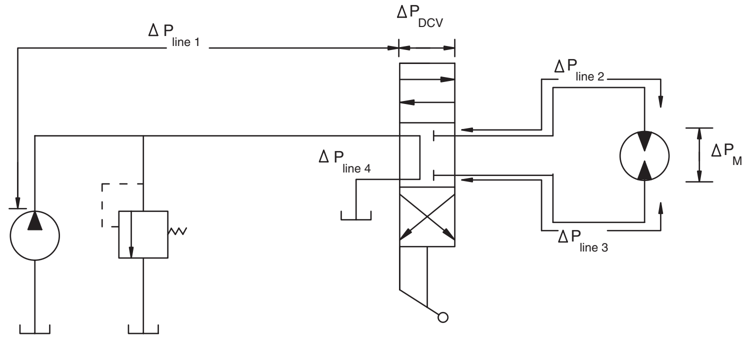


FIGURE 1.6
Circuit for Problem 1.5.

2

Fluid Power Basics

2.1 Introduction

Fluid power systems are designed using all the principles learned in fluid mechanics. It is appropriate to briefly review these principles before proceeding with our study of the applications.

It is required that a student who reads this treatment of fluid power have had an undergraduate course in fluid mechanics. One of the underlying postulates of fluid mechanics is that, for a particular position within a fluid at rest, the pressure is the same in all directions. This follows directly from Pascal's Law. A second postulate states that fluids can support shear forces only when in motion. These two postulates define the characteristics of the fluid media used to transmit power and control motion. Traditional concepts such as static pressure, viscosity, momentum, continuity, Bernoulli's equation, and head loss are used to analyze the problems encountered in fluid power systems. The reader should continuously keep in mind that the fundamental concepts are being applied. New methodology is used, but no new concepts are introduced.

Dimensions and units provide the engineer with a convenient method to track the progress of, and report the results of, analyses. Today, there is a transition occurring in the U.S. from English to metric systems of units. While the scientific community universally embraces the metric system, trade continues to occur in the English system of units in some places. Engineering students must be competent working in both U.S. Customary units and the metric SI (from the *Le Système International d'Unités*), which is also known as the *International System* .

Perhaps the main difficulty encountered by young engineers is handling mass versus force. For example, in the U.S. Customary system, it is customary to weigh objects and report the magnitude of force generated by the object in the Earth's gravitational field. On the other hand, the SI system of measurements relies on determination of mass directly.

Confusion arises when converting mass and force between these two systems. While Newton's law of gravitational mass attraction applies in either case, implementation with respect to the use of constants and the distinction between mass and force differs. For example, in the U.S. Customary system, there is a distinction between lb_m (pounds-mass) and lb_f (pounds-force). However, numerically, these values remain the same. Newton's law of gravitational mass attraction can be written as

$$F = G\frac{m_1 \cdot m_2}{\bar{r}} \tag{2.1}$$

where F is force, m_i is mass of body i, r is the distance between the centroids of the two masses in question, and G is a gravitational constant. The simplified form of this equation is often written as

$$F = m \cdot g \tag{2.2}$$

Most students get in trouble when multiplying by a gravitational acceleration of 32.2 ft/s^2 and treating this equation similar to the way they might in the SI system. In reality, most students derive the conversion between mass and force using Newton's second law.

$$F = m \cdot a \tag{2.3}$$

where a is acceleration. A better way to express Eq. (2.3) is

$$F = \frac{m \cdot a}{g_c} \tag{2.4}$$

where g_c is a conversion factor. This factor takes different values depending on the units used in the equation. For example, when working in the U.S. Customary system of units with lb_m and lb_f, g_c becomes 32.2 lb_m ft/s$^2 \cdot lb_f$. In the SI system, however, g_c is 1.0 kg m/s$^2 \cdot$ N. As one might suspect, the unity value allows one to neglect the g_c constant when doing analyses in the SI system. The same logic applied in the U.S. Customary system of units will yield disastrous results.

2.2 Fluid Statics

2.2.1 Hydrostatic Pressure

To begin, we review the concept of static pressure. Assume that a cylindrical container is filled with fluid to a depth of 1.0 m as shown in Fig. 2.1. The fluid

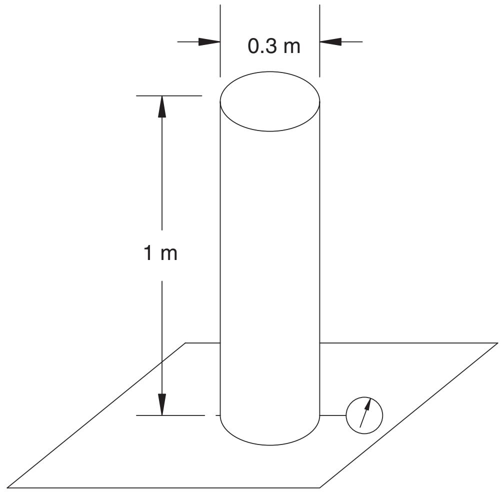


FIGURE 2.1
Static pressure developed by a column of fluid.

column is open to the atmosphere. If we measure the pressure exerted by the fluid at the bottom of the column using a gauge, we might record a pressure of 8.83 kPa (1.4 psi). The reason for this pressure acting at the bottom of the column is the mass of fluid above the pressure port. As the depth of fluid increases, so does the magnitude of pressure, regardless of the container shape.

The product of fluid mass density and the gravitational constant is the constant of proportionality between the depth of fluid in the container and the pressure acting at that depth in the fluid. This relationship can be written as

$$P = \rho \cdot g \cdot h \tag{2.5}$$

where P = pressure
ρ = fluid mass density
h = depth of fluid
g = gravitational constant

Typically, pressure is reported in units of force per unit area, such as Pa and psi. However, it is the practice within some engineering application areas to refer to pressure as *head,* specified in units of length. This length term refers to the pressure that results at the bottom of a column of water of equivalent height. For the most part, engineers in the fluid power industry convert head to pressure.

The density of a fluid is often referred to in terms of *specific gravity.* Specific gravity, by definition, is the ratio of the specific weight of the fluid in question to that of water at standard conditions (standard pressure 760 mm Hg and temperature of 4°C). We can now modify the static pressure equation and introduce specific gravity as

$$P = S_g \rho_w g h \tag{2.6}$$

where S_g = specific gravity
= density of water

Applying this equation to the original example and rearranging the terms, we can describe the S_g of the fluid in the column. Solving for S_g, we have

$$S_g = \frac{P}{\rho_w g h} \tag{2.7}$$

We know that ρ_w is 1000 kg/m^3, and the measured pressure is P = 8.83 kPa. The specific gravity of the fluid in question is

$$S_g = \frac{8830\ \text{Pa} \cdot 1\dfrac{N}{\text{Pa} \cdot \text{m}^2}}{1000\dfrac{\text{kg}}{\text{m}^3} \cdot 9.81\dfrac{\text{m}}{\text{s}^2} \cdot 1\dfrac{N \cdot \text{s}^2}{\text{kg} \cdot \text{m}} \cdot 1.0\ \text{m}} = 0.9$$

Returning to the original fluid column example, there is one additional point to be made. The hydrostatic pressure acting on the bottom of the column acts normal to the surface of the bottom of the column. We can now integrate the hydrostatic pressure over the bottom of the fluid column to determine the total weight of fluid.

$$F = \int P \cdot dA$$

or, in simplified form,

$$F = P \cdot A \tag{2.8}$$

If the column has an internal diameter of 30.0 cm,

$$F = (8830\ \text{Pa}) \cdot \frac{\pi \cdot (0.30\ \text{m})^2}{4} \cdot 1\frac{N}{Pa \cdot m^2} = 624\ N$$

Excluding the container, any support for this fluid column must be capable of withstanding a total force of 624 N (140 lb$_f$).

In most hydraulic systems, static pressure resulting from fluid columns or a depth of fluid in a reservoir is of minor importance. An exception is when the supply to a pump is restricted, and the pump begins to cavitate. Cavitation occurs when the pump does not completely fill with liquid. The incoming fluid is a mixture of gas and liquid. This condition will be discussed further at several points in the text.

An important application of fluid studies is the rather amazing force multiplication that can be achieved with a hydraulic jack (Fig. 2.2). Downward force on the jack handle creates a pressure on the fluid in the small cylinder. Because of Pascal's law, the pressure on the bottom of the large cylinder is the same as the pressure on the bottom of the small cylinder. Force that can be delivered by the large cylinder is set by the ratio of the two cylinder areas. This principle is used many times throughout an industrialized economy.

Example Problem 2.1

The small cylinder of the jack shown in Fig. 2.2 has a bore of 0.25 in., and the large cylinder has a bore of 4 in. How much can be lifted if the jack handle is used to apply 10 lb_f to the small cylinder (F_s = 10 lb_f)?

Pressure developed:

$$P_s = F_s/A_s$$

where A_s = area of small cylinder (in^2)

$A_s = 0.25^2\pi/4 = 0.049\ in^2$

$P_s = 10/0.049 = 204$ psi

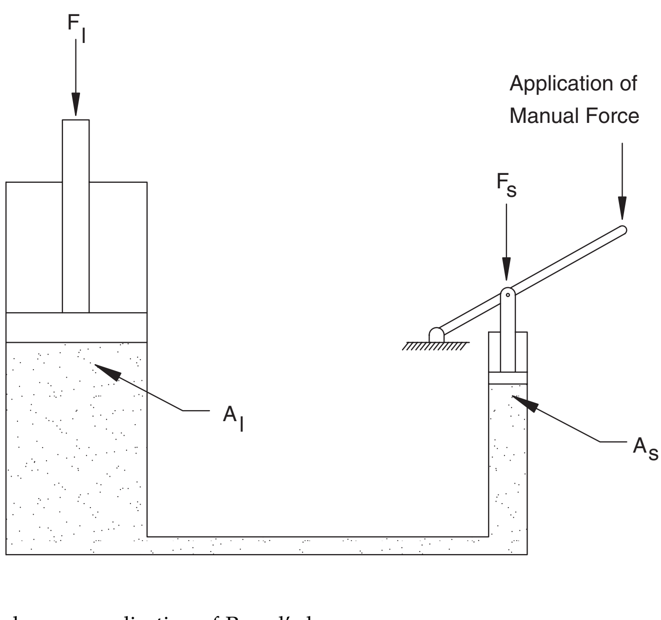


FIGURE 2.2
Hydraulic jack as an application of Pascal's law.

Pascal's law:

$$P_l = P_s$$

where P_l = pressure on large cylinder (psi)

Lift developed:

$$F_l = P_l A_l$$

where A_l = area of large cylinder (in^2)

$A_l = 4^2\pi/4 = 12.56\ in^2$

$F_l = (204)\ (12.56) = 2560\ lb_f$

If the small cylinder bore is 0.125 in., how much can be lifted?

$$A_s = (0.125)^2\ \pi/4 = 0.0123\ in^2$$

$$P_s = 10/0.0123 = 813\ psi$$

$$F_l = (813)\ (12.56) = 10200\ lb_f$$

A 10 lb_f produces a 10,000-lb_f lift.

When referring to pressure, it is important to distinguish between gauge and absolute pressures. To illustrate this difference, let us consider the example shown in Fig. 2.3.

Assume that a closed end tube 2 m in length is filled with mercury and then inverted, and the open end is placed in a reservoir filled with mercury. The reservoir is open to the atmosphere. (It should be noted that mercury poses a health risk to humans, and demonstrations of this nature are ill advised.) A void appears at the top of the tube. We learn that the closed end tube will support a column of mercury only approximately 760 mm in height. Above the supported column of mercury, we have a nearly absolute vacuum. (A total vacuum is almost impossible because of the gas exchange between the mercury and its surroundings.) The question arises as to why or how this column of fluid is supported. This is easily explained if we consider the pressure exerted by the Earth's atmosphere. The layer of air surrounding the Earth is held in place by gravitational attraction, and its weight produces a static pressure on the Earth's surface. It is this pressure that acts on the free surface of the mercury, forcing it up into the column. A height of 760 mm represents standard atmospheric pressure at sea level.

The relationship between gauge and absolute pressure is perhaps best depicted in Fig. 2.4. Gauge pressures are always measured relative to atmo-

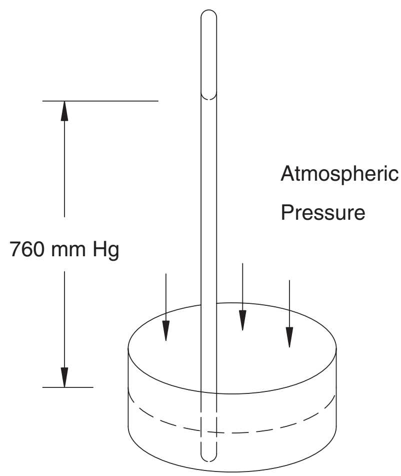


FIGURE 2.3
Barometer used to measure atmospheric pressure.

spheric pressure. With changing weather patterns, atmospheric pressure tends to fluctuate up and down about some mean value. The mean value taken at sea level is termed *standard atmospheric pressure.* This value is a standard for comparison of all units of pressure measurement. If we are merely concerned with pressure differences throughout a system, then ignoring atmospheric pressure has little effect on our analyses. However, if we are to apply the equation of state for an ideal gas, such as in the case analyzing gas-filled accumulators, then it is essential to consider absolute pressure, which is the sum of gauge and local atmospheric pressure.

$$P_{absolute} = P_{atmospheric} + P_{gauge} \tag{2.9}$$

Be careful to keep track of the sign of the gauge pressure. Gauge pressures can be reported with a negative sign or as vacuums. Absolute pressures are always positive.

Returning to Fig. 2.1, we can apply Eq. (2.9) to determine the absolute pressure at the bottom of the fluid column. Assuming a local atmospheric pressure of 770 mm Hg, we have

$$P_{abs} = 770 \text{ mm Hg} \cdot \frac{(101,325) \text{ Pa}}{760 \text{ mm Hg}} + 8830 \text{ Pa} = 111{,}490 \text{ Pa}$$

From this example, it should be clear that any of the equivalencies in Fig. 2.4 can be used to change units so that the two quantities on the right can be added together to coincide with the units desired for the answer on the left.

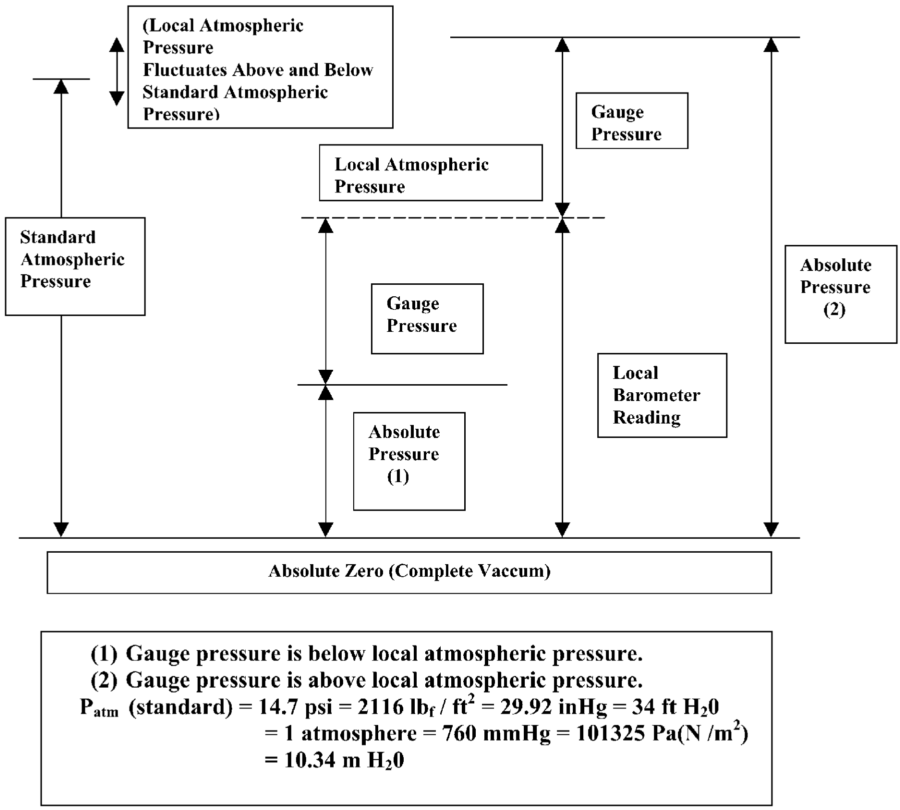


FIGURE 2.4
Datums for measurement of gauge and absolute pressures.

When measuring gauge pressure, one of the most common approaches is to use a Bourdon tube gauge. This gauge, as depicted in Fig. 2.5, operates by moving an indicator needle in proportion to changes in pressure. Needle movement results as the pressure within the curved Bourdon tube changes thereby causing a deflection in the tube. This deflection is amplified via a mechanical linkage attached to the free end of the tube and read on the dial.

2.2.2 Conservation of Mass

Nearly all analytical approaches applied to fluid power systems are based on the concept of *conservation of mass.* To apply this concept, the analyst must first identify a control volume. A control volume is simply a boundary across which the analyst will account for mass transfer. Figure 2.6 illustrates a simple control volume applied to a "T" pipe fitting. The control volume, as denoted by the dashed lines, defines where and how the mass balance will be performed. In this case, the control volume has one inlet and two outlets.

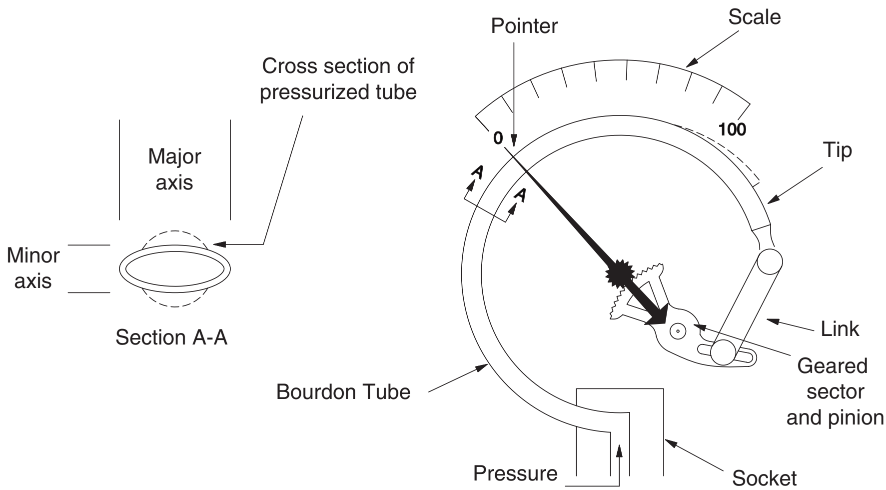


FIGURE 2.5
Bourdon tube pressure gauge.

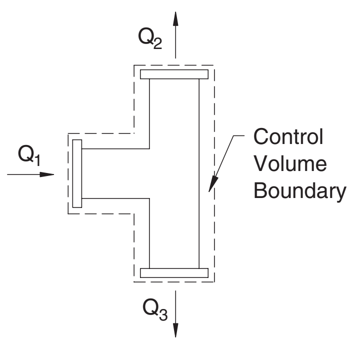


FIGURE 2.6
System with defined control volume.

The following relationship can be written to describe flow into and out of the system (i.e., control volume):

$$\left.\frac{dM}{dt}\right|_{System} = \frac{\partial}{\partial t}\int_{CV} \rho dV + \int_{CV} \rho v \cdot dA \tag{2.10}$$

where M = system mass
t = time
V = system volume
ρ = fluid density
v = velocity normal to incremental area dA
A = area perpendicular to flow streamlines

The notation $\int_{CV}$ refers to a volumetric integral with boundaries that coincide with the boundaries of the control volume, and $\int_{CS}$ refers to integration over the surface of the control volume. A special case is of particular interest in our study of fluid power. If fluid density remains constant (incompressible flow) and the boundaries of the control volume are fixed, meaning that the control volume does not expand or contract, Eq. (2.10) simplifies to

$$0 = \int_{CS} \rho v \cdot dA \tag{2.11}$$

This equation means that the total flow of mass out of the control volume equals the total flow in. For the control volume shown in Fig. 2.6,

$$Q_1 = Q_2 + Q_3$$

The practical application of the conservation of mass principle is seen with a circuit that has a fixed-displacement pump. This pump delivers a given volume of fluid for each rotation. If it is driven at a given rpm, a certain amount of fluid exits the pump. All this fluid goes somewhere; it doesn't just disappear. All of it has to be accounted for in a complete analysis of the circuit.

In a hydraulic circuit, where oil pressures can rise to 6000 psi, the fluid does compress (density is not constant) and hoses do swell (control volume boundaries are not constant). These conditions are considered in Chapter 11. In the intervening chapters, our discussion of systems with a liquid fluid will assume that the fluid is incompressible and the components are rigid. Flow out of a component (valve, actuator, etc.) will always equal the flow in minus any leakage to a drain line.

2.3 Functions of a Working Fluid

Two fluids are of interest: liquid and gas. The liquids are divided into two classes: oil-based and water-based. Petroleum oil is by far the most widely used of the oil-based fluids. Some food plants use nonpetroleum oil such as fish oil for fish processing plants and soybean oil (or other vegetable oil) for plants processing other food products. These oils have similar characteristics to those of petroleum oils. Our discussions of hydraulic systems presume a petroleum oil as the working fluid unless otherwise stated.

Pneumatic systems use air as the working fluid. The air does not have to be purchased, and it can be exhausted back to the atmosphere after use, thus avoiding the expense of a return line. A discussion of pneumatic systems is presented in Chapter 10.

The function of the fluid is to

1. Transmit power
2. Provide lubrication
3. Provide cooling
4. Seal clearances

To provide lubrication and seal clearances between moving parts, the fluid must be able to establish and maintain a continuous film between the parts. High pressures and high relative velocities affect the establishment and maintenance of the film. The design of pumps and motors involves a detailed analysis of the fluid film between the moving parts. Maximum operating speed is set by the ability of the design to maintain a minimum film thickness. Properties of the fluid become more critical as operating pressure and operating speed increase.

Friction is, of course, unavoidable, and the resultant heat raises the temperature of the fluid. The flowing fluid carries this heat to a point where it is exchanged with the atmosphere.

The main heat source is not friction but the conversion of fluid energy to heat energy when there is a pressure drop across a restriction or along a conductor, and no mechanical work is done. The load placed on an actuator is, in effect, a restriction. It produces a pressure drop across the actuator (cylinder or motor), but in this case the fluid energy is converted to mechanical energy. If the flow (Q) from the pump drops across a restriction (maybe across the relief valve) and delivers *no mechanical work*, all the fluid energy represented by this pressure drop (ΔP) is converted to heat energy.

Conversion of fluid energy to heat energy is a key issue in the design of fluid power systems. It is a necessary consequence of the operation of most valves. Often it is the "price" paid for the excellent control of load motion that can be achieved with a fluid power circuit. Yes, the conversion of fluid energy to heat energy does represent a loss of some of the energy put into the system (pump converts input mechanical energy into fluid energy), but, with a good design, this energy is "invested" to achieve the desired motion control.

Conversion of fluid energy to heat energy raises the operating temperature of the fluid. As the difference between operating temperature and ambient temperature increases, the rate of heat exchange from the fluid to the surroundings increases. Eventually, a quasi-equilibrium is achieved.

Elevated temperature changes the properties of the fluid and, because of thermal expansion, changes the clearances between moving parts. Both of these effects influence the establishment and maintenance of the lubricating film. Just as an internal combustion engine must operate in a given temperature range, a fluid power system must operate in a given temperature range. Temperature control is discussed in Chapter 8.

Fluid properties are of critical importance in a fluid power system. It is estimated that 80% of all failures are related to a fluid "failure." A basic understanding of fluid properties and how they change with temperature and, in some cases pressure, is essential for the designer of fluid power systems. The following discussion of these properties will be amplified as needed in subsequent chapters.

2.4 Fluid Properties

2.4.1 Viscosity

A reader who has attempted to pour cold syrup onto pancakes has some concept of viscosity. The cold syrup pours very slowly. It appears "thick" and forms a thick layer on the pancakes. If this syrup is warmed, it flows much more easily.

Viscosity is the fluid's resistance to shear. Cold syrup does not shear readily, thus it flows slowly. An engineering understanding of viscosity can be obtained from Fig. 2.7. The top plate is moving with velocity v relative to the bottom plate, which is stationary. The fluid molecules in contact with the bottom plate are at rest, and those in contact with the top plate are moving at velocity v. In between, a velocity profile is established. If y is the distance between the plates, the slope of the velocity profile is

$$\text{slope} = \frac{\Delta v}{\Delta y} \tag{2.12}$$

Now suppose that the moving plate has an area A and it requires a force F to keep it moving at velocity v. Shear stress in the fluid between the plates is

$$\tau = \frac{F}{A} \tag{2.13}$$

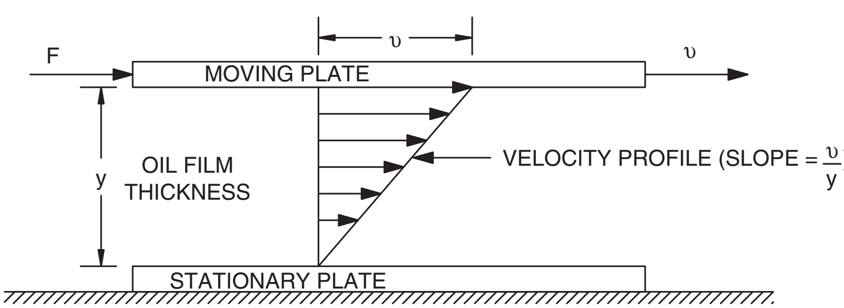


FIGURE 2.7
Fluid velocity profile between stationary and moving plates.

Dynamic viscosity (or absolute viscosity) is defined as the ratio between the shear stress and the slope.

$$\mu = \frac{\tau}{v/y} \tag{2.14}$$

or

$$\mu = \frac{F \times y}{v \times A} \tag{2.15}$$

When the units for F are dyne, y cm, v cm/s, and A cm^2, the units for dynamic viscosity are dyne · s/cm^2. 1 dyne = 1 g · cm/s^2; therefore, the units of dynamic viscosity are g/(s · cm). The name given these units is the *poise*. Dividing by 100, we obtain the common unit for dynamic viscosity, the *centipoise*.

Kinematic viscosity is simply the dynamic viscosity divided by the fluid density measured at the same temperature as the dynamic viscosity measurement.

$$\nu = \frac{\mu}{\rho} \tag{2.16}$$

where μ = dynamic viscosity [g/(s · cm)]
ρ = density (g/cm^3)

The units for kinematic viscosity are cm^2/s, which is a *stoke*. Dividing by 100, we obtain the common unit for kinematic viscosity, the *centistoke* (cS).

A standard, designated ASTM D 2422 and ISO 3448, has been adopted universally. This standard rates fluids based on their viscosity at 40°C. The ISO grades have the letter VG (viscosity grade) followed by a number. For example, most fluid power systems use VG32, VG46, or VG68. The boxes shown in Fig. 2.8 give the kinematic viscosity range (cS) for the ISO grades. Familiar SAE grades are shown for comparison. The SAE grades are based on tests run at 100°C and represent a wider range of kinematic viscosity.

Typically, the specifications provided by the manufacturer will give the kinematic viscosity of an oil at 40 and 100°C. It is appropriate to use these two data points to write the linear equation that gives kinematic viscosity as a function of temperature in the range 40°C < T < 100°C. All well designed fluid power systems operate in this temperature range. Kinematic viscosity at intermediate operating conditions is obtained by referencing the operating temperature to the linear equation.

Early measurements of viscosity were made with a Saybolt viscometer. A sample of oil was maintained in a container surrounded by a water bath. This water bath was used to set the temperature of the oil. An orifice at the bottom

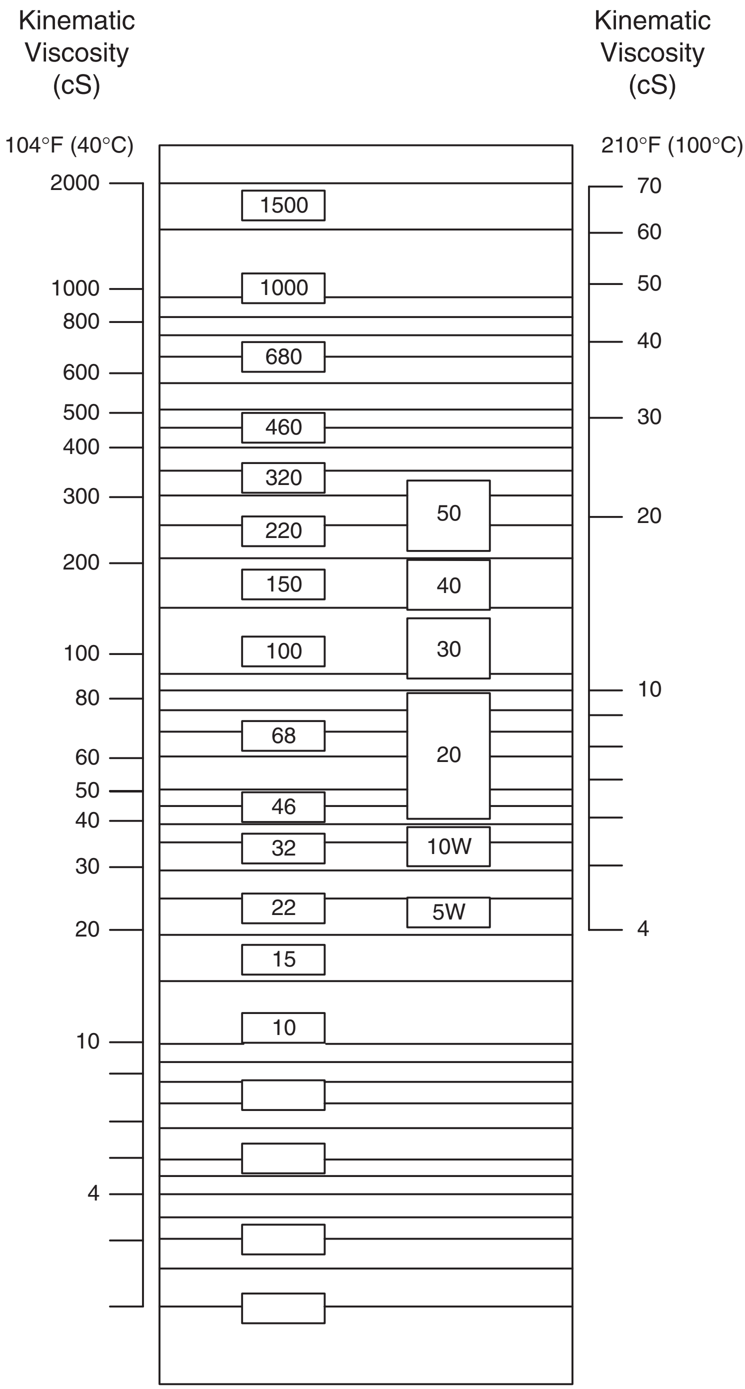


FIGURE 2.8
Comparison of ISO and SAE viscosity grades.

of the container was opened, and the time required for 60 mL of oil to flow through the orifice was measured. This measurement was reported as Saybolt universal seconds (SUS).

Traditional reporting of viscosity in SUS continues. The conversion from SUS to cS is facilitated by ASTM D 2161, which contains both formulae and tables for converting from one to the other. For reasonable estimates of viscosity in centistokes, the following equations can be used:

$$\nu = 0.226t - \frac{195}{t} \qquad 32 < t < 100 \tag{2.17}$$

$$\nu = 0.22t - \frac{135}{t} \qquad t > 100 \tag{2.18}$$

where ν = viscosity (cS)
t = time in SUS

There are many fluid formulations supplied by more than 40 different manufacturers. A few of these fluids are given in Appendix 2.1. In addition to the petroleum-base, phosphate ester, and water glycols shown, a phosphate ester blend and several emulsions are also available.

Viscosities are reported by some manufacturers at 100°F and 212°F. It is appropriate to use these points to develop an equation for a straight line and then use this equation to calculate viscosity at intermediate operating temperatures.

The use of an oil with too low a viscosity can lead to several problems.

1. It can result in a loss of pump (and motor) efficiency due to increased internal leakage. (Clearances are not sealed.)
2. It can cause increased component wear due to breakdown of the lubrication film.
3. At high operating speeds and high operating pressures, the lubrication film can breakdown completely, which will cause the moving parts to "spot weld" together and ultimately cause a complete failure.

An oil with too high a viscosity can cause the following problems:

1. Pump cavitation—the oil is so "thick" that it does not flow readily into the pump. The pump is filled partly with oil and partly with air, a condition known as *cavitation* .
2. High pressure drops occur due to friction in the lines.

It is readily apparent that oil viscosity is a key parameter in the successful operation of a fluid power circuit. Always use an oil that meets the specification of the pump (and motor) manufacturer, and make sure operating temperature is controlled within the range required to keep viscosity within the specified range.

2.4.2 Bulk Modulus

As a first approximation, we often assume liquids to be incompressible. However, there are instances in which the oil compression at the pressures found in hydraulic circuits must be taken into consideration. The degree of oil compressibility is expressed by the *bulk modulus* .

$$\beta = \frac{-\Delta P}{(\Delta V/V)} \tag{2.19}$$

where β = bulk modulus (psi)
ΔP = change in pressure (psi)
ΔV = change in volume when ΔP is applied (in^3)
V = original volume (in^3)

An example problem will illustrate a typical bulk modulus measurement.

A pressure of 2500 psi is applied to a 10 in^3 sample of oil. The measured change in volume is 0.1 in^3. What is the bulk modulus?

$$\beta = \frac{-2500}{(9.9 - 10)/10}$$

$$= 250,000 \text{ psi}$$

Typical values for hydraulic oils are 2×10^5 to 2.5×10^5 psi.

All fluids contain air in the spaces between fluid molecules. (An air molecule is smaller than a fluid molecule, so it fits in the spaces between the irregularly shaped fluid molecules.) When pressure is applied, the fluid molecules deform and compress the air. No fluid is truly incompressible. If air is entrained in the fluid, meaning that there are air bubbles surrounded by fluid, the bulk modulus is significantly reduced.

Temperature causes fluid to increase in volume, thus it has an influence on bulk modulus. The influence of temperature in the typical operating temperature range of hydraulic systems is much less than the influence of pressure, so the bulk modulus is typically not corrected for temperature.

Compressibility is the reciprocal of bulk modulus. The compressibility of the oil in the previous example is

$$E = \frac{1}{250,000} = 4 \times 10^{-6} \frac{\text{in}^2}{\text{lb}_f}$$

Typical values are 4×10^{-6} to 5×10^{-6} in^2/lb$_f$.

2.4.3 Specific Gravity

As previously mentioned, specific gravity is the ratio of the density of the fluid to the density of water at 4°C and standard atmospheric pressure and is generally denoted as

$$S_g = \frac{\rho}{\rho_w} \tag{2.20}$$

where ρ = density of fluid (slug/ft^3)
ρ_w = density of water (slug/ft^3)

Specific weight is defined as the weight per unit volume and is given by

$$\gamma = \frac{mg}{V} \tag{2.21}$$

where m = mass
V = volume
g = gravitational constant

Since $\rho = m/V$,

$$\gamma = \rho g \tag{2.22}$$

and specific gravity can be written

$$S_g = \frac{\rho g}{\rho_w g} = \frac{\gamma}{\gamma_w} \tag{2.23}$$

where γ_w = specific weight of water

Most petroleum-based fluids have a specific gravity in the range 0.85 to 0.9. Water-based fluids have a specific gravity close to 1. The main disadvantage of a higher S_g is the higher pressure drop as the fluid flows through the lines.

2.4.4 Other Fluid Properties

The chemical properties of the fluid are as important as the physical properties. Discussion in this subsection will focus on petroleum oils. Petroleum oils are hydrocarbons, meaning that the molecule has hydrogen and carbon atoms.

2.4.4.1 Oxidation

Oxidation is the reaction between the oil and oxygen. The compounds formed are referred to as *resins* and *sludges*. The rate of formation is a function of the amount of oxygen present (water-contaminated oil forms more sludges) and the temperature. Higher temperatures increase the reaction rate. As a general rule, the rate of oxidation can be expected to double with every 18°F (10°C) in fluid temperature rise above 140°F (60°C). Doubling the oxidation rate cuts the useful life of the oil in half, which increases the cost for oil replacement and the cost for used oil disposal.

The resins and sludges formed by oxidation can degrade the performance of components by plugging orifices or causing moving parts to stick, particularly the spools in spool-type valves. Also, the compounds formed tend to be acidic, and they can etch and/or corrode metal surfaces in the system.

Several steps can be taken to reduce oxidation.

1. Keep sources of oxygen (air and water) out of the system.
2. Remove particulates with a good filtration system. These particulates can act as sites for the oxidation reaction to occur.
3. Avoid the use of cadmium, zinc, and copper in contact with hydraulic oil. For example, never use galvanized pipe or fittings. These metals can act as catalysts that promote the oxidation reaction.

There is a phenomenon that can occur in a fluid power system that is not intuitive but can cause significant change in the oil chemical properties. This phenomenon is the adiabatic compression of small air bubbles in high-pressure pumps. The resulting high temperatures break the chemical bonds in the oil molecules, a process known as *cracking* in the oil refining industry. Resulting sludges and gums clog fine filters and result in the failure of precision systems using servo and proportional valves.

Yeaple (1996) gives the following estimate of localized temperature at the point where a small air bubble implodes. The calculation assumes adiabatic compression of the air from 14.7 psi and 100°F to a pressure equal to the pressure developed at the pump outlet.

Pump pressure (psi)	**Bubble temperature (°F)**
1000	1410
2000	1820
3000	2100

No one intuitively expects temperatures in this range in a hydraulic system. Given the speed at which a bubble is compressed, there is negligible time for heat transfer, and it is reasonable to then assume that the compression is adiabatic. These high temperatures can exist momentarily at point locations and thus can affect chemical change.

2.4.4.2 Corrosion and Rust Resistance

Corrosion is defined as a chemical reaction between the fluid and a metal surface. Rust is the oxidation of a ferrous metal. With corrosion, part of the metal is lost from the surface, and resultant surface pits or voids are filled with a dark-colored substance—the oxidation products that caused the corrosion. Since a portion of the metal is removed, the part becomes weaker. Also, the removed metal contaminates the oil. Rusting typically takes place in the reservoir above the oil level.

Problems with corrosion and rusting can be minimized by the following common-sense precautions.

1. Limit, to the maximum degree possible, the introduction of air, water, and chemicals into the hydraulic fluid.
2. Select a fluid with good oxidation and rust inhibiting additives. (These fluids are sometimes referred to as R and O oils.)
3. Provide filtration (i.e., a well designed and maintained system) to remove by-products of rusting and corrosion.

2.4.4.3 Fire Resistance

Fire resistance is a key consideration in hydraulic systems used on aircraft, marine and mining equipment, and some manufacturing equipment. Water-based fluids are less flammable but still should not be considered to be inflammable. Even a 95% water/5% glycol fluid will burn if the conditions are right to boil off water until the glycol concentration increases to the point at which combustion is sustained.

Three parameters are important in discussing the fire resistance of a fluid:

- Flash point
- Fire point
- Autogenous ignition temperature (AIT)

The *flash point* is the lowest temperature at which vapors rising form the oil surface will ignite in the presence of an open flame. The rate of vapor release is insufficient to sustain the flame, but it will flash, thus the name flash point. There are several tests to determine flash point, one of which is the Cleveland open cup (ASTM D 92).

When oil temperature is increased to the point at which the exiting vapors will support combustion for five seconds in the presence of an open flame, the temperature is defined as the *fire point.* Generally, the fire point is about 50°F higher than the flash point.

The temperature at which a fluid will "burst into flame" in the absence of an ignition source is called the *autogenous ignition temperature (AIT).* The AIT is usually much higher than the fire point for most oils. It is determined by the test method described in ASTM E 659.

2.4.4.4 Foam Resistance

Hydraulic oil can be formulated with antifoaming additives. These are needed when there is an opportunity for air to be entrained in the oil. A properly designed system has the reservoir sized and placed so that the entire system is always completely filled with oil.

2.5 Flow in Lines

When a fluid flows through a conductor (pipe, tube, or hose), the layer of fluid particles next to the wall has zero velocity. As distance from the wall increases, the velocity of the fluid increases and is at a maximum in the center of the conductor (Fig. 2.9). The profile develops because of viscosity. The more viscous the fluid, the greater the change in velocity with distance from the wall.

Flow is said to be *laminar* if the layers of fluid particles remain parallel as the flow moves along the conductor. It is said to be *turbulent* if the fluid layers break down as the flow moves along the conductor (Fig. 2.10). Increased movement of the fluid particles relative to each other causes an increase in the conversion of fluid energy to heat energy. This point will be discussed in more detail later.

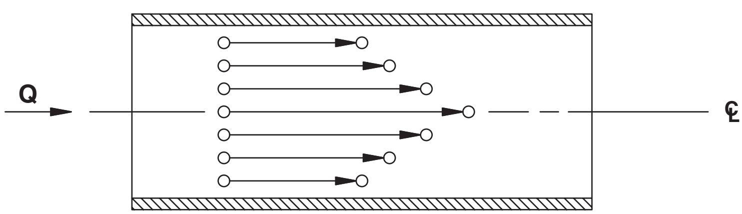


FIGURE 2.9
Velocity profile of laminar flow in a conductor.

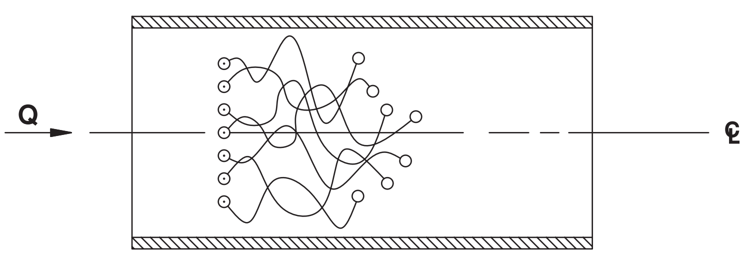


FIGURE 2.10
Flow lines when flow is turbulent.

2.5.1 Reynolds Number

The key issues in liquid flow in a fluid power circuit are forces due to fluid inertia and forces due to viscosity. In general, flow dominated by viscosity forces is laminar, and inertia dominated flow is turbulent.

Osborn Reynolds performed a series of experiments in 1833 to define the transition between laminar and turbulent flow. He found that laminar flow is a function of a dimensionless parameter now known as the Reynolds number. Reynolds number is defined by

$$N_R = \frac{vD\rho}{\mu} \tag{2.24}$$

where v = fluid velocity
D = conductor inside diameter
ρ = fluid mass density
μ = dynamic viscosity

The Reynolds number is a dimensionless ratio of inertia force to viscous force. It is convenient in many cases to use the following formula for Reynolds number:

$$N_R = \frac{7740vD}{\nu} \tag{2.25}$$

where ν = fluid velocity (ft/s)
D = conductor inside diameter (in)
ν = kinematic viscosity (cS)

Reynolds discovered the following rules with his tests:

1. If $N_R < 2000$, flow is laminar.
2. If $N_R > 4000$, flow is turbulent.
3. The region $2000 < N_R < 4000$ is defined as the transition region between laminar and turbulent flow.

Example Problem 2.2

Suppose oil with kinematic viscosity $\nu = 36.5$ cS is flowing through a tube with inside diameter $D = 0.5$ in. The flow rate is $Q = 8$ GPM. Is the flow laminar or turbulent?

First, we must determine the average fluid velocity. This is typically done by dividing volumetric flow rate by the cross-sectional area of the conductor.

$$v = \frac{Q}{A}$$

$$= \frac{8 \text{ GPM} \times 231 \text{ in}^3/\text{gal}}{\pi(0.5)^2/4 \text{ in}^2}$$

$$= 9430 \text{ in/min}$$

$$= 13.1 \text{ ft/s}$$

Substitution into Eq. (2.25) gives

$$N_R = \frac{7740(13.1)(0.5)}{36.5}$$
$$= 1390$$

$N_R < 2000$, therefore flow is laminar.

What flow of this oil can be pumped through a 0.5 in diameter tube and the flow will still be laminar? We solve Eq. (2.25) for v.

$$v = \frac{N_R \nu}{7740 D} \tag{2.26}$$

Substituting $N_R = 2000$,

$$v = \frac{2000(36.5)}{7740(0.5)}$$

$$= 18.86 \text{ ft/s}$$

$$= 13,580 \text{ in/min}$$

$$Q = vA/231$$

$$= \frac{13,580 \text{ in/min} \times 0.195 \text{ in}^2}{231 \text{ in}^3/\text{gal}} = 11.5 \text{ GPM}$$

As a general rule, the oil velocity in the lines should be less than 10 ft/s. N_R values for common inside diameters are presented below for v = 10 ft/s and kinematic viscosity, ν = 36.5 cS.

D(in.)	N_R
0.5	1060
0.75	1590
1.00	2120
1.25	2650
1.50	3180

Note that flow is laminar for the smaller diameters (D = 0.5, 0.75), but it is transitional for $D > 1.00$ in.

2.5.2 Darcy's Equation

Friction is the main cause of the loss of fluid energy as the fluid flows through a conductor. Because of friction, some fluid energy is converted to heat energy and exchanged into the surrounding atmosphere.

Fluid power at the inlet to a conductor is

$$\mathcal{P}_{hyd1} = P_1 Q_1$$

where P_1 = pressure at the inlet
Q_1 = flow at the inlet

Fluid power at the outlet is

$$\mathcal{P}_{hyd2} = P_2 Q_2$$

There is no leakage from the line, thus $Q_1 = Q_2$. For H_{hyd2} to be less than H_{hyd1}, meaning that there has been a loss of fluid power, then P_2 must be less than P_1. In other words, friction causes a pressure drop in the line. This pressure drop is sometimes referred to as a *head loss* . Head loss in a conductor is given in handbooks and is often reported as psi/ft of conductor length.

Loss along a conductor can be calculated directly using Darcy's equation.

$$h_L = f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right) \tag{2.27}$$

where h_L = head loss (ft)
f = friction factor (dimensionless)
D = conductor inside diameter (ft)
L = conductor length (ft)
v = average fluid velocity (ft/s)
g = gravitational constant (ft/s^2)

(The units for h_L are actually ft · lb_f/lb_f, but the unit is generally just reported as ft.)

The friction factor for laminar flow is given by

$$f = \frac{64}{N_R} \tag{2.28}$$

Substitution into Darcy's equation gives the Hagen-Poiseuille equation.

$$h_L = \frac{64}{N_R}\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right) \tag{2.29}$$

Remember, this equation is valid for laminar flow only.

The Hagen-Poiseuille equation is also written in the form

$$\Delta P = \frac{128\mu LQ}{\pi D^4} \tag{2.30}$$

where ΔP = pressure drop
μ = absolute viscosity
L = length
Q = volume flow rate
D = diameter

Note that the terms *dynamic* viscosity and *absolute* viscosity are used interchangeably. It is instructive to show that Eqs. (2.29) and (2.30) are equivalent.

We first use the definition for kinematic viscosity to observe that

$$\mu = \rho\nu$$

Substituting into Eq. (2.30), we obtain

$$\Delta P = \frac{128(\rho\nu)LQ}{\pi D^4} \tag{2.31}$$

Volumetric flow is given by

$$Q = Av$$

and, for a pipe with inside diameter D,

$$Q = \frac{\pi D^2 v}{4} \tag{2.32}$$

Substituting into Eq. (2.31),

$$\Delta P = \frac{64}{2}\frac{\nu\rho Lv}{D^2} \tag{2.33}$$

Reynolds number can be written in terms of kinematic viscosity.

$$N_R = \frac{vD}{\nu} \tag{2.34}$$

Solving Eq. (2.34) for ν and substituting into Eq. (2.33), we obtain

$$\Delta P = \frac{64}{2}\left[\frac{vD}{N_R}\right]\rho\frac{L}{D^2}v = \frac{64}{N_R}\left(\frac{L}{D}\right)\left(\frac{\rho v^2}{2}\right) \tag{2.35}$$

From Eq. (2.22), $\rho = \gamma/g$. Substituting into Eq. (2.35) gives

$$\Delta P = \frac{64}{N_R}\left(\frac{L}{D}\right)\left(\frac{\gamma \nu^2}{2g}\right) \tag{2.36}$$

It remains to show that $\Delta P/\gamma = h_L$, and we will have completed the proof. The units are consistent

$$\frac{\Delta P}{\gamma} = \frac{\text{lb}_f/\text{ft}^2}{\text{lb}_f/\text{ft}^3} = \text{ft} = h_L$$

Suppose we have a column of fluid with cross-sectional area A and height h. The pressure exerted by the column of fluid is simply the total weight divided by the cross-sectional area.

$$P = \frac{\gamma A h}{A} = \gamma h \tag{2.37}$$

or

$$\frac{\Delta P}{\gamma} = h$$

Use of this relation completes the proof that Eqs. (2.29) and (2.30) are equivalent.

A conversion of head loss (ft) to pressure drop (psi) is often needed. Using the definition of S_g [Eq. (2.23)],

$$\gamma = \gamma_w S_g \tag{2.38}$$

$\gamma_w = 62.4$ lb_f/ft^3. Substituting into Eq. (2.37) and dividing both sides of the equation by 144 to convert lb_f/ft^2 to lb_f/in^2,

$$\Delta P = \frac{62.4}{144} S_g h_L$$
$$= 0.433 S_g h_L \quad (2.39)$$

where S_g = specific gravity (decimal)

When flow is turbulent, the friction factor is a function of Reynolds number and the relative roughness of the conductor. Relative roughness is defined as the conductor inside surface roughness ε divided by the conductor inside diameter D.

$$\text{Relative roughness} = \frac{\varepsilon}{D} \quad (2.40)$$

The physical meaning of inside surface roughness is shown in Fig. 2.11.

The inside surface of hydraulic hose and drawn tubing is relatively smooth. A typical value given by Esposito (1988) for drawn tubing is ε = 0.000005 ft. Generally, it is acceptable to use the Moody diagram curve for smooth pipe (Fig. 2.12). To determine the friction factor in the turbulent zone, locate the Reynolds number on the horizontal axis, move vertically to the curve, and read the friction factor on the left vertical axis.

The friction factor in the turbulent flow range can also be calculated using

$$f = \frac{0.1364}{N_R^{0.25}} \quad (2.41)$$

for smooth conductors and a Reynolds number less than 100,000. Equation (2.41) was experimentally determined and is known as the Blasius equation. In most hydraulic systems, the flow velocity is kept less than 15 ft/s; thus, the Reynolds number is less than 100,000, and Eq. (2.41) is applicable.

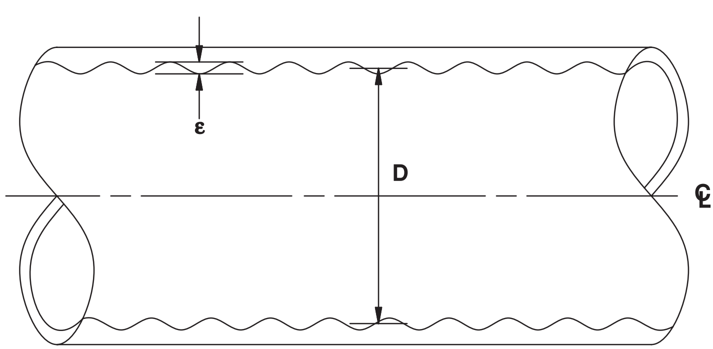


FIGURE 2.11
Physical meaning of inside surface roughness ε.

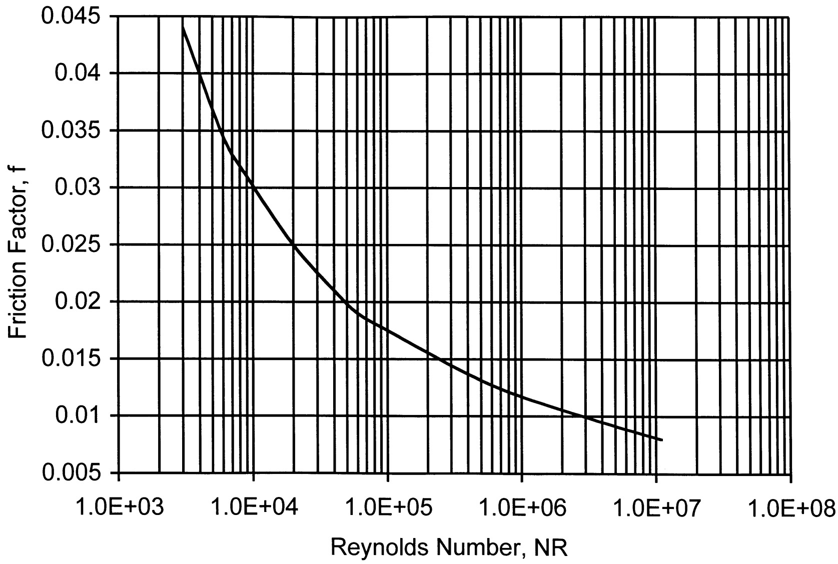


FIGURE 2.12
Reproduction of Moody diagram for smooth conductor.

Example Problem 2.3

A hydraulic motor powers a conveyor on a harvesting machine. Oil flows in a 16-ft hydraulic hose from the pump to the motor. The oil properties are S_g = 0.9 and ν = 36.5 cS. Flow rate is Q = 15 GPM, and the hose inside diameter is 0.75 in. What is the pressure drop between the pump and motor? Velocity of the fluid is

$$
\begin{aligned}
v &= \frac{Q}{A} \\
&= \frac{15\text{GPM} \times 231\ \text{in}^3/\text{gal}}{\pi(0.75)^2/4\ \text{in}^2} \\
&= 7840\ \text{in/min} = 10.9\ \text{ft/s}
\end{aligned}
$$

Reynolds number is

$$
\begin{aligned}
N_R &= \frac{7740 vD}{\nu} \\
&= \frac{7740(10.9)(0.75)}{36.5} \\
&= 1730
\end{aligned}
$$

Since $N_R < 2000$, flow is laminar, and the friction factor is

$$f = \frac{64}{N_R} = \frac{64}{1730} = 0.037$$

Using Darcy's equation,

$$\begin{aligned} h_L &= f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right) \\ &= 0.037\left(\frac{16}{0.75/12}\right)\left[\frac{10.9^2}{2(32.2)}\right] \\ &= 17.5 \text{ ft} \end{aligned}$$

Using Eq. (2.39),

$$\begin{aligned} \Delta P &= 0.433 h_L S_g \\ &= 0.433(17.5)(0.9) = 6.8 \text{ psi} \end{aligned}$$

There is a requirement to increase the speed of the motor 2.5 times. The decision is made to increase flow to $2.5 \times 15 = 37.5$ GPM without increasing hose size. What is the pressure drop?

$$\begin{aligned} v &= \frac{Q}{A} \\ &= \frac{37.5 \text{ GPM} \times 231 \text{ in}^3/\text{gal}}{\pi(0.75)^2/4 \text{ in}^2} \\ &= 19600 \text{ in/min} = 27.2 \text{ ft/s} \end{aligned}$$

$$\begin{aligned} N_R &= \frac{7740 vD}{v} \\ &= \frac{7740(27.2)(0.75)}{36.5} \\ &= 4330 \end{aligned}$$

Referencing Fig. 2.12 for $N_R = 4500$, $f = 0.038$. Using Eq. (2.41), $f = 0.039$.

$$
\begin{aligned}
h_L &= f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right) \\
&= 0.038\left(\frac{16}{0.75/12}\right)\left[\frac{27.2^2}{2(32.2)}\right] \\
&= 111.8 \text{ ft} \\
\Delta P &= 0.433 h_L S_g \\
&= 0.433(111.8)(0.90) = 43.5 \text{ psi}
\end{aligned}
$$

We can expect the pressure drop in the 16-ft hose to increase from 6.8 to 43.5 psi as flow is increased from 15 to 37.5 GPM.

In many applications, particularly on mobile machines, the conductor lengths are relatively short, and the pressure drop along the conductor is not a significant factor. Pressure drops through other components (valves, fittings, etc.) are more significant than the pressure drops in the conductor itself.

2.5.3 Losses in Fittings

In fluid mechanics, an engineer is introduced to the concept of a *K* factor for a fitting. Tests have shown that head losses in fittings are proportional to the square of the velocity of the fluid.

$$h_L = \frac{Kv^2}{2g} \tag{2.42}$$

where h_L = head loss (ft)
v = fluid velocity (ft/s)
g = gravitational constant (ft/s^2)

Typical *K* factors for common fittings are given by Esposito (1988).

Fitting	*K* Factors
Standard tee	1.8
Standard elbow	0.9
45° elbow	0.42
Return bend (U-turn)	2.2

The influence of fluid velocity on losses through fittings is illustrated with an example problem.

Example Problem 2.4

The circuit described in Example Problem 2.3 has an elbow ($K = 0.9$) at the motor. Fluid flows from the hose through the elbow into the motor. What is the pressure drop in this fitting for Q = 15 GPM, Q = 37.5 GPM.

Q = 15 GPM

$$v = 10.9 \text{ ft/s}$$

$$h_L = \frac{Kv^2}{2g}$$

$$= \frac{0.9(10.9)^2}{2(32.2)} = 1.66 \text{ ft}$$

$$\Delta P = 0.433 h_L S_g$$

$$= 0.433(1.66)(0.9) = 0.6 \text{ psi}$$

Q = 37.5 GPM

$$v = 27.2 \text{ ft/s}$$

$$h_L = \frac{Kv^2}{2g}$$

$$= \frac{0.9(27.2)^2}{2(32.2)} = 10.3 \text{ ft}$$

$$\Delta P = 0.433 h_L S_g$$

$$= 0.433(10.3)(0.9) = 4 \text{ psi}$$

Flow increased 2.5 times from 15 to 37.5 GPM, and the pressure drop across the elbow increased 6.7 times from 0.6 to 4 psi. It is now clear why recommendations on maximum fluid velocity are given. These recommendations are a trade-off between component initial cost (higher for larger components) and operating cost (lower for larger components because of lower losses). More detailed discussion on component sizing is given in Chapter 9. Intervening chapters do not consider pressure drops for individual conductors and fittings, only total ΔPs for sections of the circuit are given. It is important for the reader to remember, however, that all the principles learned in fluid mechanics are applicable even if all detail is not included with each example problem.

2.6 Leakage Flow

Spool-type valves have a cylinder with grooves machined at intervals along the length. This cylinder, known as the *spool,* slides back and forth in a bore to open and close passageways through the valve. Obviously, there has to be clearance between the spool and the bore. This section presents a brief discussion of the leakage through the annulus between the spool and bore. Other similar leakage pathways, for example, leakage between the piston and cylinder wall of a piston pump, are analyzed in a similar manner.

If the annulus is "unwrapped" from around the spool, it will have a shape as shown in Fig. 2.13. The distance a is the clearance between the spool and bore, and w is the circumference of the bore. The length L is the distance between adjacent grooves machined in the spool. These sections between grooves are called *lands*.

Dryden et al. (1956) present the following expression for leakage flow as a function of pressure drop across the land (along the length L):

$$Q = \frac{wa^3}{12\,\mu L}\Delta P \tag{2.43}$$

where Q = leakage flow
w = width of rectangular opening
a = height of rectangular opening
μ = absolute viscosity
L = length of leakage pathway
ΔP = pressure difference across land between adjacent grooves

The width is $w = \pi D$ and the absolute viscosity is $\mu = \rho\nu$. Substituting into Eq. (2.43),

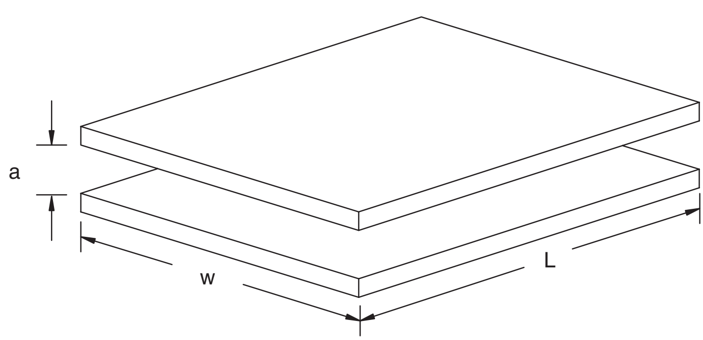


FIGURE 2.13
Geometry of leakage flow between spool and bore in spool-type valve. Annulus has been idealized as a thin rectangle.

$$Q = \frac{\pi D a^3 \Delta P}{12\rho\nu L} \tag{2.44}$$

The following example problem will illustrate the use of Eq. (2.44).

Example Problem 2.5*

The land in a spool valve separates two fluid passages. The land has a 1.0 in. length and 0.7493 ± 0.0002 in. diameter and operates in a 0.7500 ± 0.0004 in. bore. We assume that the spool is concentric in the bore. Pressure difference across the land is 3000 psi. Calculate the leakage flow rate past this land for minimum, nominal, and maximum leakage conditions assuming a fluid with minimum, nominal, and maximum viscosities of 45, 150, and 4000 SUS, respectively. The fluid has a specific gravity of 0.9.

Viscosity in Saybolt universal seconds (SUS) is converted to centistokes using Eqs. (2.17) and (2.18). The units for centistokes are mm^2/s.

t = 45 SUS

$$\begin{aligned}\nu &= 0.226(45) - \frac{195}{45} = 5.84 \text{ mm}^2/\text{s} \\ &= 6.28 \times 10^{-5} \text{ ft}^2/\text{s}\end{aligned}$$

t = 150 SUS

$$\begin{aligned}\nu &= 0.220(150) - \frac{135}{150} = 32.1 \text{ mm}^2/\text{s} \\ &= 3.46 \times 10^{-4} \text{ ft}^2/\text{s}\end{aligned}$$

t = 4000 SUS

$$\begin{aligned}\nu &= 0.220(4000) - \frac{135}{4000} = 880 \text{ mm}^2/\text{s} \\ &= 9.47 \times 10^{-3} \text{ ft}^2/\text{s}\end{aligned}$$

Minimum height of the passage is achieved for the maximum diameter of the spool and the minimum diameter of the bore.

$$\begin{aligned}a &= \frac{(0.7500 - 0.0004) - (0.7493 + 0.0002)}{2} \\ &= 0.00005 \text{ in.}\end{aligned}$$

* Appreciation is expressed to Dr. David Pacey, Kansas State University, Manhattan, Kansas, for his contribution of Example Problem 2.5.

Nominal height of the passage is achieved when both spool and bore have their nominal dimensions.

$$a = \frac{0.7500 - 0.7493}{2}$$
$$= 0.00035 \text{ in.}$$

Maximum height of the passage is achieved when the spool has a minimum diameter and the bore has a maximum diameter.

$$a = \frac{(0.7500 + 0.0004) - (0.7493 - 0.0002)}{2}$$
$$= 0.00065 \text{ in.}$$

Density of water is 1.94 slug/ft^3, thus the density of this fluid is

$$\rho = 1.94S_g$$
$$= 1.94(0.9) = 1.746 \text{ slug/ft}^3$$

Given in the problem,

$$L = 1 \text{ in}$$

and we use $D = 0.75$ in. All variables are now defined for substitution into Eq. (2.44). We have mixed units, so the appropriate conversion factor(s) will have to be identified.

Minimum leakage is achieved for the highest viscosity fluid and the minimum height of the passage.

$$Q_{min} = \frac{\pi(0.75 \text{ in})(0.00005^3 \text{ in}^3)(3000 \text{ lb}_f/\text{in}^2)}{12(1.746 \text{ slug/ft}^3)(9.47 \times 10^{-3} \text{ft}^2/\text{s})(1.0 \text{ in})}$$

$$= \frac{0.88357 \times 10^{-9} \text{in}^2 \cdot \text{lb}_f}{198.4 \times 10^{-3} \text{ in} \cdot \text{slug/ft} \cdot \text{s}}$$

$$1 \text{ lb}_f = 1 \text{ slug} \cdot \text{ft/s}^2$$

Substituting and simplifying the units,

$$Q_{min} = 4.453 \times 10^{-9} \text{ in} \cdot \text{ft}^2/\text{s}$$

We must multiply by 144 in^2/ft^2 to obtain consistent units, thus,

$$Q_{min} = 6.41 \times 10^{-7} \text{ in}^3/\text{s}$$

Q_{nom} is calculated for $a = 0.00035$ in and $\nu = 3.46 \times 10^{-4}$ ft^2/s.

$$Q_{nom} = 6.02 \times 10^{-3} \text{ in}^3/\text{s}$$

Q_{max} is calculated for $a = 0.00065$ in and $\nu = 6.28 \times 10^{-5}$ ft^2/s.

$$Q_{max} = 0.212 \text{ in}^3/\text{s}$$

For this range of passage dimensions and fluid viscosity, Q_{max} is 35 times higher than Q_{nom}.

2.7 Orifice Equation

The orifice equation states that flow through an orifice is proportional to the square root of the pressure drop across the orifice (ΔP = upstream pressure – downstream pressure).

$$Q = CA\sqrt{(2g\Delta P)/\gamma} \tag{2.45}$$

where Q = flow (in^3/s)
C = orifice coefficient (decimal)
A = area (in^2)
g = gravitational constant (in/s^2) ($g = 386$ in/s^2)
ΔP = pressure (psi)
γ = specific weight of fluid (lb_f/in^3)

This equation simplifies to

$$Q = k\sqrt{\Delta P} \tag{2.46}$$

where $k = CA\sqrt{2g/\gamma}$

Solving for ΔP, we obtain the orifice equation with Q as the independent variable, a form that is more useful for our analysis of fluid power circuits.

$$\Delta P = Q^2/k^2 = k_{eq}Q^2 \tag{2.47}$$

where $k_{eq} = 1/k^2$. This relationship shows there is a quadratic relationship between ΔP and Q. If Q is doubled, then the pressure drop will increase fourfold. All valves form some type of orifice in the line and thus have a pressure vs. flow curve similar to that shown in Fig. 2.14. Knowing the flow through the valve, the designer can estimate the pressure drop. The technical data sheets (tech sheets) supplied by almost all valve manufacturers have a curve similar to Fig. 2.14. Sometimes tabular data is supplied, but more often it is a curve.

There is a classic problem that nicely illustrates the influence of an orifice in a circuit. Fluid from an accumulator flows into a cylinder (Fig. 2.15). How fast does the cylinder extend? Before solving this problem, it is first necessary to understand how an accumulator works.

Accumulators are devices used to store fluid under pressure. There are three main types: diaphragm, bladder, and piston (Fig. 2.16). All three types work on the same principle—an inert gas is compressed as fluid enters into the accumulator. The governing equation is Boyle's law,

$$P_1 V_1 = P_2 V_2 \tag{2.48}$$

The accumulator is precharged to some pressure P_1. At this pressure, the inert gas occupies volume V_1. Fluid is pumped into the bottom of the accumulator, causing the diaphragm, bladder, or piston to deform upward, and the gas pressure increases to P_2 as it is compressed to a new volume, V_2. The total volume of fluid to perform work at some pressure greater than P_1 is $V_1 - V_2$.

Accumulators are widely used in industry to provide a supply of pressurized fluid in the event of an electric power failure. Suppose a ladle filled with molten metal is being swung into position to pour into a mold, and the power fails. Obviously, the ladle needs to be returned and emptied. An accumulator provides the stored energy to complete this emergency procedure.

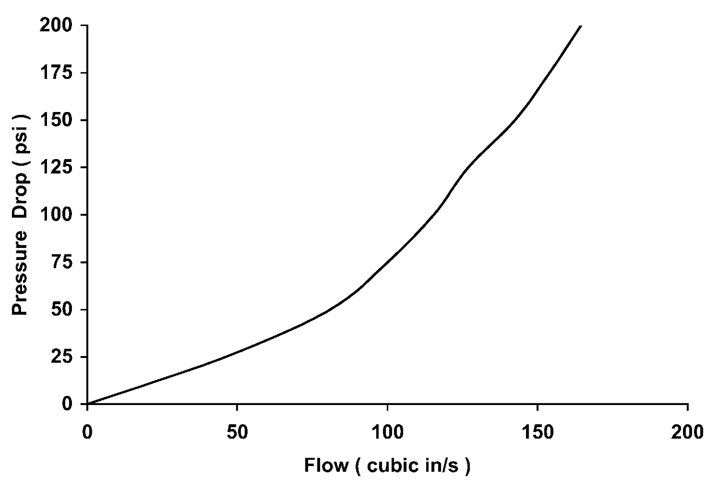


FIGURE 2.14
Typical ΔP vs. Q curve for a pressure control valve.

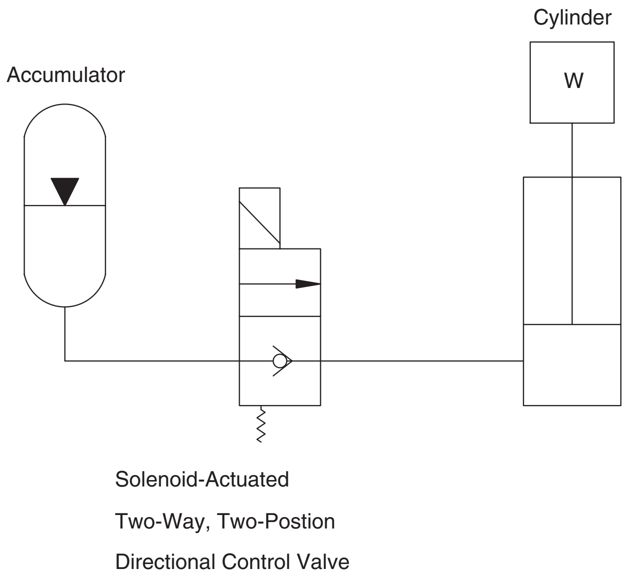


FIGURE 2.15
Accumulator used to extend cylinder.

2.7.1 Analysis to Illustrate Use of Orifice Equation

The accumulator in Fig. 2.15 is charged by adding fluid to build a pressure of 2725 psi. It takes 500 psi to extend the cylinder and lift the load. In this case, the load is the closure of a gate valve on a storm water management system. The accumulator is used to close the gate valve in the event that electric power is lost during a storm. The directional control valve (DCV) has a k_{eq} = 32.5×10^{-4}. (This value presumes that flow Q is in^3/s and ΔP is psi.) The value of k is then

$$k = 1/\sqrt{k_{eq}} = 17.5$$

When the DCV is activated, what is the velocity of the cylinder extension?

We know nothing of the mass to be moved. We only know that it takes a minimum of 500 psi to slide the gate valve in the two side channels that hold it in position. At some short Δt after the DCV is activated, the cylinder will be moving, and we assign a pressure of 500 psi on the cylinder side of the DCV. Using the orifice equation,

$$\begin{aligned} Q &= k\sqrt{\Delta P} \\ &= 17.5\sqrt{2725 - 500} \\ &= 825 \text{ in}^3\text{/s} = 214 \text{ GPM} \end{aligned}$$

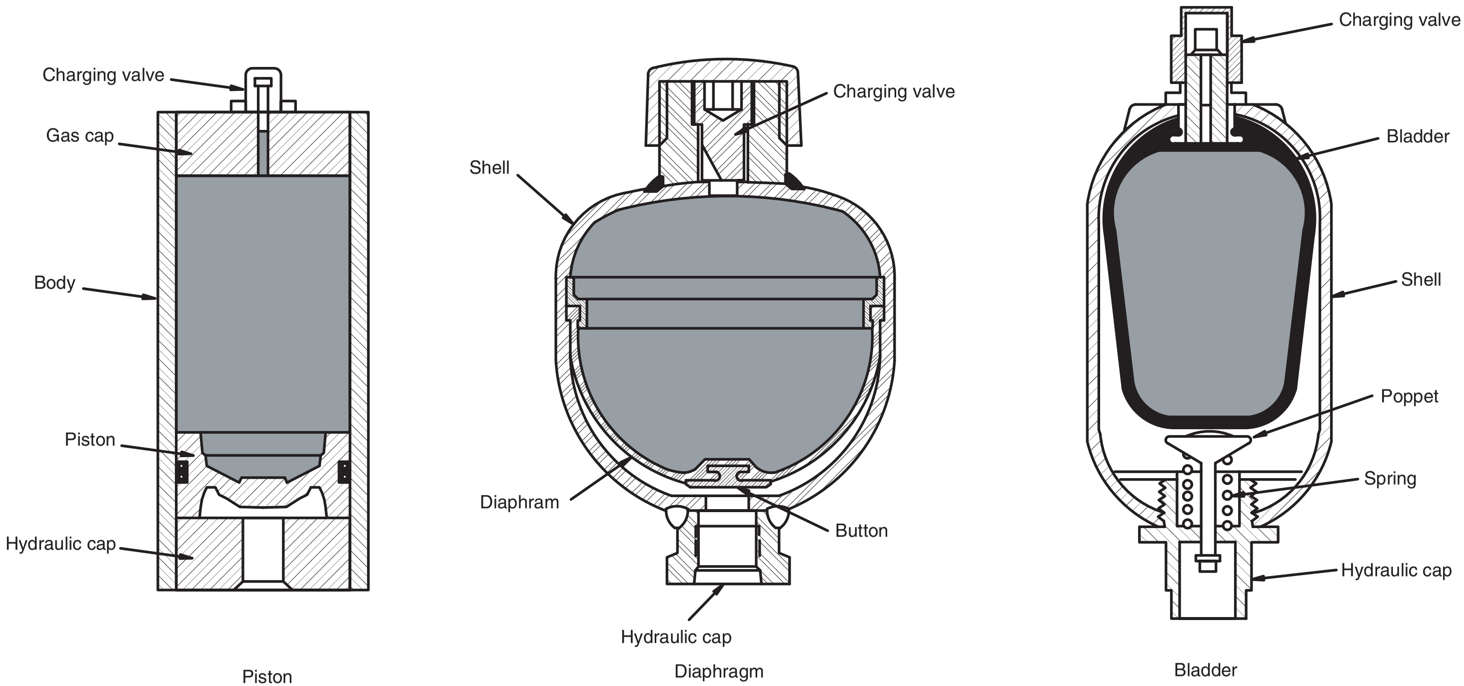


FIGURE 2.16
Three main designs for accumulators.

Will fluid flow out of the accumulator and across the orifice formed by a fully open DCV at this rate? The answer is "no" because of some factors we haven't considered. The fact remains, however, that there does need to be some means provided to control the cylinder speed. It also highlights the need for a safe method to bleed off pressure from a charged accumulator. The operator should not just open a valve and have the high pressure fluid emerge at high velocity and cause a shockwave to travel through the circuit.

Suppose the cylinder bore is 3 in. If flow from the accumulator is 825 in^3/s, the resultant cylinder velocity is

$$\begin{aligned} v &= Q/A \\ &= 825/(\pi 3^2/4) = 116.7 \text{ in/s} \end{aligned}$$

This velocity is very high; it might be described as an "explosive" extension of the cylinder. Suppose the cylinder speed must be limited to 2.4 in/s to ensure a safe emergency procedure. What changes can be made to the circuit to achieve this actuator speed?

Total force to move the load is

$$F_L = PA_c \tag{2.49}$$

where A_c = cylinder cap end area

$$\begin{aligned} F_L &= 500(\pi 3^2/4) \\ &= 3534 \; lb_f \end{aligned}$$

Suppose a cylinder bore equal to half the original 3 in. bore is selected.

$$\begin{aligned} A_c &= \pi(1.5^2)/4 \\ &= 1.767 \text{ in}^2 \end{aligned}$$

Pressure required to extend the cylinder is

$$\begin{aligned} P_L &= F_L/\mathrm{A}_c \\ &= 3534/1.767 \\ &= 2000 \text{ psi} \end{aligned} \tag{2.50}$$

Now the flow is

$$\begin{aligned} Q &= k\sqrt{\Delta P} \\ &= 17.5\sqrt{2725 - 2000} \\ &= 471 \text{ in.}^3/\text{s} = 122 \text{ GPM} \end{aligned} \tag{2.51}$$

Cylinder velocity is

$$\begin{aligned} v &= Q/A \\ &= 471/1.767 \\ &= 266 \text{ in/s} \end{aligned}$$

Velocity increased from 117 to 266 in/s when the cylinder bore was halved. Flow was reduced from 825 to 471 in^3/s, or 43%, but cylinder velocity increased 2.3 times. Our idea for reducing cylinder size did not solve the problem.

Another option is the installation of a flow control valve in the circuit as shown in Fig. 2.17. The flow control valve has a manual screw adjustment that screws a needle into the orifice, thus changing the k_{eq} value. (The needle progressively blocks the orifice. If it is screwed all the way down, it completely closes the orifice and stops all flow.) What k_{eq} value must be set for the flow control valve to limit cylinder velocity to 2.4 in/s?

Required flow to the cylinder is

$$\begin{aligned} Q &= A_c v \\ &= (\pi 3^2/4)2.4 \\ &= 17 \text{ in}^3/\text{s} = 4.4 \text{ GPM} \end{aligned}$$

Required flow is about 2% of the flow available when the DCV is opened rapidly.

Pressure drop across the DCV for a flow of 17 in^3/s is obtained by using the orifice equation written with flow as the independent variable.

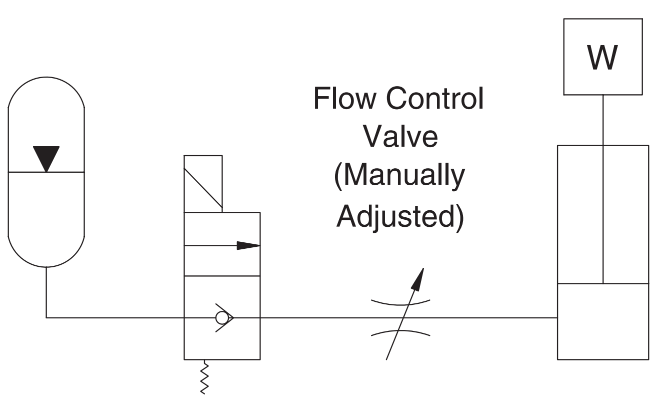


FIGURE 2.17
Flow control valve used to change cylinder extension velocity.

$$\begin{aligned}\Delta P &= k_{eq}Q^2\\ &= 32.5\times 10^{-4}(17)^2\\ &\approx 1 \text{ psi}\end{aligned}$$

This pressure drop is so low that the directional control valve can be neglected in the analysis. We assume that the full ΔP required to limit flow to 17 in^3/s must be developed across the flow control valve. The required k is

$$\begin{aligned}k_{fc} &= Q/\sqrt{\Delta P_{fc}}\\ &= 17/\sqrt{2725-500}\\ &= 0.36\end{aligned} \tag{2.52}$$

The flow control valve must be turned down almost to the closed position to obtain this k.

A flow control valve converts some of the fluid energy to heat energy, thus reducing the energy that flows to the cylinder. This is an undesirable waste of energy stored in the accumulator, but it is unavoidable in this instance. When the energy stored in the accumulator is required infrequently (only in an emergency situation), the energy loss due to the flow control valve is accepted.

2.7.2 Use of Orifice Equation to Analyze Pressure Reducing Valve

Another gate valve closure problem illustrates how the orifice equation is used to analyze a pressure reducing valve. In this case, a large gate valve is being closed with a hydraulic cylinder. The cylinder bore is 7 in., and the rod diameter is 3.5 in. Breakout force, the force to initially move the valve, is 41, 700 lb_f. The desired velocity of the gate value is 2.4 in/s. The applied pressure must be limited to prevent the gate valve from being damaged when it is seated. A pressure-reducing valve is used in the circuit (Fig. 2.18) to accomplish this pressure limiting function.

Before proceeding with the solution to this problem, it is necessary to gain some understanding of the operation of a pressure-reducing valve. A schematic is shown in Fig. 2.19. The spring is set to provide a certain downward force (F_s) on the valve spool. Equal hydraulic force is exerted in both directions by the inlet pressure, because the area of the spool skirt is equal at the top and bottom (Fig. 2.20). A hydraulic force is applied to the bottom of the spool by the downstream pressure. For purposes of discussion, we will denote this force as F_{hb}. When the downstream pressure rises to the point where $F_{hb} = F_s$, the spool is displaced upward, and the bottom skirt partly closes the *orifice* where the fluid flows out of the valve. The higher the down-

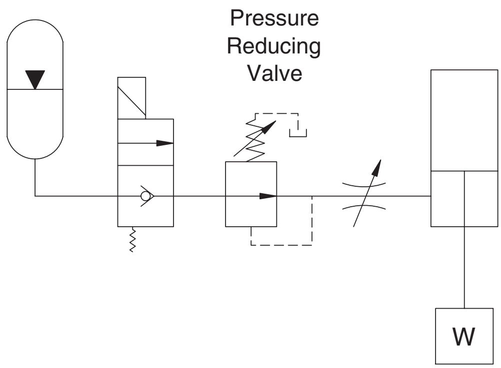


FIGURE 2.18
Circuit to close a large gate valve with a hydraulic cylinder.

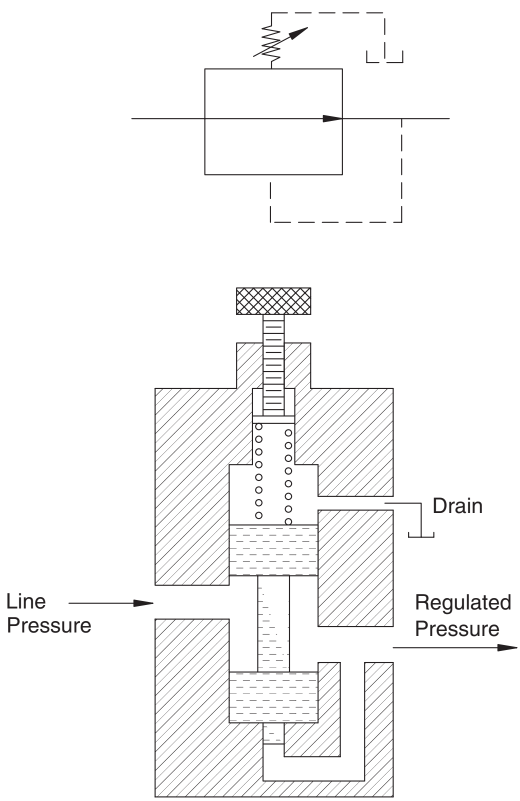


FIGURE 2.19
Schematic of a pressure-reducing valve.

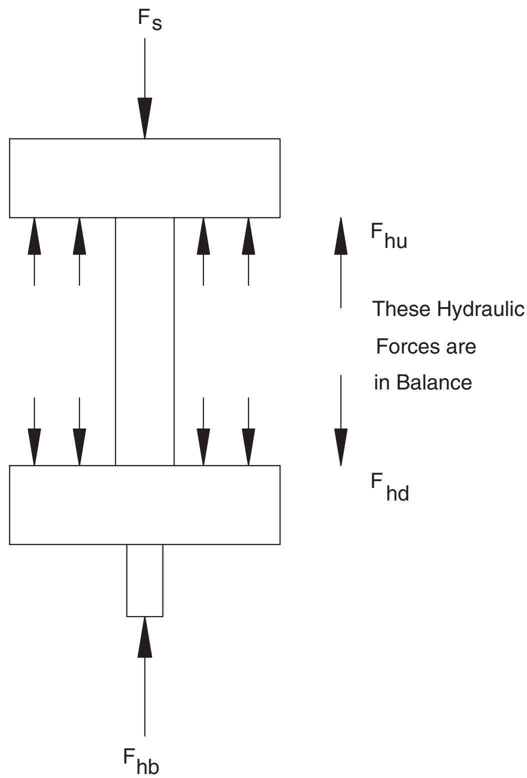


FIGURE 2.20
Free body diagram of spool in pressure-reducing valve shown in Fig. 2.19.

stream pressure, the more the valve displaces upward, and the more the spool closes the orifice. The orifice area is reduced, so k is reduced, and the ΔP across the valve is increased. The spool adjusts the ΔP in this manner to maintain a downstream pressure equal to the valve setting.

Required pressure at the rod end of the cylinder to initiate movement of the gate valve is

$$P_L = F/A_r$$

where A_r = rod end area (in^2); A_r is given by

$$A_r = \left(\frac{\pi 7^2}{4} - \frac{\pi 3.5^2}{4}\right) = 28.86 \text{ in.}^2$$

$$P_L = (41,700)/28.86$$

$$= 1445 \text{ psi}$$

Flow to lift the valve at the desired velocity is

$$
\begin{aligned}
Q &= A_r v \\
&= 28.86(2.4) \\
&= 69.3 \text{ in}^3/\text{s} = 18 \text{ GPM}
\end{aligned}
$$

A pressure-reducing valve works well in the circuit shown in Fig. 2.18, because it automatically changes its spool position as the accumulator pressure drops. (Accumulator pressure drops toward the precharge pressure as it empties.) The pressure-reducing valve opens the orifice, thus reducing the pressure drop, as the inlet pressure falls.

The setting of the pressure-reducing valve must be carefully chosen to ensure that enough pressure is available for *break-out,* the force required to get the load moving. Force required for acceleration is small when the required cylinder velocity is low. In this case, a breakout force is required to overcome friction and move the gate valve off its seat, and acceleration of the load is a minor factor.

A flow control valve is included in the circuit (Fig. 2.18) to provide a means for "fine tuning" the flow to the cylinder. It is set during the testing phase, and this setting remains fixed. In this case, it functions as a fixed orifice and the pressure-reducing valve functions as a variable orifice. The two in series constitute an interesting circuit to analyze.

Example Problem 2.6

For the circuit in Fig. 2.18, the following operating parameters are given:

Accumulator full charge pressure	2725 psi
Accumulator precharge pressure	1710 psi
Pressure-reducing valve setting	1810 psi
Maximum cylinder speed	2.4 in/s

A bank of accumulators have been manifolded together to provide enough stored fluid to fully extend the cylinder. For this problem, we do not have to consider the quantity of stored fluid.

Recapping parameters given earlier, required breakout force to begin to lift the valve is 41,700 lb_f. The cylinder parameters are:

Bore	7 in.
Rod diameter	3.5 in.
Stroke	96 in.

A pressure-reducing valve rated for 0 to 30 GPM was selected from a manufacturer's catalog (Sun Hydraulics Corp.). The characteristic curve for this valve in the *full open* position, with no downstream load pressure, is given in Fig. 2.21. Any set of coordinates along this curve can be used to compute a k value for this valve. We choose the points, $Q = 57.8$ in^3/s, $\Delta P = 50$ psi, since this flow is in the range of interest. Using the orifice equation,

$$k_{pr} = \frac{Q}{\sqrt{\Delta P_{pr}}}$$

$$= \frac{57.8}{\sqrt{50}} = 8.17$$

At the instant the directional control valve is shifted, the pressure at the upstream side of the pressure-reducing valve is 2725 psi. The reducing valve closes such that the downstream pressure is 1810 psi, the valve setting. The input pressure to the flow control valve is then 1810 psi, and the load pressure (downstream pressure) is the breakout pressure, 1445 psi. What k value must be set on the flow control valve to limit the flow to $Q = 69.3$ in^3/s, corresponding to a maximum cylinder velocity of 2.4 in/s? Again, using the orifice equation,

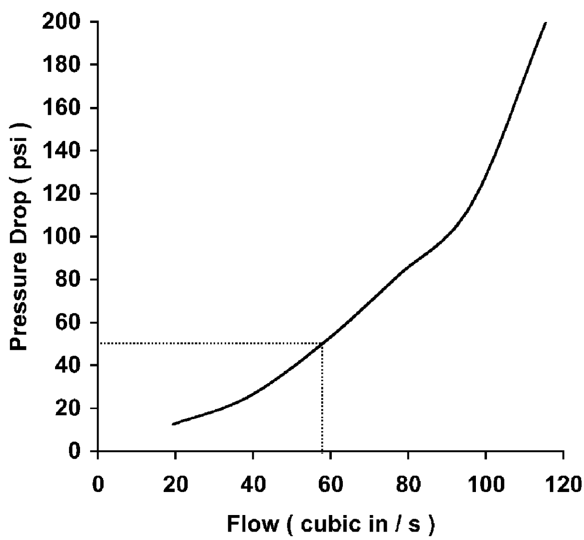


FIGURE 2.21
Characteristic curve for pressure reducing valve used in circuit shown in Fig. 2.18. Valve is in fully open position.

$$
\begin{aligned}
k_{fc} &= \frac{Q}{\sqrt{\Delta P_{fc}}} \\
&= \frac{69.3}{\sqrt{1810 - 1445}} \\
&= 3.63
\end{aligned} \tag{2.53}
$$

The flow control valve is adjusted to get this k value, and the initial velocity of the cylinder is set at 2.4 in/s.

It is now necessary to calculate the cylinder velocity when the accumulator is almost empty, meaning that the cylinder has almost reached the end of its stroke. We assume that the accumulator pressure has fallen to 1810 psi, and the force to move the valve is 80% of the breakout force.

$$
\begin{aligned}
F &= 0.8 \times 41,700 \\
&= 33,360 \text{ lb}_f
\end{aligned}
$$

The load pressure is

$$
\begin{aligned}
P_L &= F/\mathrm{A}_r \\
&= 33{,}360/28.86 \\
&= 1155 \text{ psi}
\end{aligned}
$$

Since the pressure reducing valve is in the full open position, the k factor is that computed from Fig. 2.21, $k_{pr} = 8.17$. Flow through this valve is given by

$$
Q = k_{pr}\sqrt{\Delta P_{pr}} = 8.17\sqrt{\Delta P_{pr}} \tag{2.54}
$$

where ΔP_{pr} = pressure drop across reducing valve (psi).

Flow through the flow control valve, a fixed orifice, is

$$
Q = k_{fc}\sqrt{\Delta P_{fc}} = 3.63\sqrt{\Delta P_{fc}} \tag{2.55}
$$

where ΔP_{fc} = pressure drop across flow control valve (psi).

Since the same fluid flows through both valves, Eqs. (2.54) and (2.55) can be set equal to each other, and

$$
8.17\sqrt{\Delta P_{pr}} = 3.63\sqrt{\Delta P_{fc}}
$$

or

$$\Delta P_{fc} = 5.06\Delta P_{pr} \tag{2.56}$$

As has been done for all the analysis, the pressure drop in the lines and across the directional control valve is neglected. Total pressure drop is

$$\Delta P_{tot} = \Delta P_{pr} + \Delta P_{fc} \tag{2.57}$$

The total pressure drop is the difference between the accumulator and load pressures.

$$\Delta P_{tot} = 1810 - 1155 = 655 \text{ psi}$$

We now have two equations and two unknowns.

$$655 = \Delta P_{pr} + \Delta P_{fc}$$

$$\Delta P_{fc} = 5.06 \ \Delta P_{pr}$$

Solving,

$$\Delta P_{pr} = 108 \text{ psi}$$

$$\Delta P_{fc} = 547 \text{ psi}$$

To verify these answers, the flow is

$$Q = 8.17\sqrt{108} = 84.9 \text{ in}^3\text{/s}$$

$$Q = 3.63\sqrt{547} = 84.9 \text{ in}^3\text{/s}$$

Corresponding cylinder velocity is

$$v = Q/A_r$$

$$= 84.9/28.86$$

$$= 2.94 \text{ in/s}$$

Cylinder speed is 23% faster than the design maximum velocity, 2.4 in/s. It is not possible to get the same cylinder velocity at the beginning and end of gate valve closure, because the accumulator pressure decreases as it empties, and the pressure-reducing valve adjusts accordingly.

Since the maximum allowable cylinder speed is 2.4 in/s (Q = 69.3 in^3/s), the flow control valve must be closed further to limit flow. Trial and error calculations show that k_{fc} = 2.87gives

$$\Delta P_{pr} = 72 \text{ psi}$$

$$\Delta P_{fc} = 583 \text{ psi}$$

and the flow is

$$Q = 2.87\sqrt{583} = 69.3 \text{ in}^3/\text{s}$$

Flow at the beginning of valve closure is now

$$Q = 2.87\sqrt{1810 - 1445}$$
$$= 54.8 \text{ in}^3/\text{s}$$

and corresponding cylinder velocity is 1.9 in/s, or 21% slower than the allowed maximum.

2.8 Summary

The function of the fluid in a fluid power system is to transmit power, provide lubrication, provide cooling, and seal clearances between moving parts. Pneumatic systems use air as the working fluid, and hydraulic systems use either an oil-based or water-based liquid.

In a hydraulic system, the establishment and maintenance of a lubricant film between the moving parts in a pump or motor is essential, thus the properties of the fluid are very important. Viscosity is the fluid's resistance to shear. It decreases as operating temperature increases. A hydraulic system must be designed to operate in a given temperature range so as to maintain viscosity in the range needed to ensure that the lubrication film between parts is continuous.

The compressibility of an oil is defined by the bulk modulus. Other properties of interest are oxidation potential and corrosion resistance. Fire resistance is a key consideration in some applications.

Pressure drop along a conductor (hose, tube, or pipe) is proportional to length and velocity squared and inversely proportional to inside diameter. In general, pressure drops across fittings are more significant than pressure drops along the conductor.

All valves form some type of orifice in the line and thus have a quadratic relationship between the pressure drop across the valve (ΔP) developed by a given flow through the valve (Q). This relationship is fundamental to most of the analysis done to describe circuit performance. When a pressure drop is created and no mechanical work is done, all the fluid energy is converted to heat energy. For the vast majority of circuits, this source of heat is greater than the heat produced by pressure drops in the lines and fittings.

Heat increases the temperature of the fluid, which reduces the viscosity. If viscosity gets too low, the lubrication film will break down, and metal-to-metal contact will result. Maintenance of operating temperature in a given range is a key factor in the design of fluid power circuits.

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Problems

2.1 The inside diameter of a hydraulic hose is 0.375 in. It is recommended that the fluid velocity in a supply line not exceed 10 ft/s. What is the maximum flow (GPM) for this line?

2.2 What flow of oil (GPM) can be pumped through a 0.75 in. diameter tube and the flow will still be laminar? The kinematic viscosity of the oil is $\nu = 35$ cS.

2.3 The line length between a pump and motor is 6.0 ft. A crossover relief valve is placed at the midpoint between the pump and motor as shown in Fig. 2.22. Total pressure drop through the crossover relief valve at a flow of 27 GPM is 25 psi. The line has an elbow at the pump and an elbow at the motor. The following parameters are given.

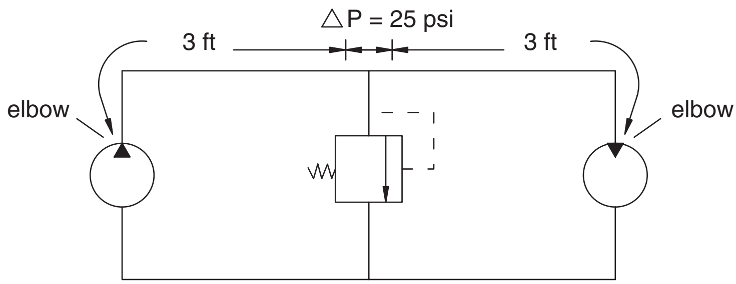


FIGURE 2.22
Circuit for Problem 2.3.

Line diameter	$D = 0.75$ in
Fluid viscosity	$\nu = 35$ cS
Fluid specific gravity	$S_g = 0.9$

Find the total pressure drop between the pump and motor.

2.4 A pressure-reducing valve is set to limit the force exerted by a clamping cylinder. The 1.5-in. bore cylinder must not exert more than 2900 lb_f to prevent damage to the part being held. At the moment the DCV is shifted to close the clamp cylinder, the supply pressure is 2750 psi.

a. What pressure drop must the pressure reducing valve maintain?
b. If the flow to the clamp cylinder is 8 GPM, what orifice coefficient (k) must be developed by the pressure reducing valve?

2.5 Molten metal is poured from a ladle. Total weight of the full ladle is 18 tons. Just as the pour begins, there is a power failure. Fortunately, the hydraulic circuit has a bank of accumulators charged to 2750 psi that will supply the fluid to move the ladle back to a safe position. Acceleration of the load must be limited to 290 in/s^2. An operator manually activates a DCV to direct the accumulator fluid to the 7-in. diameter cylinder moving the ladle. This operator cracks the DCV and watches the ladle to achieve the correct motion.

a. What pressure is required to accelerate the load to 290 in/s^2?
b. What pressure drop across the DCV must be maintained during the period when the load is being accelerated?
c. If the force to move the load at constant velocity is F_L= 2700 lb_f, what pressure drop across the DCV is required? Assume the accumulator pressure has dropped to 1950 psi by the time the load has just reached constant velocity.

APPENDIX 2.1

Data for Selected Hydraulic Fluids

Brand Name	Designation	Specific Gravity	Viscosity (cS) 100°F	Viscosity (cS) 212°F
		Petroleum Base		
Amoco				
American Ind.	15	0.893	34.1	5.5
	51	0.891	110.2	11.1
	95	0.892	209.3	16.7
	175	0.893	380.7	24.8
BP Canada				
Energol CS	40	0.890	14.4	2.7
	55	0.868	27.6	5.2
	100	0.882	67.8	8.3
	200	0.882	142.8	13.6
Castrol Oils				
Hyspin	40	0.902	132.2	2.0
	55	0.866	23.7	4.5
	100	0.880	62.5	8.0
	175	0.882	119.2	12.5
Conoco				
Super Hyd. Oil	15	0.860	32.1	5.5
	21	0.869	45.6	6.8
	31	0.872	67.8	8.6
Exxon				
Nuto	43	0.870	33.9	5.5
	53	0.878	67.1	8.3
	93	0.888	232.2	18.9
	146	0.894	470.1	30.5
Mobil Oil Corp.				
Mobil DTE	Light	0.871	32.1	5.2
	Med.	0.876	48.9	7.1
	Hvy. med.	0.879	64.4	8.3
	Heavy	0.882	94.3	10.3
	Ex. heavy	0.887	140.6	13.0

Brand Name	Designation	Specific Gravity	Viscosity (cS) 100°F	212°F
Shell Oil. Co.				
Hydraulic	21	0.865	21.8	4.2
	29	0.871	46.7	6.5
	41	0.873	102.0	11.1
	71	0.882	219.9	17.2
Texaco				
Regal Oil	A R&O	0.868	32.3	5.5
	C R&O	0.904	68.7	6.5
	E R&O	0.912	109.7	8.9
	H R&O	0.892	269.0	19.7
Phosphate Ester				
Chevron				
FR Fluid	8	1.165	32.3	4.2
	10	1.135	48.5	5.2
	13	1.135	71.5	5.8
	20	1.135	130.9	6.8
Water Glycol				
Bel-Ray Co., Inc.				
No-Flame G	15	1.06	32.1	–
	20	1.07	43.3	–
	30	1.08	65.6	–
Houghton				
Houghto-Safe	271	1.045	43.3	15.6@150°F
	416	1.08	34.4	23.0@150°F
	520	1.08	43.3	18.1@150°F
	620	1.07	43.3	17.9@150°F
Stauffer				
Fyrguard	150	1.076	32.1	19.4@150°F
	200	1.079	43.3	27.6@150°F

3

Pressure Control

3.1 Introduction

Pressure control is a key element in the design of any circuit. Not only is it the key to achieving a given functional objective, it is also the key to safe operation.

Components are designed to operate at a given maximum pressure and will withstand pressure peaks up to some burst pressure. Failure of a component can be dangerous to nearby workers. They can be injured by shrapnel, or they may be injured when they are hit by a stream of high-pressure, high-temperature oil. Injuries received when oil penetrates the skin are very difficult to treat and require specialized medical knowledge. Often, there is also potential for worker injury by losing control of a load held against gravity.

The fundamental pressure control problem in circuit design is the limiting of pressure to a level below the working pressure of the lowest-rated component in the circuit. If a piece of hose, rated at 1500 psi working pressure, is used in a circuit where all other components are rated at 3000 psi, then maximum pressure in this circuit must be limited to 1500 psi. It is important to remember the review of fluid mechanics given in Chapter 2. Pressure can build to the relief valve setting at *all* points between the pump and the load.

Six pressure-control valves will be discussed in this chapter. These valves are:

1. Relief valves
2. Unloading valves
3. Sequence valves
4. Pressure-reducing valves
5. Counterbalance valves
6. Brake valves

Each of these valves works on the same principle; a spring force balances a hydraulic force. The hydraulic force is produced by pressure acting on a given area. When the hydraulic force becomes greater than the spring force, the valve spool moves. There are many different ways in which this principle is used in valve design. The construction of some valves is intricate, but the principle of operation is simple. It is appropriate to re-emphasize the principle; a spring force opposes a hydraulic force.

3.2 Review of Needed Symbols

As in previous chapters, ANSI symbols are introduced here as they are needed. The directional control valve symbol is the most intuitive and self-explanatory of the symbols. Some experience with the directional control valve symbol has already been gained. At this point, it is necessary to review the three most common center configurations (Fig. 3.1) for spool-type directional control valves.

An open-center (float) valve allows flow between all four ports when the valve is in the center (nonactuated) position. The actuator (downstream from the valve) is not held in position but is free to float.

The open-center valve also allows free flow from the inlet port to the return (or tank) port, but it blocks the actuator ports. The actuator cannot move (neglecting leakage) when the open-center valve is in the center position.

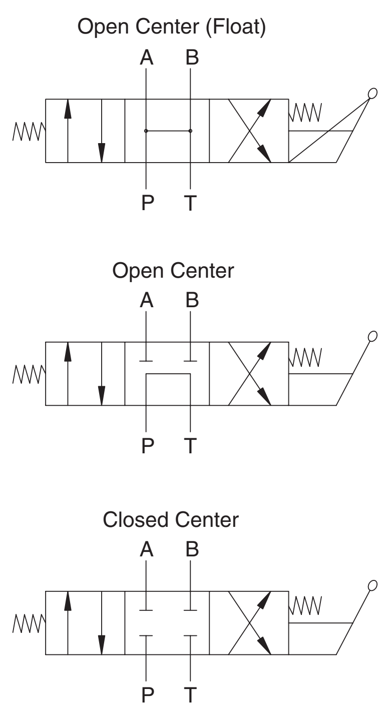


FIGURE 3.1
Three main center configurations in directional control valves. (P is the pressure port, T the tank port, and A and B are the actuator ports.)

The closed-center valve has all four ports blocked when it is in the center position. There is no pathway through the valve between any of the four ports.

There are many configurations for directional control valves. Two are shown in Figs. 3.2 and 3.3. The check valves ensure that flow can go only from the pump to the circuit. Thus, the pump is isolated from pressure spikes that may occur due to load dynamics. Both figures show two directional control valves stacked in the same housing. Ten or more valves can be stacked in this manner. (The reader may have observed a bank of handles on a machine for manual actuation of individual valves stacked in this manner.) As is often the case, the figures show a relief valve built into the valve housing, a feature which simplifies circuit assembly.

Figure 3.2 shows a valve stack where the flow passes directly through the valve and returns to the reservoir when neither spool is actuated. The bottom half of this figure shows both spools activated simultaneously. If the pressure required by both cylinders is approximately equal, the pump flow will divide, and some flow will go to each cylinder. However, if one cylinder requires more pressure, the flow will always take the path of least resistance and goes to the lower-pressure cylinder first. When this cylinder is fully extended or hits a stop, the pressure will rise to the level required to extend the second cylinder.

The valves in Fig. 3.3 are still open-center valves, but they are configured differently. Both spools are shown in the actuated position in the bottom half of the figure. Flow goes to Cylinder 1 and no flow to Cylinder 2. If Cylinder 1 is returned to the center position, then flow will go to Cylinder 2. This spool design ensures that only one circuit can be actuated at any time.

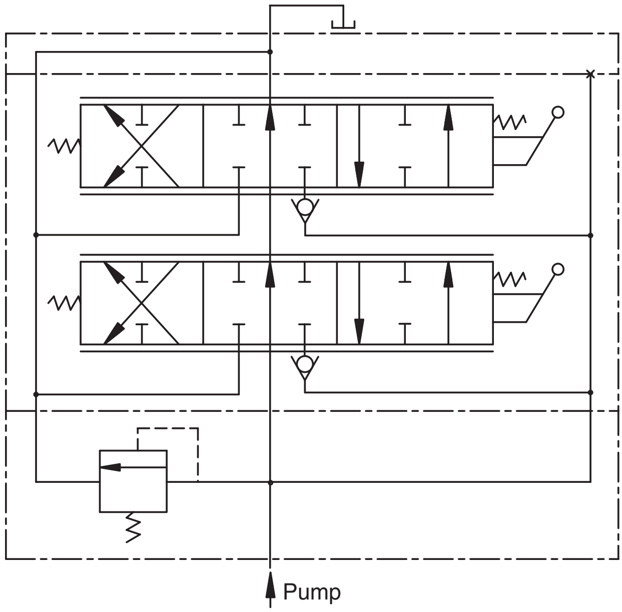
Manufacturers can assemble different spool sections in the same stack. Many options are available. Simplified diagrams are shown in Chapters 4, 5, and 7 where they are needed to understand circuit function. The manufacturer's literature should be referenced for options not included in this text.

3.3 Relief Valve

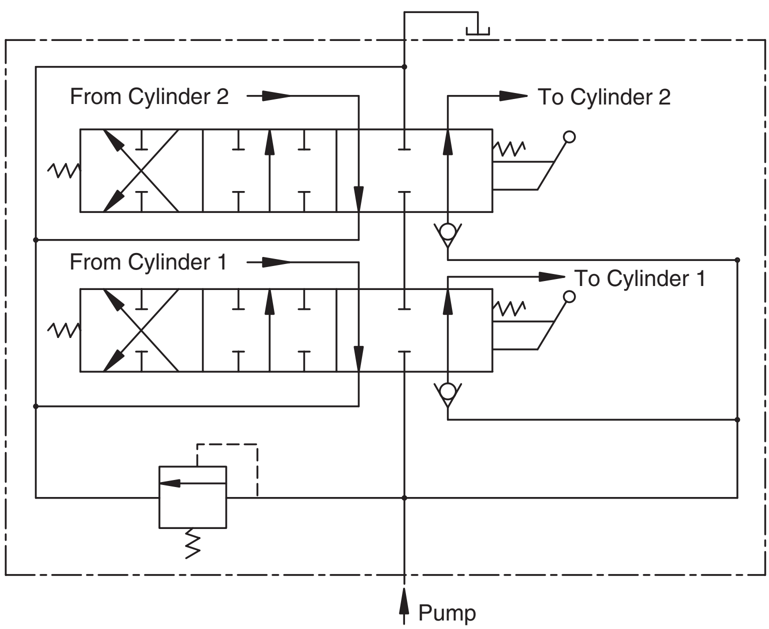
The relief valve is used to limit pressure in an entire circuit. It is generally the first component downstream from the pump. Relief valves can be direct acting or pilot operated.

3.3.1 Direct-Acting Relief Valve

A schematic of a direct-acting relief valve is shown in Fig. 3.4. Pressure acts on the annular area of the valve spool. The hydraulic force is given by



Unactuated Position



Actuated Position

FIGURE 3.2
Stacked directional control valve with two open-center spools configured such that both circuits can be actuated simultaneously.

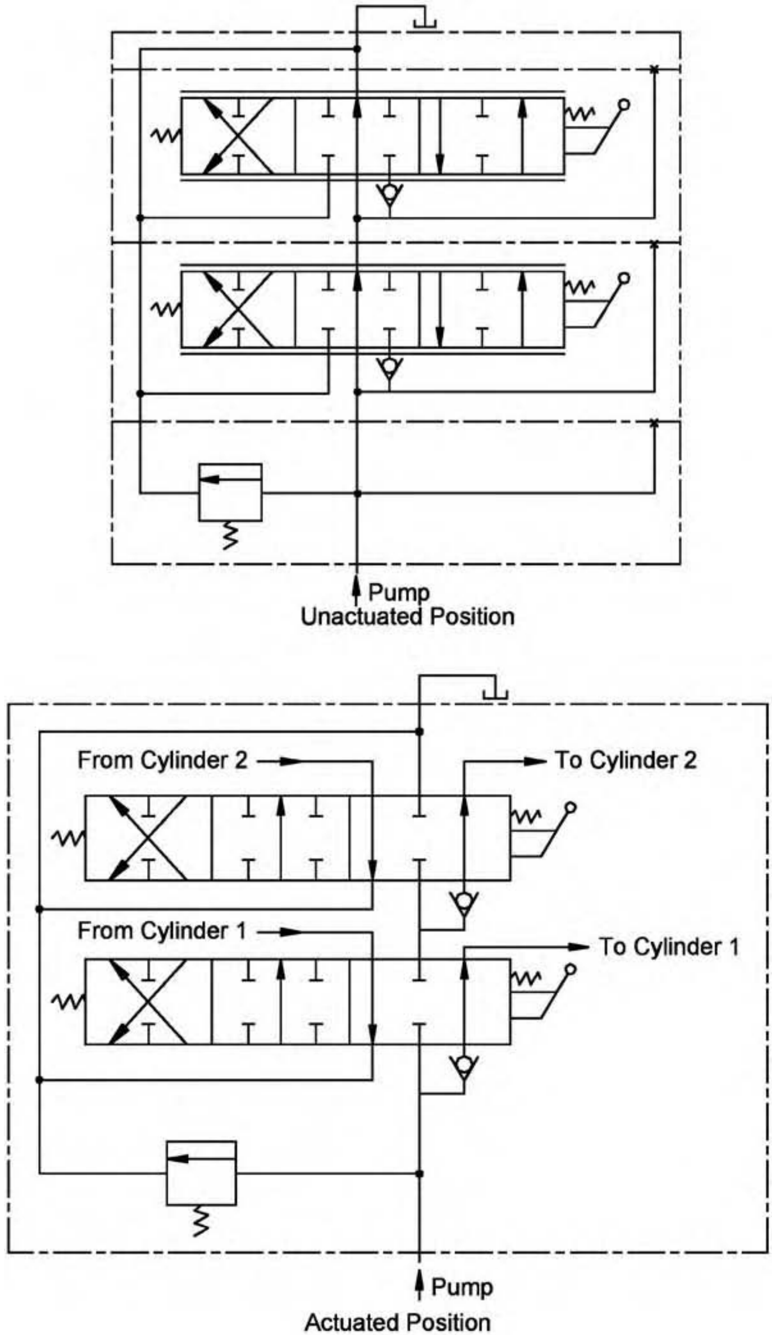


FIGURE 3.3
Stacked directional control valve with two open-center spools configured such that only one circuit can be actuated at any time.

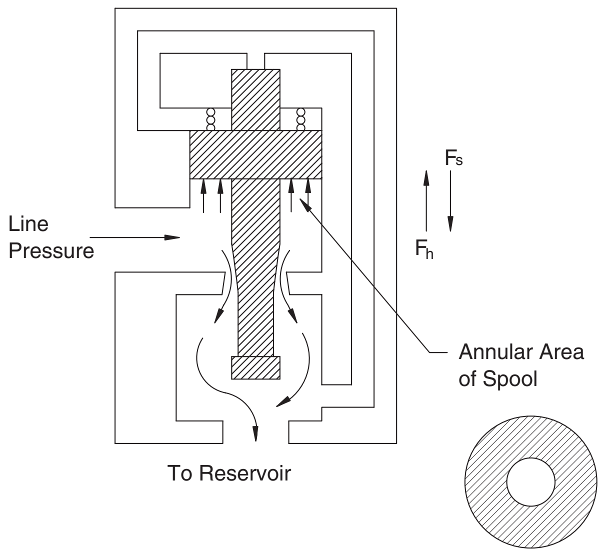


FIGURE 3.4
Functional diagram of direct-acting relief valve. (Reprinted with permission from Parker Hannifin Corp.)

$$F_h = PA_a \tag{3.1}$$

where F_h = hydraulic force (lb_f)
P = pressure (psi)
A_a = annular area (in^2)

The notation F_s will be used for the spring force. When F_h equals F_s, the valve cracks open, meaning that the spool lifts off its seat and allows fluid to flow to the reservoir. As pressure increases, the spool lifts higher, allowing more flow to bypass to the reservoir. At some pressure level, the total flow bypasses to the reservoir.

A typical flow vs. pressure curve for a direct-acting relief valve is shown in Fig. 3.5. The valve is set to open at 1500 psi. This pressure is known as the *cracking pressure.* When pressure reaches 2000 psi, the valve is fully open, and all flow is bypassed to the reservoir; no flow goes to the remainder of the circuit. The 500 psi differential between cracking and full bypass is needed for a direct-acting valve when it has a functional role in flow control in addition to its pressure limiting function. Pilot-operated relief valves have a much lower differential and are used when the sole function of the relief valve is overpressure protection for the circuit.

The characteristics of a direct-acting relief valve can be used in a simple circuit to control the speed of the actuator. In the circuit shown in Fig. 3.6, the flow control valve is simply an adjustable orifice in the circuit. When the flow

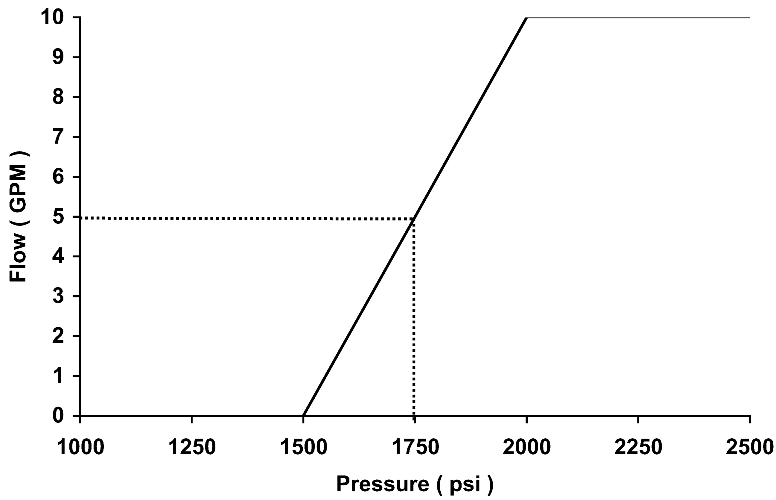


FIGURE 3.5
Typical flow vs. pressure curve for a direct-acting relief valve.

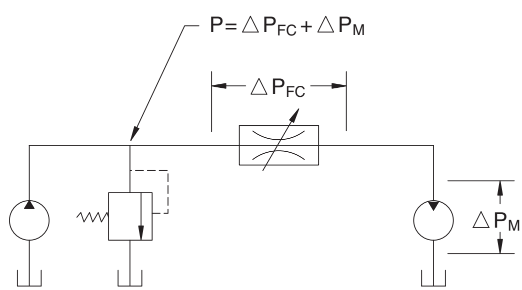


FIGURE 3.6
Circuit in which motor speed is controlled with a flow control valve.

control valve is partly closed, a pressure drop is created across the valve. Pressure at the relief valve is the sum of the pressure drop across the flow control valve plus the pressure drop across the motor. (For this simple example, pressure drops in the lines are neglected.) To slow the motor, the flow control valve is closed to create enough pressure at the relief valve to cause it to crack open. Part of the pump output now bypasses to the reservoir; thus, flow to the motor is reduced, and the speed decreases.

A simple analysis will illustrate the performance of the circuit in Fig. 3.6. Suppose the relief valve has the characteristics shown in Fig. 3.5. The fixed displacement pump is delivering 10 GPM to the motor. The flow control valve is fully open, and the pressure at the relief valve is 1000 psi. To reduce the motor speed to one-half its current value, what pressure drop must be created at the flow control valve?

Flow to the motor must be reduced to 5 GPM to cut the speed by half, which means that 5 GPM must flow across the relief valve. As shown in Fig. 3.5, pressure must rise to 1750 psi before 5 GPM bypasses through the relief valve. Pressure drop across the motor is only 1000 psi; therefore, the required pressure drop across the flow control valve must be 1750 – 1000 = 750 psi.

No mechanical energy is output at the relief valve; consequently, all the hydraulic energy in the flow across the valve is converted to heat energy. The circuit in Fig. 3.6 is simple but not energy efficient.

It is instructive to calculate the energy flow in this simple circuit. The pump is a fixed-displacement unit; consequently, the delivered flow is constant. (At this time, we neglect that pump leakage increases as pressure increases, and therefore, pump output decreases as pressure increases.) Total hydraulic power delivered by the pump is

$$\mathcal{P}_{hyd} = \frac{PQ}{1714} \tag{3.2}$$

where $\mathcal{P}_{hyd}$ = hydraulic power (hp)
P = pressure (psi)
Q = flow (GPM)

For this example,

$$\begin{aligned}\mathcal{P}_{hyd} &= \frac{(1750)(10)}{1714} \\ &= 10.2 \text{ hp}\end{aligned} \tag{3.3}$$

Hydraulic power converted to heat when 5 GPM flows across the relief valve is

$$\begin{aligned}\mathcal{P}_{rv} &= \frac{1750(5)}{1714} \\ &= 5.1 \text{ hp}\end{aligned} \tag{3.4}$$

Hydraulic power converted to heat at the flow control valve is

$$\begin{aligned}\mathcal{P}_{fc} &= \frac{(1750-1000)(5)}{1714} \\ &= 2.19 \text{ hp}\end{aligned} \tag{3.5}$$

Hydraulic power converted to mechanical power at the motor is

$$\mathcal{P}_m = \frac{1000(5)}{1714}$$
$$= 2.91 \text{ hp} \tag{3.6}$$

Total hydraulic power delivered by the pump is used in the following way:

$$\mathcal{P}_{hyd} = \mathcal{P}_{rv} + \mathcal{P}_{fc} + \mathcal{P}_m$$
$$10.2 = 5.1 + 2.19 + 2.91 \tag{3.7}$$

The efficiency of the circuit is

$$eff = \frac{\mathcal{P}_m}{\mathcal{P}_{hyd}} \times 100$$
$$= \frac{2.91}{10.2} \times 100 = 28.5\% \tag{3.8}$$

Only 28.5% of the hydraulic power is delivered as mechanical power by the motor. The remainder is converted to heat. Operating temperature of this circuit will be high. Obviously, it is a poor design; however, the analysis does reinforce an important concept in pressure control. *Any time there is a pressure drop across a valve and no mechanical power is output, heat is generated and circuit efficiency is reduced.* Simple circuits may have a lower initial cost, but the higher operating costs over their design life often offset this advantage.

3.3.2 Pilot-Operated Relief Valve

A pilot-operated relief valve has the same function as a direct-acting relief valve; however, it has a different pressure vs. flow curve. The performance curves for the two types of relief valve are given in Fig. 3.7. The pilot-operated valve opens completely over a narrow pressure range. This allows the circuit to operate over a wider pressure range without loss of fluid over the relief valve.

A functional diagram of a pilot-operated relief valve is shown in Fig. 3.8. The main spool has a small hole (orifice) drilled in the skirt. Because of this hole, pressure is the same on the top and bottom of the skirt. As long as there is negligible flow through the orifice, there is no pressure drop across the orifice.

The pilot section of the valve is the top section. A dart is held in place by the pilot spring. When the hydraulic force on this dart becomes greater than the pilot spring force, the dart is unseated, and fluid flows from the cavity above the skirt, through an internal drain to the valve outlet. Flow through

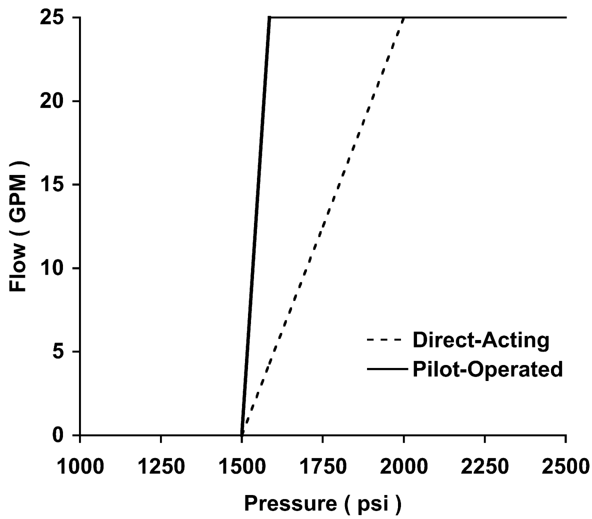


FIGURE 3.7
Comparison of performance curves for direct-acting and pilot-operated relief valves.

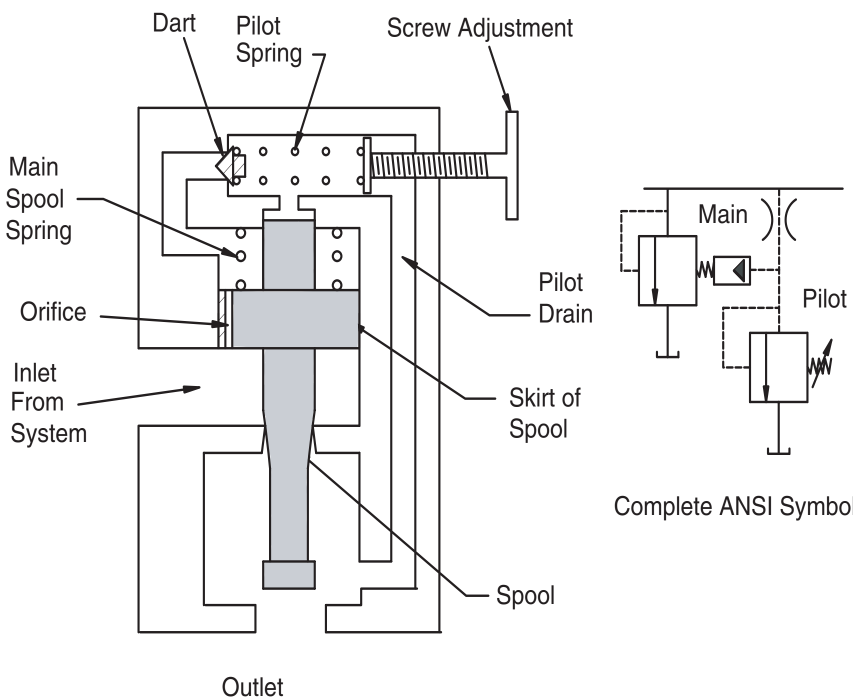


FIGURE 3.8
Functional diagram of pilot-operated relief valve. (Reprinted with permission from Parker Hannifin Corp.)

the orifice replaces the fluid lost from the cavity above the skirt. The spool is still held in position by the main spool spring.

At this point, discussion is facilitated if the two springs are assigned values. Suppose the pilot spring is a 1425-psi spring, and the main spool spring is a 75-psi spring. When pressure at the valve inlet reaches 1425 psi, the dart is unseated. Pressure in the upper cavity cannot increase above 1425 psi. The hydraulic force on the top and bottom of the skirt is equal, and these two forces balance. The main spool is held in position only by the spring force produced by the 75 psi spring.

What happens when the pressure at the valve inlet reaches 1425 psi? The relief valve stays closed. If pressure continues to increase and reaches 1500 psi, the spool lifts, and fluid is bypassed to the reservoir. As pressure continues to increase above 1500 psi, the main spool opens further until it is completely open. Only a small pressure increase is needed to completely compress the 75 psi spring. (In Fig. 3.7, the pilot-operated valve is fully open at 1585 psi.) In other words, the valve goes from cracking to full open with a very small increase in pressure. When the load is changing quickly, and sharp pressure spikes are created, the quick opening feature of a pilot-operated relief valve is needed to protect the circuit.

The key advantage of a pilot-operated valve is that it allows the designer to use pressure to within 100 psi of the valve setting to meet the functional objective of the circuit. In comparison, the direct-acting valve cracks open at 1500 psi, and pressure must increase to 2000 psi before it is fully open.

A pilot-operated relief valve can be used with a remote pilot as shown in Fig. 3.9. The remote pilot functions like the pilot built into the top of the main relief valve. It allows the designer to set two pressure levels with one main relief valve.

Suppose the pilot on top of the main valve has a 1925 psi spring, and the remote pilot has a 925 psi spring. If the pressure at the remote pilot reaches 925 psi, the dart unseats, and the pressure in the cavity above the skirt is limited to 925 psi. The main valve cracks open at 925 + 75 = 1000 psi. If pressure at the remote pilot location stays below 925 psi, then circuit pressure can continue to build until it reaches 1925 psi, the setting of the pilot built into the housing of the main valve. The main valve now cracks open at 1925 + 75 = 2000 psi. In this illustration, no information is given on where the remote pilot is connected in the circuit. We only know that it is someplace other than the main pressure line from the pump.

3.3.3 Example Circuits Using Pilot-Operated Relief Valves

A pilot-operated relief valve can be used to unload the pump at low pressure during periods between work cycles. The schematic in Fig. 3.10 shows a solenoid-actuated directional control valve connected to the remote pilot port on the side of the valve. In the position shown, the port is connected to the reservoir. The pilot spring cavity is vented, and the main relief valve opens at

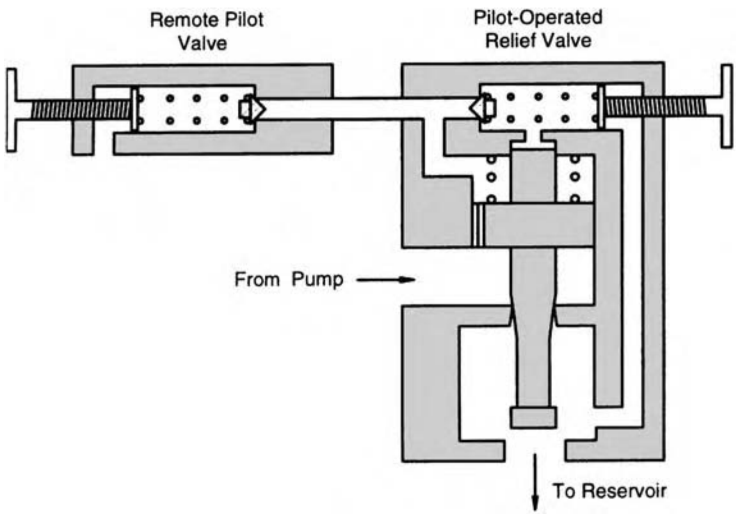


FIGURE 3.9
Pilot-operated relief valve with remote pilot. (Reprinted with permission from Parker Hannifin Corp.)

the main spring setting. In the shifted position, the directional control valve blocks the port, and the pilot-operated valve operates as previously described. Repeating the description, when the control valve is shifted to the left (port blocked), the integral pilot relief valve is operational, and the main relief valve acts like a pilot-operated relief valve. In the unactuated position, the port is connected to the reservoir, and the main valve is held closed by the 75 psi spring only. When pressure increases above 75 psi, the valve opens, thus the pump builds only 75 psi pressure during the periods between work cycles.

A diagram of a circuit that uses a pilot-operated relief valve to unload the pump at low pressure is shown in Fig. 3.11. The relief valve symbol designated with the letter "A" refers to the main spool of the pilot-operated relief valve. The orifice in the skirt is orifice B, and the symbol designated with letter "C" is a symbol that shows that the valve is held closed with spring pressure and a pilot pressure. The relief valve symbol designated with a letter "D" refers to the pilot stage of the valve (dart held in place with the pilot spring). The circuit operates in the following manner. When the four-way, three-position directional control valve is shifted to extend (or retract) the cylinder, the three-way, two-position directional control valve is simultaneously shifted. The pilot-operated relief valve is thus set to open at the pilot spring setting plus the main spring setting, 1925 + 75 = 2000 psi. When the three-position directional control valve shifts back to the center position, the two-position directional control valve shifts to connect the pilot pressure line to the reservoir. The main relief valve now opens at 75 psi.

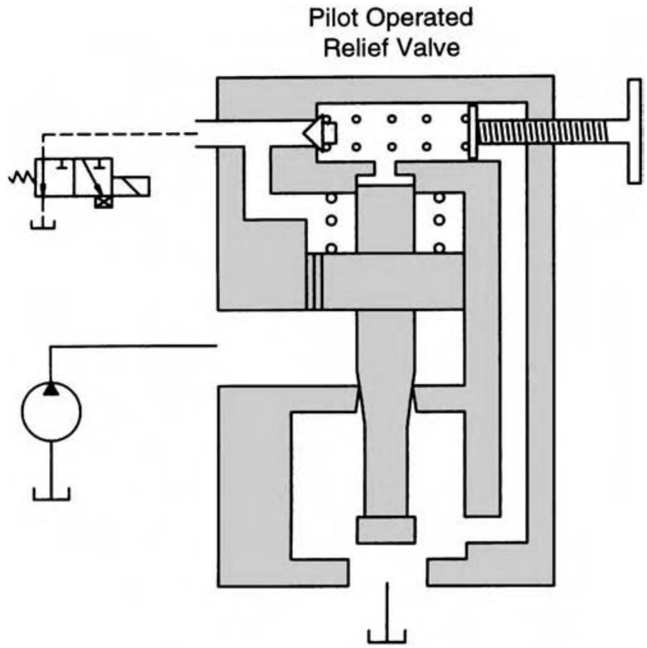


FIGURE 3.10
Two-way, two-position directional control valve connected to remote pilot location on pilot-operated relief valve. (Reprinted with permission from Parker Hannifin Corp.)

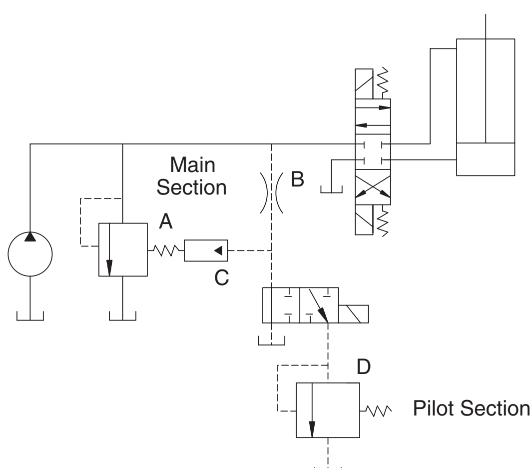


FIGURE 3.11
Use of pilot-operated relief valve to unload pump at low pressure.

When drawing circuit diagrams, a designer will often use a simplified symbol to designate a pilot-operated relief valve. The complete symbol (components A, B, C, and D in Fig. 3.11) is used only when it is needed to explain circuit operation.

A second method for using the pilot-operated relief valve to unload the pump between work cycles is shown in Fig. 3.12. Here, a special directional control valve is used with a fifth port. This port provides a pathway for the pilot line to be connected to the reservoir when the directional control valve is centered. When the directional control valve is shifted, the pilot line is blocked, and the pilot-operated relief valve will not open until the pressure equals the pilot spring pressure plus the main spring pressure.

The circuit in Fig. 3.13 is designed to provide high-pressure relief during extension and low-pressure relief during retraction. (The functional requirements of the circuit are unexplained at this point. The low-pressure relief may be needed to prevent damage or injury, if the workpiece strikes an obstruction during retraction.) Use of the pilot-operated relief valve is similar to the use in Fig. 3.11. The check valve is held in place by high pressure, thus blocking the remote pilot connection on the pilot-operated relief valve. The valve then opens at the high-pressure setting (pilot spring + main spring pressure). During return, the pilot line is connected to the reservoir (as shown in Fig. 3.10); therefore, the valve opens at the low-pressure setting (main spring pressure). It is understood that the low-pressure setting must be high enough to provide the force needed for normal retraction.

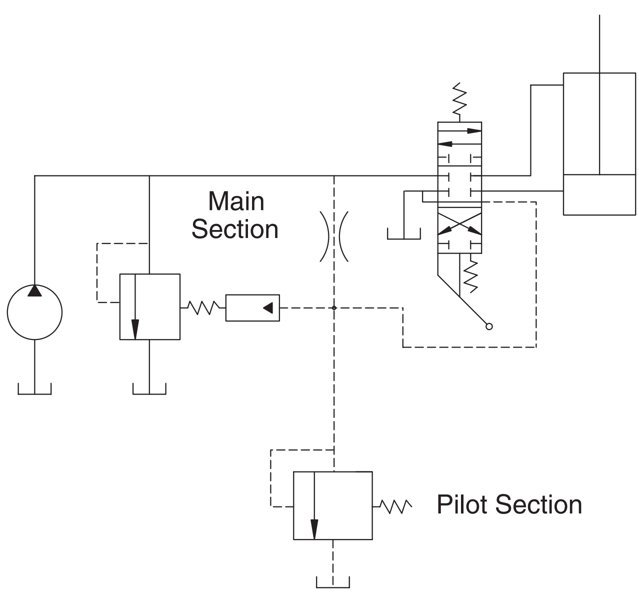


FIGURE 3.12
Circuit showing pilot-operated relief being vented through a directional control valve with a fifth port.

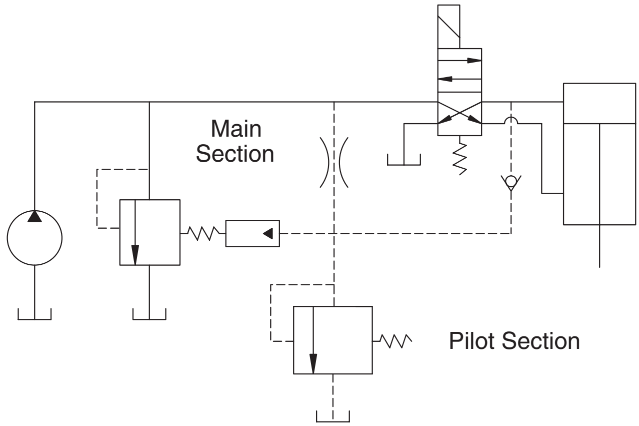


FIGURE 3.13
Use of pilot-operated relief valve to provide high-pressure relief during extension and low-pressure relief during retraction.

3.4 Unloading Valve

The symbol for an unloading valve is similar to the symbol for a relief valve except that the pilot line is not connected to sense pressure at the valve inlet. The two symbols are compared in Fig. 3.14.

A circuit with an unloading valve is shown in Fig. 3.15. It is necessary to first discuss the operation of this circuit and understand the *function* of the unloading valve before studying the operation of the valve.

The accumulator is a key component in the Fig. 3.15 circuit. As mentioned in Chapter 2, there are three types of accumulators: bladder, diaphragm, and

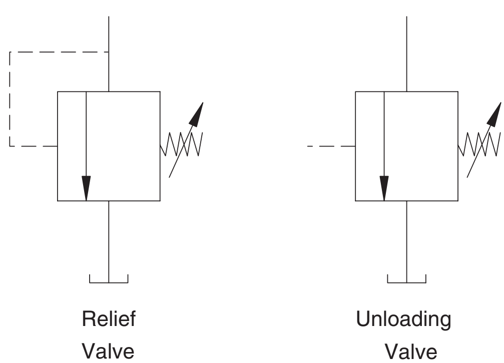


FIGURE 3.14
Comparison of symbols for relief valve and unloading valve.

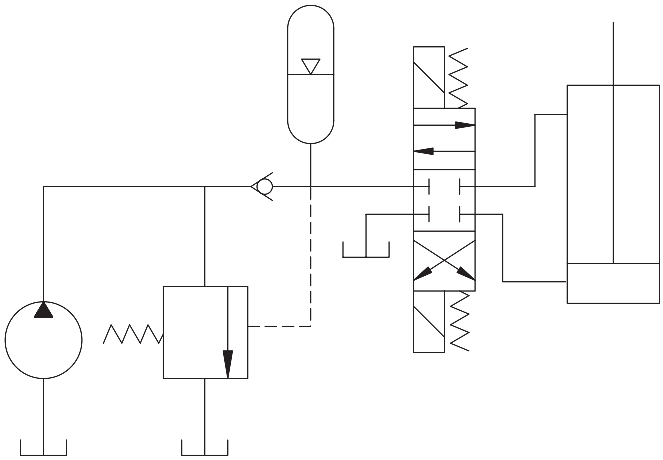


FIGURE 3.15
Circuit illustrating use of unloading valve to unload pump at low pressure between cycles of cylinder.

piston. The diaphragm accumulator is a pressure vessel divided into two compartments by a flexible diaphragm. The top half is precharged with a gas, generally nitrogen, and sealed. The bottom half is connected to the hydraulic circuit. Fluid is pumped through the check valve into the bottom of the accumulator. The bottom half is filled, and extra fluid is pumped in as the diaphragm bulges upward. Pressure increases as the gas is compressed. The accumulator is designed for some rated pressure, and pressure must be controlled to ensure that it does not exceed this rating. The unloading valve accomplishes this task.

An accumulator provides pressure to the actuator (cylinder in Fig. 3.15) at the moment the directional control valve is shifted. Pressure does not have to build from a low pressure as it does in a circuit with a open-center directional control valve (See Fig. 3.1.). Often, acceleration of the load is a significant issue in circuit design. In most manufacturing applications, profitability is increased when the number of cycles per unit time is increased. The cylinder must extend and retract as quickly as possible. If a large load is being moved, pressure must build to achieve enough force to overcome static friction and provide the inertial force to accelerate the mass. It takes an interval of time to build this pressure. This time delay can be eliminated if pressure is already available at the moment the directional control valve is shifted.

An accumulator also provides another important feature. The extra fluid stored in the accumulator allows the desired actuator speed to be achieved with a smaller displacement and, therefore, a lower-cost pump. An example will illustrate how this can be an advantage.

The actuator cycle has a 5-second active part and 20-second passive part, meaning that flow is needed for 5 seconds, and there is a 20-second interval before flow is needed again. The accumulator can supply 924 in^3 of oil. If it supplies this fluid during the active part of the cycle, it must be recharged during the passive part. Pump flow to accomplish the recharge is

$$Q_p = 924 \text{ in}^3/20\text{s} = 46.2 \text{ in}^3/\text{s}$$

$$= 12 \text{ GPM}$$

If the accumulator rating is 1000 psi, maximum power required for the pump is

$$\mathcal{P}_1 = \frac{PQ}{1714}$$

$$= \frac{1000 \times 12}{1714} = 7 \text{ hp}$$

Suppose no accumulator is used. The pump must be large enough to supply 924 in^3 in 5 s.

$$Q = 924 \text{ in}^3/5\text{s} = 184.8 \text{ in}^3/\text{s}$$

$$= 48 \text{ GPM}$$

To supply this flow at a maximum pressure of 1000 psi, the peak power required for the pump is

$$\mathcal{P}_2 = \frac{PQ}{1714}$$

$$= \frac{1000 \times 48}{1714} = 28 \text{ hp}$$

The pump and prime mover must be four times larger if the accumulator is not included in the circuit.

The analysis done for this example does not include all factors. Design of a circuit with an accumulator requires knowledge of the pressure vs. volume curve as the accumulator is being filled.

In the circuit in Fig. 3.15, the pump builds pressure in the accumulator until the setting of the unloading valve is reached. At this point, the unloading valve opens, and flow bypasses to the reservoir. The pressurized fluid is trapped in the accumulator by the check valve and the closed-center directional control valve.

A functional diagram of an unloading valve is shown in Fig. 3.16. Two features are added to a pilot-operated relief valve to create the unloading valve.

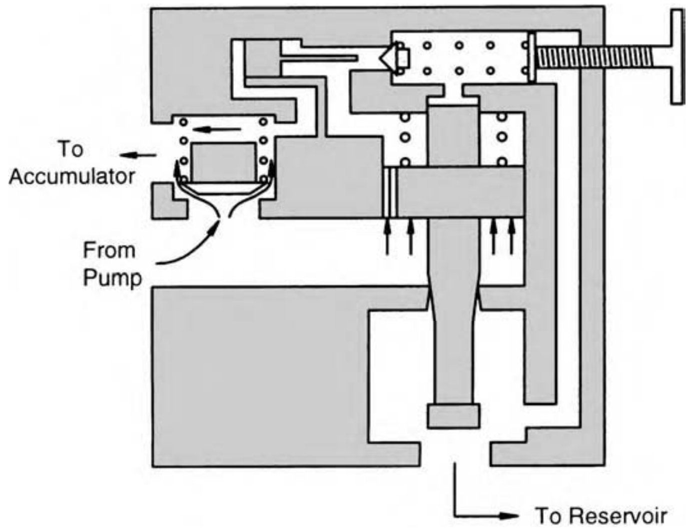


FIGURE 3.16
Functional diagram of an unloading valve showing flow when accumulator is being charged. (Reprinted with permission from Parker Hannifin Corp.)

A check valve is built in, and a small piston is included in the top section in line with the dart and pilot spring. When the unloading valve is closed, fluid flows through the check valve to charge the accumulator.

As with the pilot-operated relief valve, it is helpful to assign values to the springs. For our discussion, we assume the accumulator has a 1000 psi rating. The pilot spring is assigned a value of 975 psi, and the spool spring is assigned a value of 25 psi. When pressure reaches 975 psi, the dart is unseated, allowing fluid to flow through the internal drain to the reservoir. Pressure in the upper chamber cannot increase above 975 psi. The spool is held in place by the 25-psi spring. The small piston has balanced hydraulic forces, because the same pressure acts on both sides. The projected areas of both sides are equal, thus the hydraulic forces are equal.

As pressure continues to build and reaches 975 + 25 = 1000 psi, the unloading valve opens and flow bypasses to the reservoir. The pressure drops, and the check valve closes. Pressure on the accumulator side of the piston pushes it to the right (Fig. 3.17), where the rod pushes the dart off its seat. As long as the dart is held off its seat, the unloading valve is vented, and the pump is unloaded at 25 psi, the pressure required to compress the main spool spring.

When the directional control valve is shifted, fluid drains from the accumulator and the pressure drops. The hydraulic force on the piston drops and, when the pilot spring force becomes greater, the dart reseats. At this point, pressure equalizes on both sides of the spool skirt. The spool spring reseats the spool and the pump begins to build pressure.

Pressure drops at the directional control valve as the accumulator empties. Simultaneously, the pump is building pressure. The resulting pressure vs.

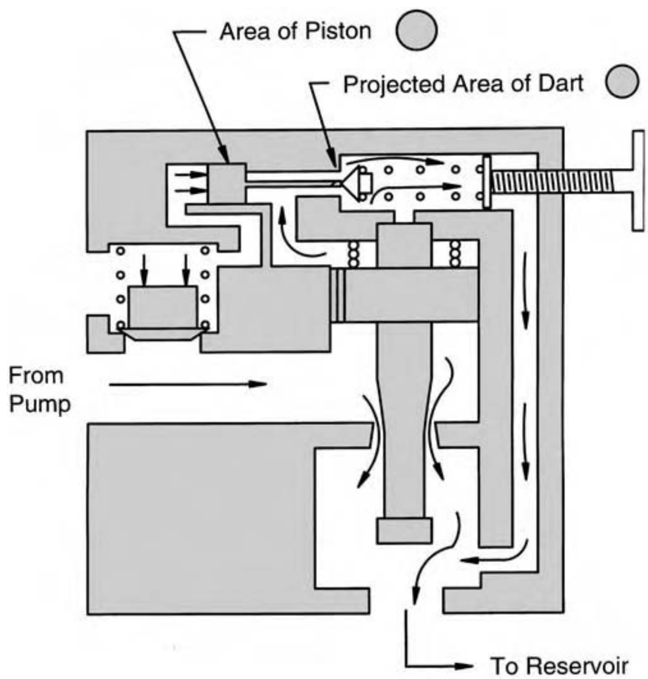


FIGURE 3.17
Schematic of an unloading valve showing flow when valve opens. (Reprinted with permission from Parker Hannifin Corp.)

time curve has a shape as shown in Fig. 3.18. The minimum pressure is a function of the load, the characteristics of the accumulator, and the characteristics of the pump.

There is a type of unloading valve identified as a *differential* unloading valve. This valve is designed to allow the accumulator to partially discharge before the valve is unvented. Generally, the valve is designed to unvent when the pressure drops 15%. Differential unloading valves are also available that unvent when pressure drops 30%. *Unventing* means that the piston moves enough to allow the dart to reseat. Once this occurs, the unloading valve functions like a pilot-operated relief valve.

Referencing Fig. 3.17, consider the moment when pressure just reaches 975 psi, and the dart just begins to unseat. Once it cracks open, the pressure in the top chamber cannot increase above 975 psi, so the pressure on one side of the piston (right side in Fig. 3.17) cannot increase above 975 psi. The pressure on the left side can continue to increase up to 1000 psi. This pressure difference causes a force imbalance, so the piston moves to the right, the rod unseats the dart, and the valve is vented.

In a differential unloading valve, the area of the piston is 15% greater than the projected area of the dart. This means that the hydraulic force holding the dart unseated is 15% greater than the hydraulic force that initially unseated

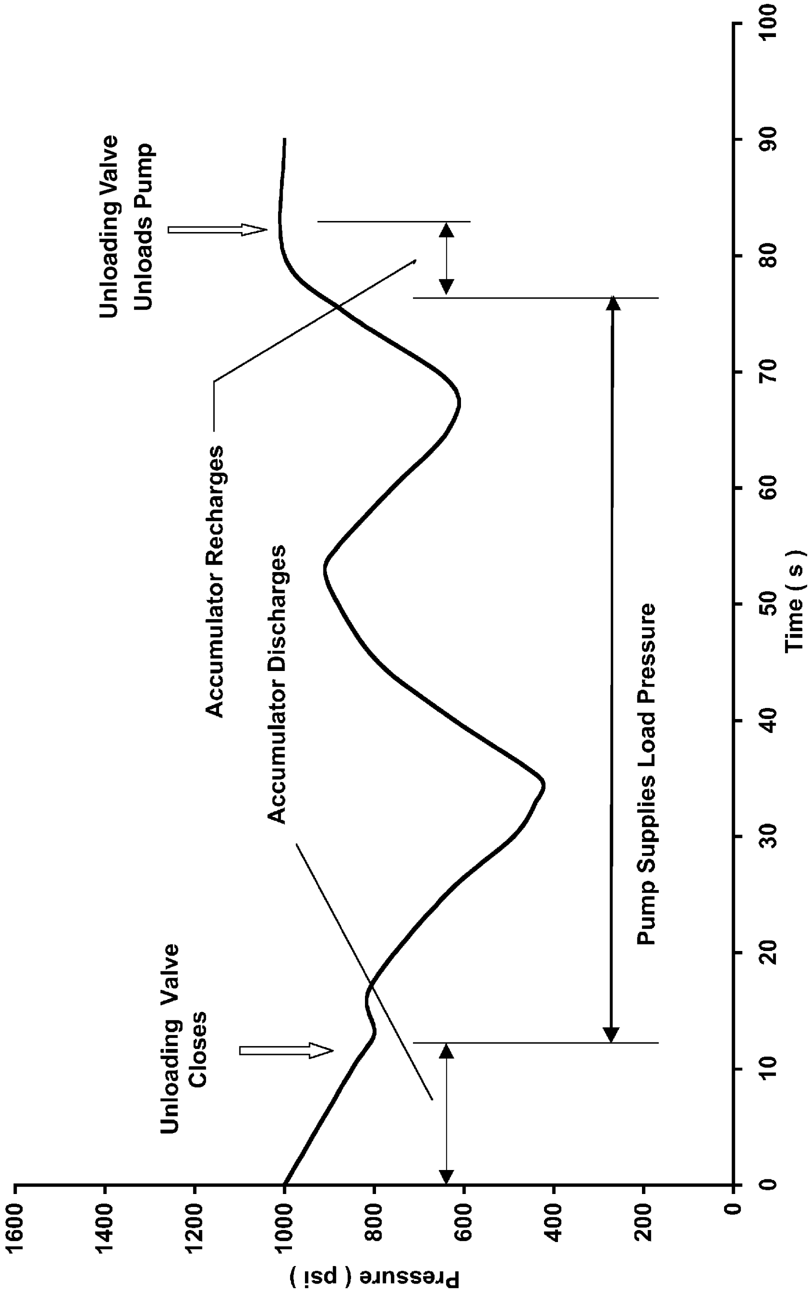


FIGURE 3.18
Typical pressure curve for circuit in Fig. 3.15.

the dart. Pressure on the accumulator side must drop 15% below 975 psi before the dart spring can reseat the dart. The force balance is

$$P_a A_d (1 + 0.15) = P_d A_d \tag{3.9}$$

where P_a = pressure on accumulator side (psi)
A_d = projected area of dart (in^2)
P_d = pressure on dart side (psi)
$P_a = P_d/1.15$
$= 975/1.15 = 848$ psi

Pressure on the accumulator side must drop to 848 psi before the valve is unvented.

The symbol used for a differential unloading valve is shown in Fig. 3.19. The vent line connected downstream from the check valve denotes the function of the piston. An accumulator is shown with the symbol in Fig. 3.19 to clarify that the differential unloading valve works with an accumulator.

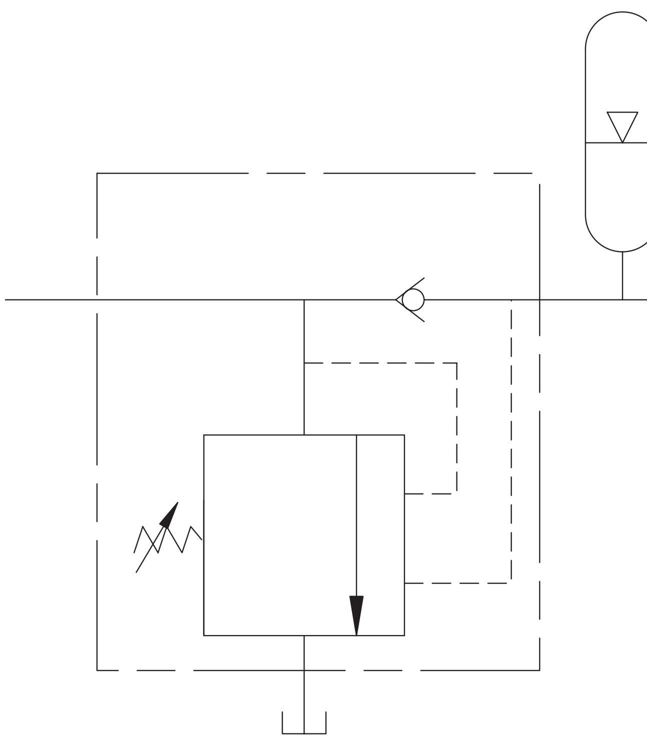


FIGURE 3.19
Symbol for differential unloading valve.

3.5 Sequence Valve and Pressure-Reducing Valve

The sequence valve and pressure-reducing valve have some similar features. It is instructive to compare their symbols side by side (Fig. 3.20). Both valves are externally drained, meaning that there is a separate line from the valve back to the reservoir. The relief valve and unloading valve are both internally drained. They have a passageway machined into the housing, which allows leakage from the spring cavity to flow to the outlet. Since the outlet is connected to the reservoir, a separate line is not needed. The outlets of the sequence and pressure-reducing valves are not connected to the reservoirs, thus they cannot be internally drained.

The check valve built into both valves is there to provide free flow in the reverse direction. In effect, it takes the valve out of the circuit when flow is reversed.

As shown in Fig. 3.20, the sequence valve and pressure-reducing valve symbols are similar. One (sequence valve) is a normally closed valve with pilot line to sense inlet pressure, and the other (pressure-reducing valve) is a normally open valve with pilot line to sense outlet pressure.

3.5.1 Sequence Valve

The sequence valve is used to ensure that a certain pressure level is achieved in one branch of the circuit before a second branch is activated. Consider a machining operation where the workpiece must be clamped with a certain force before it is extended to make contact with the cutting tool. If the piece is not securely fastened, it can slip and damage both the tool and the piece.

In the circuit shown in Fig. 3.21, the sequence valve is set on 600 psi, meaning that pressure must build to 600 psi before the valve opens. This setting ensures that the clamp cylinder exerts a 600-psi clamp force before the extend

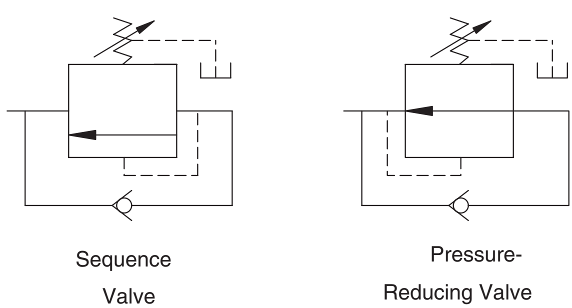


FIGURE 3.20
Symbols for sequence valve and pressure-reducing valve.

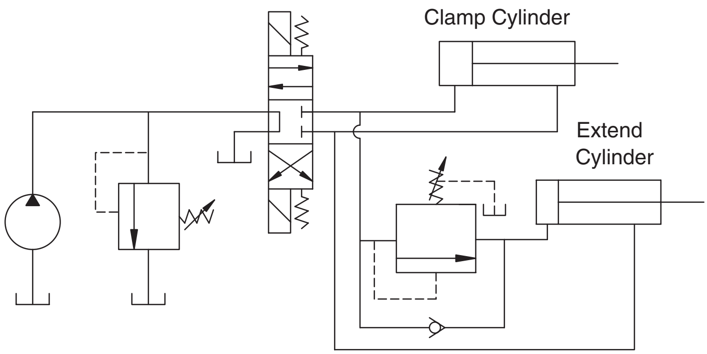


FIGURE 3.21
Circuit illustrating the use of a sequence valve.

cylinder moves. When the directional control valve is shifted for reverse flow, the check valve provides free flow, and there is no sequencing of the cylinders. Either one can retract before the other, depending on the pressure required for retraction. The cylinder with the lowest pressure requirement always retracts first.

Proper sizing of the cylinders will minimize energy loss in a sequence valve circuit. Suppose the maximum pressure to extend the workpiece in Fig. 3.21 is 400 psi. Pressure drop across the sequence valve is 600 – 400 = 200 psi. If a larger clamp cylinder is used, such that the pressure required to achieve the clamp force is only 500 psi, the sequence valve can be set at 500 psi. Pressure drop across the valve is 500 – 400 = 100 psi, and energy loss is reduced.

A functional diagram of a sequence valve with flow through to a primary circuit is shown in Fig. 3.22. When pressure at the inlet creates a hydraulic force large enough to offset the spring force, the spool shifts to open a passage to the secondary circuit and close the primary circuit.

3.5.2 Pressure-Reducing Valve

A pressure-reducing valve does not allow pressure downstream of the valve to exceed the set point. Suppose the workpiece must be clamped with two clamps. The second clamp is placed at a point where too much clamping force will damage the workpiece. A pressure-reducing valve is used to limit clamping pressure as shown in Fig. 3.23.

Suppose the valve in Fig. 3.23 is set on 500 psi. If pressure at the outlet of the valve increases above 500 psi, the pressure-reducing valve partially closes to create an orifice. Pressure drop across this orifice reduces the downstream pressure to 500 psi.

A functional diagram of the basic pressure-reducing valve was shown in Fig. 2.19. A second type of pressure-reducing valve, the reducing/relieving valve, is shown in Fig. 3.24. This valve operates like the pressure-reducing

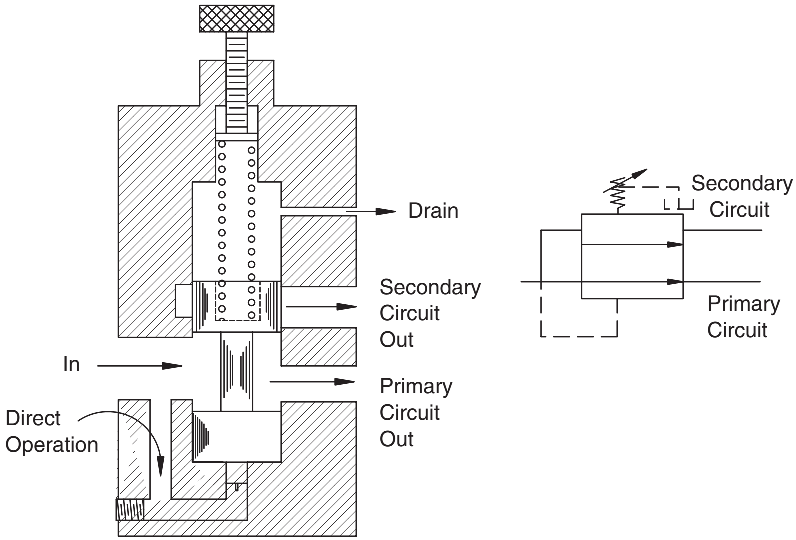


FIGURE 3.22
Functional diagram of sequence valve with flow-through to primary circuit.

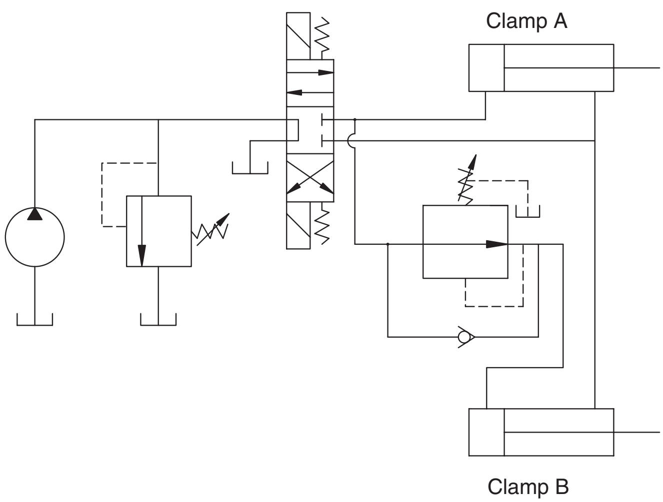


FIGURE 3.23
Pressure-reducing valve used to limit clamping force.

valve except that it bypasses fluid to the reservoir when the spool is shifted upward by the hydraulic force. The orifice between the inlet and bypass (to reservoir) opens as pressure increases, thus the valve functions like a relief valve. It combines the functions of the pressure-reducing valve and the relief valve, thus the name *reducing/relieving* valve. Note that the reducing/reliev-

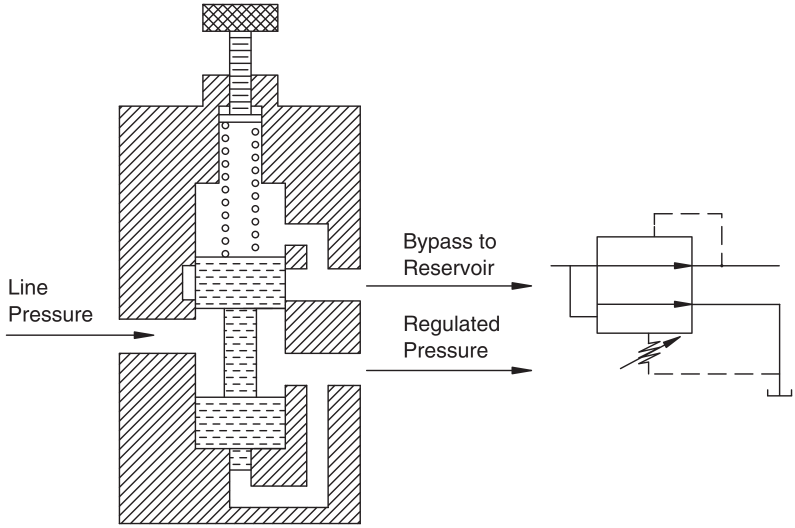


FIGURE 3.24
Schematic of pressure-reducing/relieving valve.

ing valve is internally drained, but the reducing valve (Fig. 2.19) has to be externally drained.

As an additional example of pressure-reducing valve operation, consider Fig. 3.25. In this case, the rotary actuator is turning a screw to tighten a connection. If too much torque is applied, the threads will be damaged. Torque is given by

$$T = \frac{PV_{ra}}{2\pi} \tag{3.10}$$

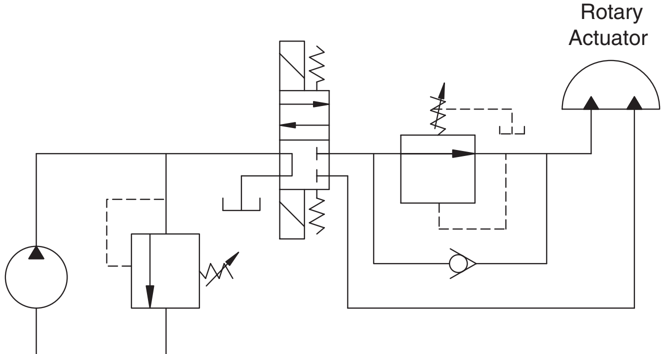


FIGURE 3.25
Use of pressure-reducing valve to limit torque developed by a rotary actuator.

where T = torque (lb_f-in)
P = pressure (psi)
V_{ra} = displacement of rotary actuator (in^3/rev)

Limiting pressure limits the torque. A description of rotary actuators is given in Chapter 5.

3.6 Counterbalance Valve and Brake Valve

The counterbalance valve and brake valve have similar symbols (Fig. 3.26). The single difference is that the brake valve has a remote pilot line in addition to an upstream pilot line. As with the sequence and pressure-reducing valves, a check valve is built in to allow free flow in the reverse direction.

A couple of definitions are required before continuing our discussion of counterbalance and brake valves.

Resistive load: A load that acts in the opposite direction to actuator motion.

Overrunning load: A load that acts in the same direction as actuator motion.

3.6.1 Counterbalance Valve

The counterbalance valve, also called a holding valve, is used to prevent a weight from falling uncontrollably. As an example, consider a press with a two-ton platen (Fig. 3.27). When the directional control is shifted, the platen will fall unless there is a means for creating an opposing hydraulic force. A counterbalance valve accomplishes this task.

Suppose the cylinder in Fig. 3.27 has a six-inch bore and 2:1 area ratio. What pressure setting should be used for the counterbalance valve? The area of the rod end is 14.1 in^2. Pressure to hold a two-ton platen is

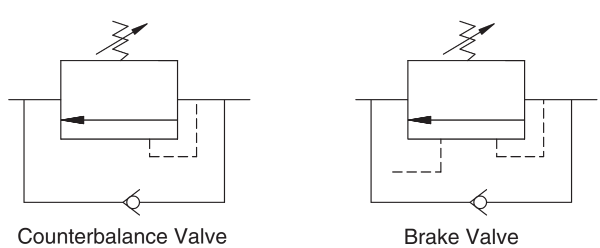


FIGURE 3.26
Symbols for counterbalance and brake valves.

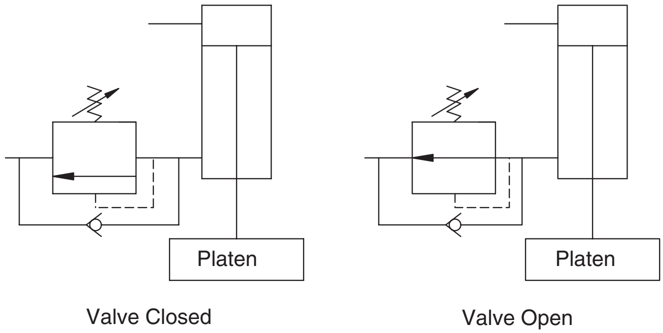


FIGURE 3.27
Use of counterbalance valve to prevent uncontrolled descent of platen.

$$\begin{aligned} P &= F/A_r \\ &= 2 \times 2000/14.1 \\ &= 284 \text{ psi} \end{aligned} \tag{3.11}$$

Generally, the valve is set about 50 psi higher than the pressure required to hold the load. A setting of 340 psi is selected.

What pressure must be developed to extend the cylinder? The hydraulic force on the rod end is

$$\begin{aligned} F_r &= PA_r \\ &= 340(14.1) \\ &= 4790 \text{ lb}_f \end{aligned} \tag{3.12}$$

The force that must be developed on the cap end before the platen begins to descend is

$$\begin{aligned} F_c &= F_r - F_1 \\ &= 4790 - 4000 = 790 \text{ lb}_f \end{aligned} \tag{3.13}$$

The required pressure is

$$\begin{aligned} P_c &= F_c/A_C \\ &= 790/(14.1 \times 2) \\ &= 28 \text{ psi} \end{aligned} \tag{3.14}$$

Any pressure above 28 psi causes the platen to move downward. Note that the cylinder friction force was not considered in this problem.

Often, it is desirable to use the weight of the platen to help build the total pressing force. In this case, a counterbalance valve with remote pilot line connection is used (Fig. 3.28). If pressure at the cap end, and thus the pilot line, drops below the set point, the counterbalance valve partly closes to slow the platen.

There is an additional reason for using a counterbalance valve other than to prevent uncontrolled motion of the actuator. When the platen is moving down too fast, the pump cannot keep the cap end filled with fluid, and a negative pressure can develop. It is possible to *suck* the fluid out of the pump fast enough to cause void spaces in the pump. The condition when the pump is not completely filled with fluid is called c*avitation*. One of the problems caused by cavitation is a breakdown of the lubrication film between moving parts. A higher pressure setting on the counterbalance valve is needed to achieve the desired counterbalance and limit actuator speed.

3.6.2 Brake Valve

A brake valve performs the same function as a counterbalance valve, but it is designed to overcome a key disadvantage. An example will illustrate this point. Suppose a direct-operated counterbalance valve is used in a motor circuit. There are times when the motor load can overrun, and braking is needed for those times. This valve is set at 350 psi, and the pressure drop across the motor to supply required torque is 350 psi. The pump must develop 350 + 350 = 700 psi. The pressure drop across the counterbalance valve is converted to heat; consequently, half the hydraulic power is wasted. A brake valve overcomes this disadvantage.

As shown in the functional diagram in Fig. 3.29, a brake valve has an internal pilot passage and a remote pilot passage. Suppose the spring is set for

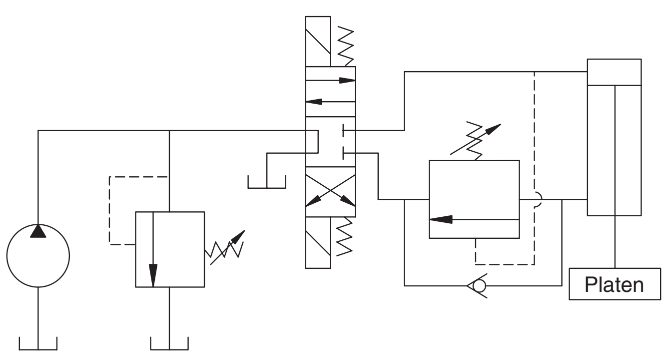


FIGURE 3.28
Counterbalance valve with remote pilot line.

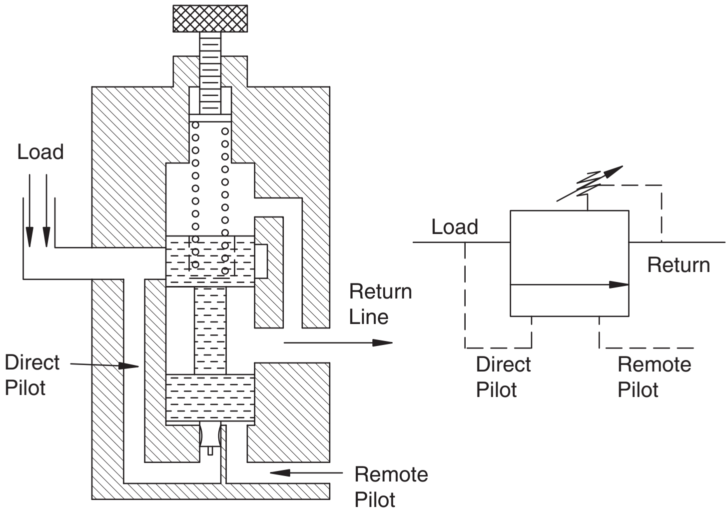


FIGURE 3.29
Brake valve functional diagrams showing construction required for valve to open at different internal and remote pilot pressures.

1000 psi. When pressure at the internal pilot reaches 1000 psi, the piston pushes the spool upward to open the valve.

The area of the piston is much less than the area of the bottom of the spool. A typical area ratio might be 10:1. The remote pilot applies pressure directly to the bottom of the spool; consequently, only 100 psi is required to compress the spring and open the valve. Pressure required to open the valve is 1000 psi at the internal pilot and 100 psi at the remote pilot.

A brake valve is used in a circuit as shown in Fig. 3.30. It requires 100 psi at the motor inlet to keep the valve open. As long as the load on the motor requires more than 100 psi, the brake valve does not affect circuit efficiency. If the load starts to overrun, and the pressure drops below 100 psi, the brake valve closes. It requires 1000 psi at the direct (or internal) pilot (Fig. 3.29) to open the valve. This 1000-psi pressure drop across the brake valve converts the mechanical energy of the overrunning load to heat energy and slows the load. When pressure at the inlet builds to 100 psi again, the brake valve opens.

3.7 Summary

Pressure control limits the force/torque produced by an actuator. Because of the high power density of a fluid power circuit, pressure control is a critical safety, as well as functional, issue.

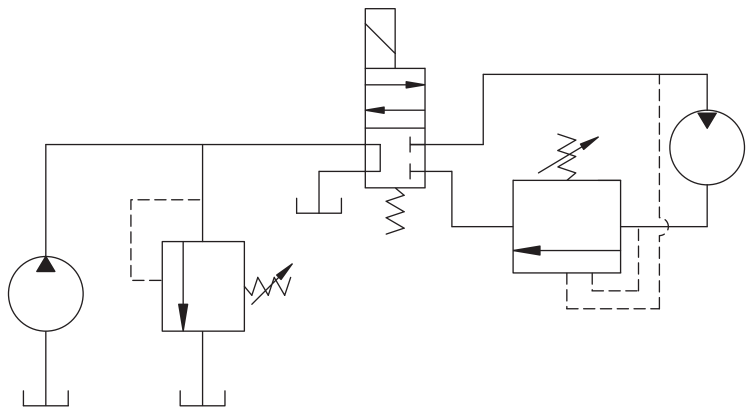


FIGURE 3.30
Brake valve used in a circuit.

Six pressure-control valves (relief, unloading, sequence, pressure reducing, counterbalance, and brake) were discussed in this chapter. All work on the principle of a hydraulic force opposed by a spring force. When pressure builds to the point where the hydraulic force is greater than the spring force, the valve spool is shifted. The relief, unloading, sequence, counterbalance, and brake valves are all normally-closed valves. Pressure rise at the valve *inlet* causes it to open. The pressure-reducing valve is a normally-open valve. Pressure rise at the valve *outlet* causes it to close.

There is always some leakage past the spool into the spring cavity. If this leakage is not drained, the valve will not function properly. In some instances, it will not open at all, and no control of pressure is achieved. If the valve outlet connects directly to the reservoir, it can be internally drained, meaning that a passage is machined in the valve body between the spring cavity and the valve outlet. Externally drained valves have a separate line from the spring cavity to the reservoir. Sometimes, a return line can be used for the drain line. However, if there is an obstruction in the return line, for example a return-line filter, then a separate drain line is required.

Pressure control is a key safety issue. Correct selection and functioning of the pressure control elements cannot be overemphasized.

Problems

3.1 A backhoe operator controls the speed of the cylinder, which extends the bucket by cracking open the DCV controlling flow of the cylinder. Describe the difference in the "feel" of the controls using the two different relief valves (a) and (b) shown in Fig. 3.31.

3.2 A 1.79-in^3 displacement motor is turning at 804 rpm and delivering 340 lb_f-in of torque. The operator notices that motor speed

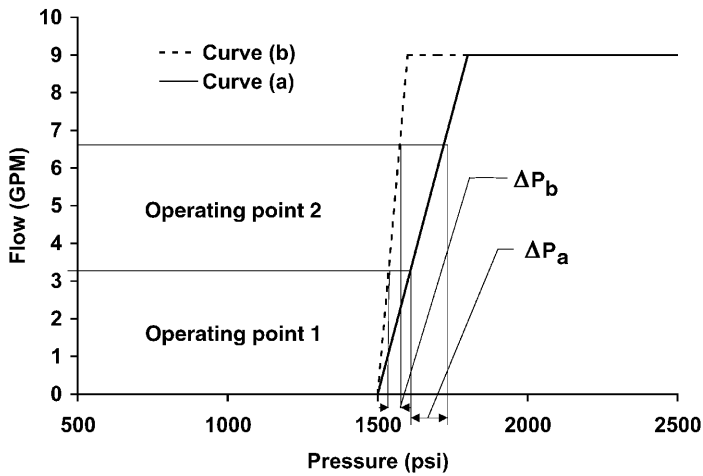


FIGURE 3.31
Characteristics of two different relief valves used for the bucket extend cylinder on a backhoe (Problem 3.1).

decreases as load increases and the torque demand is greater than 340 lb_f-in. The operator collected the following data:

Torque Output (lb_f–in)	Speed (rpm)
340	804
356	687
370	536
385	302
399	67

Use this data to deduce the shape of the relief valve curve. Plot this curve [flow across relief valve (in^3/s) vs. pressure (psi)].

3.3 A fixed-displacement pump is delivering 3.5 GPM to a circuit such as the one shown in Fig. 3.21. The workpiece must be held by the clamp cylinder with a minimum force equivalent to 600 psi during the operation. It has been determined that a clamping force equivalent to 750 psi will damage the workpiece.

Pressure at the cap end of the extend cylinder was measured during extension. An idealized plot of this pressure is shown in Fig. 3.32. (No data are presented for the retraction part of the cycle.)

a. Will this circuit operate satisfactorily? If so, briefly explain how all functional requirements are met.

b. Calculate the hydraulic energy (hp-h) dissipated across the sequence valve during extension. Remember energy is power × time.

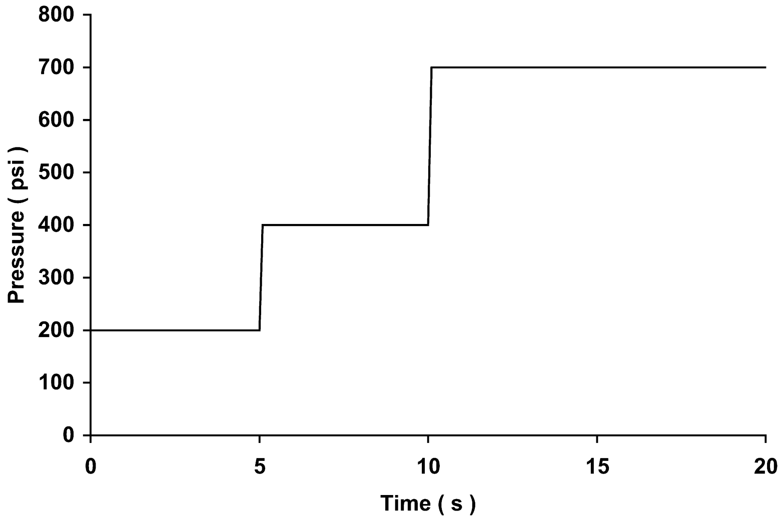


FIGURE 3.32
Pressure measured during extension (Problem 3.3).

3.4 A peanut shelling plant uses a truck dump to unload truck-trailers loaded with peanuts. The truck is backed onto a platform, and the entire vehicle (tractor and trailer) is raised to pour the peanuts out through the back unloading gate. The hose on the cap end of one lift cylinder failed at the fitting, and this caused the entire load to be held by the other cylinder. The resulting high pressure caused the hose on this cylinder to burst, and a $105,000 truck was destroyed.

You are assigned the task of redesigning the hydraulic circuit such that a similar accident could never happen again. Your circuit is shown in Fig. 3.33. A counterbalance valve is installed directly on the cap end port of the cylinder. There is no line (hose or hydraulic tubing) between the port and the valve.

For simplicity, there is only one cylinder shown. The actual design has two cylinders with a counterbalance valve on each one.

Given:

Cylinders	8-in. bore
	5-in. rod diameter
Maximum lift force	100 tons

a. Find the setting on the counterbalance valve (psi) that will ensure that the load can never fall. Valve setting should be 10%

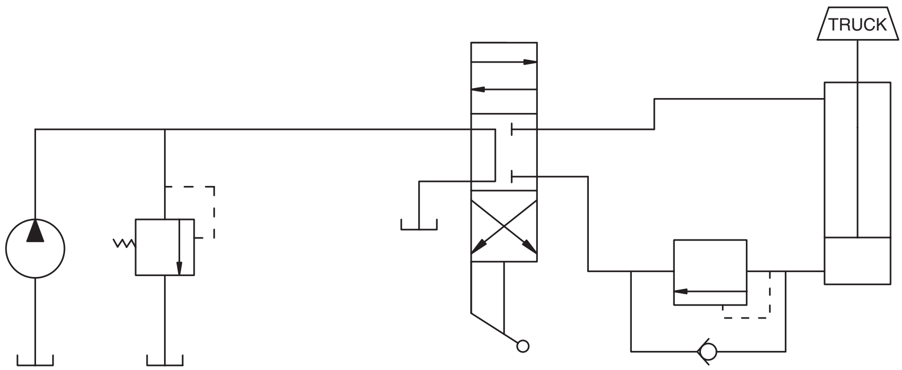


FIGURE 3.33
Circuit showing counterbalance valve mounted on lift cylinder (Problem 3.4).

higher than the pressure produced by the expected maximum load.

b. What pressure must be developed at the rod end of the cylinder to lower the load?

3.5 The stroke of the cylinders in Problem 3.4 is 18 ft. The design states that the truck must be lowered in 10 min. The counterbalance valve is a cartridge mounted in an aluminum block known as a *line body.* Total mass of the counterbalance valve is 9.4 lb. The pressure setting of the valve (inlet pressure) is 2200 psi, and the pressure drop from the outlet to the reservoir is 50 psi. Use the following assumptions and calculate the temperature rise of the counterbalance valve during a retraction event.

a. The valve is mounted on the cylinder. 20% of the total heat generated is conducted to the cylinder.

b. 40% of the total heat generated is exchanged to the oil as it passes through the valve.

c. 30% of the total heat is convected or radiated to the surrounding atmosphere during the event.

Specific heat of aluminum is 0.214 Btu/$lb_m \cdot$ F. 1 hp = 2547 Btu/h.

Hint: Remember that hydraulic power is converted to heat energy when there is a pressure drop and no mechanical work is delivered.

4

Creation and Control of Fluid Flow

4.1 Introduction

The delivery of fluid power is based on the movement of pressurized fluid. A clamping cylinder can hold a workpiece by building pressure to a certain level. After the clamp closes, a force is applied, but no work is delivered to the workpiece. Contrast this example with the extension of a cylinder to move a load. When the load is contacted, pressure builds to overcome the static friction and inertia, and the load begins to move. The initiation of movement is the beginning of work delivery.

The primary flow control device in any circuit is the pump. The pump converts mechanical power to fluid power. Pumps can be classified into two general classifications: fixed displacement and variable displacement units.

Theoretically, a fixed displacement pump delivers a fixed volume of fluid for each revolution. This volume is defined as the *displacement* and is generally reported as in^3/rev. Manufacturers publish their pump displacements based on tests run at no-load. (There is always some measurable pressure at the pump outlet, but the test conditions are selected to minimize this pressure.) The pump is driven at a given rpm, and the flow is measured with a flow meter. Pump displacement is defined by

$$\frac{\text{in}^3/\text{min}}{\text{rev/min}} = \text{in}^3/\text{rev}$$

Variable displacement pumps are designed such that the output of fluid per revolution (displacement) can be changed as the pump is operating. Flow can be continuously varied from zero to maximum flow.

Flow from all pumps is a function of pressure. As pressure increases, leakage through the clearances between moving and stationary parts increases, and the output flow decreases. A flow vs. pressure curve is the key performance data for any pump. There is a trend toward higher operating pressures, and pump designs have been refined to reduce leakage at higher pressures.

Pump volumetric efficiency is defined as

$$e_{vp} = \frac{Q_a}{Q_t} \tag{4.1}$$

where Q_a = actual output (in^3/min)
Q_t = theoretical output (in^3/min)

Theoretical flow is the flow based on the displacement published by the manufacturer. For example, suppose a pump has a displacement of 0.9 in^3/rev and is driven at 2000 rpm. The theoretical flow is

$$Q_t = NV_{pth} \tag{4.2}$$

where N = pump speed (rpm)
V_{pth} = pump displacement (in^3/rev)

$$Q_t = 0.9 \times 2000 = 1800 \text{ in}^3/\text{min}$$

If actual flow is 1500 in^3/min, pump volumetric efficiency is

$$e_{vp} = \frac{1500}{1800} \times 100 = 83.3\% \tag{4.3}$$

Overall pump efficiency is reported as

$$e_{op} = \frac{\text{Output power}}{\text{Input power}} \tag{4.4}$$

Output power is, of course, hydraulic power, and input power is mechanical power. Suppose the input torque, measured with a torque transducer, is 140 lb_f-in for the pump in the previous examples.

Input power is

$$\mathcal{P}_{in} = \frac{TN}{63,025} = \frac{140 \times 2000}{63,025} = 4.4 \text{ hp} \tag{4.5}$$

Output power is

$$\mathcal{P}_{out} = \frac{PQ/231}{1714} \tag{4.6}$$

where P = pressure (psi)
Q = flow (in^3/min)

$$\mathcal{P}_{out} = \frac{1000(1500)/231}{1714} = 3.8 \text{ hp} \tag{4.7}$$

Overall efficiency is

$$e_{op} = \frac{3.8}{4.4} \times 100 = 85.5\% \tag{4.8}$$

Pump volumetric efficiency and overall efficiency both decrease as pressure increases. The rate of decrease depends on pump design.

4.2 Fixed Displacement Pumps

Vane and piston pumps are available as fixed or variable displacement pumps. If the cam ring is fixed in position (vane pump), or the swashplate is fixed in position (piston pump), the displacement is fixed. Gear pumps, however, are available only with a fixed displacement. These pumps are less expensive, so they are widely used.

As will be seen in the next section, the leakage in some gear pumps is high, and this leakage increases as the pump wears over time. Ownership cost of fixed-clearance gear pumps is lower, but operating cost is higher over the life of the pump. Total cost (ownership + operating) for the entire design life may or may not be lower for a pump with lower initial cost.

Pressure-balanced gear pumps have volumetric efficiencies that exceed, or at least, rival those of piston pumps. Some of these gear pumps are rated for pressures in excess of 4000 psi.

An external gear pump is shown in Fig. 4.1. One gear is driven with the input shaft, and it drives the second gear. Fluid is captured by the teeth as they pass the inlet, and this oil travels around the housing and exits at the outlet. The design is simple, and it is inexpensive. It is also apparent that there are opportunities for leakage all along the housing as the fluid travels from the inlet to the outlet.

A gear pump design is available with a moveable side plate. When this side plate is out, the fluid is not confined, and no pressure is built. When it is pressed in against the rotating gear, the fluid is confined, and pressure builds. These pumps are used on over-the-road trucks. They are driven by an auxiliary shaft from the engine referred to as the *power-takeoff (PTO)* shaft. There is no clutch, so the PTO) shaft turns whenever the engine turns. The pump is engaged (pumps fluid) when the side plate is pressed in against the rotating gear.

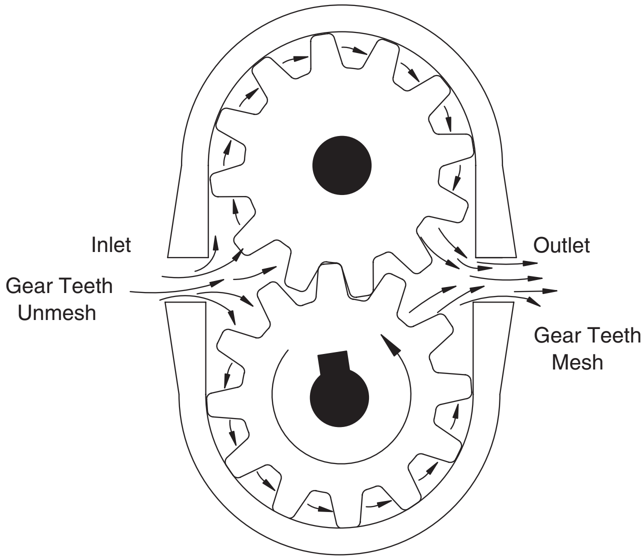


FIGURE 4.1
Diagram of external gear pump.

A gerotor (gear) pump has an inner drive gear and an outer driven gear (Fig. 4.2). As the inner gear rotates, it drives the outer gear. The inner gear has one less tooth than the outer gear, and this feature creates chambers of decreasing volume, and thus the "pumping action." A port plate ensures that fluid enters the chamber when it is largest and exits when it is smallest.

4.3 Fixed Displacement Pump Circuits

A simple circuit with fixed displacement pump is shown in Fig. 4.3. Circuits with fixed displacement pumps are called *constant-flow* circuits. The key concept is stated below.

> Each revolution of a fixed displacement pump delivers a certain volume of fluid to the circuit. This fluid ultimately returns to the reservoir, either as a return flow, or a leakage flow.

In the circuit shown in Fig. 4.3, the fluid passes through the open center directional control valve (DCV) back to the reservoir. The only pressure developed is that required to overcome pressure drops in the lines and through the DCV. When an operator shifts the DCV, flow is diverted to the cylinder. At this instant, the pressure in the circuit is the pressure to overcome the pressure drops. We now see a key disadvantage of a constant-flow circuit. The pressure

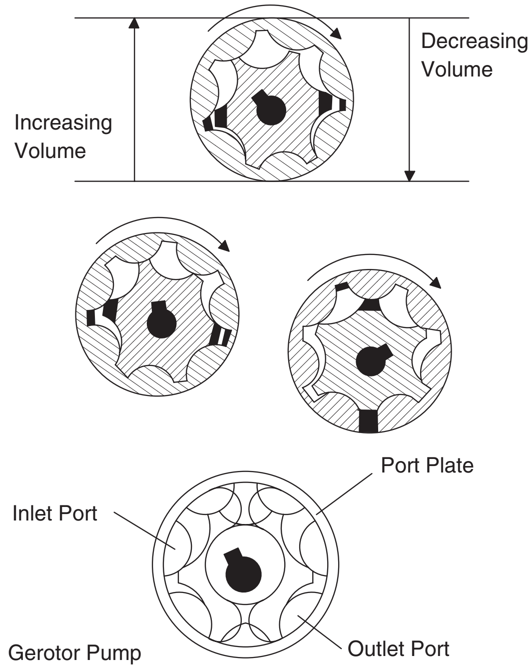


FIGURE 4.2
Diagram of gerotor pump. (Reprinted with permission from Parker Hannifin Corp.)

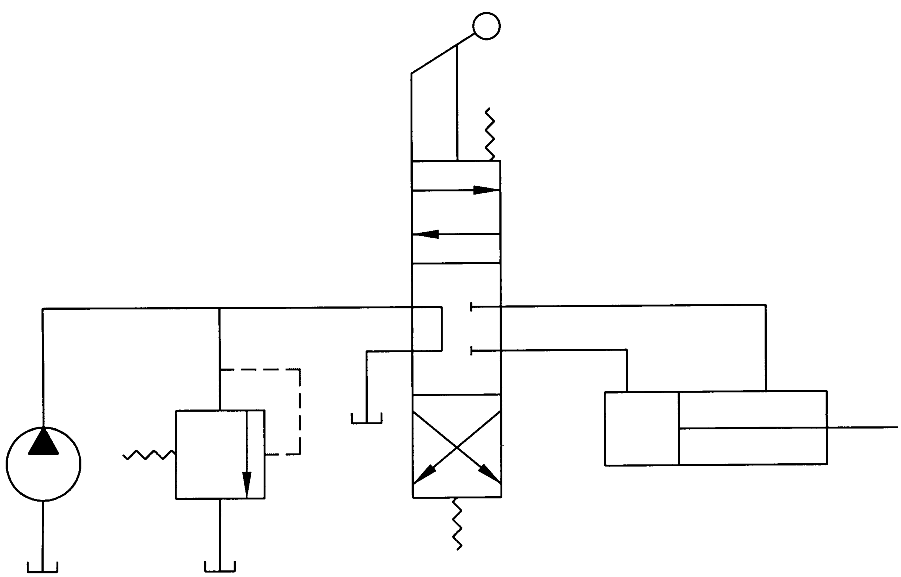


FIGURE 4.3
Simple circuit with fixed displacement pump.

must build from a very low level to the level required to accelerate the load. In a manufacturing application, cycle time is important; thus, acceleration of the load is a key functional requirement. Several techniques for supplying pressure at the instant the DCV is shifted were discussed in Chapter 3.

Most readers will have observed an operator controlling the speed of cylinder extension by controlling the opening of the DCV. An example would be a backhoe operator who swings the boom around quickly until it nears the target and then smoothly decelerates to stop the boom at the correct location. If the boom circuit is the circuit shown in Fig. 4.1, what happens to the “constant” flow of fluid being delivered by the pump?

To answer the above question, it is necessary to consider the operation of a spool-type directional control valve (DCV). When the spool is shifted to connect pump flow to port A and connect the reservoir to port B, orifices are created at ports A and B (Fig. 4.4). The pressure drops across these orifices are designated ΔP_{DCVA} and ΔP_{DCVB}, respectively. These orifices raise the total pressure at the relief valve until it cracks open and diverts part of the flow back to the reservoir, thus slowing the actuator speed. It is helpful to consider a specific example.

The relief valve in Fig. 4.3 has the characteristics shown in Fig. 4.5. The cracking pressure is 900 psi, and full flow pressure is 1000 psi, meaning that, when the system pressure reaches 1000 psi, all flow is diverted over the relief valve back to the tank.

The bore of the cylinder is d = 4 in., and the cylinder area ratio is 2:1. Thus, the cap end area is A_c = 12.56 in^2, and the rod end area is A_r = 12.56/2 = 6.28 in^2. Total flow from the pump is Q = 50 in^3/s. When the DCV is fully open to extend the cylinder, the pressure drops in the circuit are as follows:

$$\Delta P_{line\ 1} = 10 \text{ psi}$$

$$\Delta P_{DCVA} = 10 \text{ psi}$$

$$\Delta P_{DCVB} = 10 \text{ psi}$$

$$\Delta P_{line\ 2} = 5 \text{ psi}$$

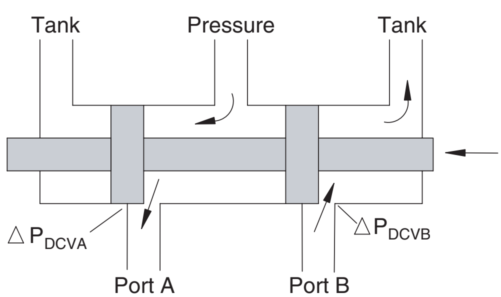


FIGURE 4.4
Diagram of orifices created when a directional control valve is shifted.

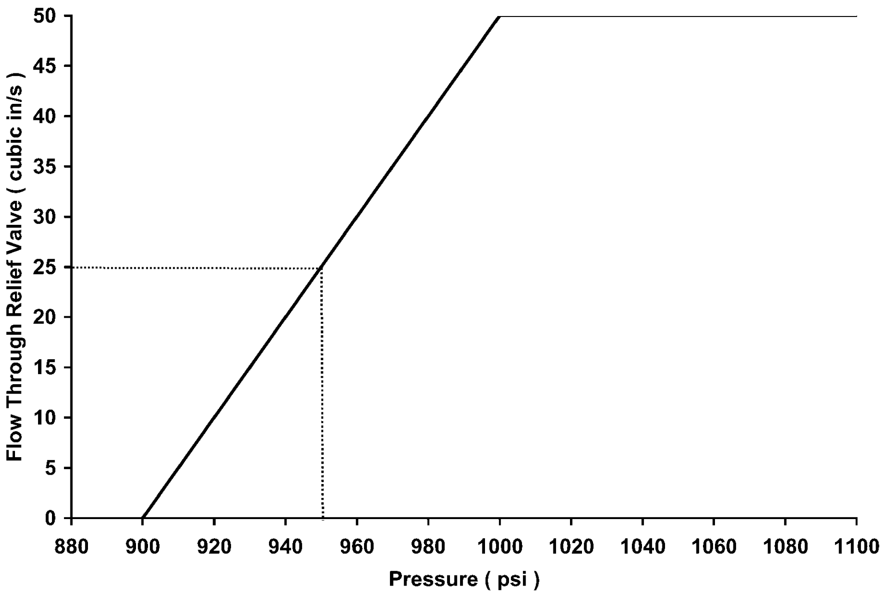


FIGURE 4.5
Operating characteristics of relief valve shown in Fig. 4.3.

where $\Delta P_{line\,1}$ = ΔP in line between pump and DCV
ΔP_{DCVA} = ΔP between ports P (pressure) and A in DCV
ΔP_{DCVB} = ΔP between ports B and T (tank) in DCV
$\Delta P_{line\,2}$ = ΔP in line between DCV and tank

A no-load test was conducted to determine the friction force. (The seals around the piston and rod produce a friction force that opposes movement of the piston. This force is referred to as the *friction force*.) During no-load extension, the following pressures were measured at the cylinder ports.

$$P_i = \text{pressure at inlet} = 25 \text{ psi}$$

$$P_o = \text{pressure at outlet} = 15 \text{ psi}$$

The friction force is

$$F_f = P_i A_c - P_o A_r$$

$$= 25\,(12.56) - 15(6.28)$$

$$= 314 - 94 = 220 \text{ lb}_f$$

For this example, we assume that the friction force is constant. A more detailed analysis would require that we deal with the fact that it is not consistent from initiation of movement to full extension.

The load is 10,000 lb_f. We are now ready to determine the total pressure required to extend the cylinder. Pressure at the cylinder outlet port is

$$\begin{aligned} P_o &= \Delta P_{DCVB} + \Delta P_{line2} \\ &= 10 + 5 = 15 \text{ psi} \end{aligned} \tag{4.9}$$

A force balance gives the total pressure required at the cylinder inlet port.

$$P_i A_c - (F_L + F_f + P_o A_r) = 0$$

or

$$P_i = (F_L + F_f + P_o A_r)/A_c \tag{4.10}$$

where F_L = load force (lb_f)

$$\begin{aligned} P_i &= [10{,}000 + 220 + 15(6.28)]/12.56 \\ &= 826 \text{ psi} \end{aligned}$$

Pressure at the relief valve is

$$P_{rv} = \Delta P_{line1} + \Delta P_{DCVA} + P_i = 10 + 10 + 821 = 841 \text{ psi}$$

The relief valve doesn't crack until pressure reaches 900 psi, so the cylinder extends at full velocity.

Cylinder velocity at full flow is

$$v_{cyl} = \frac{Q}{A_c} = \frac{50 \text{ in}^3/\text{s}}{12.56 \text{ in}^2} = 4 \text{ in/s} \tag{4.11}$$

Assuming that the operator wishes to reduce this speed to 2 in/s, what must occur? Flow to the cylinder must be reduced from 50 in^3/s to 25 in^3/s to reduce the cylinder speed to 2 in/s. Referencing Fig. 4.5, the pressure at the relief valve must increase to 950 psi for 25 in^3/s (half the pump flow) to be diverted to the tank.

The operator partially closes the DCV to reduce the orifice size at ports *A* and *B*. The new pressure drops are

$$\Delta P_{DCVA} = 80 \text{ psi}$$

$$\Delta P_{DCVB} = 80 \text{ psi}$$

The force balance on the cylinder now gives the following pressure at the inlet:

$$\begin{aligned} P_i &= [F_L + F_f + (\Delta P_{DCVB} + \Delta P_{line2})A_r]/A_c \\ &= [10000 + 220 + (85)6.28]/12.56 \\ &= 856 \text{ psi} \end{aligned} \tag{4.12}$$

The pressure at the relief valve is

$$\begin{aligned} P_{rv} &= P_i + \Delta P_{DCVA} + \Delta P_{line1} \\ &= 856 + 80 + 10 = 946 \text{ psi} \end{aligned} \tag{4.13}$$

With a slight adjustment to the DCV spool position, the operator will be able to increase the ΔP_{DCV} pressure drops and get $P_{rv} = 950$ psi. The speed of the cylinder will then be 2 in/s. Experienced operators develop such a "feel" for the system that they can operate a cylinder almost like they move their own arms.

In cases where the load is such that operating pressure is close to the cracking pressure of the relief valve, it is possible to have the pump flow divided into three flows.

1. Flow through relief valve back to reservoir
2. Flow to load
3. Leakage flow through DCV back to reservoir (smallest of three flows)

When this happens, the operator, functioning as a feedback loop, has to readjust the DCV position.

It is important to re-emphasize a point made in Chapter 3. Flow across an orifice converts hydraulic energy to heat energy and thus reduces the efficiency of the circuit. To achieve speed control with the circuit shown in Fig. 4.1, this loss is unavoidable. Since the fluid is being heated by the speed control method, heat rejection from the circuit must be planned accordingly.

4.4 Variable Displacement Pump Circuits

Variable displacement pump circuits are called *demand flow* circuits. A simple example is shown in Fig. 4.6. (The astute reader will quickly observe that this

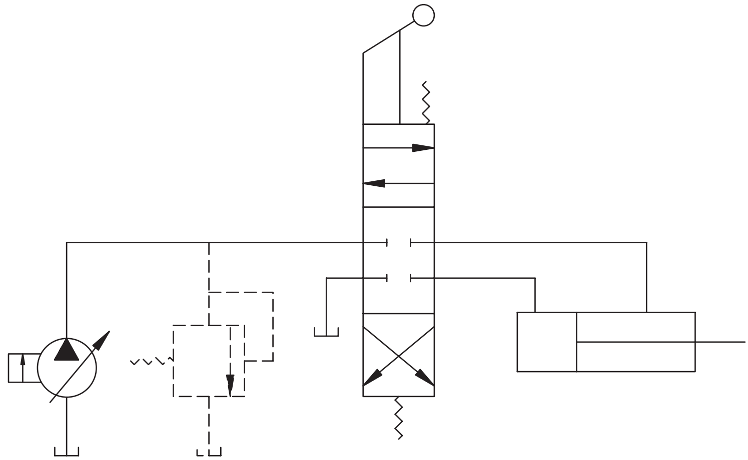


FIGURE 4.6
Simple circuit with variable displacement pump.

circuit is the first circuit we have studied with no relief valve. Although no relief valve is shown, it is recommended that a relief valve always be used to ensure that pressure can never reach an unsafe level. Rapid deceleration of a large load can produce a dangerous pressure spike.) Before understanding how a demand flow circuit operates, it is necessary to learn about the operation of a particular variable displacement pump—the vane pump.

4.4.1 Vane Pump

A vane pump (Fig. 4.7) has a series of vanes that slide back and forth in slots. There are springs in these slots that push the vanes out until the tip contacts the cam ring. (Some designs port pressurized fluid into the slots to force the vanes out.) A chamber is formed between adjacent vanes and the cam ring. As the rotor turns, the chamber decreases in size. Fluid flows into this chamber when it is a maximum size and exits during some $\Delta\theta$ of rotation when it is a minimum size. This change in chamber size provides the *pumping action.*

The principle of operation of a vane-type variable displacement pump is shown in Figs. 4.7 and 4.8. These illustrations are not to scale and are incomplete. Certain features are not shown. The cam ring is held in position in Fig. 4.7 with a threaded rod turned with a hand wheel. This ring will slide to the left when the hand wheel is turned. In the position shown in Fig. 4.6, the cam ring is centered on the axis of rotation. The chambers are equal size at the inlet and outlet, so no fluid is pumped (displacement is zero.) The rotor turns at the same speed, the vane tips are in contact with the cam ring, but no fluid is pumped.

The vane pump can be converted to a pressure-compensated pump by replacing the hand-wheel adjustment with a spring as shown in Fig. 4.9. A

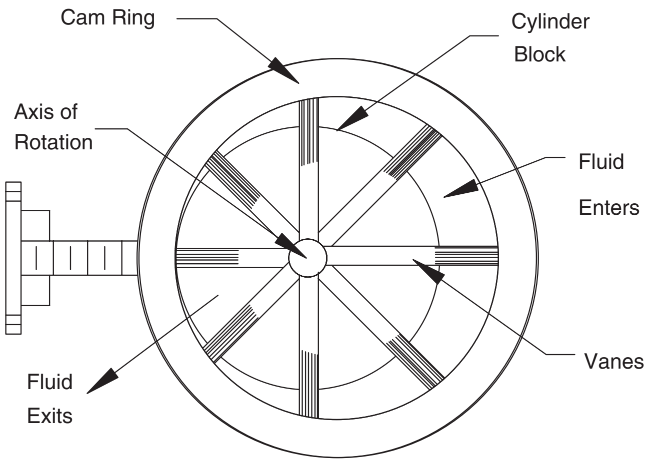


FIGURE 4.7
Vane-type variable displacement pump with cam ring held in position for maximum displacement.

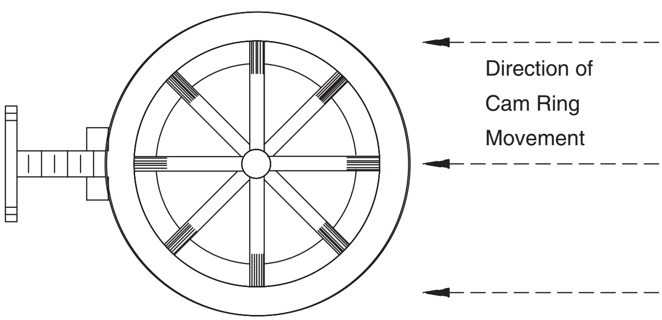


FIGURE 4.8
Vane-type variable displacement pump with cam ring held in position for minimum displacement.

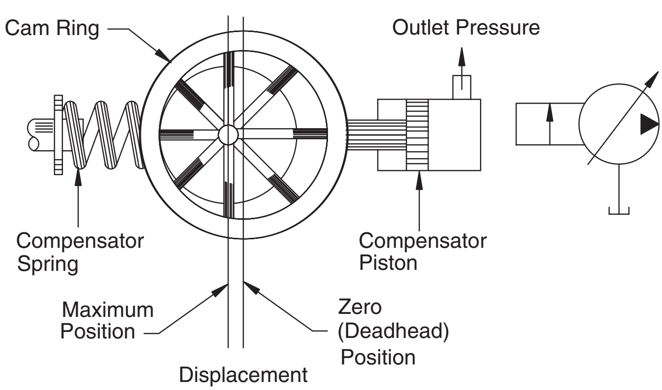


FIGURE 4.9
Pressure-compensated vane pump.

small cylinder, identified as the *compensator,* is placed on the opposite side. Outlet pressure acting on the compensator piston creates a hydraulic force that opposes the spring force. When the outlet pressure rises to a certain point, the hydraulic force becomes greater than the spring force, and the cam ring shifts to the left. As pressure continues to rise, the ring shifts more to the left until it eventually is centered on the axis of rotation (as shown in Fig. 4.8). At this pressure, known as the *deadhead pressure,* the pump displacement is almost zero. Some flow is produced to replace leakage.

It is now clear how the circuit in Fig. 4.6 can operate without a relief valve. The maximum pressure the pump can develop is limited by the compensator spring in the pressure-compensated pump.

A pressure-compensated pump can maintain deadhead pressure with very little energy input. Hydraulic power output is proportional to pressure × flow. If flow is zero, then hydraulic power output is zero. Some input energy is required to maintain deadhead pressure because of friction and leakage. The advantage of a demand-flow circuit (Fig. 4.6) as compared to a constant-flow circuit (Fig. 4.3) is that pressure is available at the instant the DCV is shifted; it does not have to build from zero.

Typical flow vs. pressure characteristics for a pressure -compensated variable displacement vane pump is shown in Fig. 4.10. When pressure reaches 2900 psi, identified as the *cutoff pressure,* the cam ring begins to shift, and the pump flow decreases. The rate of decrease (slope of the curve) is set by the spring constant of the compensator spring.

Figure 4.11 shows a pressure compensated variable displacement pump in an exploded view. The cam ring (pressure ring), vanes, and compensator are readily visible. Modern designs, like the one shown, do not use a compensa-

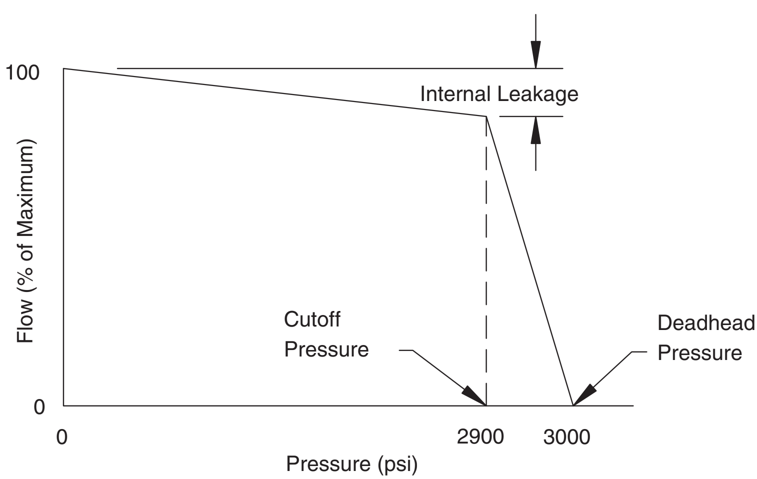


FIGURE 4.10
Typical flow vs. pressure performance curve for a pressure-compensated variable displacement vane pump.

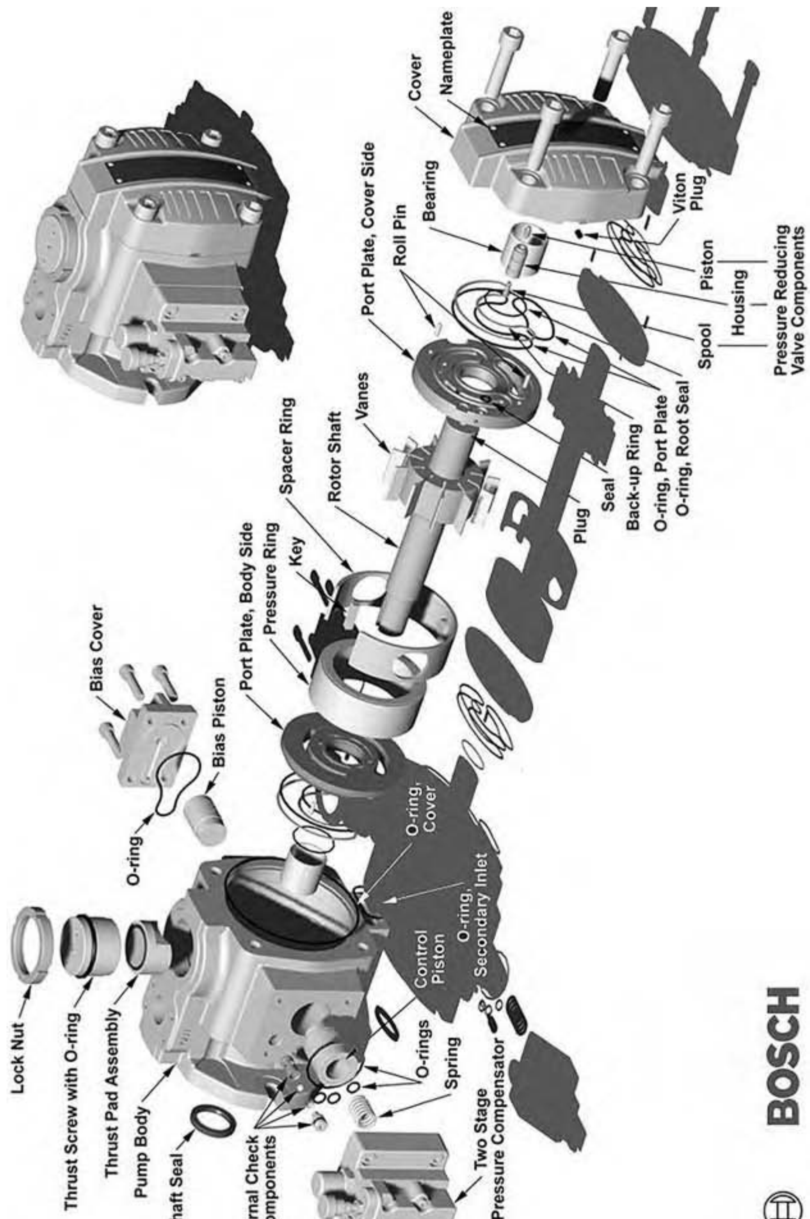


FIGURE 4.11
Exploded view of a VPV vane pump.

tor spring; rather, a specific pilot pressure on a bias piston that holds the pressure ring in place. As the pressure developed at the pump outlet rises, the force developed by the compensator piston eventually becomes large enough to equal the bias piston force and center the pressure ring.

4.4.2 Piston Pump

There are two piston pump designs:

1. Axial piston pump
2. Radial piston pump

Different manufacturers implement these two designs in different ways. This discussion will focus on the principle of operation and give only the detail required to understand that principle.

4.4.2.1 Axial Piston Pump

The axial piston pump has a series of cylinders (typically 7 or 9) mounted parallel to the axis of rotation. (The arrangement is similar to shell chambers in a revolver.) Pistons are installed in the cylinders. Each piston has a spherical end that mounts in a shoe (Fig. 4.12).

The shoe is held against a swashplate by a spring in the cylinder block (not shown in Fig. 4.12). The swashplate remains stationary as the cylinder block rotates with the input shaft. When the swashplate is at an angle to the shaft (as shown in Fig. 4.12), it moves the pistons back and forth in the cylinders as the cylinder block rotates. This movement provides the "pumping action."

It is helpful to follow the motion of one piston as the cylinder block makes one revolution. As shown in Fig. 4.13, the piston moves to the left as the cyl-

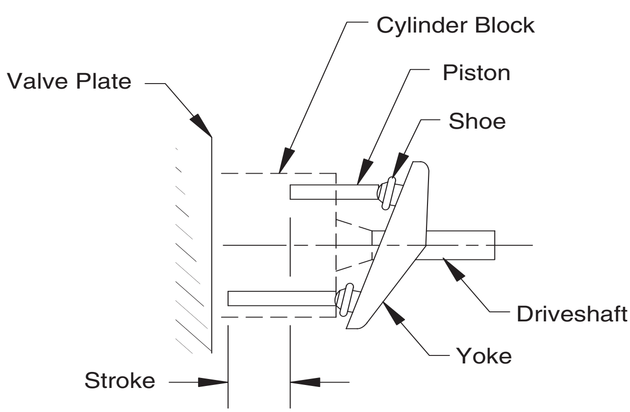


FIGURE 4.12
Schematic of axial piston pump.

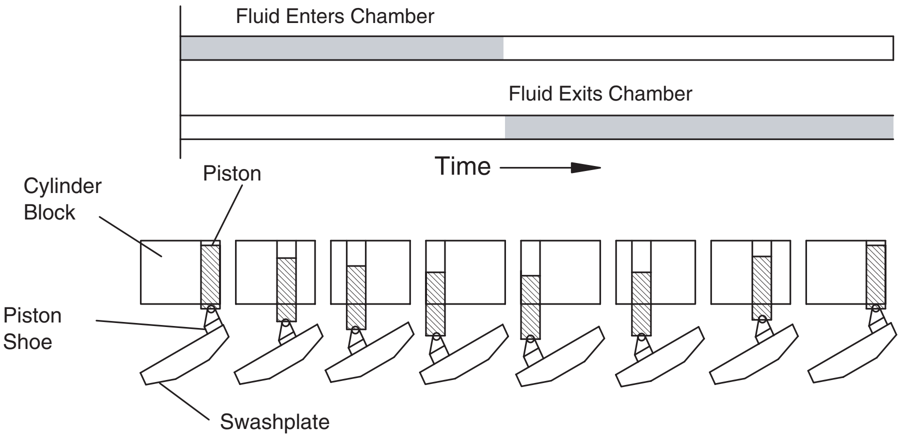


FIGURE 4.13
Schematic illustrating the motion of one piston during a single rotation of the cylinder block. Axis of rotation is in the plane of paper.

inder block rotates 180° to place the piston at the bottom of the cylinder. It moves to the right as the cylinder block returns to the original position. The reader can readily visualize that, if proper porting is provided, fluid will flow into the cylinder during the first 180° of rotation, and this fluid will be forced out during the second 180° of rotation. Note the bar graph in the figure that shows when the inlet port is open and when the outlet port is open.

Implementation of the design in an actual pump is shown in Fig. 4.14. Three key components, not previously discussed, are identified in this figure.

1. *Cylinder block spring.* This spring holds the block in position so that the piston shoes are always held in contact with the swashplate. This spring rotates with the cylinder block.
2. *Yoke spring assembly.* This spring holds the swashplate against the actuator piston.
3. *Activator piston.* This activator functions like a small hydraulic cylinder. When fluid flows into the cylinder, the piston extends and reduces the angle of the swashplate. The yoke spring assembly is compressed when the swashplate angle is reduced. Like the swashplate, both the actuator piston and the yoke spring assembly are stationary. [Smaller pumps (<15 hp) are actuated directly and do not have the activator piston, unless they are pressure compensated.]

Like the vane pump, an axial piston pump can be configured as a pressure-compensated pump (Fig. 4.15). The outlet pressure (high pressure), P_s, is incident on the end of the compensator valve spool. This pressure multiplied by the area of the spool gives a hydraulic force, F_h, which is opposed by the spring force, F_s, produced by the compensator valve spring. When P_s

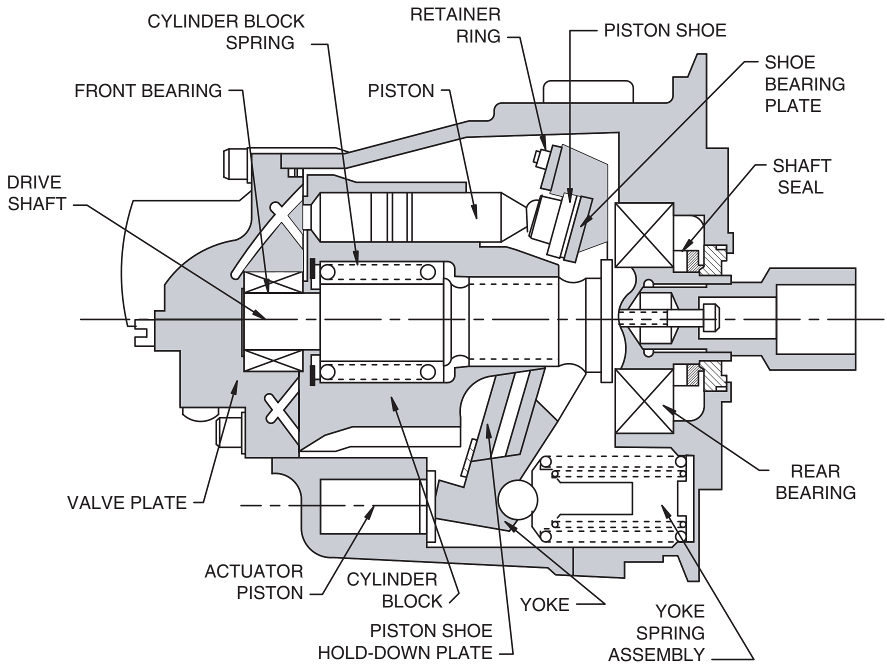


FIGURE 4.14
Implementation of axial piston pump design. (Reprinted with permission from Eaton Hydraulics.)

increases to the point where F_h equals F_s, the spool shifts downward, and fluid flows to the actuator piston.

The pressure at the actuator piston is $P_c = P_s - \Delta P$, where ΔP is the pressure drop across the orifice formed when the compensator valve cracks open. As P_s increases, the compensator valve opens more, ΔP decreases, and P_c approaches P_s. The increase in P_c increases the hydraulic force produced by the actuating piston, and eventually it rotates the yoke until it is perpendicular to the shaft and pump displacement is zero. The pump will hold this pressure and deliver no flow until something is done to lower the pump outlet pressure.

The pressure-compensated axial piston pump, like the pressure-compensated vane pump, can be used in a circuit without a relief valve. It is good design practice to include the relief valve. The first reason a relief valve is needed is to clip pressure spikes due to load dynamics. The second reason is readily apparent with a more careful examination of Fig. 4.15. Suppose the spool in the compensator valve sticks. [Spool-type valves will "silt-up" if they are actuated infrequently. The silting phenomenon is caused by tiny particles in the fluid (contaminants) being forced into the clearances in the valve. Eventually, the valve spool sticks and can be shifted only with the application of a sizeable force.] If this happens, the system pressure can continue to

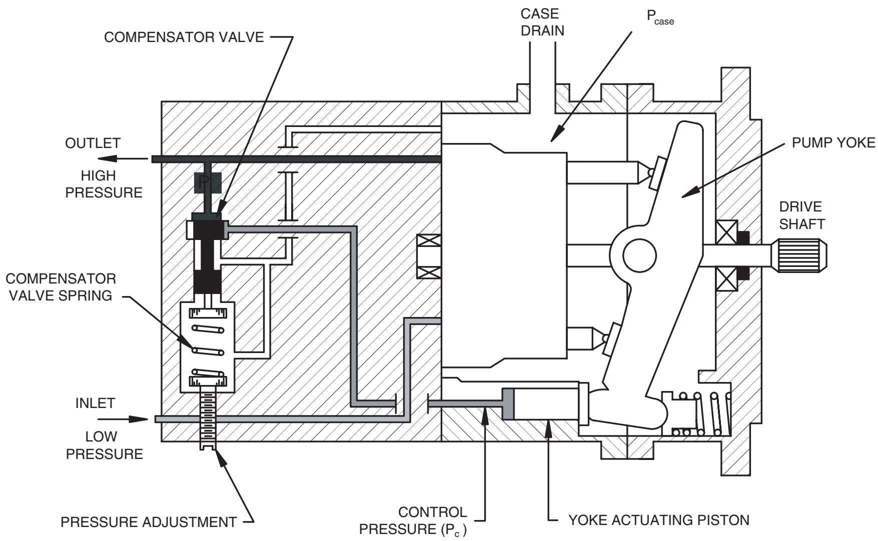


FIGURE 4.15
Schematic showing axial piston pump with pressure compensation.

increase above the deadhead pressure set by the compensator valve spring. A relief valve protects the circuit if this happens.

4.4.2.2 Radial Piston Pump

The principle of operation of the radial piston pump is shown in Fig. 4.16. In this case, the cylinders are positioned radially around the axis of rotation. As the shaft rotates, the connecting rods push the pistons back and forth in the cylinders to develop the pumping action. The design is used to pump both liquids and gases but, in the fluid power industry, it is more commonly used for pneumatic systems.

4.4.3 Improvement in Efficiency with Load Sensing

Load sensing was developed to improve the efficiency of a circuit. It requires a variable displacement pump; thus, it is appropriate to discuss it here.

An example will illustrate the advantage provided by load sensing. An application requires a maximum flow of 20 GPM at a maximum pressure of 2500 psi. To meet this requirement, a pump must be selected that can supply 20 GPM at 2500 psi. Most of the time, however, the application requires less than 20 GPM at less than 2500 psi. The percentage of total operating time the system operates under reduced load, and the way the system responds to this condition, is a key issue. The specific reduced load situation we will consider is the activation of a cylinder. A metered flow rate of 9 GPM at 1300 psi is required. To understand the advantage of load sensing, it is helpful to review

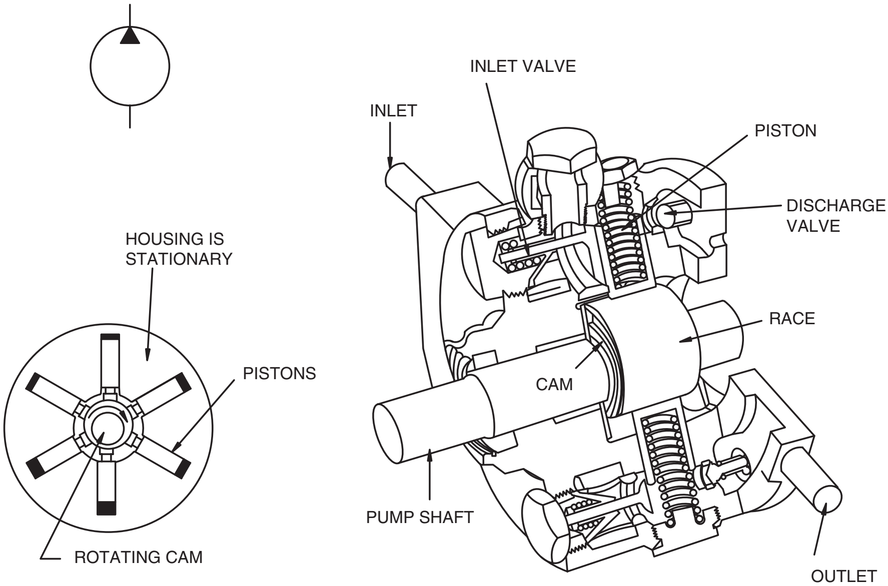


FIGURE 4.16
Schematic of radial piston pump.

the operation of an open-center and closed-center circuit that meets the functional objective and then consider a closed-center circuit with load sensing.

4.4.3.1 Open-Center Circuit

The fixed displacement pump shown in Fig. 4.3 delivers 20 GPM continuously. The relief valve is set on 2500 psi. When the DCV is manually shifted, the pump builds pressure to 1300 psi, and the load begins to move. At this point, the operator notices that the load is moving too fast. Remember, the required flow is only 9 GPM, and the pump is delivering 20 GPM. The operator partly closes the DCV, thus creating a restriction. The operator continues to close the valve manually until the restricted flow gives the desired load speed. The restriction produced by partly closing the DCV builds pressure at the relief valve to the point where it opens sufficiently to dump 11 GPM back to the reservoir. This flow is dumped at some pressure less than 2500 psi, the full open position of the relief valve. To produce the graph at the bottom of Fig. 4.17, we assume that the 11 GPM is dumped at 2500 psi.

Total system capacity is compared to the required capacity in the power diagram shown at the bottom of Fig. 4.17. The power loss for the open circuit is

$$\mathcal{P}_{loss\ \mathrm{oc}} = \frac{2500(11)}{1714} = 16 \text{ hp}$$

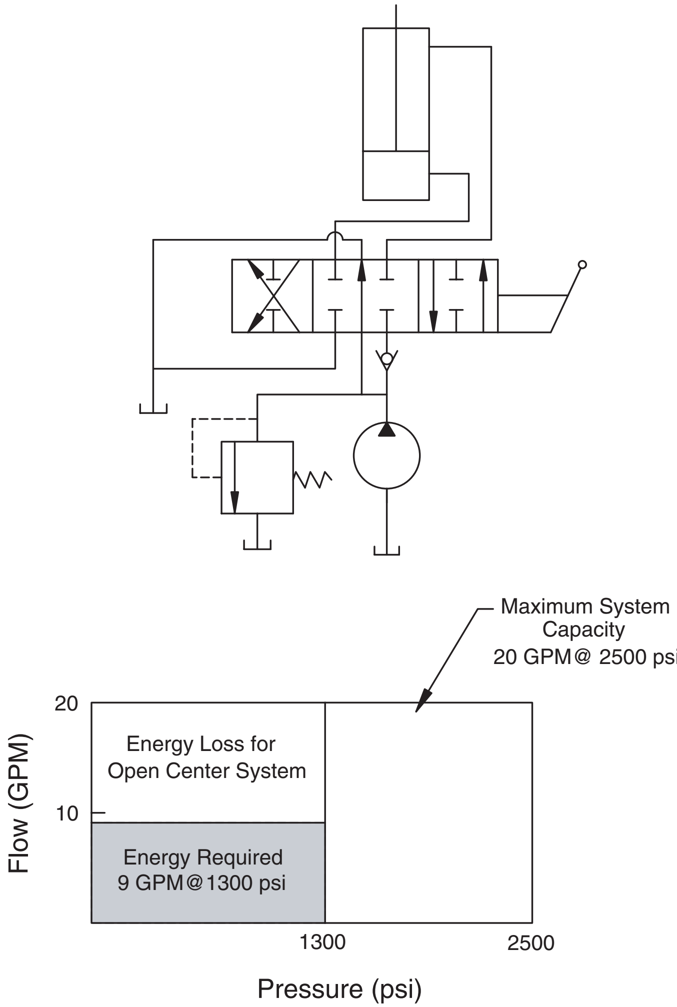


FIGURE 4.17
Open-center system used to supply flow to an actuator moving a partial load at less than maximum velocity.

4.4.3.2 Closed-Center Circuit

The closed-center circuit shown in Fig. 4.18 has a variable displacement pump like that shown in Fig. 4.15. The compensator valve spring is a 2500 psi spring. The pump builds 2500 psi before the compensator valve moves to open a pathway for fluid to flow to the yoke actuating piston. The piston extends and forces the swashplate to the zero displacement position. The pump then maintains the 2500 psi and supplies only enough flow to replace leakage.

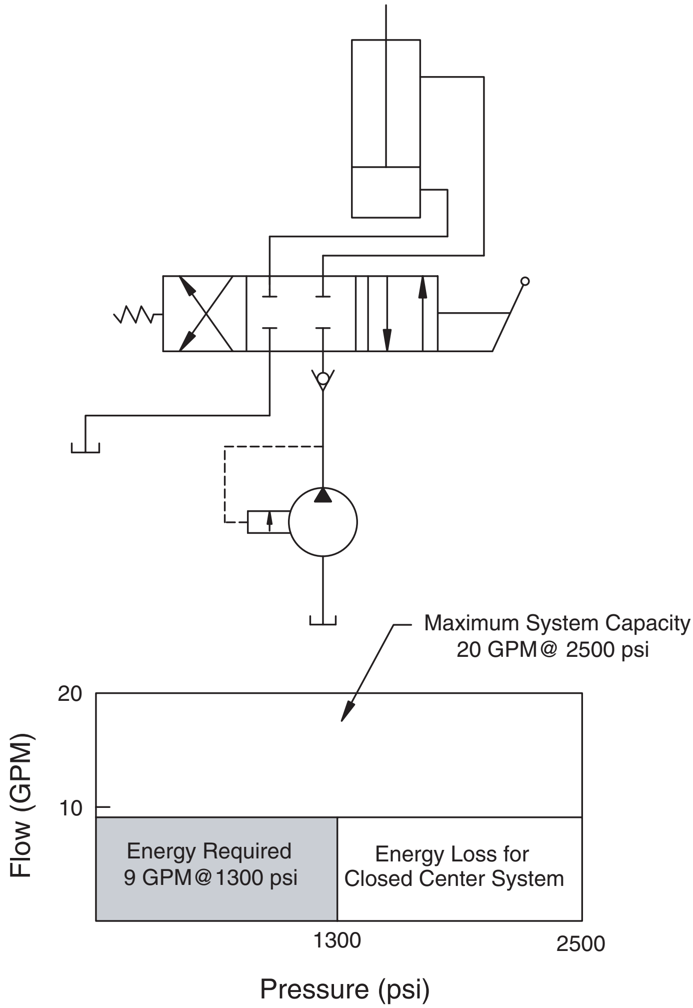


FIGURE 4.18
Closed-center system used to supply flow to an actuator moving a partial load at less than maximum velocity.

The operator cracks open the DCV to start the load. Pressure drops slightly at the pump, and the compensator valve moves slightly to partly close the pathway to the yoke actuating piston. This piston retracts slightly, and the swashplate tilts slightly so that the pump is now delivering fluid to the DCV and thus the load. This sequence of events continues until the operator has opened the valve to the position where 9 GPM is flowing to the load. The pressure is something less than 2500 psi, depending on the characteristics of the compensator valve spring. To produce the power diagram, we again assume that the pressure is 2500 psi.

Load pressure is 1300 psi, thus the pressure drop across the DCV is 2500 – 1300 = 1200 psi. The pump is delivering 9 GPM, thus the power loss is

$$\mathcal{P}_{loss\ cc} = \frac{1200(9)}{1714} = 6.3 \text{ hp}$$

This loss is considerably less than the 16 hp for the open-center circuit but is still quite large. We are now ready to see what load sensing can accomplish.

4.4.3.3 Closed-Center Circuit with Load Sensing

Load sensing is achieved with a special compensator mounted on a variable displacement pressure-compensated pump. The reader should envision this special compensator mounted on the pump shown in Fig. 4.15. For this analysis, we assume that the special compensator is set to destroke the pump at 200 psi. (The pump develops 200 psi and is destroked to deliver essentially zero flow at this pressure.) As the DCV is shifted to extend the cylinder, a pilot line senses the 1300 psi required to move the load. This work port pressure is added to the "destroke" pressure, and the pump delivers the required 9 GPM at 1300 + 200 = 1500 psi. How does this actually work? The details are a bit complicated. The reader is directed to Jarboe (1983) for an explanation of how the load-sensing compensator, pump, and DCV work together to deliver less than full flow at less than full pressure with low losses. Actual power loss in this case is

$$\mathcal{P}_{loss\ ls} = \frac{200(9)}{1714} = 1 \text{ hp}$$

A general understanding can be obtained by considering the load-sensing circuit shown in Fig. 4.19. The three-way, two-position DCV is shifted with pilot pressure. The pump is destroked when this valve is in the position shown in Fig. 4.19. There is no pilot pressure on the left side. The pump pressure applied on the right side shifts the two-position DCV to the position shown. There is no hydraulic pressure to add to the spring force (200 psi) in the compensator (spring cavity is vented through the two-position DCV to the reservoir), thus the pump builds only 200 psi pressure. It remains in this operating condition until the operator shifts.

When the operator shifts the three-position DCV, the pilot line is connected to supply pressure, which is 1300 psi in this example. This pilot pressure acts on the left side of the two-position DCV, causing it to shift to the right. Now supply pressure is added to the spring force in the compensator, and the pump builds 1300 + 200 = 1500 psi.

The means for controlling pump flow to 9 GPM is not shown in Fig. 4.19. The reader is referred to Jarboe (1983) for this detail. Figure 4.19 does have a power diagram similar to Figs. 4.17 and 4.18 to complete the visual comparison of power losses for the three circuits.

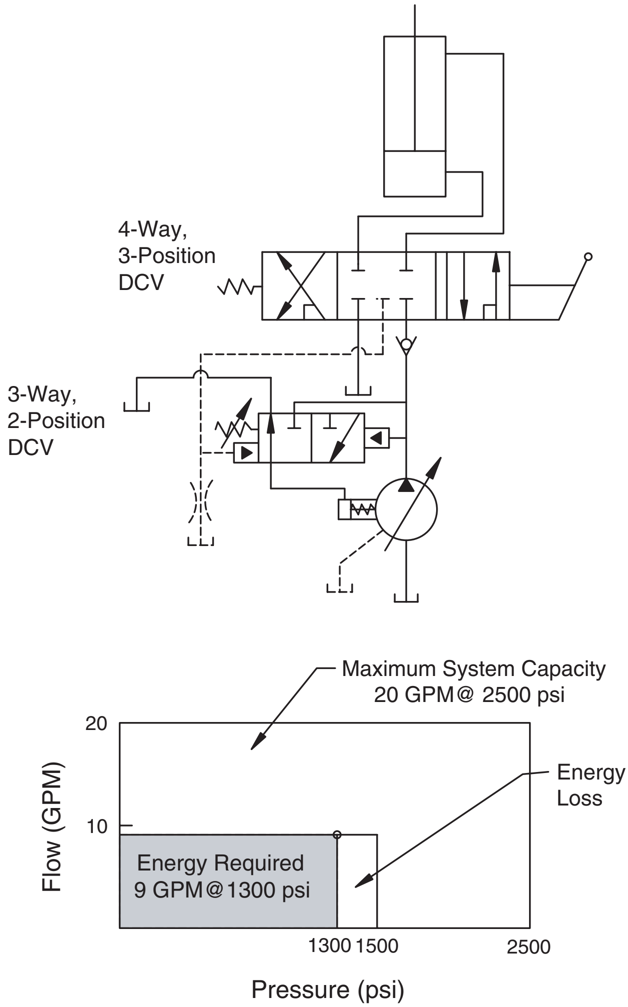


FIGURE 4.19
Open-center system with load sensing used to supply flow to an actuator moving a partial load at less than maximum velocity.

Response of a closed-center system with load sensing is slower than that of a regular closed-center system, because pressure must build from 200 psi to 1300 psi before the load begins to move. (With a regular closed-center system, the pressure is 2500 psi at the moment the four-way, three-position DCV is shifted.) Time to build pressure is reduced by designing the four-way, three-position DCV such that it connects the pilot line to the work port a short interval before the pressure line is connected. The pump compensator is already shifted (pump is building pressure) when the pressure line is fully connected to the work port.

A cross section of a solenoid-actuated DCV with load-sensing pilot line is shown in Fig. 4.20. Several definitions are needed before discussion of the operation of this valve can proceed.

The spool and bore in the valve body are machined to a sliding fit. The spool is machined with lands (sections that have the same diameter as the bore minus clearance) and undercuts (sections where material has been removed from the spool). Spool overlap is the distance along the spool land that separates the internal passages in the valve body. When the spool is shifted, the spool undercuts move into a position where they connect adjacent passages in the valve body. For example, when the spool in Fig. 4.20 is shifted to the left, pump flow (P) is connected to Port A and Port B is connected to the return (R).

Considering the DCV shown in Fig. 4.20, when the spool (1) is in the neutral position, ports A and B are blocked at spool land (2) and metering notches (3). Pump flow enters the valve at (4) and is blocked at (5). No flow can go to ports A and B or the return for this type of closed-center valve. Overlap at (2) is less than the overlap at metering notch (6). When the spool is shifted to the left, Port A is connected to area (7) before the pressure line at (4) is connected to area (7). *[Remember, the overlap at (2) is less than the overlap at metering notch (6).]* There is a direct passage from area (7) to the load-sensing port (8). Pressure at area (7) is immediately communicated to the pump compensator. The pump begins to build pressure to a level 200 psi above the area (7) pressure (work pressure) before the pressure port (4) is connected to Port A via metering notch (6). The orifice formed by metering notch (6) produces a 200 psi ΔP.

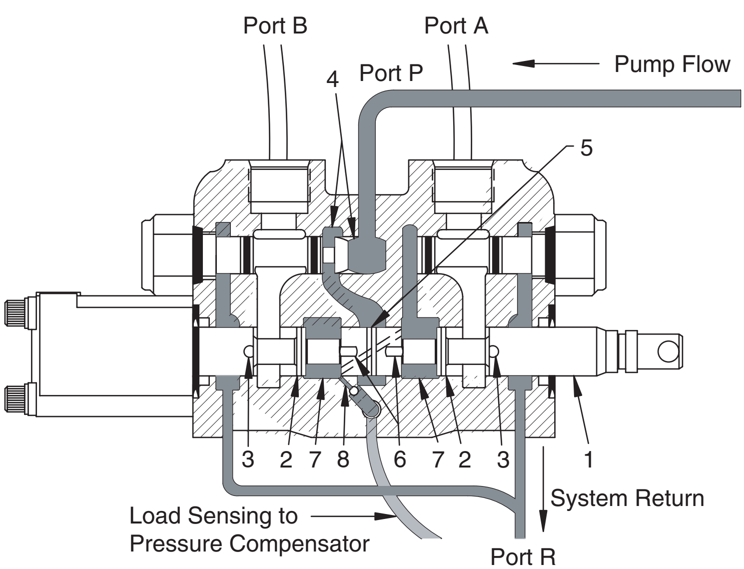


FIGURE 4.20
Four-way, three-position directional control valve with load-sensing pilot line.

This ΔP ensures that pump output pressure is 200 psi above the pressure required at Port *A*.

It should be clear that the load-sensing DCV is symmetrical. A shift of the spool to the right will connect Port *B* to area (7), and thus to load-sensing port (8), before the pressure port is connected to Port *B*.

A load-sensing circuit requires a special compensator on the variable displacement pump and a load-sensing DCV. These two components work together to minimize losses when the circuit needs to provide less than maximum flow at less than maximum pressure.

4.5 Comparison of Pump Performance Characteristics for Three Main Designs

Data was selected from manufacturers' literature for a fixed displacement pump of each design: gerotor (gear), vane, and piston. Pumps with approximately the same displacement were selected from three different manufacturers. (The selections should not be considered an endorsement or a criticism of a particular model. No attempt was made to select an optimal example of each design.) Performance data in the technical information supplied for each pump was used for the following comparison. In all cases, the test data were collected when the pump was pumping petroleum oils at a stated viscosity. All curves are plotted with pressure as the independent variable.

4.5.1 Gerotor Pump

The Hydreco Model 1919 gerotor pump has a displacement of 4.53 in^3/rev. The manufacturer gives performance data up to a maximum speed of 3000 rpm. The maximum pressure curve is 2500 psi.

When driven at 1200 rpm, the volumetric efficiency drops from 92% at 500 psi to 78% at 2500 psi (Fig. 4.21a). At 1800 rpm, the volumetric efficiency drops from 93% at 500 psi to 84% at 2500 psi (Fig. 4.21b). Why is the volumetric efficiency higher at the higher speed?

This question can be answered by comparing performance at a specific pressure. In this case, the comparison will be made at 1500 psi. The leakage flow is

$$Q_1 = Q_t - Q_a \tag{4.14}$$

where Q_1 = leakage flow (GPM)
Q_t = theoretical flow (GPM)
Q_a = actual flow (GPM)

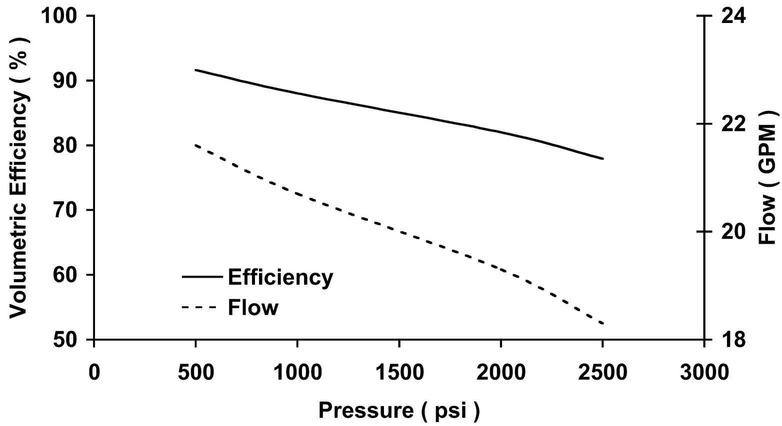


FIGURE 4.21a
Volumetric efficiency for a gerotor pump driven at 1200 rpm.

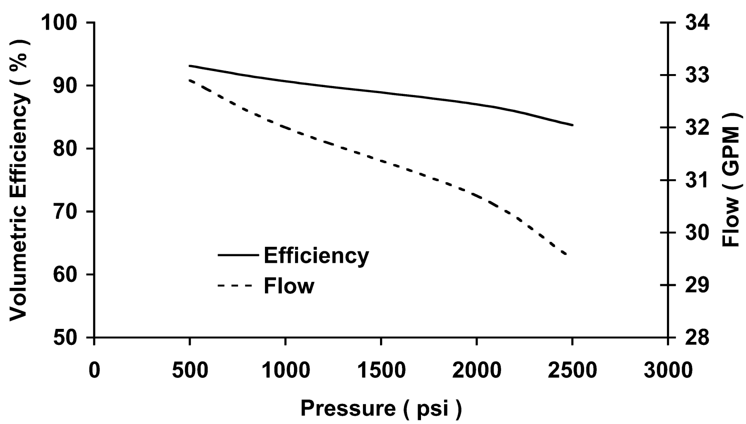


FIGURE 4.21b
Volumetric efficiency for a gerotor pump driven at 1800 rpm.

Theoretical flow at 1200 rpm is given by

$$Q_t = \frac{1200(4.53)}{231} = 23.53 \text{ GPM} \tag{4.15}$$

Volumetric efficiency at 1500 psi is $e_{vp} = 0.85$, so

$$\begin{aligned} Q_a &= Q_t e_{vp} \\ &= 23.53(0.85) = 20.0 \text{ GPM} \end{aligned} \tag{4.16}$$

Substitution into Eq. (4.14) gives

$$Q_1 = 23.53 - 20 = 3.53 \text{ GPM} \tag{4.17}$$

Leakage flow is primarily a function of pressure, with rotational speed being a much less significant factor. Suppose that leakage flow is the same for the performance test run at 1800 rpm, 1500 psi.

$$\begin{aligned} Q_t &= \frac{1800(4.53)}{231} = 35.3 \text{ GPM} \\ &= (Q_t - Q_1) \\ &= 35.3 - 3.53 = 31.77 \end{aligned} \tag{4.18}$$

The volumetric efficiency is

$$e_{vp} = \frac{Q_a}{Q_t} = \frac{31.77}{35.3} \times 100 = 90\% \tag{4.19}$$

Measured volumetric efficiency was e_{vp} = 88.7%, slightly less than the efficiency calculated by assuming constant leakage flow. There is some increase in leakage at higher speeds due to the increased turbulence of the fluid. This explains why the volumetric efficiency is less than that calculated by assuming constant leakage flow as speed increases. It is generally good design practice to select a smaller pump and operate it at higher speed to achieve a higher volumetric efficiency.

A volumetric efficiency in the 85 to 90% range is achievable with a gerotor pump if pressures are below 1500 psi. Remember, however, that volumetric efficiency decreases as operating temperature increases. Viscosity of the fluid decreases, and more leakage occurs through the clearances in the pump. This issue will be dealt with more fully in Chapter 8.

Overall efficiency accounts for the loss of mechanical energy due to friction, and the loss of hydraulic energy due to the leakage flow. The clearances between moving parts can be thought of as an orifice. Fluid on one side of the orifice has a high pressure (up to the maximum pressure that exists at the pump outlet), and fluid on the other side of the orifice has a pressure equal to the pressure in the pump housing (generally less than 50 psi). This ΔP multiplied by the leakage flow gives an estimate of the hydraulic power loss.

Overall efficiency for the Hydreco Model 1919 gerotor pump is plotted in Fig. 4.22a (1200 rpm) and Fig. 4.22b (1800 rpm). Efficiency is near a maximum at 1000 psi, so the comparison will be made at this pressure. Efficiency is 87.5% at 1200 rpm and 88% at 1800 rpm. The trend is the same as that found for volumetric efficiency; however, the increase in overall efficiency at the higher rpm is smaller.

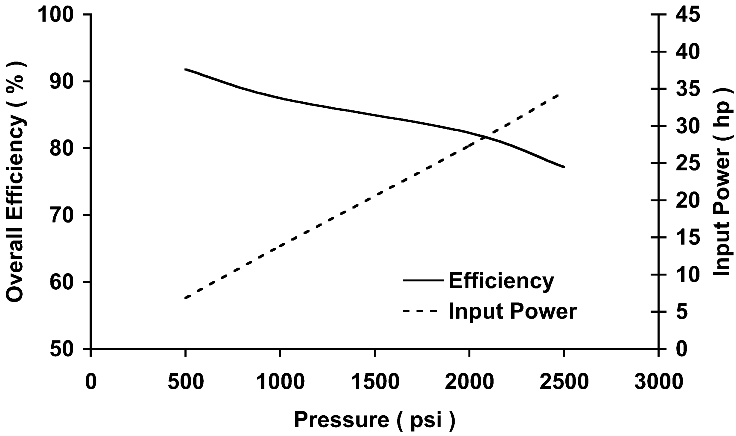


FIGURE 4.22a
Overall efficiency for a gerotor pump driven at 1200 rpm.

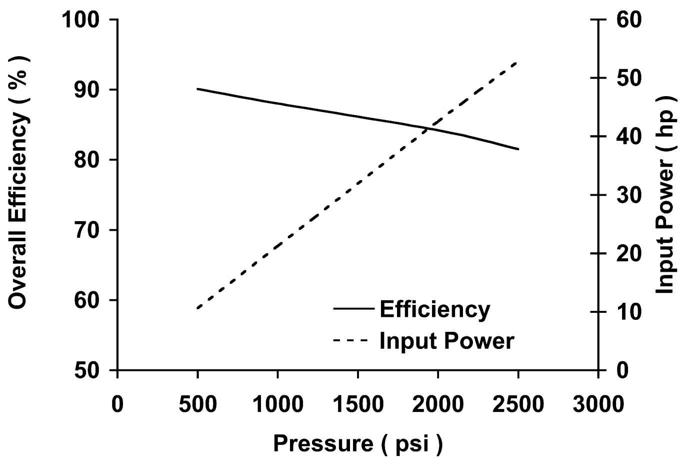


FIGURE 4.22b
Overall efficiency for a gerotor pump driven at 1800 rpm.

The energy loss (fluid energy converted to heat) due to fluid moving past a stationary part, or a part moving in a fluid, is proportional to velocity squared (Chapter 2). We expect the energy loss due to friction to increase as operating speed increases. This friction loss, however, is only part of the total power loss. Leakage flow represents a loss of hydraulic power. This flow is raised to some pressure, but it is not part of the output flow, and thus it is not included in the output power from the pump. Realizing that those who have done an in-depth study of fluid machinery will question the simplification, total power loss will now be discussed as the sum of two losses,

$$\text{Total power loss} = \text{Friction power loss} + \text{Leakage power loss}$$

A simple experiment might help the reader better understand the partitioning of total loss in this manner. A schematic for this experiment is shown in Fig. 4.23. Suppose a variable displacement pump is set for zero displacement and operated at a given speed. Fluid at the outlet has an unimpeded pathway back to the inlet, so developed pressure (ΔP across the pump) will be very small. Outlet flow will be small, because the pump displacement is set at zero. Delivered hydraulic power (small pressure × small flow) is very small and can be taken to be zero. The measured input power is the friction power loss.

Input power from the electric motor is given by

$$\mathcal{P}_{in} = \frac{TN}{63,025} \tag{4.20}$$

where T = torque measured with torque transducer (in -lb$_f$)
N = input speed (rpm)

Since no hydraulic power is delivered by the pump, this input power is the total power required to replace losses. (Speed is constant, so inertia is not a factor.). Friction power loss is defined by

$$\mathcal{P}_f = \mathcal{P}_{in} \tag{4.21}$$

where $\mathcal{P}_{in}$ = measured input power [Eq. (4.20)]

It is instructive to calculate the overall efficiency for a gerotor pump based on leakage power loss only (neglecting the friction loss component) and compare this calculated efficiency with the overall efficiency reported by the manufacturer. This comparison reveals the magnitude of the two components for the gerotor design. Data from the 1200 rpm, 1500 psi test will be used.

As previously calculated, Q_l = 3.53 GPM. Because of the gerotor pump design, it is difficult to measure pressure inside the housing at the point

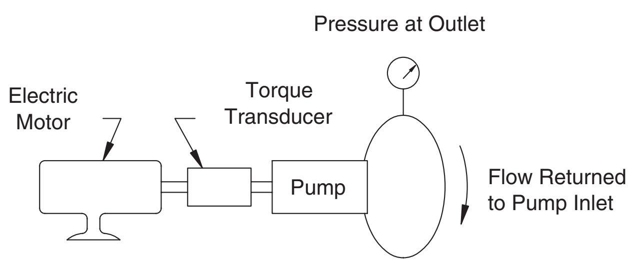


FIGURE 4.23
Diagram of experiment done to illustrate the partitioning of total power loss in a hydraulic pump.

where the fluid is leaking from the outlet back to the inlet. We simply assume that the leakage ΔP equals the pressure rise across the pump. Leakage power loss is

$$\mathcal{P}_{lk} = \frac{\Delta PQ}{1714} = \frac{1500(3.53)}{1714} = 3.09 \text{ hp} \tag{4.22}$$

Hydraulic power delivered, assuming the pressure at the pump inlet is atmospheric pressure, is

$$\mathcal{P}_{out} = \frac{\Delta PQ_a}{1714} = \frac{1500(20)}{1714} = 17.5 \text{ hp} \tag{4.23}$$

Total input power is

$$\begin{aligned}\mathcal{P}_{in} &= \mathcal{P}_{out} + \mathcal{P}_{lk} \\ &= 17.5 + 3.09 = 20.59 \text{ hp}\end{aligned} \tag{4.24}$$

and the calculated overall efficiency is

$$\begin{aligned}e_{op} &= (\mathcal{P}_{out}/\mathcal{P}_{in})100 \\ &= (17.5/20.59)100 = 85.4\%\end{aligned} \tag{4.25}$$

The overall efficiency based on the input horsepower curves given by the manufacturer is 85.4%. This overall efficiency includes both leakage and friction losses. When we calculated leakage loss, we assumed that leakage ΔP equaled the ΔP across the pump. Actual leakage ΔP is smaller, thus our estimate of leakage loss was too high. It is true that the leakage power loss is by far the dominant term for the gerotor design.

Key comparisons between volumetric and overall efficiencies for the gerotor pump are given in Table 4.1. The fact that the overall and volumetric efficiencies are approximately equal for each speed and pressure reinforces our conclusion that leakage loss is by far the dominant term for the gerotor design.

Overall efficiency at 2500 psi increases from 77% at 1200 rpm to 81.5% at 1800 rpm for the gerotor pump. Because leakage losses are such a significant term for this type of gear pump design, it is generally good policy to use a smaller displacement pump and operate at higher rpm to get a given flow.

If a gerotor motor with similar efficiency, around 80%, is used to convert the hydraulic power back to mechanical power, the overall efficiency (pump and motor combination) is

$$\begin{aligned}e_{ot} &= (e_{op} \times e_{om})100 \\ &= (0.8 \times 0.8)100 = 64\%\end{aligned} \tag{4.26}$$

TABLE 4.1

Comparison of Volumetric and Overall Efficiency for a Gerotor Pump Operated at 1200 and 1800 RPM

Pump speed (rpm)	Operating pressure (psi)	Volumetric efficiency (%)	Overall efficiency (%)
1200	500	91.5	91.5
	1000	88	87.5
	2500	78	77
1800	500	93	90
	1000	91	88
	2500	83.5	81.5

Less than two-thirds of the input power is delivered to the load. Mechanical transmissions can have efficiencies of 95% or better, so it is readily apparent that a decision to transmit mechanical energy with fluids must carefully weigh the advantages and disadvantages.

4.5.2 Vane Pump

The Vickers Model 25V (T) vane pump has a displacement (cam ring fixed for maximum displacement) of 4.81 in^3/rev. The specifications are:

Maximum speed 1800 rpm

Maximum pressure 2500 psi

As was done with the gerotor pump, performance at 1200 and 1800 rpm is compared.

Volumetric efficiency at 1200 rpm decreases from 97.3% at 500 psi to 84% at 2500 psi (Fig. 4.24a). For comparison, the gerotor pump at the same speed and pressure decreased from 91.5 to 78%. The vane pump volumetric efficiency is 5 to 6% higher than the gerotor pump. Leakage flow is less, thus the volumetric efficiency is higher.

Volumetric efficiency at 1800 rpm decreased from 97.8% at 500 psi to 88% at 2500 psi (Fig. 4.24b). As with the gerotor pump, the volumetric efficiency is higher at higher shaft speed across the whole pressure range.

The overall efficiency of the vane pump is quite low at low pressures. As shown in Figs. 4.25a and 4.25b, the overall efficiencies at both 1200 and 1800 rpm increase to a maximum at 1000 psi. The decrease in overall efficiency as pressure increases from 1000 psi to 2500 psi, is less for the vane pump than the gerotor pump. The vane pump efficiency curve is "flatter," indicating that vane pump performance will change less as load pressure changes in the 1000 to 2500 psi range.

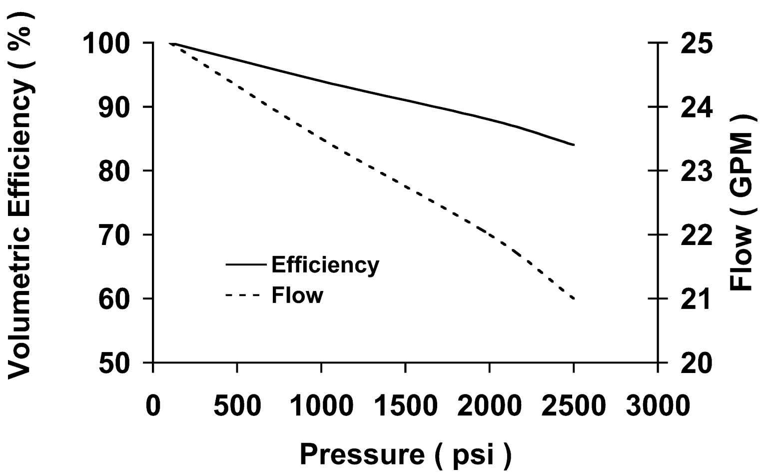


FIGURE 4.24a
Volumetric efficiency for a vane pump driven at 1200 rpm.

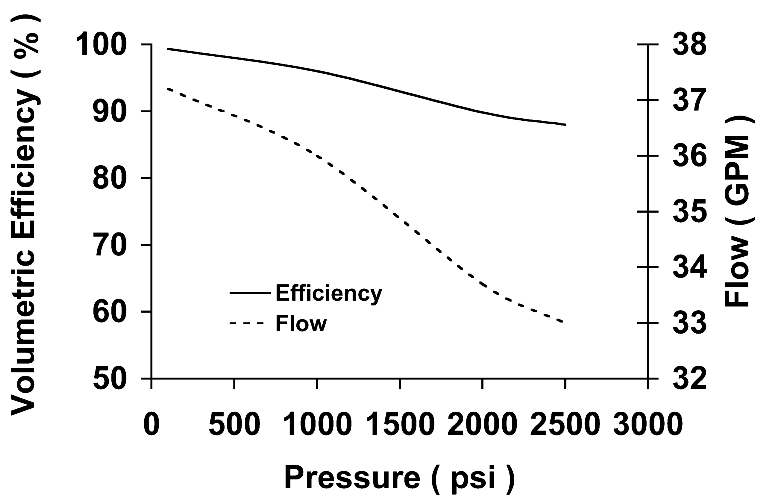


FIGURE 4.24b
Volumetric efficiency for a vane pump driven at 1800 rpm.

Overall efficiency is much lower than volumetric efficiency for the vane pump (Table 4.2). As previously stated, overall efficiency includes both the friction and the leakage power losses. For the gerotor pump, the friction power loss was small in comparison to the leakage power loss, so the two efficiencies are approximately equal. The vane pump, on the other hand, has a higher friction loss, so the overall efficiency is less than the volumetric efficiency.

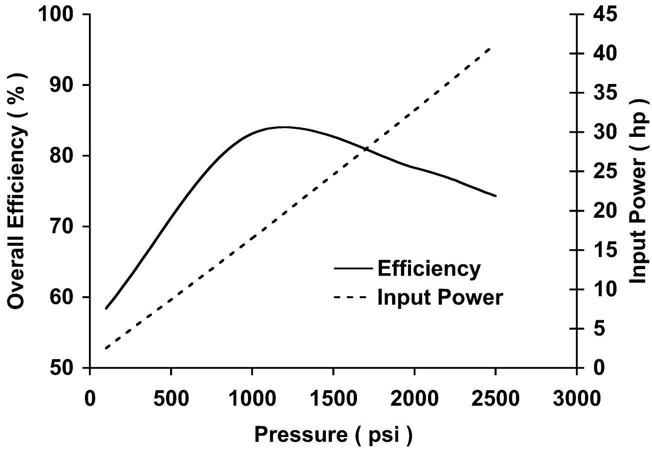


FIGURE 4.25a
Overall efficiency for a vane pump driven at 1200 rpm.

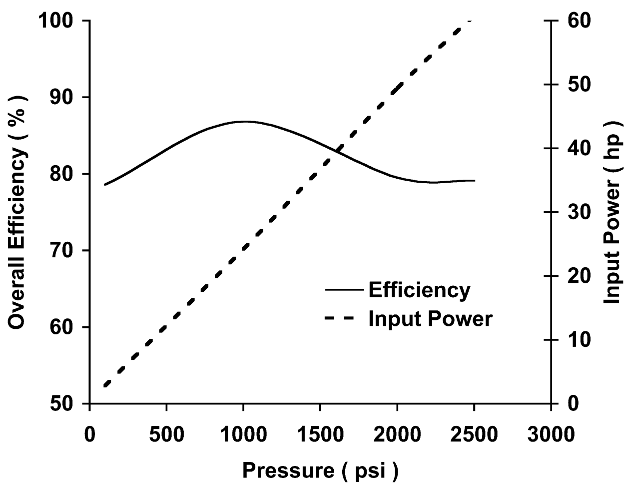


FIGURE 4.25b
Overall efficiency for a vane pump driven at 1800 rpm.

4.5.3 Axial Piston Pump

The Sauer-Danfoss Series 90–075 variable displacement axial piston pump (swashplate locked in maximum position to simulate fixed displacement) has a displacement of 4.57 in^3/rev. The specifications are:

Maximum speed 3950 rpm

Maximum pressure 7000 psi

TABLE 4.2

Comparison of Volumetric and Overall Efficiency for a Vane Pump Operated at 1200 and 1800 RPM

Pump speed (rpm)	Operating pressure (psi)	Volumetric efficiency (%)	Overall efficiency (%)
1200	500	97	69
	1000	94	83
	2500	84	74
1800	500	98	82
	1000	96	87
	2500	88	79

This pump has a higher maximum speed than the gerotor pump (3000 rpm) or the vane pump (1800 rpm). More significantly, it is rated for a much higher pressure (7000 psi vs. 2500 psi). Careful design is required to achieve the 7000 psi maximum pressure. Each component in the system must have a maximum operating pressure rating of 7000 psi or above for safe operation.

Although the piston pump is rated for much higher speeds, its performance at 1200 and 1800 rpm is presented for direct comparison with the gerotor and vane pumps. The independent variable is scaled from 1000 to 6000 psi instead of 0 to 3000 psi as was done for previous plots.

Volumetric efficiency at 1200 rpm decreases from 99% at 1000 psi to 89% at 6000 psi (Fig. 4.26a). For comparison, the gerotor pump had a volumetric efficiency of 88% at 1000 psi, and the vane pump efficiency was 94%.

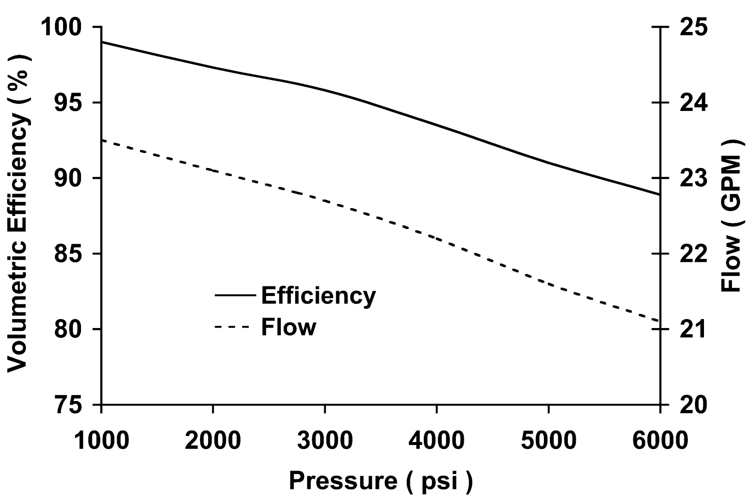


FIGURE 4.26a
Volumetric efficiency for a piston pump driven at 1200 rpm.

Volumetric efficiency at 1800 rpm decreased from 98.8% at 1000 psi to 90.3% at 6000 psi (Fig. 4.26b). The increase in volumetric efficiency at higher speed was less for the piston pump than the gerotor or vane pumps. Piston pumps are expensive, but this cost is justified by its exceptional performance, better than 90% volumetric efficiency at 6000 psi. The vane pump operating at 1800 rpm cannot attain this efficiency at pressures above 2000 psi. If the higher pressures are going to be used, and the extra cost incurred, the system operating condition is of critical important. Fluid temperature and contamination control will be discussed in Chapter 8.

The overall efficiency of the piston pump is quite high (>90%) for pressures below 4000 psi. Overall efficiency at 1800 rpm (Fig. 4.27b) is slightly higher than at 1200 rpm (Fig. 4.27a) because, as discussed for the gerotor pump, *leakage flow is predominantly a function of pressure, not speed.*

The gerotor and piston pumps are on opposite ends of the efficiency scale and the initial cost scale, thus it is instructive to repeat the same analysis of leakage power loss for the piston pump. Theoretical flow at 1200 rpm is given by

$$Q_t = \frac{1200(4.57)}{231} = 23.74 \text{ GPM} \tag{4.27}$$

Volumetric efficiency at 1500 psi is $e_{vp} = 0.9815$, so

$$Q_a = Q_t e_{vp} = 23.74(0.9815) = 23.3 \text{ GPM} \tag{4.28}$$

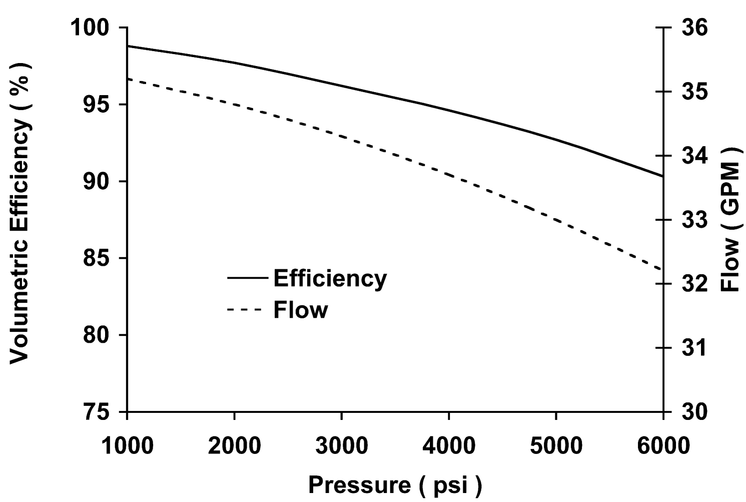


FIGURE 4.26b
Volumetric efficiency for a piston pump driven at 1800 rpm.

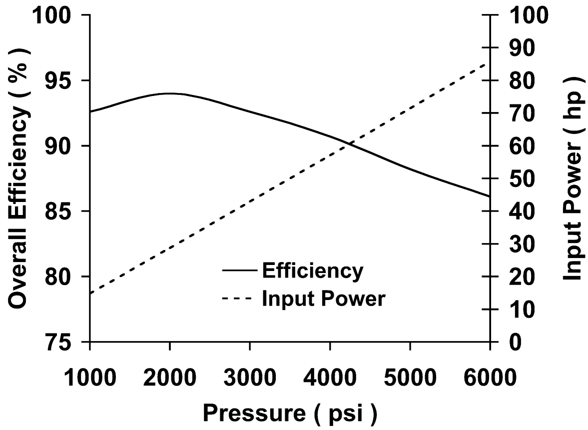


FIGURE 4.27a
Overall efficiency for a piston pump driven at 1200 rpm.

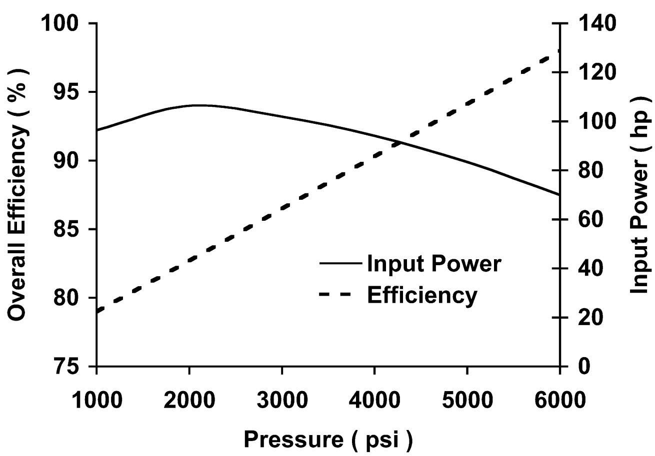


FIGURE 4.27b
Overall efficiency for a piston pump driven at 1800 rpm.

and $Q_l = 23.74 - 23.3 = 0.44$ GPM. The hydraulic power loss due to this leakage, assuming that the pump housing pressure is 50 psi, is

$$\mathcal{P}_{lk} = \frac{\Delta PQ}{1714} = \frac{(1500-50)(0.44)}{1714} = 0.37 \text{ hp} \tag{4.29}$$

Hydraulic power delivered, assuming the pressure at the pump inlet is atmospheric pressure, is

$$\mathcal{P}_{out} = \frac{(1500-0)(23.3)}{1714} = 20.39 \text{ hp} \tag{4.30}$$

Overall efficiency at 1500 psi is $e_{op} = 0.933$, thus total input power is

$$\mathcal{P}_{in} = \frac{\mathcal{P}_{out}}{e_{op}} = \frac{20.39}{0.933}$$

$$= 21.85 \text{ hp} \tag{4.31}$$

Total power loss is

$$\mathcal{P}_{loss} = (\mathcal{P}_{in} - \mathcal{P}_{out})$$

$$= (21.85 - 20.39) = 1.46 \text{ hp} \tag{4.32}$$

The two components of this total loss are friction and leakage. The leakage power loss has already been calculated at 0.37 hp, so the total loss divides as follows:

	Friction	**+**	**Leakage**	**=**	**Total**
hp	1.09	+	0.37	=	1.46
%	75	+	25	=	100

Tolerances in the piston pump are designed to provide the correct lubricating film while minimizing leakage flow. Because of the number of moving parts, and, consequently, the total area of lubricating film in the piston pump, friction accounts for 75% of the power loss.

It is interesting to compare the partitioning of power loss for the three pump designs. These losses, expressed as a percent of input power, were calculated at 1200 rpm, 1500 psi and are given in Table 4.3. The trend in losses due to leakage is as expected; the vane pump is about mid way between the gerotor and piston pumps. Losses due to friction are higher in the vane pump than the piston pump. Both pumps have a relatively high number of moving parts as compared to the gerotor pump. The vane pump does have a larger lubricating film area than the piston pump, and this explains why the friction

TABLE 4.3

Comparison of Power Losses in Three Pump Designs

	Power Loss as a Percentage of Input Power[a]		
Pump Design	**Due to Friction**	**Due to Leakage**	**Total**
Gerotor	–	15.0	15.0
Vane	11.0	7.6	18.6
Piston	5.0	1.7	6.7

a. Comparison made at 1200 rpm operating speed and 1500 psi pressures power loss is 11% for the vane pump and only 5% for the piston pump. All three designs are effective solutions for different applications. An applications engineer must consider the pump performance characteristics when designing *any* fluid power circuit.

4.6 Multiple Pump Circuits

The multiple pump design places two or more pumps in the same housing and drives them with a single input shaft. Sometimes, only one inlet is provided, but each pump has its own outlet. The advantage of a multiple pump is isolation of circuits.

The pump shown in Fig. 4.28 supplies flow to a hydraulic motor and cylinder. When the solenoid-activated, two-position DCV is shifted, flow is directed to the motor. If the manually activated three-position DCV is then shifted, flow is diverted to the cylinder. The pressure in the motor and cylinder circuits dictates how the flow will divide. If the motor pressure is much higher, then all the flow will go to the cylinder, and the motor will stop until the three-position DCV is recentered or the cylinder reaches its full extension and the pressure builds to the pressure required by the motor circuit.

It may be undesirable to interrupt the motor operation to activate the cylinder. In this case, a multiple pump is an attractive solution (Fig. 4.29). The same electric motor is used, and all the hydraulic components are the same, except the three-position DCV now has a open center rather than a closed center. Note also that each pump has a relief valve to protect the circuit.

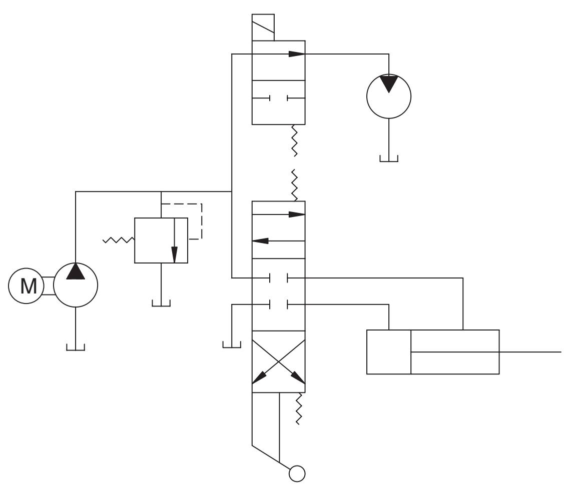


FIGURE 4.28
Circuit showing one pump supplying flow to two actuators.

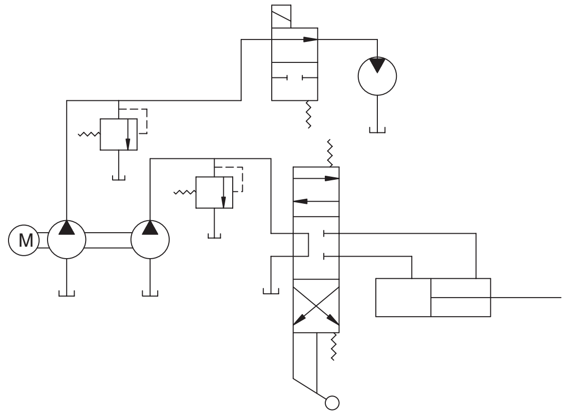


FIGURE 4.29
Circuit showing one pump supplying flow to two actuators.

Gear pumps are available with more than two pumps in the same housing. No manufacturer known to the author supplies more than two piston pumps in the same housing. Two pumps in the same housing are commonly referred to as a *tandem* pump.

Many manufacturers of all three designs supply models with a pad for mounting a second pump on the primary pump. A cover plate is removed from this pad to reveal a splined coupling on the end of the primary pump shaft. The input shaft of the second pump mates into the splined coupling, and the bolt holes on the pad are used to bolt the second pump in place. The designer must ensure that

1. The splines match.
2. The mounting specifications match.
3. The required torque for the second pump does not exceed the rating given by the primary pump manufacturer.

It is possible to mount a tandem piston pump on the auxiliary mount of a primary piston pump and thus power three piston pumps from one prime mover. Different pump designs can be mixed. For example, a multiple gear pump can be mounted on the auxiliary mount of a piston pump.

4.7 Pump Mounts

Pump mounts are gearboxes that are designed to power hydraulic pumps. These gearboxes have one input and provisions for mounting one to four pumps on the output side (Fig. 4.30). Mounts are available with or without clutches, and different gear ratios are available. When an internal combustion engine is the prime mover, it is desirable to run the engine at a speed that will give maximum torque output, typically around 2000 rpm for a diesel engine. It may be desirable to run the pump at a higher speed to produce a higher output flow. Selecting a pump mount with a 1:1.25 gear ratio will convert a 2000 rpm input speed to a 2500 rpm pump drive speed.

Many mobile applications require a number of circuits on an individual machine. Separate pumps for each circuit can be provided with a number of combinations. Suppose the requirement is to provide four pumps on the machine. A single pump could be mounted on each mount of the four-pump mount shown in Fig. 4.30, or we might use a two-pump mount with a single pump on one mount and a three-pump multiple on the second mount. A designer typically considers several options before finalizing the design. In addition to the functional requirements, cost, space, and weight (on mobile machines) are key factors.

It is possible to drive some pumps with a belt drive. Belt drives place a radial load on the pump bearings. Manufacturers list an allowable radial load for their pumps, and this load should not be exceeded.

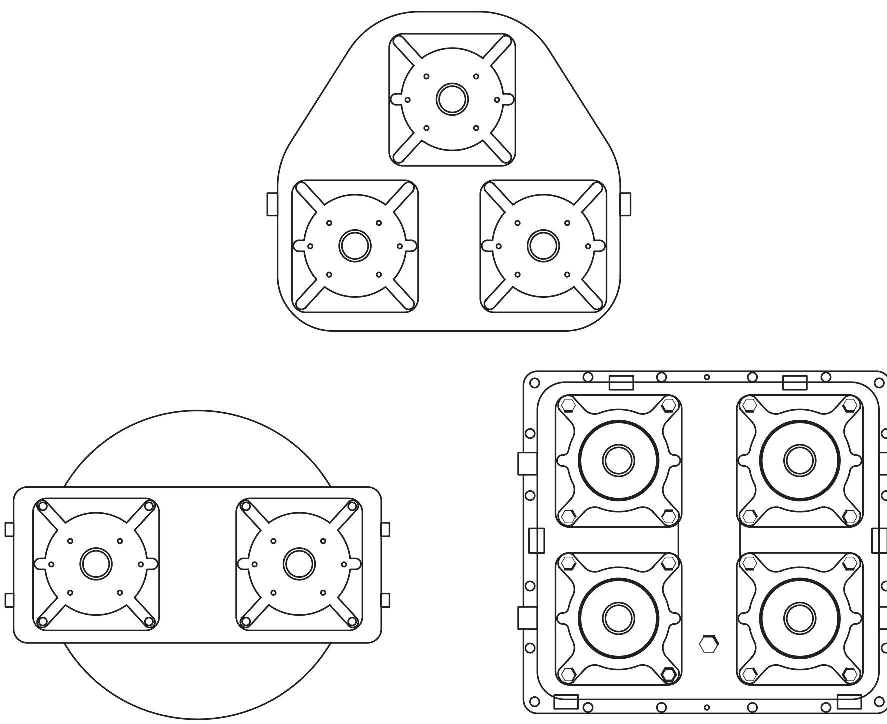


FIGURE 4.30
Hydraulic pump mount designed to power several individual pump units with one prime mover.

4.8 Flow Control Valves

Flow control valves are used in constant-flow (fixed displacement pump) circuits to control actuator speed. The simplest type of flow control valve is a *needle* valve. Another name for a needle valve is a *non-pressure-compensated flow control valve.* Turning the manual adjustment on a needle valve causes the needle to move down into the orifice, thus reducing the orifice area. Pressure drop across the valve (ΔP_{fc}) is increased by continuously restricting the orifice until enough pressure is produced to cause the relief valve to crack open. At this point, extra turns will further reduce the orifice, increase the ΔP_{fc}, increase the pressure at the relief valve, dump more fluid to the reservoir, and thus slow the actuator. The sequence of events is exactly the same as using a DCV to control actuator speed. If load pressure changes over a narrow range, the needle valve will give fairly good flow control. Again, a reminder is given. Flow across the relief valve represents an energy loss. A needle valve is inexpensive, but the operating cost is high because of the energy loss.

A pressure-compensated flow control valve has a provision for changing the ΔP_{fc} as the load pressure changes. Total pressure at the relief valve,

$$P_r = \Delta P_{fc} + \Delta P_L$$

is maintained nearly constant. As ΔP_L increases, ΔP_{fc} decreases, and vice versa. A constant P_r means a constant load on the pump (leakage is constant) and a constant flow across the relief valve.

A partial schematic of a flow control valve is shown in Fig. 4.31a, and the full schematic in Fig. 4.31b. It is instructive to first do a force balance on the spool of the valve shown in Fig. 4.31a. Suppose the spring is a *100 psi spring,* meaning that it produces a force equivalent to a 100 psi pressure. The force balance on the spool is

$$(P_c A_c - P_c A_r + P_c A_r - 100 A_c) = 0 \tag{4.33}$$

where A_c = area of spool cap end (in^2)
A_r = area of spool rod end (in^2)
P_c = pressure in the cavity between the two spool ends

The pressure P_c must equal 100 psi for the spool to be in force balance. The spool finds the position that maintains 100 psi in the center cavity. If the inlet pressure is 500 psi, this means that the pressure drop across the orifice shown in Fig. 4.31a is ΔP_o = 400 psi. This pressure drop represents an energy loss that is characteristic of this type of valve. The pressure drop between the center

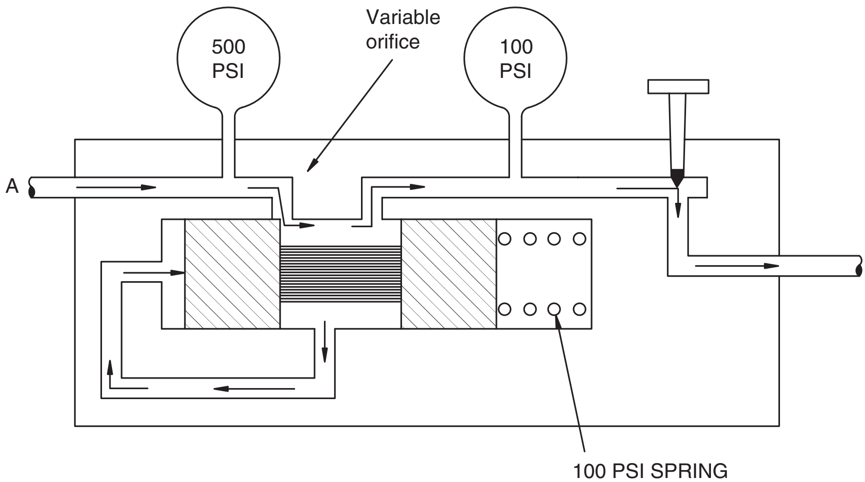


FIGURE 4.31a
Partial diagram of pressure-compensated flow control valve. (Reprinted with permission from Parker Hannifin Corp.)

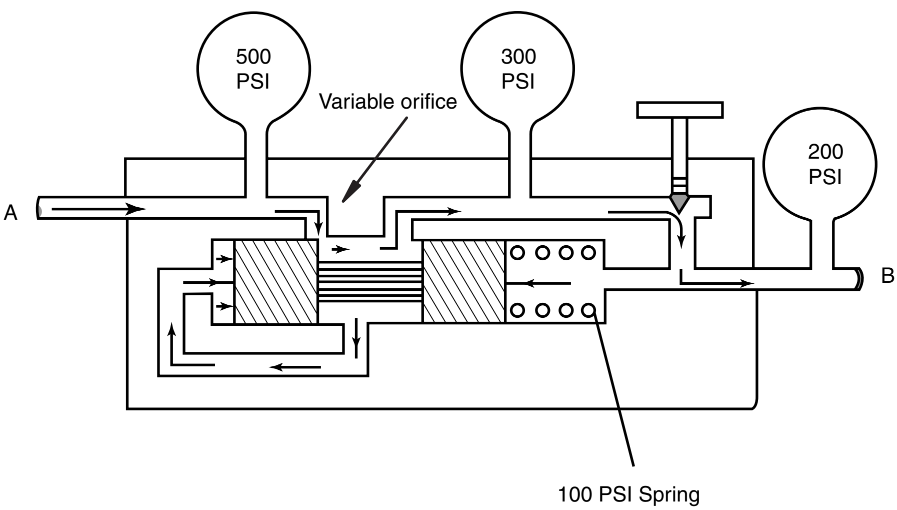


FIGURE 4.31b
Diagram showing the operation of a pressure-compensated flow control valve. (Reprinted with permission from Parker Hannifin Corp.)

cavity and the outlet to the valve is 100 psi. This pressure drop sets the flow through the orifice created by the position of the handwheel adjustment. Repeating the orifice equation given in Chapter 2,

$$Q = k\sqrt{\Delta P}$$

If ΔP is constant, then Q will be constant.

The complete schematic for the flow control valve is shown in Fig. 4.31b. Here, the spring cavity opens to the downstream pressure. Now, the downstream pressure adds to the spring pressure to give the total pressure in the center cavity. Using the pressures shown in Fig. 4.31b,

$$\begin{aligned}\text{Cavity pressure} &= \text{Downstream pressure} + \text{Spring pressure}\\ &= 200 + 100\\ &= 300 \text{ psi}\end{aligned}$$

The spool finds a position where the orifice ΔP is

$$\Delta P_o = 500 - 300 = 200 \text{ psi}$$

If the downstream pressure goes as high as 400 psi,

$$\begin{aligned}P_c &= 400 + 100\\ &= 500 \text{ psi}\end{aligned}$$

then,

$$\Delta P_o = 500 - 500 = 0 \text{ psi}$$

This valve can maintain 100 psi ΔP across the handwheel orifice when load pressure fluctuates from 0 to 400 psi. At higher load pressures, the ΔP across the handwheel orifice will be less than 100 psi, and flow will not be constant.

Denoting the downstream pressure as P_L and the pressure drop across the handwheel orifice as ΔP_{hw}, total pressure at the valve inlet is

$$P_{fc} = \Delta P_o + \Delta P_{hw} + P_L \tag{4.34}$$

Case 1 $P_L = 0$
$P_{fc} = 400 + 100 = 500$ psi

Case 2 $P_L = 200$
$P_{fc} = 200 + 100 + 200 = 500$ psi

Case 3 $P_L = 400$
$P_{fc} = 0 + 100 + 400 = 500$ psi

Note that, for all three cases (P_L varies from 0 to 400 psi), the pressure at the valve inlet is held constant. The pump and relief valve "see" the same pressure, thus flow is held constant.

Some flow control valves have a bypass to a secondary circuit as shown in Fig. 4.32. The valve sleeve moves to set the total pressure drop required to maintain the set flow to the primary circuit. Excess flow is bypassed to the secondary circuit. The sleeve position in Fig. 4.32 shows the pathway to the secondary circuit partly open. If it moves farther to the right, it will close off the secondary circuit.

4.8.1 Flow Dividers

Valves are available that divide the flow from a single pump to supply two circuits that operate at different pressures. These valves are supplied with orifices that give a 50:50 split up to a 90:10 split, generally in increases of 10%.

A cross section of a flow divider is shown in Fig. 4.33. Flow enters Port 1 and exits Port 6. The key component of the flow divider, as with most valves, is the spool, identified in Fig. 4.33 as component 2. This spool has a passage drilled down the center. Fluid enters the spool at 3 and splits to flow along passage 4 in both directions (to the right and to the left). There is an orifice at both ends of passage 4. These orifices are identified as component 5 in the figure. If the orifices are the same size, the flow divider is designed for a 50:50 split.

When the flow through both orifices is the same, the pressure drop is the same at both ends, and the spool is in force balance. Under these conditions, the spool is centered in the valve. Now suppose that the pressure at the left port is lower than at the right port. Fluid entering at 1 takes the path of least resistance and flows toward the left port. Higher flow at the left orifice produces a higher pressure drop at the orifice. The greater pressure on the upstream side of the left orifice creates a force imbalance on the spool and shifts it to the left. The spool moves closer to the end plate and thus partially

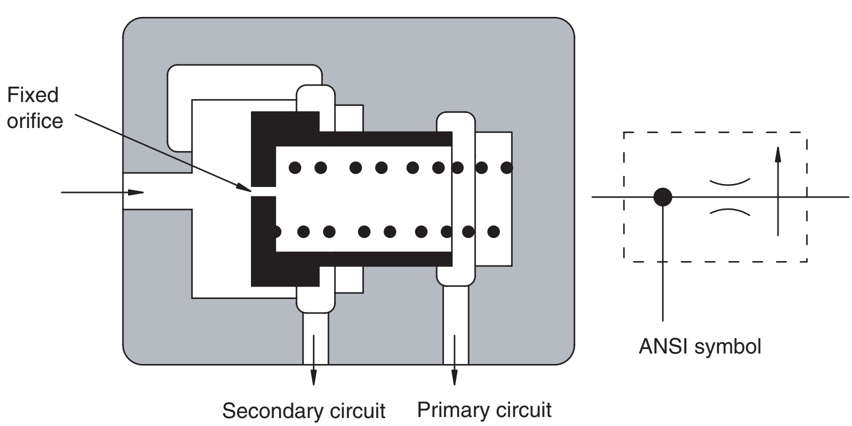


FIGURE 4.32
Flow control valve with bypass.

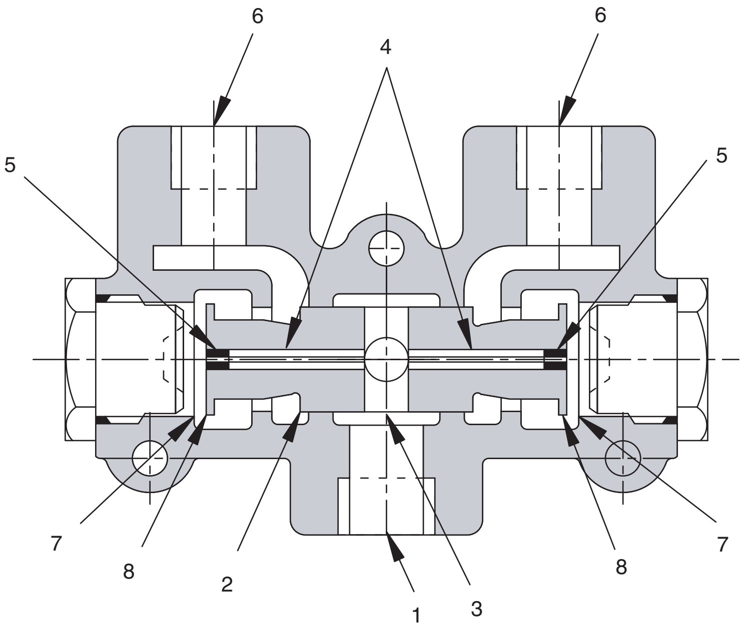


FIGURE 4.33
Cross section of flow divider.

blocks the orifice. *The end of the spool (8) moves toward the end plate (7).* The higher pressure drop across the orifice reduces the flow. The spool moves until it finds the position where flow is equal in both directions. Equal flow produces equal pressures, and the spool is returned to force balance. As pressure changes at the two output ports, the spool continuously moves back and forth to maintain the desired split between the two ports.

Keep in mind that any pressure drop through a flow divider converts hydraulic energy to heat energy. Operating cost for a flow divider is relatively high. The decision to use a flow divider should not be made until the option of using a tandem pump has been discarded. A separate pump for each circuit will always give better isolation of the two circuits.

4.9 Circuits Using Flow Control Valves

A flow control valve can be used to meter into the actuator (Fig. 4.34), meter flow out of the actuator (Fig. 4.35), or to bleed flow from the circuit (Fig. 4.36). A flow control valve with a built-in check valve is used for the meter-in and meter-out circuits so that the cylinder can be retracted at full speed. For the bleed-off circuit (Fig. 4.36), no check valve is needed, because a return flow

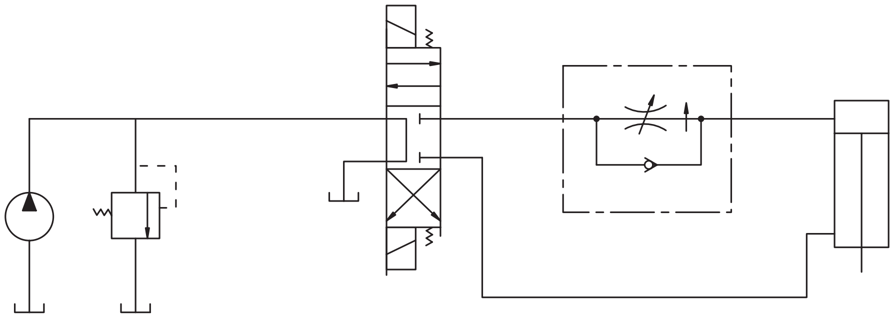


FIGURE 4.34
Circuit illustrating meter-in flow control of cylinder speed.

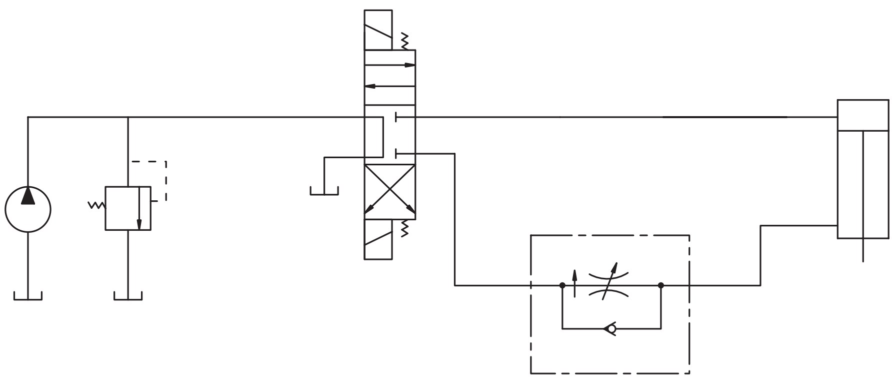


FIGURE 4.35
Circuit illustrating meter-out flow control of cylinder speed.

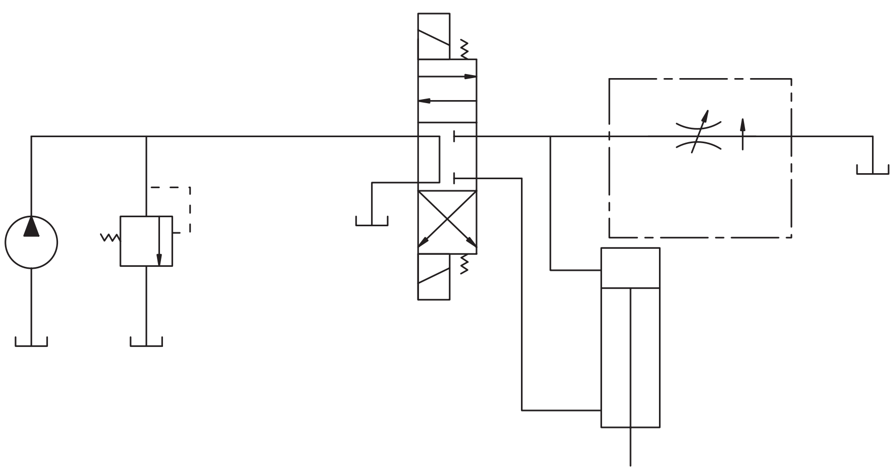


FIGURE 4.36
Circuit illustrating bleed-off control of cylinder speed.

path through the DCV is provided. Flow will take the path of least resistance; therefore, it will go through the DCV rather than through the flow control valve.

4.10 Summary

The first step in the utilization of fluid power is the conversion of mechanical energy into fluid energy. The pump does this conversion and, in that sense, it is the "heart" of a fluid power circuit.

There are three main pump designs: gear, vane, and piston. Volumetric efficiency (volume of fluid delivered based on the theoretical delivery) is primarily a function of pressure. The gear pump has the highest leakage flow at a given pressure; consequently, it has the lowest volumetric efficiency. Piston pumps have the highest volumetric efficiency and are rated for maximum pressures up to 7000 psi. The pumps range in cost from gear (lowest) to piston (highest). All three designs can provide acceptable performance when they are placed in a properly designed circuit.

Overall efficiency, defined as output hydraulic power divided by input mechanical power, is highest for the piston pump and lowest for the gear pump. Total power loss is the sum of friction power loss and leakage power loss. Friction power loss is higher in the vane pump than in the piston pump.

Both the vane and piston pumps are available as variable displacement pumps. The displacement can be changed manually or by providing feedback of the outlet pressure to a small cylinder in the pump. This cylinder positions the cam ring (vane pump) or swashplate (piston pump) to set the displacement. A pressure-compensated pump reduces output flow to zero when pressure rises to a given level. Both the vane and piston pumps are available as pressure-compensated pumps.

When two or more pumps are included in the same housing, the design is referred to as a *multiple pump.* Multiple pumps have separate outputs for each pump and are used to provide isolation between circuits. Each circuit has its own pump; consequently, pressure fluctuations in one circuit do not affect flow in the other circuit.

Pump mounts are gearboxes that allow the mounting of one to four separate pumps (single or multiple) on the same prime mover. They are available with or without clutches and with different gear ratios.

Many pumps have auxiliary mounts to drive a second pump off the same input shaft. Multiple pumps can be mounted on these auxiliary mounts to power several pumps with one mechanical power input.

Pump designs with a wide range of operating characteristics are available. A designer must select carefully to achieve a circuit design that meets the functional objective while minimizing total cost (ownership cost + operating cost) over the life of the components. Pump selection is a very important decision in achieving this goal.

Flow control valves are used to divide the flow from one pump to supply more than one actuator simultaneously. If two actuators require different pressures, the flow will tend to go to the lowest pressure branch of the circuit first. A pressure-compensated flow control valve will do a reasonable job of dividing the flow when the circuit branches have unequal pressures. Ownership cost for the flow control valve option is usually less as compared to the multiple pump option, but flow control accuracy is also less.

References

Jarboe, H.R. 1983. Agricultural load-sensing hydraulic systems. ASAE Distinguished Lecture Series. Tractor Design No. 9. Am. Soc. of Agric. Eng., 2950 Niles Rd., St. Joseph, MI 41085-9659.

Problems

4.1 The diameter of the cylinder shown in Fig. 4.37 is d = 4 in., and the cylinder ratio is 2:1. The load being moved is W = 11,000 lb_f, and the extension velocity is 6 in/s. The flow vs. pressure characteristics of the circuit were measured and are plotted in Fig. 4.38. What pressure drop must be developed across the flow control valve to reduce the extension velocity to 5 in/s? (Neglect the pressure drop in the lines and across the directional control valve.)

Hint: The most accurate method for calculating the pressure to give the desired flow is to write the equation for the Q vs. P curve given in Fig. 4.38.

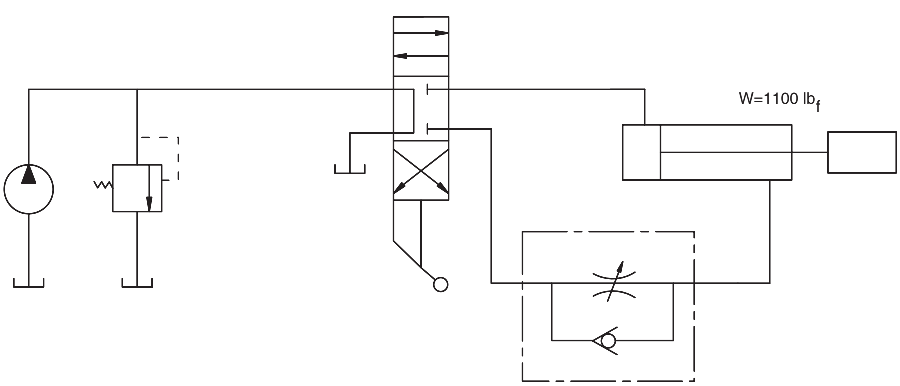


FIGURE 4.37
Circuit for Problem 4.1.

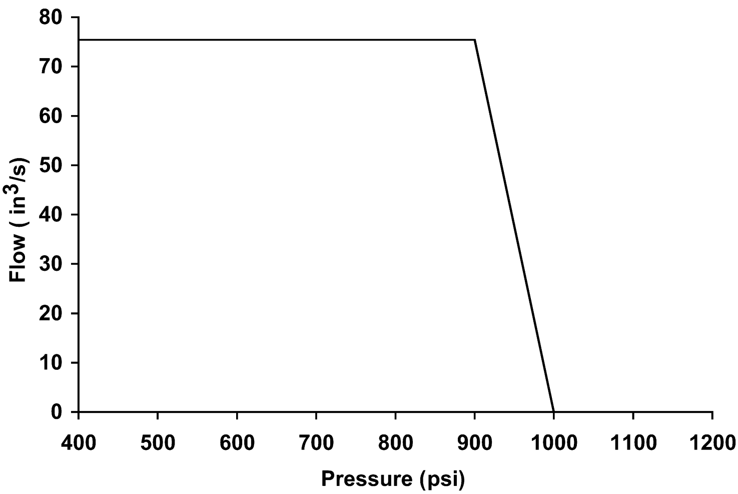


FIGURE 4.38
Characteristics of relief valve used in the circuit for Problem 4.1.

$$P_1 = 900 \text{ psi} \qquad P_2 = 1000 \text{ psi}$$
$$= Q_1 = 75.4 \text{ in}^3/\text{s} \qquad Q_2 = 0$$

4.2 The drain line on a hydraulic pump was allowed to drain into a container. (The drain line provides a pathway for fluid that leaks into the housing to flow back to the reservoir.) The container was weighed every two minutes, and these measurements used to calculate the leakage flow rate. Pressure drop across the pump was measured to obtain the data given in Table 4.4. Pump displacement is 3.15 in^3/rev, and it is operated at nominal speed of 2925 rpm. Measured pump speed at each loading (each pressure reading) is given in Table 4.4. Note that the pump speed slows as the load on the prime mover increases.

TABLE 4.4
Measured Leakage into Pump Housing

Pressure ΔP (psi)	Leakage flow Q_l (in^3/s)	Pump speed (rpm)
500	1.54	2925
1000	1.84	2925
1500	2.26	2875
2000	2.92	2780
2500	4.17	2650
3000	6.56	2500
3500	9.10	2250

Deduce the volumetric efficiency from the data in Table 4.4 and present these data in tabular and graph form.

4.3 A gear pump with 4.53 in^3/rev displacement is operated at 1800 rpm. The volumetric efficiency was measured for two test conditions. For Test 1, the temperature of the fluid was controlled at 140°F. For Test 2, the temperature was maintained at 160°F. Data collected is given in Table 4.5 and plotted in Fig. 4.39. Pump speed was held constant. Use these results to deduce the orifice coefficient for the leakage flow. You may assume that the case pressure is 50 psi; thus, the pressure drop across the "orifice" in the leakage pathway is

TABLE 4.5

Measured Volumetric Efficient for Tests Run on a Gear Pump Operated at a Constant 1800 RPM Input Speed

	Volumetric efficiency (%)	
Pressure	**Test 1 (140°F)**	**Test 2 (160°F)**
500	93.1	92.5
1000	90.7	90.2
1500	88.7	88.0
2000	87.0	86.2
2500	83.7	82.7

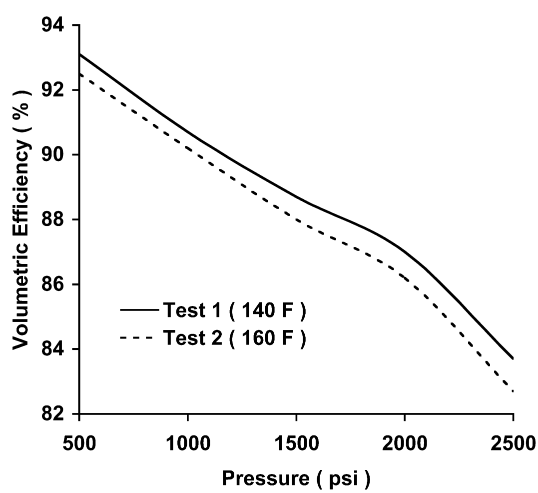


FIGURE 4.39
Volumetric efficiencies calculated from test data collected on gear pump operated at 1800 rpm with fluid temperatures of 140°F (Test 1) and 160°F (Test 2).

$$\Delta P = P_s - 50$$

where P_s = supply pressure (pressure at pump outlet) (psi).

Use the form of the orifice equation

$$Q = k\sqrt{\Delta P}$$

or

$$k_{eq} = \frac{Q}{\sqrt{\Delta P}}$$

Use in^3/s for flow and psi for pressure and calculate the k for each pressure in Tests 1 and 2. Plot the values of k vs. pressure (two curves) and discuss the influence of pressure and temperature on the equivalent orifice coefficient, k.

4.4 A variable displacement axial piston pump is operated at 1800 rpm with the swash plate set at a 4° angle. Full displacement, 4.57 in^3/rev, is achieved at a 19° swash plate angle. The displacement is

$$V_{p4} = V_{p19}\frac{\tan 4\%}{\tan 19\%}$$

$$= 4.57\left(\frac{0.0699}{0.3443}\right) = 0.928 \text{ in}^3\text{/rev}$$

Tests were run at 2500 psi, and a leakage flow of 4.25 in^3/s was measured by collecting the case drainage and weighing it at measured time intervals. (As a frame of reference, it would take 0.9 min to collect a gallon of case drain flow.) Leakage flow in any axial piston pump is almost constant as displacement is increased. Assume that Q_l = 4.25 in^3/s and is constant as displacement increases.

a. Calculate the volumetric efficiency (e_{vp}) for the displacements corresponding to swash plate angles of 4, 7.8, 11.7, 15.4, and 19°. All other factors are held constant. The pump speed is 1800 rpm and the pressure is 2500 psi.

   Assume that friction power increases with displacement and that this increase is given by the following equation:

$$\mathcal{P}_f = 1.676 + 0.0407V_p$$

where $\mathcal{P}_f$ = friction power (hp)
V_p = pump displacement (in^3/rev)

Assume that pump case pressure is 50 psi.

b. Calculate the overall efficiency (e_{vp}) for the same displacements as in part (a).

c. Present your results in a table and discuss the changes in efficiency as displacement is increased.

4.5 A diesel engine powers a dual pump mount. The pump mounted on the right is a fixed displacement gear pump with a displacement of 1.25 in^3/rev. The pump mounted on the left is a 2.3 in^3/rev axial piston pump. Load on the two pumps varies such that the required pressure is given by the idealized curves in Fig. 4.40. The pumps are driven at 2000 rpm, and the overall efficiency is given by the following equations:

$$\text{Gear pump: } e_{op} = D_1P + E_1 + (A_1 + B_1P + C_1P^2)^{\frac{1}{2}}$$

$$\text{Piston pump: } e_{op} = D_2P + E_2 - (A_2 + B_2P + C_2P^2)^{\frac{1}{2}}$$

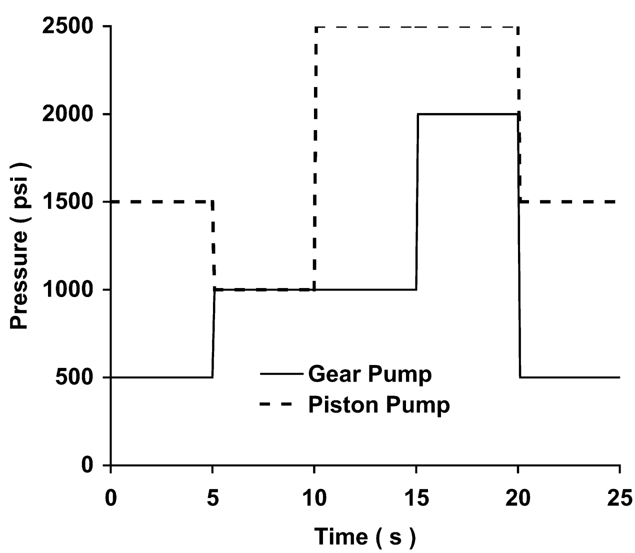


FIGURE 4.40
Idealized curves for two pumps mounted in a dual pump mount and driven by a diesel engine (Problem 4.5).

where P = pressure (psi) developed by pump.

$A_1 = 0.4262647 \times 10^{-1}$ $A_2 = 0.3259060 \times 10^{2}$

$B_1 = -0.1004802 \times 10^{-3}$ $B_2 = -0.3380966 \times 10^{-1}$

$C_1 = 0.5930394 \times 10^{-7}$ $C_2 = 0.1032614 \times 10^{-4}$

$D_1 = -0.4061440 \times 10^{-2}$ $D_2 = 0.1288487 \times 10^{-2}$

$E_1 = 0.9204993 \times 10^{2}$ $E_2 = 0.9392986 \times 10^{2}$

Assume that the mechanical efficiency of the dual pump mount is 97%.

a. Plot the torque (lb_f-in) requirement on the engine for each time interval.
b. Find the required engine power (hp) for each time interval and discuss the range of power fluctuation.

5

Rotary Actuators

5.1 Introduction

Many of the concepts developed in Chapter 4 for pumps are applicable to hydraulic motors. Motors convert fluid energy back into mechanical energy and thus are the mirror image of pumps. It is not surprising that the same mechanisms are used for both. The typical motor designs are gear, vane, and piston.

Motor performance is a function of pressure. As pressure increases, leakage increases, speed decreases, and thus the quantity of mechanical energy delivered to the load decreases. Motor volumetric efficiency is defined as

$$e_{vm} = \frac{\text{Actual motor speed}}{\text{Theoretical motor speed}} \tag{5.1}$$

Repeating Eq. (4.1), pump volumetric efficiency is

$$e_{vp} = \frac{\text{Actual flow}}{\text{Theoretical flow}}$$

In the most basic sense, the purpose of the pump is to produce flow. The load sets the pressure. The purpose of the motor is to receive this flow and reproduce rotary motion. The definitions for volumetric efficiency present the same concept but reflect the difference in function.

It is appropriate to present a simple example. Suppose a motor has a displacement of 3.9 in^3/rev. Measured flow to this motor is 10 GPM. The theoretical output speed is

$$N_{mth} = \frac{231Q}{V_{mth}} \tag{5.2}$$

where N_{mth} = theoretical motor speed (rpm)

Q = flow (GPM)
V_{mth} = motor displacement (in^3/rev)

Substituting,

$$N_{mth} = \frac{231(10)}{3.9} = 592 \text{ rpm}$$

Assuming the measured output speed is found to be 536 rpm, the motor volumetric efficiency is

$$e_{vm} = \frac{536}{592} \times 100$$

$$= 90.5\% \tag{5.3}$$

For a given motor design (gear, vane, or piston), the change in motor volumetric efficiency with an increase in pressure is very similar to the performance curves shown in Chapter 4. Discussion of this performance need not be repeated here.

The overall efficiency of a hydraulic motor is defined by

$$e_{om} = \frac{\text{Actual output power}}{\text{Input power}} \tag{5.4}$$

The input power is the hydraulic power measured at the motor inlet port, and the output power is mechanical power delivered by the motor output shaft. A simple example will illustrate how motor overall efficiency is calculated.

The motor in the previous example (V_{mth} = 3.9 in^3/rev) is operating with a 2000 psi pressure drop across the ports. Measured flow to the motor is 10 GPM, therefore hydraulic power input is

$$\mathcal{P}_{in} = \frac{\Delta PQ}{1714} \tag{5.5}$$

where $\mathcal{P}_{in}$ = input hydraulic power (hp)
ΔP = pressure drop (psi)
Q = flow (GPM)

Substituting,

$$\mathcal{P}_{in} = \frac{2000(10)}{1714}$$

$$= 11.67 \text{ hp}$$

Measured output torque is 1080 lb_f-in at a speed of 536 rpm. Consequently, the output mechanical power is

$$\mathcal{P}_{out} = \frac{TN}{63025} \tag{5.6}$$

where $\mathcal{P}_{out}$ = output mechanical power (hp)
T = measured output torque (lb_f-in)
N = measured output speed (rpm)

Substituting,

$$\mathcal{P}_{out} = \frac{1080(536)}{63025}$$

$$= 9.18 \text{ hp}$$

The motor overall efficiency is

$$e_{om} = \frac{\mathcal{P}_{out}}{\mathcal{P}_{in}} \times 100$$

$$= \left(\frac{9.18}{11.67}\right) \times 100$$

$$= 78.7\% \tag{5.7}$$

Torque efficiency is another parameter that is sometimes used to describe hydraulic motor performance. It is defined as follows:

$$e_{tm} = \frac{\text{Actual output torque}}{\text{Theoretical output torque}} \tag{5.8}$$

As derived in Chapter 1, theoretical torque output is

$$T_{mth} = \frac{\Delta P V_{mth}}{2\pi} \tag{5.9}$$

where ΔP = pressure drop across motor (psi)
V_{mth} = displacement (in^3/rev)

Substituting,

$$T_{mth} = \frac{2000(3.9)}{2\pi}$$

$$= 1241 \text{ lb}_f\text{-in}$$

Actual output torque is 1080 lb_f-in, so the torque efficiency is

$$e_{tm} = \left(\frac{1080}{1241}\right) \times 100$$

$$= 87\%$$

It can be shown that the overall efficiency is the product of the volumetric and torque efficiencies. Volumetric efficiency is

$$e_{vm} = \frac{N}{N_{mth}} \tag{5.10}$$

where N = actual output speed (rpm)
N_{mth} = theoretical output speed (rpm)

Substitution of

$$N_{mth} = 231 \frac{Q}{V_{mth}}$$

into Eq. (5.10) gives

$$e_{vm} = \frac{NV_{mth}}{231Q} \tag{5.11}$$

Torque efficiency is

$$e_{tm} = \frac{T}{T_{mth}} \tag{5.12}$$

where T = measured output torque (lb_f-in)
T_{mth} = theoretical output torque (lb_f-in)

Substitution of

$$T_{mth} = \frac{\Delta P V_{mth}}{2\pi}$$

into Eq. (5.12) gives

$$e_{tm} = \frac{2\pi T}{\Delta P V_{mth}} \tag{5.13}$$

Multiplying these two efficiencies together,

$$\begin{aligned} e_{vm}e_{tm} &= \frac{NV_{mth}}{231Q} \times \frac{2\pi T}{\Delta P V_{mth}} \\ &= \frac{2\pi TN}{231\Delta PQ} \\ &= \frac{TN}{36.76\Delta PQ} \end{aligned} \tag{5.14}$$

By definition,

$$\begin{aligned} e_{om} &= \frac{\mathcal{P}_{out}}{\mathcal{P}_{in}} \\ &= \frac{2\pi TN}{231\Delta PQ} \\ &= \frac{TN}{36.76\Delta PQ} \end{aligned} \tag{5.15}$$

It is clear then that

$$e_{om} = e_{vm}e_{tm} \tag{5.16}$$

Using the data from the previous example,

$$\begin{aligned} e_{om} &= 0.905(0.87) \\ &= 0.787 \end{aligned}$$

which agrees with Eq. (5.7).

5.2 Stall Torque Efficiency

Stall torque efficiency is defined by

$$e_{sm} = (T_{ms}/T_{mth})100 \tag{5.17}$$

where e_{sm} = stall torque efficiency (%)
T_{ms} = measured torque developed at stall (lb_f-in)
T_{mth} = theoretical torque (lb_f-in)

For practical purposes, *stall* is defined as an output speed less than 1 rpm.

Typical performance of a high-speed hydraulic motor is shown in Fig. 5.1. Our previous discussion of motor characteristics has always used pressure as the independent variable. In Fig. 5.1, motor output speed is the independent variable. Intuitively, we do not expect speed to affect output torque because

$$T_{mth} = \Delta P \frac{V_{mth}}{2\pi}$$

and N does not appear in the equation. In fact, torque output for most high-speed motor designs decreases when speed drops below 200 rpm. A "family" of motors (same basic features but a range of displacements) with torque efficiencies greater than 85% can have stall torque efficiencies ranging from 85% down to 55%.

Stall torque is of critical importance for mobile applications. Because of inertia, a high torque is required to start a stationary vehicle. Sometimes hydraulic motors must be sized on the basis of *stall torque* rather than *operating torque* characteristics.

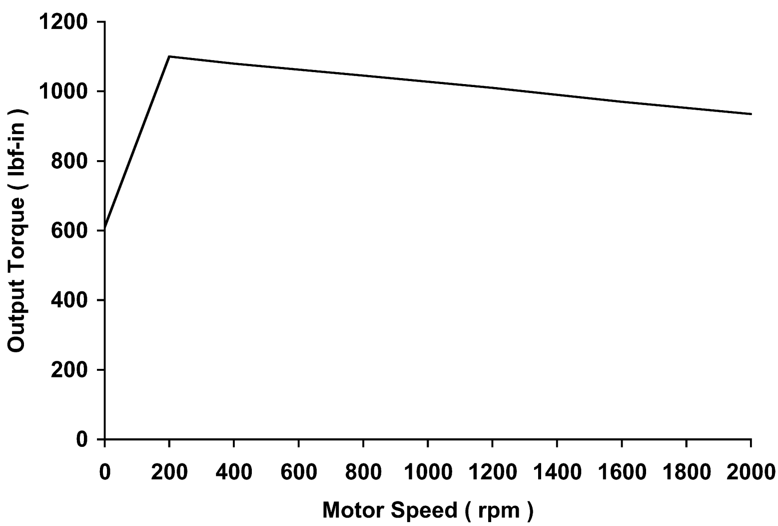


FIGURE 5.1
Torque output vs. motor output speed for typical high-speed hydraulic motor.

There is another characteristic of high-speed motors that causes problems at low output speeds. It is the irregular output speed known as *cogging*. At very low speed, the output speed is jerky. In certain instances, this phenomena is readily visible.

Most axial piston motors have an odd number of pistons, typically nine. This design is used to avoid *hydraulic lock-up,* which could occur if the same number of pistons were at high and low pressure. The cogging phenomenon is the result of the 5:4, 4:5 ratio of pistons at high versus low pressure. This happens nine times per revolution and gives a *torque ripple.* (The phenomenon is also observed with radial piston motors.) Manufacturers generally quote a smooth output speed beginning at 100 rpm for both axial piston and bent axis motors.

Low-speed, high-torque motors were developed to address the low-speed problems observed with conventional designs. In this text, the high-speed designs, comparable to pump designs discussed in Chapter 4, will be discussed first.

5.3 Typical Performance Data for a Gear Motor

Manufacturer's data for a gear motor are given in Fig. 5.2. These data serve as an example of typical data supplied by a manufacturer. The sloping horizontal curves are pressure drop (ΔP across motor), and the sloping vertical curves are flow to the motor. The vertical grid lines are output speed, and the horizontal grid lines are output torque. The operating point for the previous example (10 GPM, 2000 psi) is shown in the figure.

Data from Fig. 5.2, published by the manufacturer, was used to calculate the efficiencies given in Table 5.1. The 10 GPM curve was used for all three operating points, 1000, 2000, and 3000 psi. The volumetric and overall efficiencies follow the same trend seen for the gear pump (Chapter 4).

TABLE 5.1

Performance Data for Gerotor-Type Motor Supplied with 10 GPM Flow

	Efficiency (%)		
Pressure drop across motor (psi)	**Volumetric, e_{vm}**	**Torque, e_{tm}**	**Overall, e_{om}**
1000	99.7	86.5	86.0
2000	90.5	87.0	78.7
3000	73.5	85.5	62.8

Leakage increases as pressure increases; consequently, efficiency decreases. For this particular design of gear motor, the efficiency decreases significantly when pressure increases above 2000 psi. A secondary effect is the deforma-

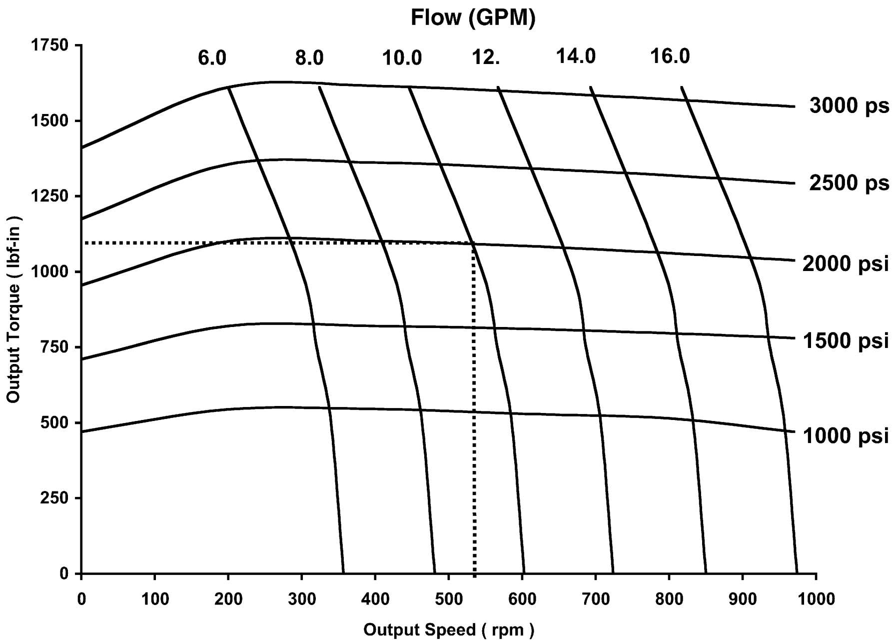


FIGURE 5.2
Performance data for gerotor-type motor (Vickers Model CR-04).

tion of the components. Clearance between parts increases with pressure, thus the effective area of the leakage pathway increases.

Torque loss is defined by

$$T_\ell = T_{mth} - T \tag{5.18}$$

where T_{mth} = theoretical torque (lb_f-in)
T = measured torque (lb_f-in)

The torque loss at the various load pressures is given in Table 5.2. Since torque loss for this motor design is approximately a linear function of pressure, the torque efficiency is approximately constant (Table 5.1).

TABLE 5.2

Torque Loss Data for Gerotor-Type Motor Supplied with 10 GPM Flow

Pressure drop across motor (psi)	Torque loss T_l (lb_f-in)
1000	83.8
2000	161.4
3000	270.0

5.4 Comparison of Motor Performance Characteristics for Three Main Designs

Motors can be broadly classified into two main groups: high-speed motors and low-speed, high-torque (LSHT) motors. As was done for pumps in Chapter 4, it is instructive to compare the three main designs of high-speed motors (gear, vane, and piston). The comparison is made at a nominal 1800 rpm operating speed, and manufacturer's data are used for the efficiency calculations.

5.4.1 Gear Motor

The gear pump design studied in Chapter 4 can also be used as a motor. The Hydreco Model 1919 motor has a theoretical displacement of 4.53 in^3/rev, a maximum pressure rating of 2500 psi, and 3000 rpm maximum speed rating. Efficiencies were calculated for a 36 GPM input flow to the motor (Fig. 5.3). Volumetric efficiency decreases linearly as pressure increases. Torque efficiency is almost constant above 1500 psi. Overall efficiency is a maximum at 1500 psi and decreases to 76% at 2500 psi. If the Model 1919 pump in Chapter 4 and the Model 1919 motor are used together as a hydrostatic transmission, the overall efficiency at 2500 psi (mechanical energy out at the motor divided by mechanical energy in at the pump) would be 62%. No losses in the lines are considered.

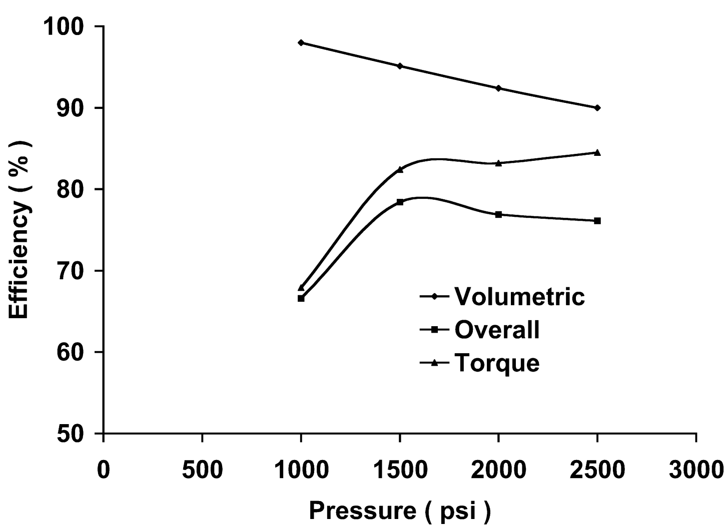


FIGURE 5.3
Efficiencies for gear motor with 36 GPM input flow.

5.4.2 Vane Motor

A Vickers Model 25M (65) vane motor was chosen for the comparison. This motor is rated for 3000 rpm at 2500 psi maximum pressure. It has a displacement of 4.19 in^3/rev. Efficiencies were calculated for a 35 GPM input flow.

As shown in Fig. 5.4, the torque efficiency for this design is higher than for the gear motor, and it is relatively constant over the pressure range 500 to 2500 psi. Overall efficiency for the motor tends to be a little higher than for a comparable vane pump (Fig. 4.24b).

Vane motors are available with two displacement settings. These motors have two rotors. A given flow directed to only one of the rotors will produce twice the speed but only half the torque. When maximum torque is needed, the input flow is sent to both rotors, thus doubling the displacement and doubling the output torque. Speed is, of course, halved. The operator adjusts a valve on the outside of the motor to switch from low-speed, high-torque (two rotors) to the high-speed, low-torque (one rotor) configuration. One application for such a design is a wheel motor for an agricultural machine. The machine needs low speed and high torque for field operations, and the operator shifts to high speed, low torque for road travel between fields.

5.4.3 Piston Motor

The fixed displacement axial piston motor (Sauer-Danfoss Model 90-075 MF), comparable to the variable displacement axial piston pump (Sauer-Danfoss

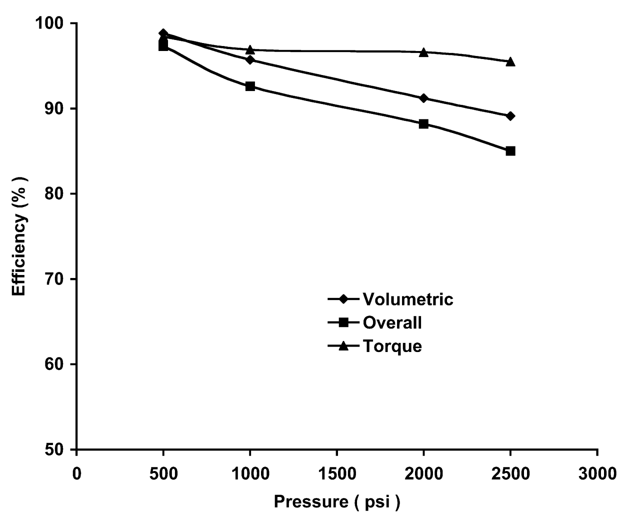


FIGURE 5.4
Efficiencies for vane motor with 35 GPM input flow.

Model 90-075 PV) studied in Chapter 4, was chosen. This motor has a maximum speed of 3950 rpm and a rated pressure of 6000 psi. Displacement is 4.57 in^3/rev. Efficiencies were calculated for a 36 GPM input flow.

Torque efficiency for this design (Fig. 5.5) increases to a maximum at 3000 psi and remains constant at higher pressures. Volumetric efficiency decreases from 99% at 1000 psi to 90.5% at 6000 psi. For comparison, the volumetric efficiency of the equivalent pump (Fig. 4.26b) decreases from 99% at 1000 psi to 90.5% at 6000 psi. At 2500 psi, the overall efficiency of the motor is 92.5% and for the pump it is 93.3%. If this pump and motor are used as a hydrostatic transmission, the overall efficiency (mechanical energy out divided by mechanical energy in), neglecting line losses, is

$$0.933 \times 0.925 = 0.86$$

The gear pump and motor combination had an overall efficiency of 62% at the same operating pressure. This efficiency range, 60 to 85%, is typical for the transfer of mechanical energy with the various designs of pumps and motors available.

A direct comparison of the overall efficiency for the three designs is given in Fig. 5.6. At pressures less than 1000 psi, the vane motor has the highest overall efficiency. At higher pressures, the piston motor has a higher overall efficiency. The gear motor has a lower efficiency but, remember, it is the lowest-cost motor of the three designs. There are applications where a gear motor is the optimal choice, just as there are applications where a vane or piston motor is the optimal choice.

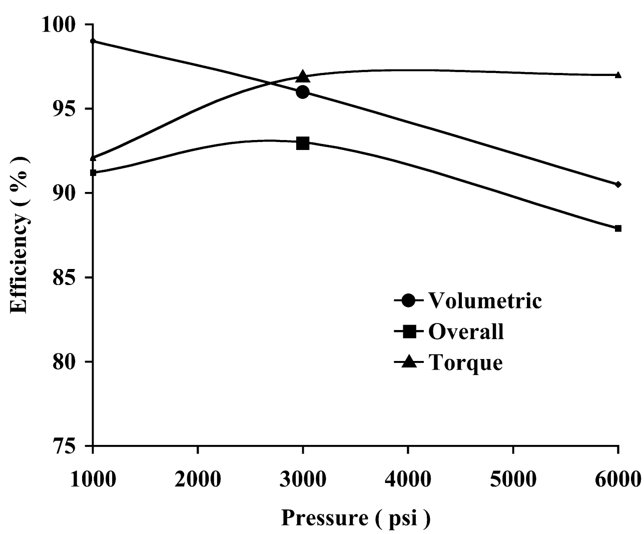


FIGURE 5.5
Efficiencies for piston motor with 36 GPM input flow.

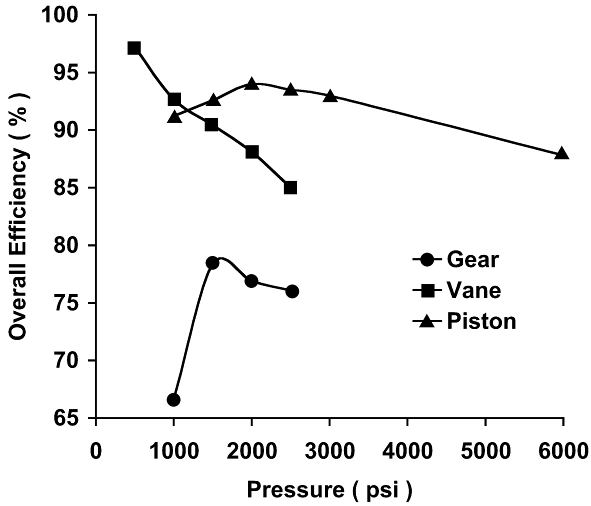


FIGURE 5.6
Comparison of overall efficiencies for gear, vane, and piston high-speed motor at a constant input flow of 35 GPM (nominal).

5.5 Performance Characteristics of Low-Speed, High-Torque Motors

Two typical prime movers used to supply the mechanical energy input for a fluid power circuit are electric induction motors (stationary applications) and diesel engines (mobile applications). The induction motor operates in the range 1700 to 1800 rpm. The diesel engine operating range might typically be 1800 to 2000 rpm. Often the machine operating speed needed is much lower. Low-speed, high-torque motors were designed for these applications.

The "geroler" motor is similar in concept to the gerotor motor. Rather than a gear operating inside a gear (Fig. 4.2), the gear operates inside a housing with rollers in place of the outer gear teeth (Fig. 5.7). This design will be used as the baseline comparison for the low-speed, high-torque family of motors.

One manufacturer uses a disc valve to distribute fluid to the geroler pockets. This valve provides improved performance at low speeds. Some of these motors can run effectively at speeds as low as 1 rpm.

The geroler design is reversible. Changing the direction of fluid flow through the motor changes the direction of shaft rotation.

5.5.1 Geroler Motor (Disc Valve)

The Char-Lynn 10,000 series motor with 40.6 in^3/rev displacement was chosen for an example. (Notice that this displacement is approximately 10 times

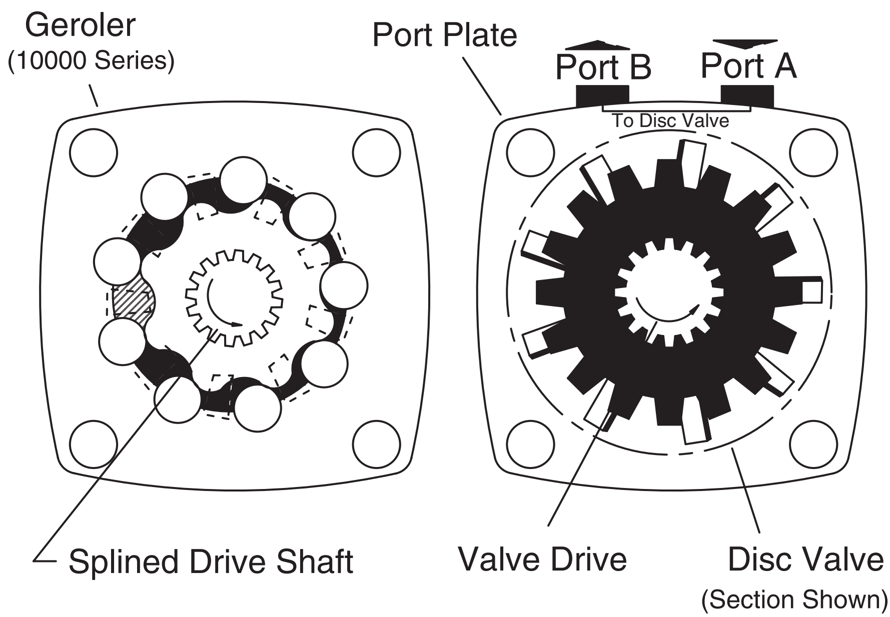


FIGURE 5.7
Geroler motor design. (*Source:* courtesy of Eaton Corp., Hydraulic Div.)

the displacement of the high-speed motors previously studied.) This motor has a maximum speed of 254 rpm for continuous operation. Maximum pressure is 3000 psi. Efficiencies were calculated for an input flow of 36 GPM, the same flow used for the high-speed motor comparison.

Volumetric efficiency for this motor was higher than for the gear motor, ranging from 98% at 500 psi to 93.3% of 2500 psi (Fig. 5.8). Overall efficiency increased from 79.5% at 500 psi to 86% at 1000 psi and remained nearly constant for higher pressures. The speed of this motor is 201 rpm at a pressure drop of ΔP = 500 psi and 191 rpm at 2500 psi, a drop of only 5%.

5.5.2 Vane Motor (Low-Speed, High-Torque)

The Vickers Model MHT50 vane motor with 38 in^3/rev maximum displacement was chosen for an example. This motor is rated for a maximum pressure of 4000 psi. Maximum continuous speed at 3000 psi is 200 rpm and maximum speed at 2000 psi is 350 rpm. Efficiencies were calculated for an input flow of 35 GPM.

Volumetric efficiency declined linearly as pressure increased from 1000 to 3000 psi (Fig. 5.9). Torque efficiency increased as pressure increased from 1000 to 2000 psi and remained approximately constant at higher pressures. The decrease in volumetric efficiency was offset by the increase in torque efficiency in the 1000–2000 psi range, and overall efficiency decrease was mod-

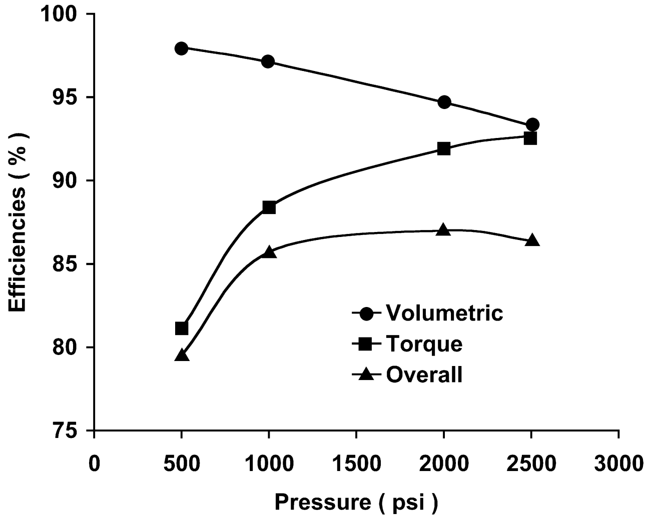


FIGURE 5.8
Efficiency for geroler motor (low-speed, high-torque) with 36 GPM input flow.

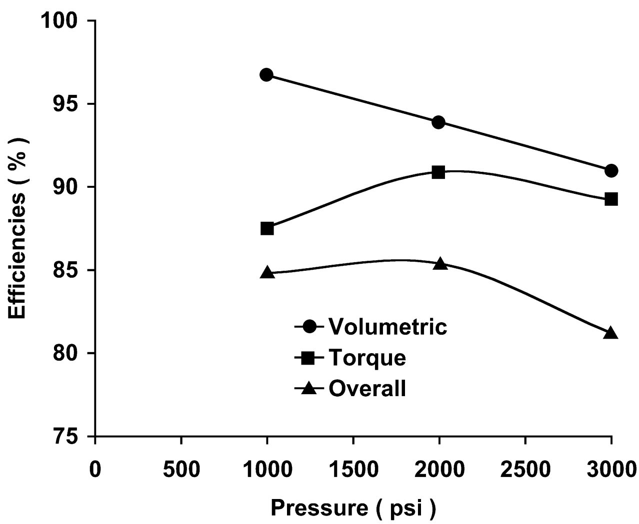


FIGURE 5.9
Efficiency for vane motor (low-speed, high-torque) with 35 GPM input flow.

erated. Overall efficiency declined from 85.4 to 81.2% as pressure increased from 2000 to 3000 psi.

As was done with the high-speed motors, it is instructive to compare the overall efficiency for the two designs (Fig. 5.10). These two designs have approximately equal performance in the 1000–2000 psi range. Other factors

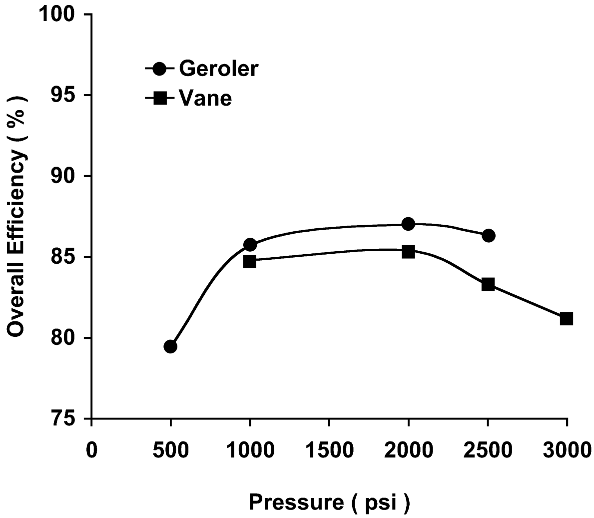


FIGURE 5.10
Comparison of overall efficiency for geroler and vane motor designs (low-speed, high-torque) at a constant flow of 35 GPM (nominal).

(for example, sensitivity to contamination) would be considered in making a choice between the two designs. For higher pressures (2000–3000 psi), the vane motor may offer an overall advantage.

Axial piston motors have high efficiencies at high pressures but are not offered in the low-speed, high-torque classification. As will be seen when we discuss hydrostatic transmissions in Chapter 6, the piston motor can be used to power a planetary gear drive to achieve the low-speed, high-torque output. This choice gives the highest overall efficiency, but the initial cost is higher, because the piston motor is higher in cost, and a planetary gear set is an extra component to be purchased. Another key consideration is the stall torque efficiency of the piston motor. Remember, high torque is required to get a vehicle started.

5.6 Design Example for Gear Motor Application

Gear motors can be an optimum selection for a given application. Care must be used to ensure that a displacement is chosen that will give an operating pressure in the range in which these motors have a competitive efficiency. The following example illustrates the design process that is used for the selection of any motor. The example uses a "family" of gear motors for the illustration.

5.6.1 Functional Requirements

A motor load is expected to average 1000 lb_f-in, with peaks as high as 1500 lb_f-in. The desired speed is 300 rpm, and quality control requires that this speed not fluctuate more than ±5%, equivalent to ±15 rpm. The objective of this exercise is to select a motor that will meet these functional requirements.

Trial No. 1

The Model CR-04 motor discussed in Sec. 5.2 is the smallest in the "family," so we will consider it first. Find the intersection of the 1000 lb_f-in line and the 300 rpm line in Fig. 5.2. An input flow of 6 GPM is required. Interpolating between the 1500 and 2000 psi curves, the pressure drop will be $\Delta P = 1810$ psi. What happens when the torque requirement increases to 1500 lb_f-in? (At this point, we are not considering the influence of pressure rise on the pump, so we assume the flow stays at 6 GPM). Projecting the intersection of the 1500 lb_f-in line and the 6-GPM curve, we find that speed is $N_o = 218$ rpm. This output speed represents a 27% speed droop, which does not meet our ±5% speed fluctuation criteria.

Trial No. 2

Performance data for the Model CR-08 motor ($V_{mth} = 7.7$ in^3/rev) are given in Fig. 5.11. The intersection of the 300 rpm and 1000 lb_f-in lines show that the flow requirement is 10.8 GPM and $\Delta P = 975$ psi. If the torque requirement increases to 1500 lb_f-in, the output speed drops to 290 rpm, a speed drop of only 3.3%. The speed fluctuation constraint is met by doubling the motor size.

5.6.2 Other Design Considerations

The Model CR-08 motor requires 10.8 GPM as compared to the 6 GPM required by the Model CR-04 motor. Selection of the larger motor requires the selection of a larger pump and also larger capacity lines and other components. A cost comparison must consider the extra cost of these components, in addition to the higher cost of the larger pump.

5.7 Interaction of Pump and Motor Characteristics

The previous design example considered only the characteristics of the motor. It is appropriate now to consider an example for which the characteristics of the pump *and* motor are considered. We learned in Chapter 4 that the pump output decreases as load pressure increases. Thus, the flow to the motor does not stay constant, as was assumed in the previous example; it

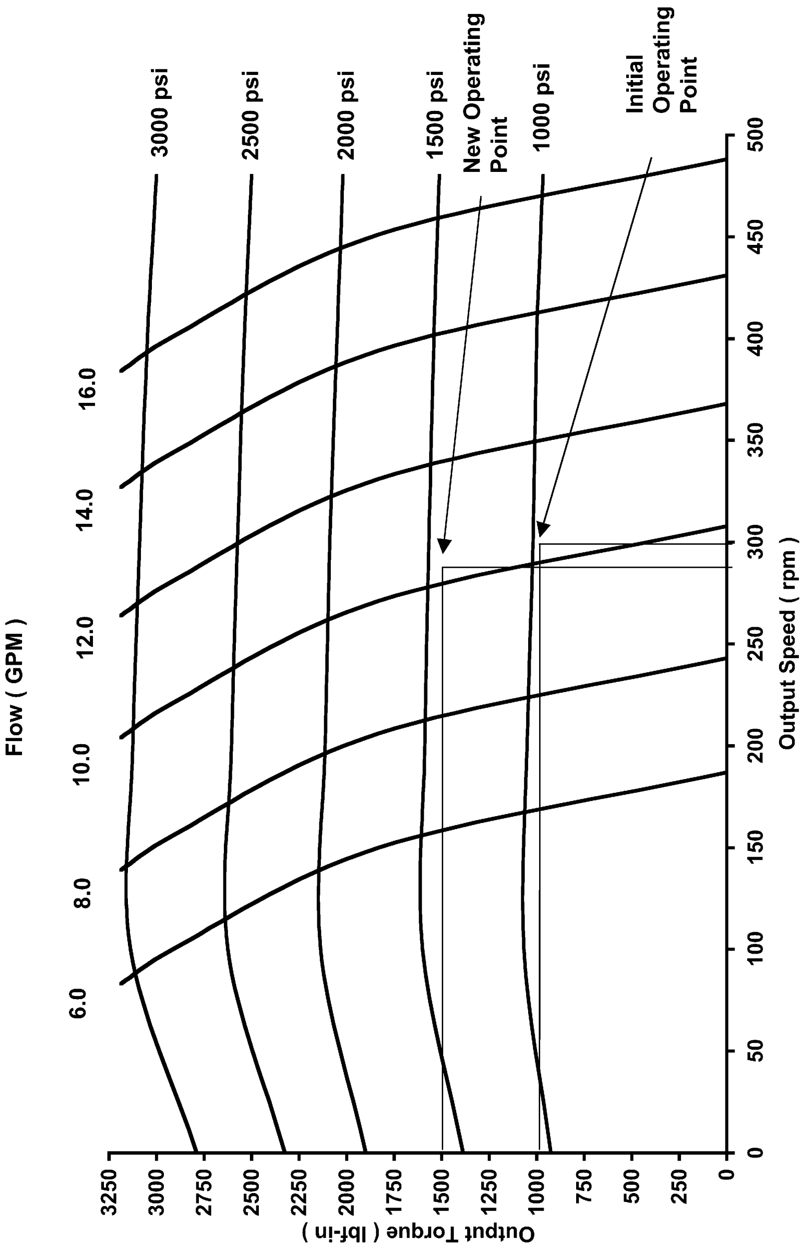


FIGURE 5.11
Manufacturer's test data for low-speed, high-torque motor (Vickers Model CR-08).

decreases as load pressure increases. This example will deal with the issue of speed change as a function of load change.

A hydraulic motor is used to drive a time-varying load (Fig. 5.12). In "real-world" situations, all loads are time-varying, particularly in a manufacturing situation where operations are completed in a sequence. There is generally a cycle where the pressure increases and then decreases. Sometimes, speed change during this cycle is critical.

Load pressure as a function of time [$P(t)$] is given in Fig. 5.13. The requirement is to plot the percentage change in motor output speed as pressure varies. No characteristics of the prime mover are considered; it turns at a constant 1800 rpm independent of the torque required to drive the pump.

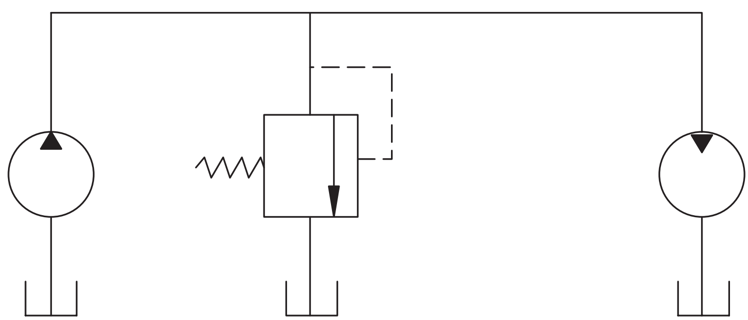


FIGURE 5.12
Open-circuit configuration for a hydraulic motor powering a time-varying load.

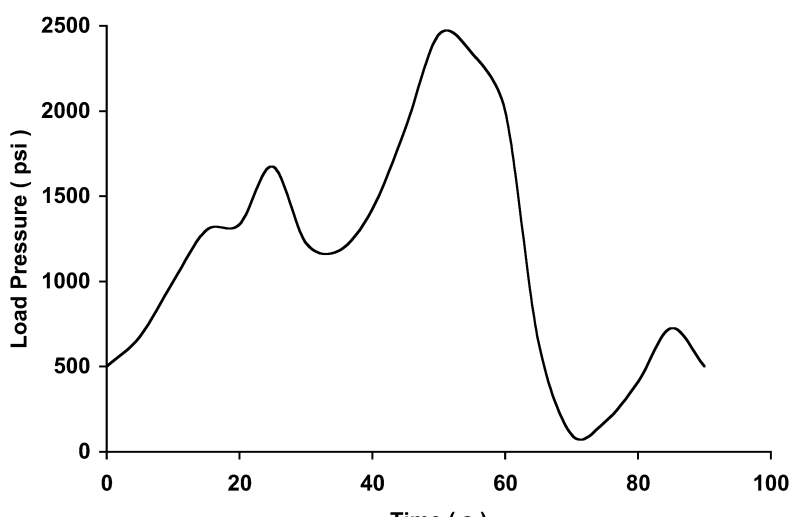


FIGURE 5.13
Load pressure as a function of time.

The Hydreco Model 1919 pump and motor were chosen to represent the gear design. The axial piston design is represented by the Sauer-Danfoss Model 90-075 pump and motor. (The Sauer-Danfoss Model 90-075 is a variable displacement pump, and for this example, it is operated at maximum displacement.)

The equation for pump volumetric efficiency is

$$e_{vp} = DP + E + (A + BP + CP^2)^{1/2} \tag{5.19}$$

where e_{vp} = pump volumetric efficiency (%)
P = pressure (psi)
A, B, C, D, E = constants (Table 5.3)

The reader is referred to Appendix 5.1 for an explanation of the procedure used to calculate the A, B, C, D, and E constants.

TABLE 5.3

Coefficients for Volumetric Efficiency Equations

	Gear		Piston	
Coefficient	**Pump**	**Motor**	**Pump**	**Motor**
A	0.0	0.0	0.0	0.0
B	0.0	0.0	0.0	0.0
C	0.0	0.0	0.0	0.0
D	-4.7×10^{-3}	-5.0×10^{-3}	-1.7×10^{-3}	-1.7×10^{-3}
E	9.54×10^{1}	1.025×10^{2}	1.005×10^{2}	1.007×10^{2}

The equation for the motor volumetric efficiency is

$$e_{vm} = D\,P + E + (A + BP + CP^2)^{1/2} \tag{5.20}$$

where e_{vm} = motor volumetric efficiency (%)
P = pressure (psi)
A, B, C, D, E = constants (Table 5.3)

Equations of this form were chosen because, generally, any volumetric efficiency curve can be represented with the proper choice of the constants A, B, C, D, and E. The curvature of the plotted curve is so small that the A, B, and C constants are negligible. In this case, Eqs. (5.19) and (5.20) reduce to the equation for a straight line.

Pump output flow is given by

$$Q = N_p\, V_{pth}\, e_{vp}/100 \tag{5.21}$$

where Q = flow delivered by pump (in^3/min)
V_{pth} = pump displacement (in^3/rev)

N_p = pump speed (rev/min)
e_{vp} = pump volumetric efficiency (%)

Corresponding motor speed is

$$N_m = \left(\frac{Q}{V_{mth}}\right)\frac{e_{vm}}{100} \tag{5.22}$$

where N_m = motor speed (rev/min)
Q = flow to motor (flow delivered by pump) (in^3/min)
V_{mth} = motor displacement (in^3/rev)
e_{vm} = motor volumetric efficiency (%)

The reference chosen for motor speed change is the speed when pressure is 500 psi. (This is the pressure at time $t = 0$ in Fig. 5.13.)

$$\Delta N_m = \left(\frac{N_m - N_{mo}}{N_{mo}}\right)100 \tag{5.23}$$

where ΔN_m = motor speed change (%)
N_m = motor speed ($t = t$)
N_{mo} = motor speed ($t = 0$)

Motor speed change for both designs is given in Fig. 5.14. For the gear pump-motor combination, the speed change ranges from –23 to + 4%, a total change of 27% from maximum to minimum speed. The piston pump-motor combination has a maximum speed change of –7% as pressure varies as shown in Fig. 5.12. Maximum pressure is 24.5 times minimum pressure, and the speed change is only 7% over this pressure range. This performance is quite good for a hydraulic circuit. Remember, however, that no speed change of the prime mover is considered.

Motor output speed also depends on the characteristics of the prime mover. Typically, the prime mover is an electric motor for a stationary application or a diesel engine for a mobile application. Both these prime movers slow down as the torque requirement increases. As the prime mover slows, pump output decreases, and the motor output speed decreases. The reader can readily understand the importance of analyzing the whole system to predict performance. System interactions will be revisited in later chapters.

5.8 Bent Axis Motors

Bent axis motors were developed to improve the operating and stall torque efficiencies of high-speed motors. The design (Fig. 5.15) operates in a manner

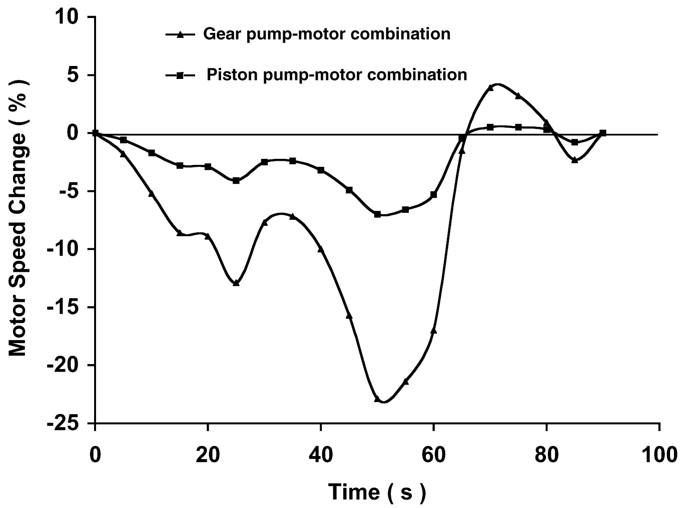


FIGURE 5.14
Motor output speed calculated as a percentage of the speed when load pressure drop across the motor is 500 psi.

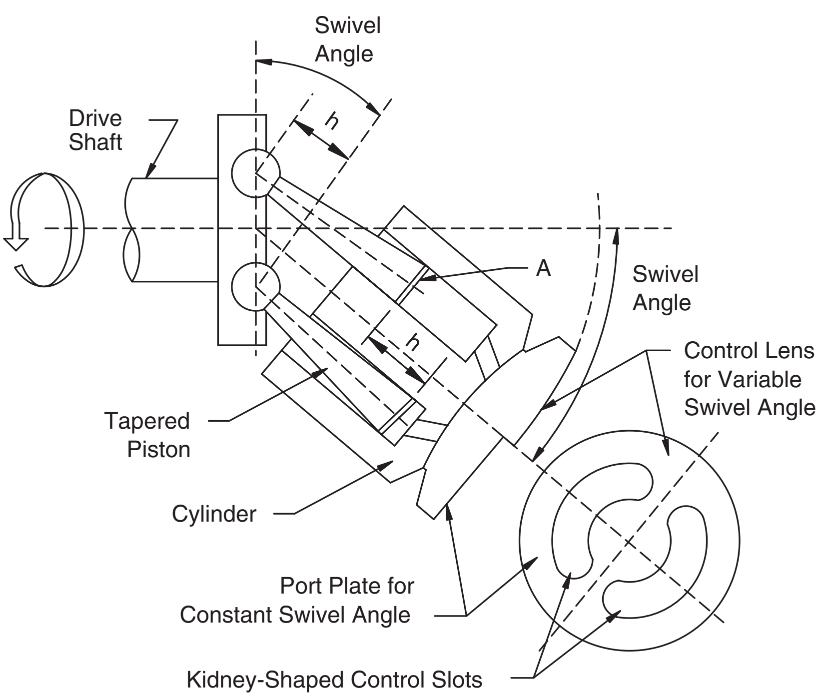


FIGURE 5.15
Diagram of bent axis motor (Dimension A is the area of the individual piston, and h is the stroke.)

similar to that of the axial piston design. A series of cylinders are mounted around the center line of the bent axis. The pistons in these cylinders have a spherical end that fits in a plate attached to the output shaft. Springs hold the pistons against the plate. (These springs are not shown in Fig. 5.15.) Fluid enters the motor and flows into the cylinder that is aligned with the inlet port. The piston extends, pressing against the plate, causing it to rotate. This rotation causes the cylinder carrier to rotate, and the next cylinder is aligned with the inlet port. This piston extends and produces the next increment of rotation. Continuous rotation is produced by the rapid sequencing of these events.

The Rexroth Series 6 axial piston, bent axis motor, Size 80, was chosen as an example of the bent axis design. This motor has a 4.91 in^3/rev displacement and maximum speed of 4500 rpm (5000 rpm intermittent). Note that this speed is the highest speed rating of any design discussed. Maximum pressure rating is 5800 psi.

The efficiencies for the bent axis motor (Fig. 5.16) are similar to the efficiencies for the axial piston motor (Fig. 5.5). For both designs, the torque efficiency reaches a maximum at about 3000 psi and remains approximately constant at higher pressures. Volumetric efficiency declines linearly with pressure, so overall efficiency decreases after reaching a maximum of 91% between 3000 and 4000 psi.

Bent axis motors are available as both fixed and variable displacement units. A variable displacement design is shown in Fig. 5.17. The servo cylinder moves the cylinder block in an arc to change the angle between the cylinder block center line and the output shaft center line. When this angle is 0°, shaft output speed is 0. For the first increment of angle increase, shaft speed

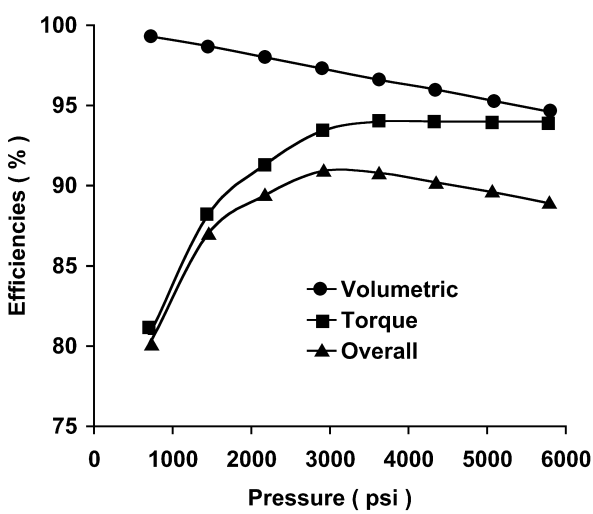


FIGURE 5.16
Efficiencies for bent axis motor operated at 1800 rpm.

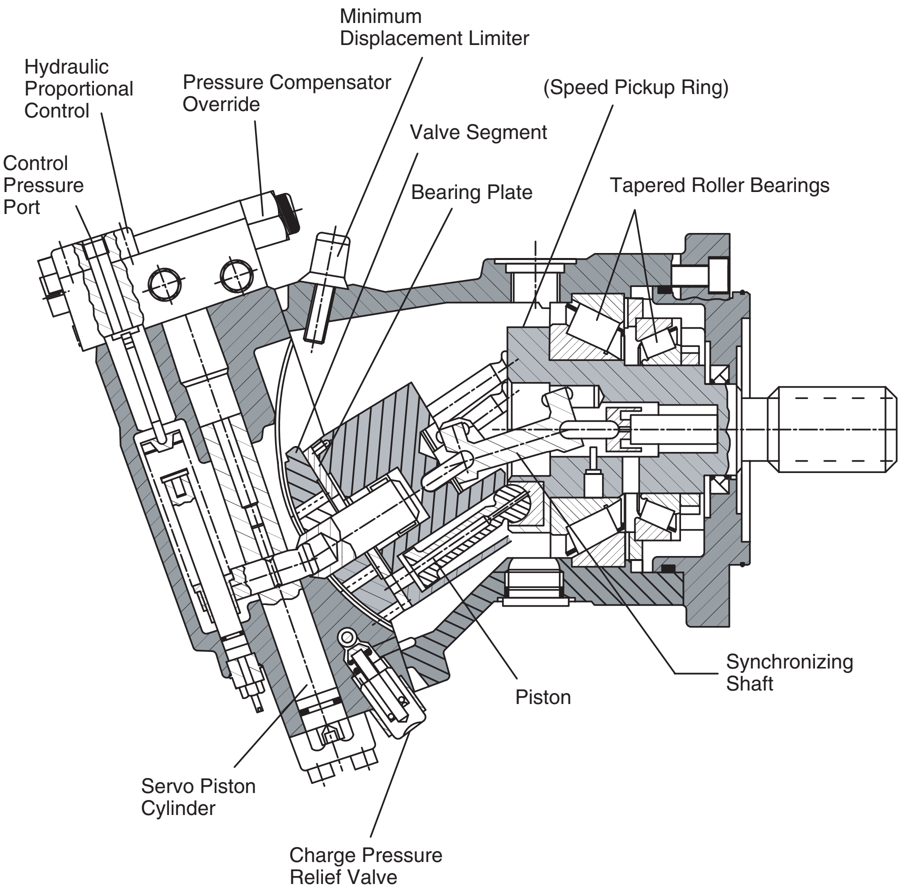


FIGURE 5.17
Diagram of a variable displacement, bent axis motor. (Reprinted with permission from Sauer-Danfoss Corp.)

increases as the angle increases. If flow to the motor is held constant, a further increase in angle will increase motor displacement and thus reduce speed. Potential torque output increases, because motor displacement increases.

5.8.1 Design Considerations for Bent Axis Motors

The piston shaft (tapered section shown in Fig. 5.15) does not provide the linkage that ensures that the cylinder block rotates with the output shaft. This linkage is provided with another mechanism. Different manufacturers use various mechanisms to develop this synchronizing torque, and this detail is beyond the scope of this text. The key point is that the full output torque of the motor is *not* transmitted through the synchronizing mechanism, only the torque to keep the cylinder block turning in synch with the output shaft.

Generally speaking, the maximum speed of an axial piston motor (in-line design or bent axis design) is limited by the requirement to maintain an oil film between the piston and wall of the cylinder. As the cylinder block rotates, centrifugal force throws the pistons out against the cylinder wall. A heavier piston produces a higher centrifugal force and thus a higher potential for the oil to be squeezed out of the clearance between the piston and cylinder bore. Loss of this film results in metal-to-metal contact and a "burn" of the piston, or cylinder bore, or both.

Two design features have been incorporated in bent axis motors to reduce the potential for loss of the oil film between piston and cylinder bore.

1. Lighter pistons are used. Also, the end in contact with the cylinder bore is shaped to minimize the contact area between the piston and cylinder bore.
2. A synchronizing mechanism minimizes the side load on the pistons. This mechanism transmits the torque required to keep the cylinder block in synch with the output shaft.

Given the lighter-weight piston, coupled with the small contact area of the spherical end, and the elimination of the requirement for the piston to transmit much (if any) side force, the bent axis design can run at higher speeds for a given displacement than can the in-line design.

The reader can appreciate that the lightweight pistons (with needed tight tolerances) and the synchronizing mechanism are not easy to build. Bent axis designs are expensive. There are certain performance advantages, and, if these advantages are fully utilized, the bent axis design can be very cost competitive.

5.8.2 Performance Advantage of Bent Axis Design

Both the in-line axial and bent axis piston motors are more expensive, but they both have higher operating speeds and higher operating pressures. A designer who needs a higher-performance motor will typically choose between these two designs. It is appropriate, then, to focus our discussion of motor performance on a comparison of the bent axis and in-line designs.

Stroke Ratio

Most bent axis motors are "bent" to a maximum angle of 30 to 40°. The swashplate angle limit on most in-line axial motors (and pumps) is less than 20° (typically 17 to 19°). Torque efficiency considerations suggest that 6° is a practical minimum angle for comparison of the designs.

The ratio of the maximum to (practical) minimum displacement for the in-line axial design is the ratio of the angle tangents.

$$\text{Ratio} = \frac{\tan(19°)}{\tan(6°)} = 3.28$$

The maximum to minimum displacement ratio is just over 3:1. The ratio of the maximum to minimum displacement for the bent axis design is the ratio of the sine of the angles.

$$\text{Ratio} = \frac{\sin(40°)}{\sin(6°)}$$

Maximum to minimum displacement ratio for the bent axis design is just over 6:1, or twice the in-line axial design. A variable displacement bent axis motor has *twice* the displacement range of a variable displacement in-line axial motor. This displacement range is referred to as the *stroke ratio.* Most designers use a 2.5:1 stroke ratio for the in-line axial design and a 5:1 ratio for the bent axis design.

The difference in stroke ratio gives a big advantage to the bent axis design for mobile applications. Many mobile machines, agricultural machines being prime examples, need a certain speed range for field operations and a higher speed for road travel. In certain applications, a variable displacement bent axis motor can be selected with small enough displacement at 6° angle to give sufficient wheel speed for the desired road travel. This motor would be set for a larger displacement to give the higher torque required for field operations. The in-line axial design, because of its smaller stroke ratio, would need a larger flow, and thus a larger pump, to get the same maximum wheel speed.

Minimum Displacement

In-line axial motors can be taken to zero displacement, but bent axis motors cannot. This limitation might not be thought of as a disadvantage. When would a variable displacement motor need to be taken back to zero displacement? One case is a machine that has four-wheel drive (4WD) for field operations and is shifted to two-wheel drive (2WD) for road travel. It two of the wheel motors are variable displacement in-line axial motors, they can be set to zero stroke and thus effectively taken from the circuit. (In actual practice, the swashplate is set back to about 1°, so there is always some piston movement in the bore.) This shifting can be done *on-the-fly.* There are no extra valves to set and no need for a detailed analysis of circuit dynamics that might occur if two motors are suddenly cut out of the circuit. Variable displacement bent axis motors can be set back to a minimum displacement, but not back to zero. They cannot be taken out of the circuit like the in-line axial motors.

Maximum Operating Speed

The bent axis motor, particularly designs with the lighter pistons, can be operated at a little higher maximum operating speed than the in-line design. In the case of a vehicle, this means that a variable displacement bent axis motor can be destroked to obtain a minimum displacement and thus a very high speed for a given pump flow. This high motor speed driving through a gearbox generates the necessary wheel speed for road travel. Sometimes, this high motor speed can give the desired road speed without increasing pump size.

Stall Torque Efficiency

The major advantage touted by bent motor manufacturers is higher stall torque efficiency. Because the bent axis motor is "bent" at a steeper angle, less of the total torque generated by a piston sliding down the inclined plane of the swashplate is consumed as friction. Some laboratory tests have been run at motor output speeds as low as 1 rpm when measuring stall torque. It is reported that bent axis motors have a stall torque efficiency about 5 percentage points higher than an in-line axial motor.

Stall torque is a very important issue in the design of a vehicle drive. The motor must be able to start the vehicle, even if it is on an incline with high rolling resistance. The designer must make a choice.

1. Choose the bent axis design over the in-line axial design to get the slightly higher stall torque efficiency.
2. Choose a larger displacement motor that can produce the needed starting torque at the lower stall torque efficiency.

Life cycle cost becomes a factor in the decision. Knowledge on how this decision is made is sequestered in design teams throughout a number of companies. This type of in-depth analysis is beyond the scope of this text.

Size

Bent axis motors, because of the ability to bend the axis further, generally have a smaller physical size, at least in one plane. In-line axial motors tend to be cube shaped, whereas bent axis motors tend to be "skinny" rectangles. In some applications, the bent axis shape fits better than the in-line axial shape.

Summary

Ranking of the various bent axis design advantages is problematic. One designer (Hull, 1999) gives the following guideline for justifying the added cost of a bent axis variable displacement motor.

1. *Stroke ratio.* If the wider displacement range from minimum to maximum is needed, this is a powerful reason to choose the bent axis design.
2. *Maximum operating speed.* If the higher speed can be used to meet a functional objective, this advantage adds to the cost competitiveness.

3. *Other factors.* Factors such as stall torque efficiency and size limitation are not as significant as the stroke ratio or operating speed. They add to the decision to choose a bent axis design but are not often the deciding factor.

5.9 Radial Piston Motors

Radial piston motors produce very high torque at low speed. They are used as wheel motors for large equipment. Pistons operate in radial bores in a stationary cylinder block. The surrounding housing rotates (Fig. 5.18). The rotating housing has two identical cam rings. The pistons each have two cam rollers. An extending piston forces the two rollers against the two cam rings causing the housing to rotate.

A distributor valve is installed in the cylinder block. This valve rotates with the housing. It directs high-pressure oil to the pistons during their work stroke and collects the low-pressure oil during their return stroke.

The distributor valve has two concentric rings. When the valve is set to supply high-pressure oil to both rings, half of the eight pistons are supplied

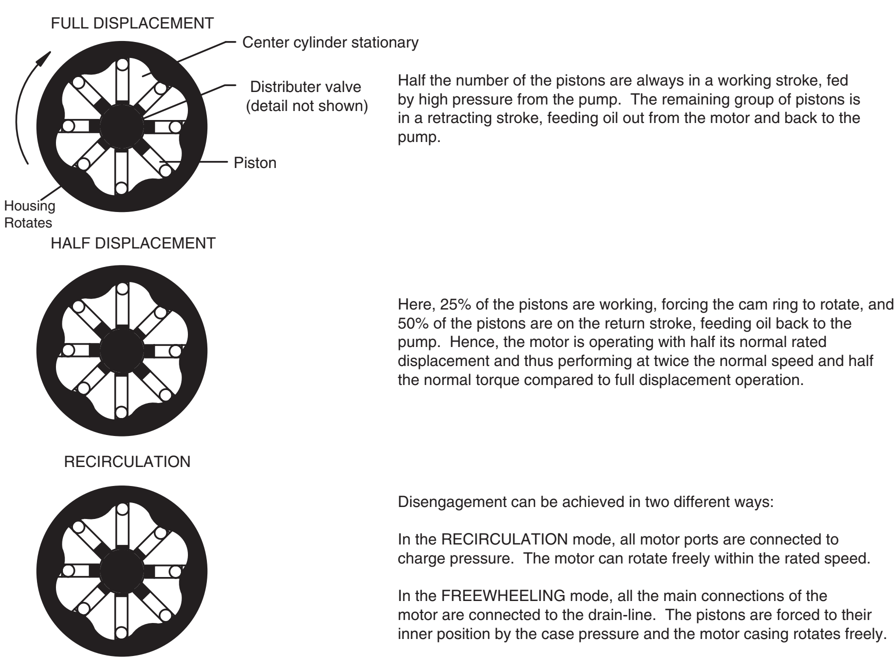


FIGURE 5.18
Radial piston motor. (*Source:* courtesy of Reidville Hydraulics & Manufacturing.)

with high-pressure oil at any time. This configuration is identified as full displacement in Fig. 5.18. It gives the maximum torque output.

When the distributor valve is set to supply high-pressure oil to only one ring, 25% of the eight pistons are supplied with high-pressure oil. The four "return stroke" pistons feed oil to the remaining two "power" pistons, and the excess oil feeds back to the pump. Since the two idling power pistons get low-pressure oil, they contribute very little to the output torque. This configuration is shown as half displacement in Fig. 5.18. Speed is doubled for a given pump flow, but output torque is halved.

In the recirculation mode shown in Fig. 5.18, charge pressure is delivered to all pistons. The motor can rotate freely up to rated speed as the vehicle is towed. Freewheeling can also be achieved by connecting all motor ports to the drain line. In this case, the pistons are forced to their inner position by the case pressure, and the housing rotates freely.

Radial piston motors can operate at pressures up to 5000 psi. They tend to be robust. Under normal operating conditions, the design life is 15,000+ hours. The manufacturer states that full torque is available at any speed, including startup.

Applications include heavy load carriers with as many as 124 wheel motors. These motors are also used for a wide range of agricultural, forestry, mining, and construction equipment.

5.10 Motor-Gearbox Combinations

Many hydraulic motors are used for applications in which the desired output is in the 50 to 500 rpm range rather than the 500 to 5000 rpm range. High-speed motors typically drive a gearbox that reduces the speed and increases the torque.

Testing has been done to compare a low-speed, high-torque (LSHT) wheel motor with a high-speed motor driving a planetary gearbox (Clifford, 1979). The gearbox had a 21.22:1 ratio, and the motor was a 2.36 in^3/rev bent axis fixed displacement motor. Displacement of the wheel motor was 52 in^3/rev. The high-speed motor, gearbox combination had an "effective" displacement of

$$\text{Motor displacement} \times \text{Gear ratio} = \text{Total}$$

$$2.46\ \text{in}^3/\text{rev} \times 21.22 = 50.1\ \text{in}^3/\text{rev}$$

The wheel motor was designed for an output speed range of 0 to 300 rpm, and the comparison tests were conducted over this range. Input torque was varied such that total power transmitted at each output speed was approximately 30 hp.

A test of the planetary gearbox alone showed that the starting torque efficiency varied in the range 92 to 89% as input torque was increased. If efficiency data is not available for a planetary gearbox, it is appropriate to use 90% as an estimate.

Test data collected on the bent axis motor-gearbox combination was used to calculate performance of other motor designs driving the same gearbox. Performance data from the technical data sheets supplied by the different motor manufacturers were used in these calculations.

In Fig. 5.19, the experimental data for the LSHT wheel motor is compared to the experimental data for the bent axis motor-gearbox combination. Starting torque was 93% for the wheel motor and 74.5% for the combination. Interestingly, the torque efficiency over the entire operating range was higher for the wheel motor.

The dotted curve in Fig. 5.19 is the calculated torque efficiency for a 3.35 in^3/rev bent axis motor. Effective displacement for this motor driving a 21.22:1 gearbox was 71 in^3/rev. This combination had a 71% starting torque efficiency as compared to 74.5% for the smaller motor combination. The larger motor combination does provide a higher torque efficiency in the 10 to 50 rpm range.

A comparison between the measured efficiency for the wheel motor and bent axis motor-gearbox combination and the calculated efficiency for an in-

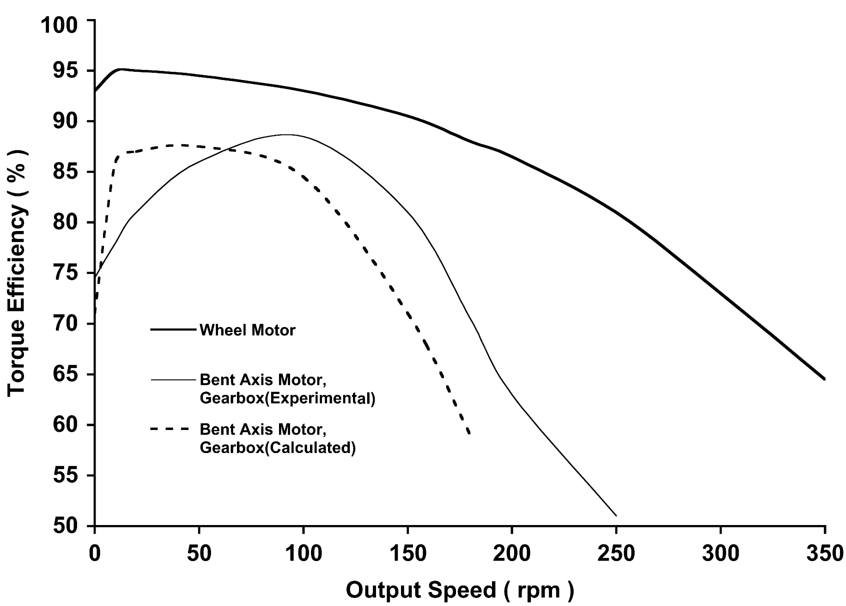


FIGURE 5.19
Comparison of measured torque efficiency for a LSHT wheel motor and a bent axis motor-gearbox combination (2.36 in^3/rev displacement, 21.22:1 gear ratio), with calculated efficiency for a bent axis motor-gearbox combination (3.35 in^3/rev displacement, 21.22:1 gear ratio).

line axial motor-gearbox combination is given in Fig. 5.20. Here, the in-line motor has a displacement of 3.15 in^3/rev; thus, the effective displacement is

$$3.15 \text{ in}^3/\text{rev} \times 21.22 = 66.8 \text{ in}^3/\text{rev}$$

Starting torque efficiency for the in-line motor-gearbox combination was 69.5%, or 5 percentage points less than the 74.5% measured for the bent axis motor combination. (Remember, the bent axis combination had an effective displacement of 50.1 in^3/rev as compared to 66.8 in^3/rev for the in-line combination.) Torque efficiency in the range 10 to 50 rpm was higher for the in-line combination. Also, the torque efficiency was higher at output speeds above 150 rpm.

Actual starting torque achieved by a motor-gearbox combination is determined by the stall torque of the motor *and* by the characteristics of the gearbox. The gearbox has backlash, and the gearing deforms when loaded, thus the hydraulic motor has some opportunity to begin turning before it must develop the full starting torque. Both of these factors benefit the motor, be it a bent axis or in-line design.

One additional point needs to be made relative to the motor-gearbox combination. A benefit from the higher operating speed of a bent axis motor with light pistons and improved synchronizing mechanism cannot be realized if it

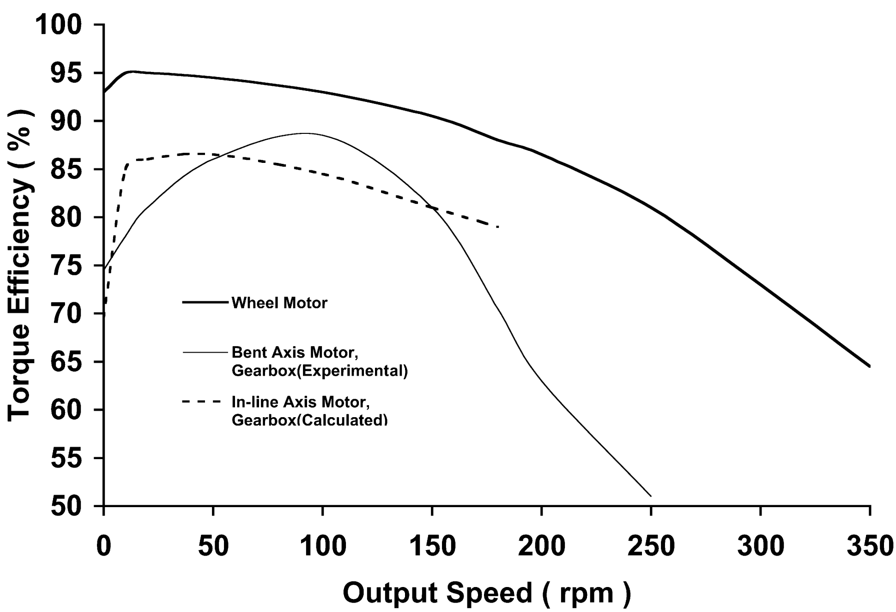


FIGURE 5.20
Comparison of measured torque efficiency for a LSHT wheel motor and a bent axis motor-gearbox combination (2.36 in^3/rev displacement, 21.22:1 gear ratio) with calculated efficiency for an in-line axial motor-gearbox combination (3.15 in^3/rev displacement, 21.22:1 gear ratio).

is driving a gearbox with a lower maximum speed rating. It does not matter that the bent axis motor can operate at 5000 rpm if the planetary gearbox has a maximum operating speed of 3500 rpm.

5.11 Oscillating Actuator

There are some applications in which continuous rotation is not needed. Two examples are industrial mechanisms performing pick-and-place operations and heavy-duty, large-payload robots.

Vane motors with one or two vanes are used for limited-rotation applications. The single-vane unit can rotate 280°, and the double-vane 150 to 160°. The direction of rotation is determined by a valve that directs fluid into one chamber or the other. As the chamber fills, it causes the moveable vane to rotate (Fig. 5.21). These motors can generate torque up 500,000 lb_f-in.

Motors with a helical spline (Fig. 5.22) are available with 90, 180, 270, and 360° of rotation. Rotation is set by the length and pitch of the helix. Units with torque ratings up to 1,000,000 lb_f-in are available.

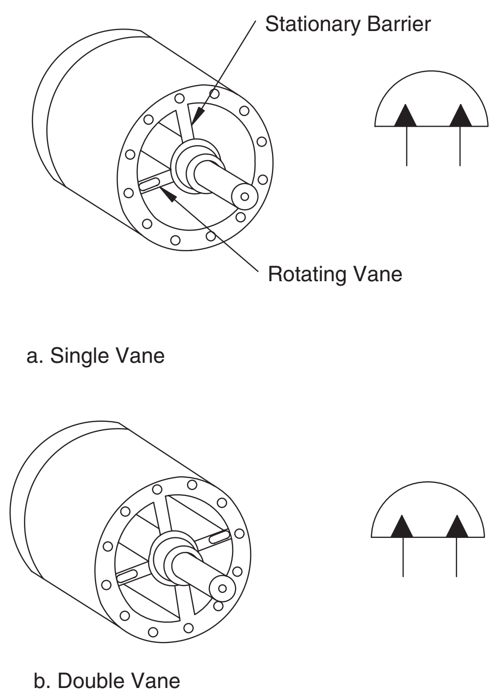


FIGURE 5.21
Single vane motor used for limited-rotation applications.

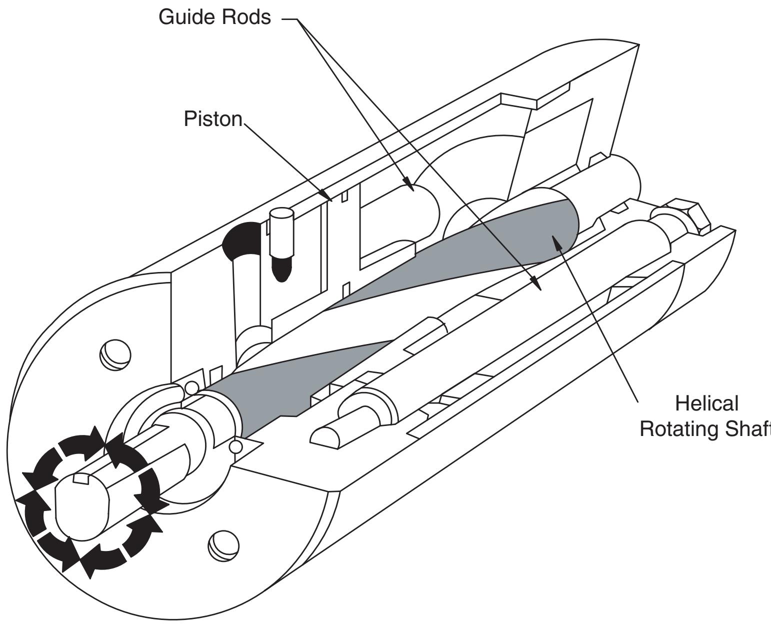


FIGURE 5.22
Limited-rotation actuator with helical shaft.

Two devices that convert linear motion to rotary motion are available, the skotch yoke (Fig. 5.23) and the rack-and-pinion actuator (Fig. 5.24). The skotch yoke is limited to 90° rotation or less, but torque output up to 45,000,000 lb_f-in is available. Two cylinders can be used to power the rack in the rack-and-pinion actuator and produce torque output in excess of 50,000,000 lb_f-in. Rotation is limited only by the cylinder stroke.

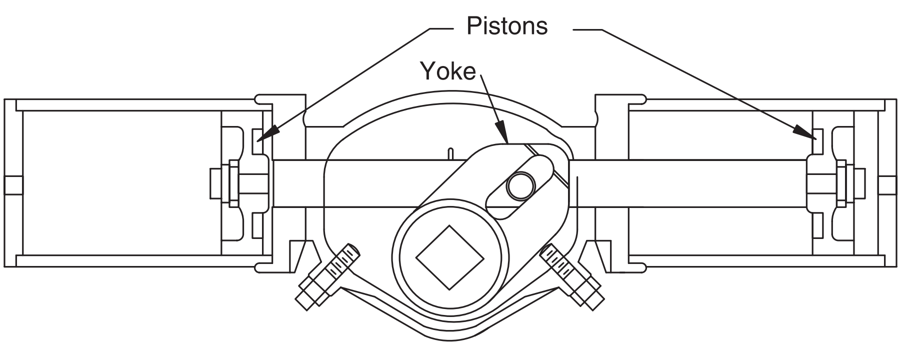


FIGURE 5.23
Skotch yoke rotary actuator.

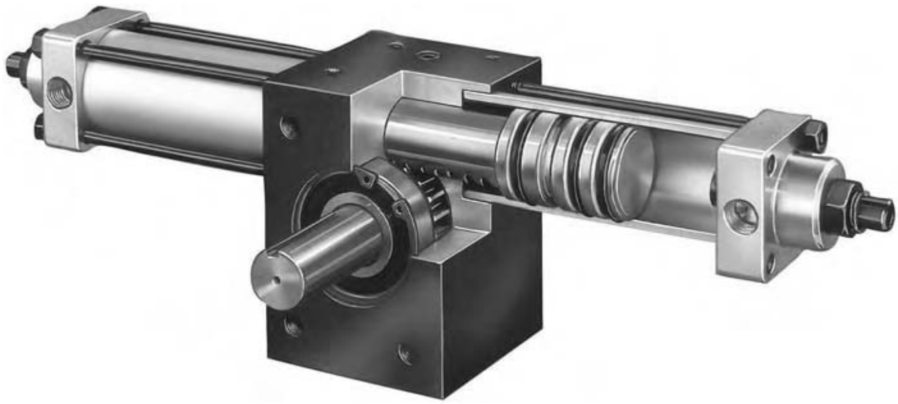


FIGURE 5.24
Rack-and-pinion rotary actuator. (*Source:* courtesy of PHD, Inc., Ft. Wayne, IN.)

5.12 Summary

Pumps convert mechanical energy into fluid energy, and motors convert this fluid energy back into mechanical energy. In function, the two are mirror images of each other; thus, it is not surprising that the same designs are used for both.

High-speed motors are available in gear, vane, and piston designs. Gear and vane motors have similar efficiencies over the pressure range 0 to 3000 psi. Axial piston motors are rated for pressures up to 6000 psi. They have a higher efficiency, and their cost is higher.

Stall torque, the torque a motor will develop at zero rpm, is of critical importance for many applications, particularly mobile applications. Because of inertia, a high torque is required to get a vehicle moving. Stall torque of a high-speed motor can be as low as 55% of theoretical torque. Also, high-speed motors have unequal rotational speed at low rpm, a condition known as *cogging*.

To overcome the limitations of high-speed motors, several designs for low-speed, high-torque motors have been developed. These motors have high displacements and higher stall torque efficiencies, and they produce a constant output speed at speeds as low as 1 rpm.

Bent axis motors are a variation of the axial piston design. These motors have operating efficiencies comparable to the in-line axial piston motors and generally have a higher stall torque efficiency. Their main advantage is a higher stroke ratio, 5:1 as compared to 2.5:1 for the in-line design.

The volumetric efficiency of all motors decreases as pressure increases. Output speed fluctuates with load pressure. For a gear pump-gear motor combination, the speed change can be as much as 30% for load pressure fluctuations in the rated pressure range. A piston pump/piston motor combination can be selected that will have less than 10% speed change over its rated pressure range.

Motors are one component of the entire hydraulic system. Their operating characteristics must be considered in combination with the operating characteristics of other components in the system. The entire system has to function in a prescribed way to meet functional objectives. Selecting the right motor from the broad range of products available is perhaps the most important decision a designer may make in sizing fluid power components. Hopefully, the material in this chapter has supplied the reader with an understanding of motor performance as influenced by pressure (output torque requirement) and flow (output speed requirement).

References

Clifford, M. 1979. An interesting and informative comparison of mobile hydrostatic wheel hub drives. SAE Paper No. 790883. SAE, Inc., 400 Commonwealth Dr., Warrendale, PA 15096.

Hull, W. E. 1999. Personal Communication. SunSource/Fauve, 5533 Washington Ave., St. Joseph, MI 49085.

APPENDIX 5.1

Curve Fitting Technique

If experimental data is plotted, and the curve has a convex shape as shown by Curve A in Fig. A5.1 or a concave shape as shown by Curve B, an equation for this data can be written as

$$y = Dx + E \pm (A + Bx + Cx^2)^{\frac{1}{2}} \tag{A5.1}$$

The constants A, B, C, D, and E are determined by solving five simultaneous equations.

The general equation for a conic section is

$$ax^2 + bxy + cy^2 + dx + ey = 1 \tag{A5.2}$$

To set up the five equations to solve for the five unknowns, a, b, c, d, and e, we need five experimental data points along the curve. We use these data points to write a matrix equation.

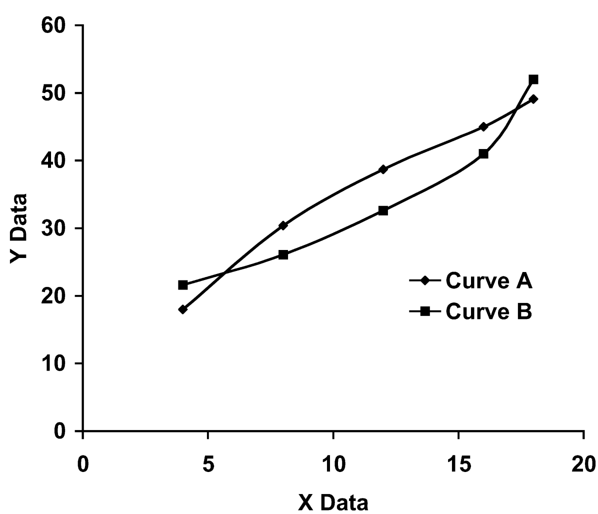


FIGURE A5.1
Examples of curves with a concave (Curve A) and convex (Curve B) slope.

$$\begin{bmatrix} x_1^2 & x_1y_1 & y_1^2 & x_1 & y_1 \\ & & \bullet & & \\ & & \bullet & & \\ & & \bullet & & \\ x_5^2 & x_5y_5 & y_5^2 & x_5 & y_5 \end{bmatrix} \begin{bmatrix} a \\ b \\ c \\ d \\ e \end{bmatrix} = \begin{bmatrix} 1 \\ 1 \\ 1 \\ 1 \\ 1 \end{bmatrix} \tag{A5.3}$$

Computer software is available to solve Eq. (A5.3). The constants in Eq. (A5.1) are given by

$$A = (e^2 + 4c)/4c^2$$

$$B = (2be - 4cd)/4c^2$$

$$C = (b^2 - 4ac)/4c^2$$

$$D = -b/2c$$

$$E = -e/2c$$

This technique can be used to write a $y = F(x)$ equation for any data where the curve shape is as shown in Fig. A5.1. It is good procedure to enter the values and verify that Eq. (A5.1) gives the corresponding values. This checking procedure reveals the correct sign, + or –, in front of the square root.

Problems

5.1 A high-speed hydraulic motor is mounted to drive the final drive (gearbox) on a vehicle. (The vehicle has two motors each mounted in a gearbox to power the two rear wheels. This problem concerns only one of the motors.) You estimate that the maximum torque required to start the vehicle (on a slope and fully loaded) is 20,850 lb_f-in. The ratio of the gearbox is 18.25:1, and the motor displacement is 2.03 in^3/rev. The relief valve is set on 3000 psi. Stall torque efficiency is 58% for this motor design, and the zero rpm mechanical efficiency of the gearbox is 90%.

Calculate the maximum torque the motor can develop to get the vehicle started (zero rpm). Is this torque sufficient to get the vehicle started?

5.2 A test stand was built to test a low-speed, high-torque (LSHT) motor with 2.8 in^3/rev displacement. A torque transducer was installed between the motor and the load to measure the actual torque delivered by the motor. Motor speed was set by increasing the displacement of a variable displacement pump. All data was collected with the motor turning at 100 rpm.

Flow of fluid to the motor was measured with a turbine-type flow meter. The output from this meter was a squarewave signal with a frequency proportional to flow. The flow meter equation is

$$Q = 0.05821075 - 0.017782533\,\nu + 8.0923806 \times 10^{-4}\,\nu^2 + 0.013329865\,F + 3.1606234 \times 10^{-5}\,\nu F - 1.2301922 \times 10^{-6}\,\nu^2 F$$

where Q = flow (GPM)
ν = viscosity (cS)
F = frequency (Hz)

Viscosity is given by the following equation:

$$\nu = -0.485\,T + 55.9$$

where ν = viscosity (cS)
T = temperature (C)

Data given in Table A5.1 were collected.

TABLE A5.1

Data Collected on Low-Speed, High-Torque Motor

Pressure (psi)	Torque (lb_f-in)	Flow (Hz)	Temperature (C)
433	89.4	62.8	37
727	185.2	67.2	37
1012	273.6	72.5	38

Pressure measured at the motor outlet port was 110 psi for all tests.

Calculate the volumetric efficiency, e_{vm}, overall efficiency, e_{om}, and torque efficiency, e_{tm}, for all three test conditions. Discuss your results.

5.3 A manufacturer gives the data shown in Table A5.2 for a geroler-type gear motor. The motor has a displacement of 4.9 in^3/rev.

a. Calculate and plot volumetric efficiency vs. pressure for the two flows on the same figure. Discuss your results.

b. Calculate and plot overall efficiency vs. pressure for the two flows on the same figure. Discuss your results.

5.4 Flow to the motor, and subsequently the output speed, does influence torque output for some motor designs. Data shown in Table A5.3 give the output torque from a "family" of motors as flow to the motor increases from 1 to 20 GPM. (A family of motors is a group of motors with the same design, but the size of the components have been increased to give a range of displacements.) Pres-

TABLE A5.2

Test Data for Geroler-Type Gear Motor

Pressure drop (psi)	Motor output torque (lb_f-in)	Motor output speed (rpm)
	Flow, Q = 1 GPM	
500	330	44
1000	670	40
1500	990	37
2000	1300	34
2500	1550	28
3000	1800	22
	Flow, Q = 8 GPM	
500	310	365
1000	660	357
1500	1015	349
2000	1345	341
2500	1685	333
3000	2020	325

sure drop across the motors was maintained constant at ΔP = 1000 psi for all tests.

Plot the torque efficiency vs. input flow for the four smaller motors (4.9 to 9.6 in^3/rev) on one plot and the four larger motors (11.9 to 24.0 in^3/rev) on a second plot. Discuss your results.

TABLE A5.3

Measured Output Torque (lb_f-in) for a Family of Motors Ranging in Displacement from 4.9 to 24.0 in^3/rev, Pressure Drop Held Constant at ΔP = 1000 psi

Flow to motor (GPM)	Motor displacement (in^3/rev)							
	4.9	6.2	8.0	9.6	11.9	14.9	18.6	24.0
1	670	830	1070	1170	1430	1850	2330	3080
2	670	830	1080	1210	1470	1900	2400	3180
4	670	820	1080	1260	1540	1980	2500	3320
8	660	810	1080	1330	1625	2080	2600	3460
12	640	800	1060	1320	1650	2100	2600	3470
16	610	780	1010	1270	1625	2040	2500	3370
20	570	730	960	1210	1550	1910	2360	3260

5.5 The test data shown in Table A5.4 were collected for a bent axis variable displacement motor set for maximum displacement, V_m = 4.92 in^3/rev. The test was conducted with the motor operating at 2500 rpm.

a. Use the data to calculate torque efficiency (e_{tm}) for this motor and present your data in a table with the given efficiencies.
b. Plot the three efficiency curves. Discuss your results.

TABLE A5.4

Efficiency Data for Bent Axis Variable Displacement Motor Set for Maximum Displacement

	Efficiency (%)	
Pressure (psi)	**Volumetric, e_{vm}**	**Overall, e_{om}**
500	99.8	76.0
1000	99.8	86.3
1500	99.7	90.5
2000	99.7	92.8
2500	99.7	94.0
3000	99.6	94.6
3500	99.5	94.9
4000	99.2	95.0
4500	98.7	94.8
5000	98.1	94.4
5500	97.1	93.7
6000	96.0	93.0

6

Hydrostatic Transmissions

6.1 Introduction

The chapter focuses on the use of hydrostatic transmissions for a vehicle. A hydrostatic transmission modifies and transmits power from the engine to the final drive, or directly to the wheels or tracks. An entire text could be devoted to the analysis required to properly match prime mover and load characteristics to achieve optimal productivity and efficiency. Our objective in this chapter is to understand the characteristics of a hydrostatic transmission so that it can be compared with mechanical transmissions. A brief review of mechanical transmissions is required to serve as a reference for our subsequent discussion of hydrostatic transmissions.

6.2 Mechanical Transmissions

A vehicle with rear-wheel drive has the components shown in Fig. 6.1. For purposes of discussion, we assume that the mechanical transmission has four gear meshes. (Functionally, a mechanical transmission can be thought of as a *black box* with gears inside. The meshing of these gears gives an output shaft speed lower than the input shaft speed. A specific gear mesh gives a specific ratio of input and output shaft speeds.) The engine can deliver torque over a given range; it will stall if the load requires torque above some maximum. A gear mesh is selected to match required output torque to available input torque. The output torque establishes the force the vehicle can exert as it moves. This force is often referred to as *drawbar pull,* but it can also be a *push* as when a bulldozer is pushing material. Typical curves for the vehicle shown in Fig. 6.1 are given in Fig. 6.2. In gear one, tremendous force is developed, but forward speed is low. In fourth gear, the speed is high, but developed force is much reduced. Power is force times velocity. A transmission does not increase power; power is set by the engine.

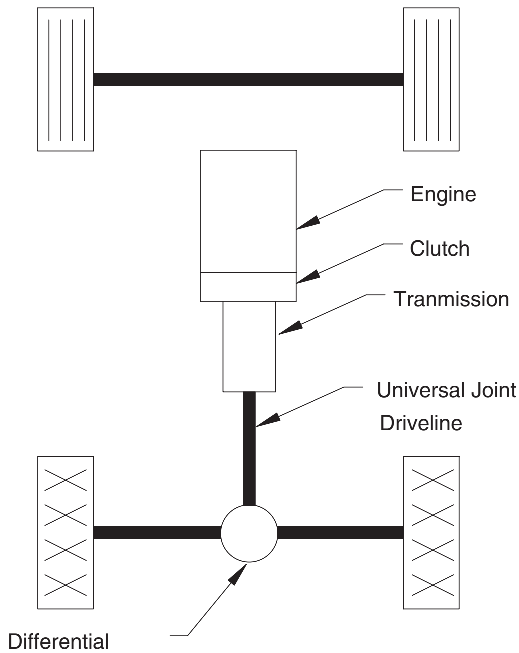


FIGURE 6.1
Diagram of rear-wheel-drive vehicle with mechanical drive.

The clutch disconnects the engine output shaft from the driveline so that the gears in the transmission can be shifted to a new mesh. With a manual clutch, the operator manually engages the clutch and manually shifts the transmission to a new gear mesh.

An automatic transmission has a series of fixed gear ratios. Depending on the design of transmission, these gear ratios each have their own unique clutch or combination of clutches. When a given clutch (or combination of clutches) is activated, the corresponding gear ratio is placed in the drive. The clutches (or combinations of clutches) are activated in appropriate sequence to move "up" through the gears. As the next clutch (or combinations of clutches) is activated, the previous clutch is deactivated. The power is transmitted through as little as one mesh, but possibly through multiple meshes, at any given time.

As engine speed increases, the torque the engine can deliver decreases. A typical curve is shown in Fig. 6.3. Shifting from first gear to second gear can produce an output power curve as shown in Fig. 6.4. In this example, the shift is made when vehicle speed is 2.1 mph. At this shift point, output power is 72 hp. The *potential power line* shows that the potential output power is 89 hp. The shaded area in Fig. 6.4 is referred to as a power *hole.* If enough gears are provided, the power holes can be minimized, and the actual output power will closely approximate the potential output power line (Fig. 6.5).

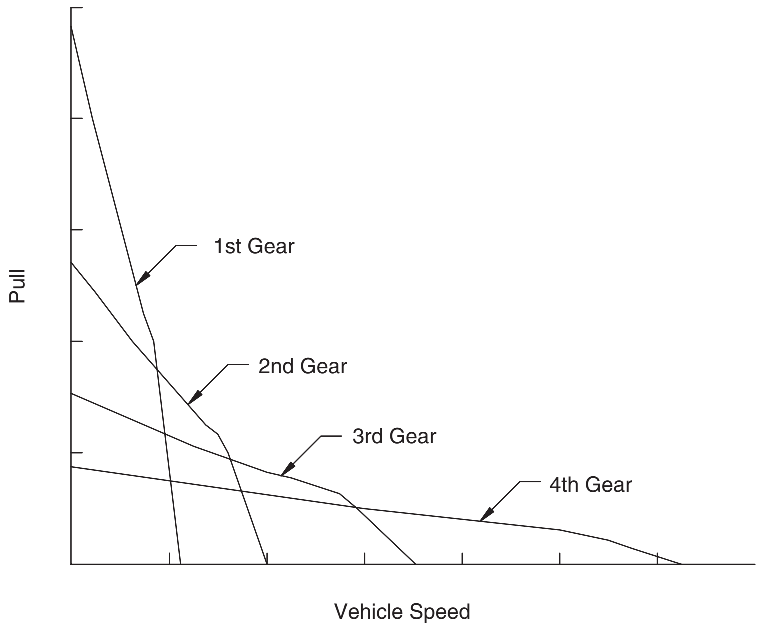


FIGURE 6.2
Typical performance curves for a mechanical drive vehicle with four gears.

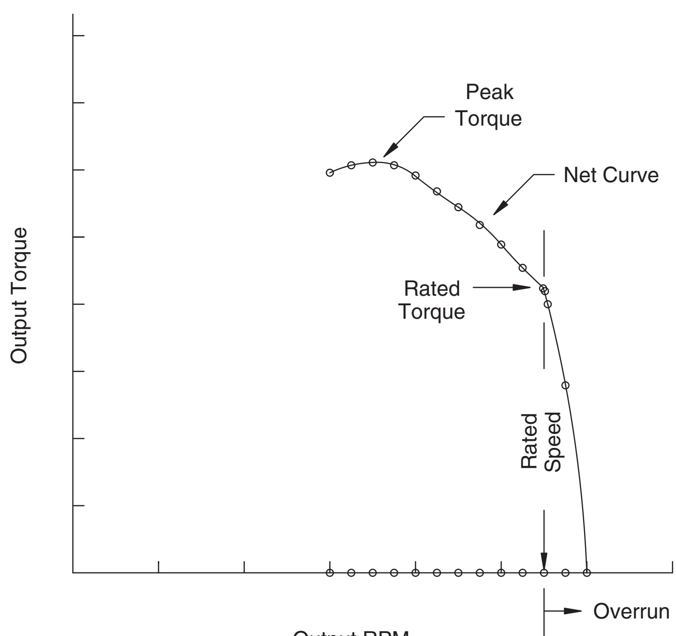


FIGURE 6.3
Typical output torque vs. speed curve for a diesel engine.

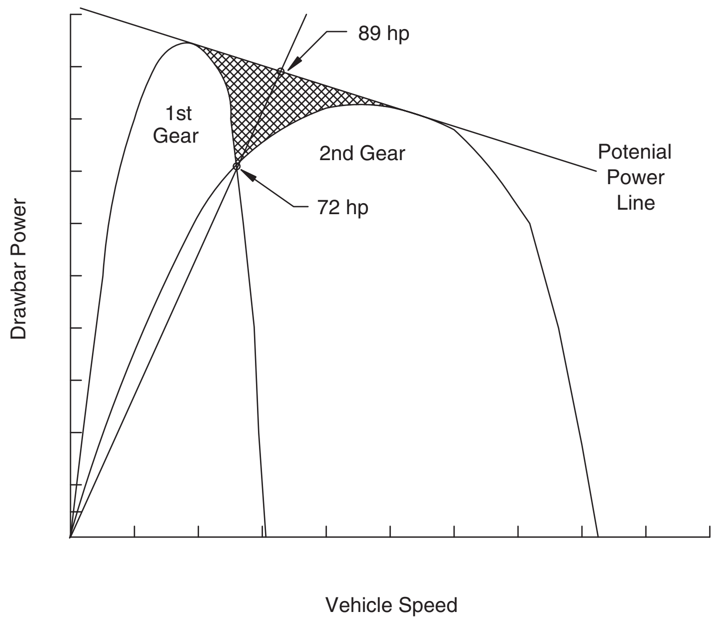


FIGURE 6.4
Output power produced when vehicle is shifted from first to second gear.

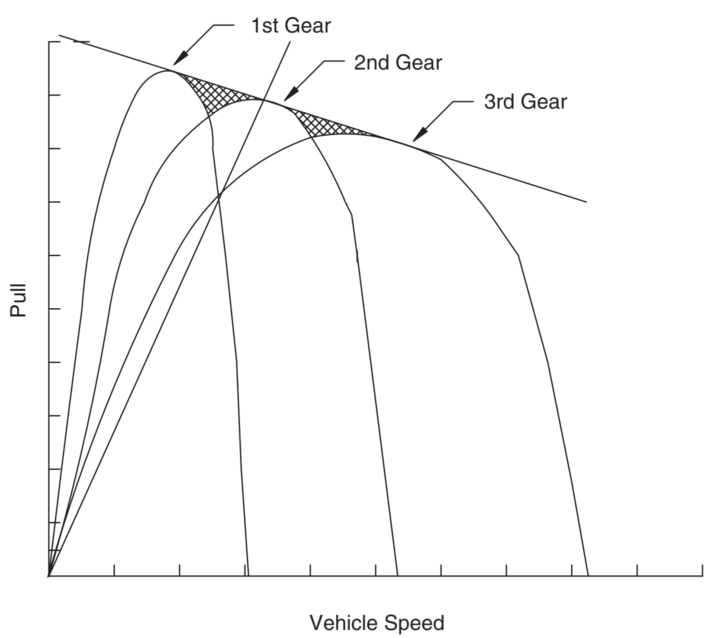


FIGURE 6.5
Additional gear shifts produce an actual power output curve that closely approximates the potential power output curve.

The cost of an automatic transmission increases as the number of gear meshes and clutches increases. It is desirable to use as few shifts as possible to obtain a suitable output power curve. A torque converter is placed between the engine and the automatic transmission (Fig. 6.6) to reduce the number of transmission shifts required. Torque converters have been designed to transmit hundreds of horsepower.

6.2.1 Torque Converters

A torque converter is a hydrodynamic drive component utilizing three (or more) annular bladed parts to absorb power from a prime mover and *automatically* adjust its operating output speed to match the load demand (Fig. 6.7). The torque converter input shaft is connected to an impeller, which imports momentum to the fluid filling the converter. This moving fluid impacts a turbine, causing it to turn. The turbine turns the output shaft, thus delivering power to the transmission. To understand the concept, visualize two window fans facing each other. If one fan is turned on, the flow of air across the blades of the opposing fan will cause this fan to turn.

There is a tendency to think of the impeller as a hydraulic pump and the turbine as a hydraulic motor. In truth, the function of the impeller is to pump (import momentum to) a fluid, and the function of the turbine is to convert fluid energy back into mechanical energy. Our discussion of pumps are as devices that generate a high P and low (relatively low) Q. The torque converter is a low P, high Q device. Remember that power is a product of P and Q. Both devices can transmit the same power. One does it with a high P and low Q and the other with a low P and high Q.

If a torque converter has only the two parts, impeller and turbine, it is known as a *fluid coupling.* The output torque equals the input torque minus losses. When the torque requirement of the load exceeds the load-carrying limit of the fluid coupling, the output shaft stalls, and no energy is delivered. The engine continues to deliver energy into the coupling. Slippage between the turning impeller and stalled turbine causes the fluid to heat up quickly. Even when proper cooling is provided, a fluid coupling cannot stay stalled

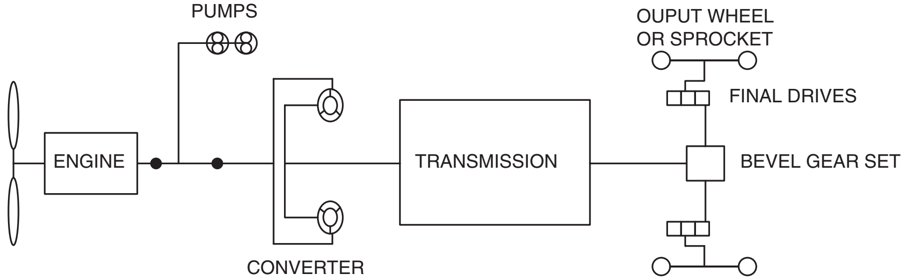


FIGURE 6.6
Schematic showing torque converter in power train for a large machine.

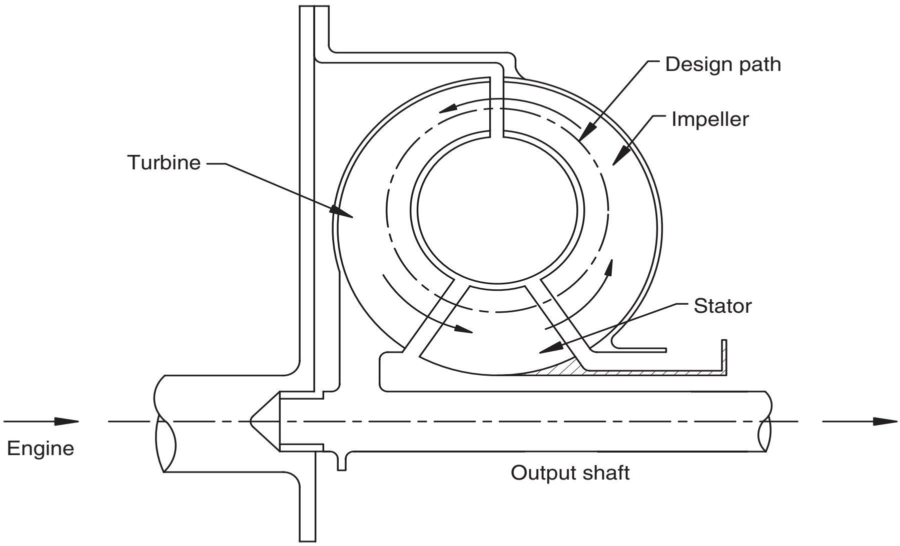


FIGURE 6.7
Functional diagram of a torque converter.

for very long before being damaged. It provides short-term stall protection but cannot remain in the stalled condition without damage.

A torque converter is designed with one more annular component than the fluid coupling: the stator. The stator provides for torque amplification by a torque converter, an extremely valuable feature. This characteristic is simply stated as

$$T_T = T_\ell + T_S \tag{6.1}$$

where T_T = turbine torque (output torque)
T_ℓ = impeller torque (input torque)
T_S = stator torque

The torque amplification feature is important, particularly for a heavy vehicle starting under load. Fig. 6.8 shows available engine torque over the operating range 1200 to 2400 rpm. Corresponding converter output torque over the range 0 to 2700 rpm is shown on the same plot. The very high output torque at 0 rpm, made possible by the torque amplification feature of the converter, is extremely important in getting the vehicle started and is the key benefit provided by the torque converter.

The four key advantages of a torque converter are listed below.

1. It provides stall protection for the engine.
2. It reduces shock transmission from the load to the engine.

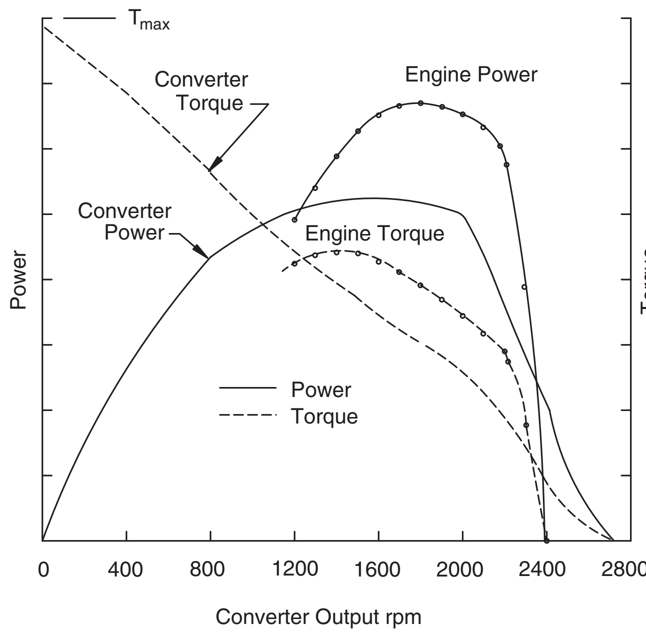


FIGURE 6.8
Comparison of available engine torque and available torque converter output torque for a typical application.

3. It increases maximum available torque at stall speed.
4. It broadens the power band for an individual gear mesh such that fewer shifts are required to achieve a smooth output power curve.

The key disadvantage is a loss of efficiency. No fluid connection can match the metal-to-metal contact of a direct drive. In many torque converter designs, the efficiency disadvantage is addressed by adding a *lockup* feature. This feature provides a means for locking the input shaft to the output shaft of the converter to provide a direct connection between the engine and the transmission.

6.2.2 Shift Control of Automatic Transmission

Performance of a direct-drive transmission with six shifts (six gears) is compared to a torque converter/automatic transmission with three shifts in Fig. 6.9. Note that the torque converter "broadens" the power transfer bands with the three shifts such that the power line is equally as smooth as the power line produced with six shifts. However, the automatic transmission power line is below the straight transmission power line, indicating that there is a higher power loss through the automatic transmission. As with any fluid device, there is an efficiency cost for any benefits.

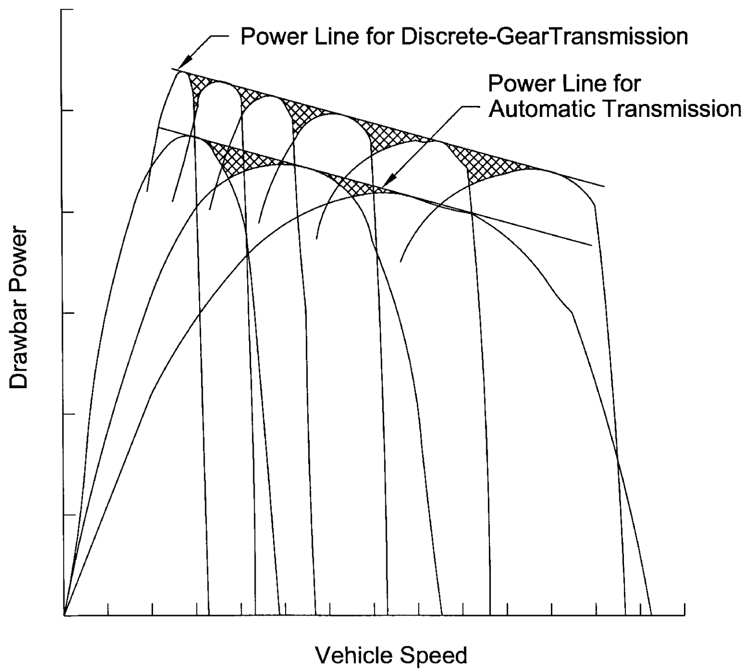


FIGURE 6.9
Comparison of drawbar power achieved with a "typical" straight transmission with six shifts to a "typical" torque converter/automatic transmission with three shifts.

Activation of the clutches in the automatic transmission (to achieve the shifts) is accomplished by activating solenoid valves to port pressurized fluid to the clutch actuator. (Typical activation pressures are in the 500 psi range.) Automatic transmissions going into vehicles today, particularly very heavy vehicles, have complex strategies to sequence activation of the solenoid valves.

A safety feature is built into valves to prevent the engine from being driven at too high a speed by an overrunning load. Other features are included to sense an operator error (or poor decision) and shift in an optimal manner to prevent damage to the engine or transmission. Using fluid power to *implement* these control strategies is an important fluid power application.

An engineer should understand the characteristics of direct-drive and automatic shift transmissions and compare these characteristics with the characteristics of hydrostatic transmissions presented in this chapter to select the best power transmission technology for the vehicle being designed. Sometimes an automatic shift transmission is the best choice based on cost (initial and operating), reliability, and service life; sometimes a hydrostatic transmission or direct-drive transmission is the best choice. This limited information on automatic shift transmissions should provide the needed background for interpretation of the following discussion of hydrostatic transmissions.

6.2.3 Summary

A machine with a direct-drive transmission is the best choice when

1. High efficiency is needed because there is a need to transfer the maximum power possible.
2. The load is relatively constant, thus minimal shifting is needed.
3. Overloads are rare, thus engine stall protection is not needed.

An automatic shift transmission is needed when

1. Load variability is high.
2. High pull is required at zero or low vehicle speed.
3. Engine stall protection is needed.
4. It is desirable to maintain a relatively constant engine speed. (An example would be a machine in which the engine powers one or more hydraulic pumps. These pumps supply flow to the actuators on the machine. An example would be a wheel loader where large cylinders are used to lift and dump the bucket.)

An automatic shift transmission also makes the operator's job much easier when a machine is being maneuvered back and forth all day. Sometimes this advantage is the controlling factor in the sale of a machine.

6.3 Introduction to Hydrostatic Transmissions

A hydrostatic transmission (HST) is simply a pump and motor connected in a circuit. Other components are added to obtain certain operating features. Each component used in a hydrostatic transmission will be discussed in some detail later in the chapter.

The four basic configurations of hydrostatic transmissions are

1. In-line (Fig. 6.10a)
2. U-shape (Fig. 6.10b)
3. S-shape (Fig. 6.10c)
4. Split (Fig. 6.10d)

Various pump and motor designs can be paired together for the split configuration. Manufacturers use the same designs to build the in-line, U-shape, and S-shape configurations.

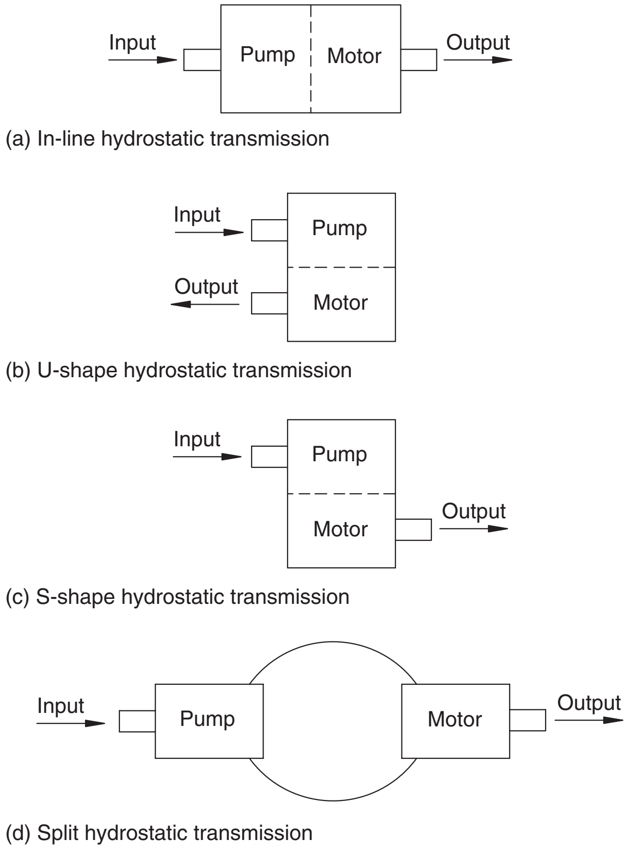


FIGURE 6.10
Four basic configurations of hydrostatic transmissions.

Hydraulic hose is available with a working pressure rating of 6000 psi. Use of these hoses allows the split transmission to be operated at the maximum pressure rating of high-performance pumps and motors. Particularly on a vehicle, the split transmission can offer many advantages. Most often, the pump is mounted on a pump mount bolted directly to the engine, but the motors can be placed at the most convenient location, thus taking full advantage of fluid power's ability to flow power "around a corner."

The split transmission, because of the flexible hoses between the pump and motor(s), will absorb some deflection caused by dynamic loads applied to the frame. Consider the situation when the pump and motor are bolted rigidly together, the pump is bolted to the engine (not driven with a universal joint driveline), and the motor is bolted to the frame. Now the pump-motor housing is subject to the dynamic loads applied to the frame. This introduction of stress into the housing is undesirable and can lead to reduced reliability.

6.4 Hydrostatic Transmissions for Vehicle Propulsion

6.4.1 Comparison of Hydrostatic and Mechanical Drives

The same vehicle shown in Fig. 6.1 is shown in Fig. 6.11 with the mechanical transmission replaced with a hydrostatic transmission; all other components remain the same. To provide a specific example, we will begin by specifying a variable displacement axial piston pump and a fixed displacement axial piston motor. (Refer to Chapter 4 for explanation of pump operation, and Chapter 5 for explanation of motor operation.) Pump output is increased by

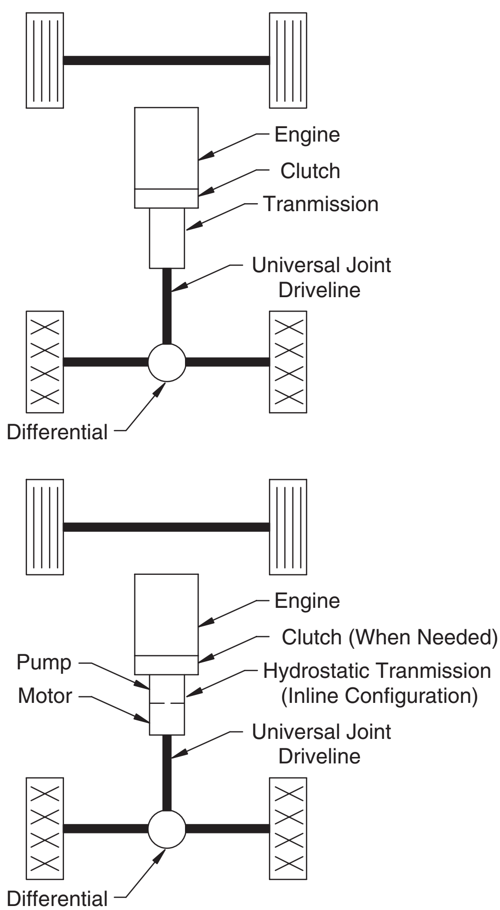


FIGURE 6.11
Comparison of rear-wheel drive vehicle with mechanical and hydrostatic drives.

stroking the pump, thereby increasing the speed of the motor. The vehicle can be speeded up and slowed down by moving the hand control that strokes the pump. The rotation of the motor shaft can be reversed by moving the swashplate control through the neutral position and displacing it in the opposite direction. The reverse position of the swashplate causes fluid to flow in the opposite direction, which causes the motor to turn in the opposite direction, thus reversing the vehicle. Vehicle motion can be changed from forward to reverse with a simple hand movement. This maneuverability is often the justification for installing a hydrostatic drive on a vehicle. A good example is a skid-steer loader, cycling back and forth, unloading a boxcar. It is tiring and time-consuming when an operator has to shift a mechanical transmission each time the direction of motion is changed. Vehicle productivity is increased with a hydrostatic transmission.

A hydrostatic transmission, like an automatic shift transmission, connects the engine and load with a fluid connection. (Remember that an automatic shift transmission has a torque converter that is a fluid power device.) Some of the same advantages are achieved with both types of transmission. The key disadvantage, as with all fluid devices, is some decrease in efficiency.

6.4.2 Advantages of Hydrostatic Transmissions

A hydrostatic transmission, as shown in Fig. 6.11, provides improved maneuverability, but at a cost. The efficiency of a hydrostatic transmission is always lower than a discrete-gear transmission. A discrete-gear transmission will typically have an efficiency of 95% or greater, meaning that 95% of the input energy is delivered to the load (wheels). A hydrostatic transmission has an efficiency of around 80%. Some well designed units will have an efficiency slightly above 85%, but none can approach the efficiency of a discrete-gear transmission. A designer always poses the question: *Does the gain in vehicle productivity offset the loss in efficiency and resultant higher fuel cost?*

In addition to increased maneuverability, a hydrostatic drive vehicle offers several other advantages:

1. *It operates over a wide range of torque/speed ratios.* Once a gear ratio is selected with a direct-drive transmission, the only speed variation available is that achieved by controlling engine speed. Once the engine speed reaches a maximum, the transmission must be shifted to a lower ratio to increase vehicle speed. With a hydrostatic transmission, vehicle speed is continuously variable from a slow creep up to a maximum.
2. *It can transmit high power with low inertia.* When a large mass is rotated at a given speed, it takes an interval of time to change this speed. A hydrostatic transmission adds little inertia to the total rotating mass associated with vehicle operation; consequently, a hydrostatic transmission vehicle tends to change speed more

quickly (have less inertia) than a direct-drive or automatic shift transmission vehicle.

3a. *It provides dynamic braking.* A hydrostatic drive vehicle can be stopped by destroking the pump. Imagine that you are traveling forward and you suddenly move the swashplate control to the neutral position. What will happen? A pressure spike will develop, and fluid will flow across the relief valve. The vehicle's mechanical energy will be converted to heat energy, and the vehicle quickly slows (probably sliding the wheels).

3b. *It remains stalled and undamaged under full load.* Vehicle hydrostatic transmissions are almost always designed for wheel slip to occur before a relief valve is actuated. The relief valve's role is to clip off peaks and attenuate shocks, as described in part 3a. If the vehicle loses traction and bogs down, the pressure increases until the relief valve opens. Stalling the vehicle in this manner does not damage the transmission. Holding it in a stalled condition causes the fluid temperature to rise, and this is undesirable. [Most HST pumps today are available with a pressure limiter function that provides the "stall and undamaged" feature with little heat generation. The pressure limiter destrokes the pump by shifting the swashplate much like a pressure compensator (Fig. 4.15).]

4. *There is no interruption of power to wheels during shifting.* Anyone who has watched the driver of a direct-drive vehicle with discrete-gear transmission shift gears while climbing a hill can appreciate the advantage of continuous power flow over a speed range.

6.5 Different Configurations of Hydrostatic Transmissions to Propel Vehicles

6.5.1 Hydrostatic Transmission with Two Wheel Motors

Wheel motors, mounted at both rear wheels (Fig. 6.12), is a variation of the configuration shown in Fig. 6.11. This arrangement eliminates the universal joint driveline, differential, and rear axle, with resultant cost and weight savings. Because the pump has low inertia, it is often possible to provide enough starting torque to start the engine with a direct-coupled pump. (The swashplate control would have to be set in the neutral position.) A clutch is needed if cold starting is a major consideration, i.e., very low ambient temperature. Later in this chapter, a gearbox with multiple pump mounts is discussed. Sometimes a clutch is not needed even when several pumps must be turned when the engine is started. In the following illustrations, the clutch is shown with the label, "for some applications."

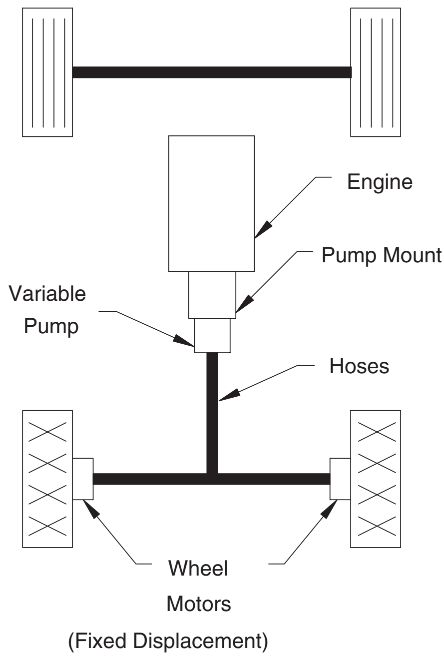


FIGURE 6.12
Hydrostatic transmission with one pump and two wheel motors.

In Fig. 6.12, the wheels are mounted directly on the shaft of the motors. The low-speed, high-torque motor designs discussed in Chapter 5 are used for wheel motors. However, the wheel motor is quite different from a standard hydraulic motor. It bolts directly to the frame of the vehicle; therefore, the housing has a structural mission relative to the vehicle, in addition to its mission relative to the operation of the motor. Wheel motors have a heavier housing. Also, the wheel motor bearings are the axle bearings for the vehicle. Wheel motors are designed for a high radial load, and care must be taken to ensure that the dynamic load during vehicle operation does not exceed the radial load rating.

On four-wheel-drive (4WD) vehicles, the wheels that steer are also powered. These wheels are subjected to thrust loads during turning; consequently, the wheel motor bearings must be designed for a thrust load in addition to the radial load. Also, hillside operation puts a thrust load on the wheel motor bearings. The loads the vehicle puts on the wheel motor bearings should be discussed with the wheel motor manufacturer before a selection is made.

Hoses or tubing carry the flow from the pump to the motors. A tee is used to divide the flow to the two wheel motors. When the vehicle turns, the pressure required to rotate the outside wheel is less; therefore, more flow goes to this wheel, and it rotates faster. This action accomplishes the same task as the differential.

With the configuration shown in Fig. 6.12, it is important to remember that speed obtained with a given flow is only one-half of the speed obtained with a single motor. On the other hand, using two wheel motors provide twice the total wheel torque. The configuration shown in Fig. 6.12 is useful for a relatively light vehicle that moves slowly but must have high tractive ability. An example would be an agricultural machine used to harvest a vegetable crop.

6.5.2 Hydrostatic Transmission with Final Drives

The configuration shown in Fig. 6.13 has a two-speed pump mount and the hydraulic motors mounted in final drives on the rear wheels. Often, it is desirable to provide a road speed for moving the vehicle from one work location to the next. The pump mount is shifted to a lower ratio to drive the pump at higher speed and thus provide the higher flow required for road speed.

Gear ratios are normally expressed as

$$\text{Ratio} = N_o/N_i \tag{6.2}$$

where N_o = number of teeth on output gear
N_i = number of teeth on input gear

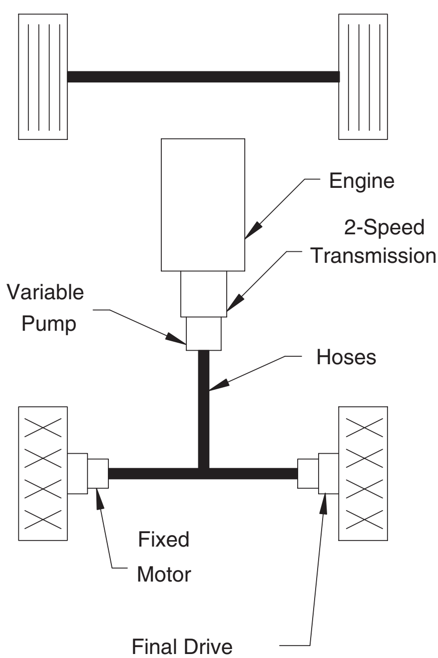


FIGURE 6.13
Hydrostatic drive with two wheel motors mounted in final drives.

If N_o is decreased relative to N_i, the ratio is *lowered*, and the output shaft speed is increased. Thus, a lower ratio in the pump mount drives the pump at a higher speed. A second way of expressing gear ratios is, for example, 20:1. In this case, an input speed is given first, followed by the output speed after the colon. Both methods are used in this text.

An example will illustrate this point. Envision a transmission where the pump has a displacement $V_p = 1.925$ in^3/rev and is driven with a 1:1 ratio. The pump volumetric efficiency is $e_{vp} = 0.92$. The engine is operated at 2000 rpm. Total flow from the pump is

$$\begin{aligned} Q_p &= \frac{V_p N_p e_{vp}}{231} \\ &= \frac{1.925(2000)(0.92)}{231} \\ &= 15.3 \text{ GPM} \end{aligned} \tag{6.3}$$

If the pump mount is shifted to provide a 0.67 ratio, pump speed will be 2000/0.67 = 3000 rpm, with a corresponding flow of

$$Q_p = \frac{1.925(3000)(0.92)}{231} = 23 \text{ GPM} \tag{6.4}$$

The final drive on the rear wheels is generally some type of planetary gear set. One name given to this gear set is *planetary wheel drive,* and other names are *wheel drive,* and *power wheel.* The housing of the planetary wheel drive mounts to the frame, and the wheel mounts to the planet gear carrier, the output side of the final drive. The hydraulic motor bolts to the drive housing, and the shaft drives the sun gear, the input side of the final drive. Since the final drive carries the radial load, the hydraulic motor can be a standard high-speed motor.

An example will illustrate how the final drive influences vehicle performance. Suppose the two-wheel motors have displacements of $V_m = 1.84$ in.^{3}/rev and volumetric efficiencies of $e_{vm} = 0.90$. The rear wheels are 28 inches in diameter, and the final drive ratio is 18:1. If the pump output is 15.3 GPM, how fast will the vehicle travel?

Motor speed is

$$N_m = \frac{231 Q_m e_{vm}}{V_m} \tag{6.5}$$

where N_m = motor speed (rpm)
Q_m = flow to motor (GPM)

V_m = motor displacement (in^3/rev)
e_{vm} = motor volumetric efficiency (decimal)

One-half the pump output goes to each wheel motor. For a 15.3 GPM pump output, motor speed is

$$N_m = \frac{231(15.3/2)(0.9)}{1.84}$$

$$= 864 \text{ rpm} \tag{6.6}$$

Wheel speed is

$$N_w = N_m/G_r \tag{6.7}$$

where N_w = wheel speed (rpm)
N_m = motor speed (rpm)
G_r = gear ratio

$$N_w = 864/18 = 48 \text{ rpm} \tag{6.8}$$

Forward speed is

$$v = \frac{2\pi(R/12)N_w(60)}{5280} \tag{6.9}$$

where v = forward speed (mph)
R = wheel radius (in.)
N_w = wheel speed (rpm)

$$v = \frac{2\pi(14/12)(48)(60)}{5280}$$

$$= 4 \text{ mph} \tag{6.10}$$

The vehicle can operate at a maximum speed of 4 mph when the pump mount is set in the 1:1 gear ratio. If the pump mount is shifted into the 1:1.5 ratio, then the pump flow will be 23 GPM, and the wheel motor speed will be

$$N_m = \frac{231(23/2)(0.9)}{1.84}$$

$$= 1300 \text{ rpm} \tag{6.11}$$

Wheel speed is

$$N_w = 1300/18 = 72 \text{ rpm} \tag{6.12}$$

and forward speed is

$$v = \frac{2\pi(14/12)(72)(60)}{5280}$$

$$= 6 \text{ mph} \tag{6.13}$$

Therefore, the maximum speed for road travel is 6 mph.

6.5.3 Hydrostatic Transmission with Variable Speed Motors

As previously mentioned, it is often necessary to provide a *road speed,* so the machine can be moved on the highway between job sites. (Generally, these travel distances are only a few miles, so road speeds in the range of 12 to 17 mph are satisfactory.) Shifting a transmission to increase pump speed is one method to increase vehicle speed, and a variable displacement motor provides another method. Reducing the motor displacement increases wheel speed. Available wheel torque decreases as displacement decreases, but this is not generally a problem for road travel.

A two-speed variable displacement motor has a high displacement position to provide high torque at the work site and a low displacement position for road travel (see Sec. 5.4.2). An infinitely variable displacement motor can also be used, but the cost is higher. In this case, a range of speeds from work speed to road speed is available. Both the two-speed and infinitely variable motors must be sized to prevent driving the wheel at higher than rated speed when they are shifted to the minimum displacement position.

6.5.4 Vehicle with Two Hydrostatic Transmissions

The vehicle shown in Fig. 6.14 has a separate in-line hydrostatic transmission for each drive wheel. The engine delivers power via a universal joint driveline to a right-angle drive gearbox. Each side of this gearbox powers an in-line hydrostatic transmission. The wheel is connected to the hydraulic motor. A final drive may or may not be used, depending on the vehicle performance criteria.

Steering of this vehicle is an issue. It will not operate like the vehicle shown in Fig. 6.13. Suppose the drive wheels are the rear wheels, and the front wheels are for steering. The swashplate controls on both pumps are set for forward travel. When the front wheels turn, both pumps continue to deliver flow for forward travel. There is no differential action. With both rear wheels

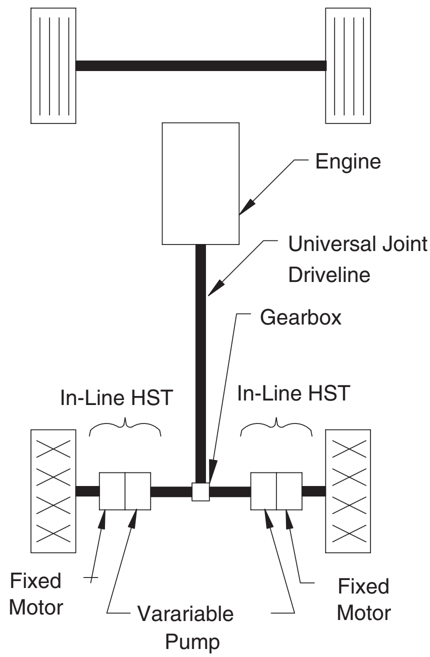


FIGURE 6.14
Vehicle with separate in-line hydrostatic transmissions for each drive wheel.

powering the vehicle forward, they will tend to slide the front wheels sideways, and turning will be defeated.

A typical application for this configuration is an agricultural machine called a *windrower*. This machine cuts hay and rolls into a continuous pile known as a *windrow*. The cutting mechanism, or header, is mounted in front of the drive wheels, which are the front (forward) wheels of the machine. The back (rear) wheels are non-steered caster wheels. There is a mechanical linkage from the steering wheel to the swashplate control on both pumps. For straight-ahead travel, the swashplate on both pumps is set at the same position. When the steering wheel is turned, the control on one side is pushed forward, and the control on the other is pushed backward. One pump delivers more flow (wheel on that side turns faster), and the other pump delivers less flow (wheel on that side turns slower). If the steering wheel is turned far enough, one pump swashplate will be in the full forward position, and one will be in the full reverse position. One drive wheel turns forward, and the other turns in reverse. In effect, the vehicle "walks" itself around in a tight circle.

The vehicle shown in Fig. 6.15 has a separate split hydrostatic transmission for each drive wheel. It does not have the universal joint driveline or the gearbox. Power is transferred via the hoses rather than mechanically.

The vehicle shown in Figs. 6.16a and 6.16b is a skid-steer machine, and it is a variation of the vehicle in Figs. 6.14 and 6.15. Skid steering can be used for

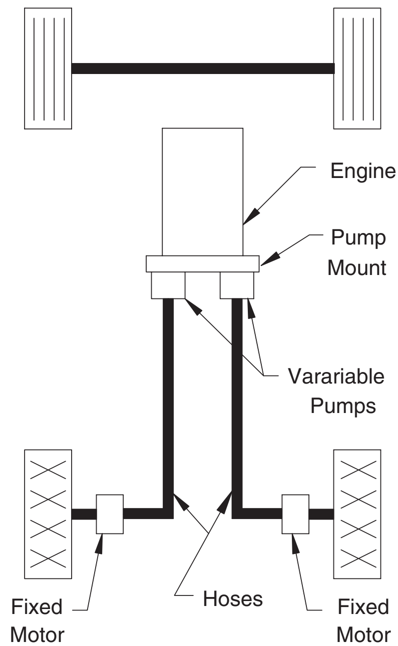


FIGURE 6.15
Vehicle with separate split hydrostatic transmissions for each drive wheel.

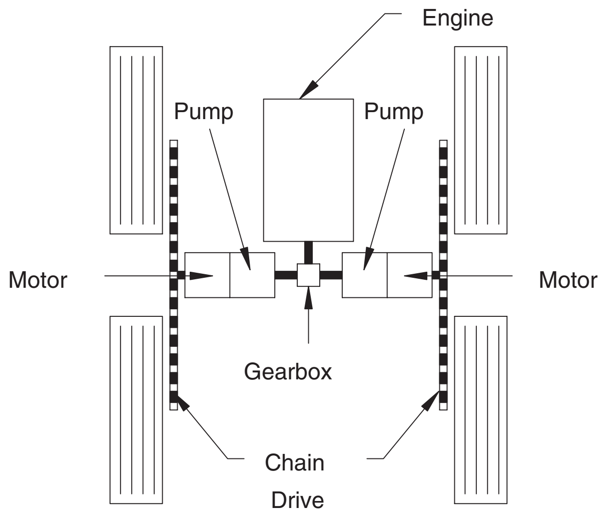
short-wheelbase machines. Both wheels on each side are connected with a chain drive; consequently, both wheels are powered. The machine is steered as previously described: one swashplate is shifted for forward travel, and one is shifted for reverse travel. The wheels slide as the vehicle pivots around. Typically, the swashplates are shifted with a hand lever. The operator can push one lever forward and pull the other back to pivot the machine in a tight circle.

6.5.5 Hydrostatic Drive for Three-Wheel Vehicle

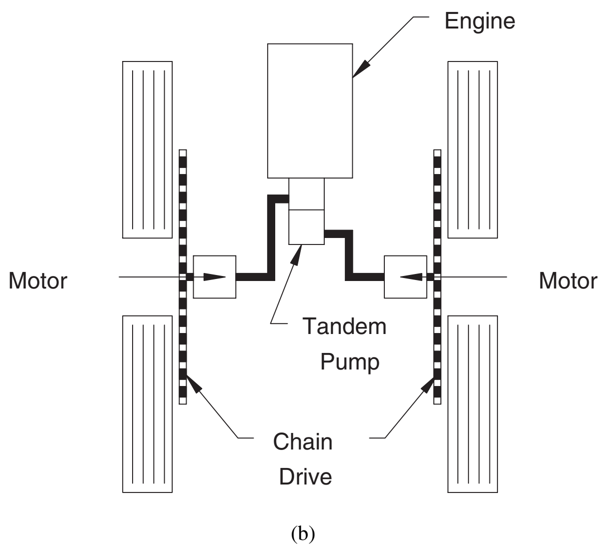
The vehicle shown in Fig. 6.17 has three wheel motors supplied by the same pump. In the design of such a vehicle, care must be taken to size the motors and final drives such that the tangential velocity of the front and rear wheels is approximately equal. The flow does divide in such a manner that the pressure drop across the front and rear motors is approximately equal. There are considerations in the design which are beyond the scope of this discussion.

6.5.6 Hydrostatic Transmission for Four-Wheel Drive Vehicle

A configuration where all four wheels are powered is shown in Fig. 6.18. A single pump provides flow to four motors. This machine can be built with two or four steerable wheels.



(a)



(b)

FIGURE 6.16
Skid-steer vehicle with (a) two in-line hydrostatic transmissions and (b) two split hydrostatic transmissions.

On three-wheel- and four-wheel-drive vehicles, the front wheels are configured to have a tangential velocity slightly higher (1 or 2%) than the rear wheels. This improves steering and helps to improve tractive effort. A large-capacity pump is required to operate all of these motors connected in parallel.

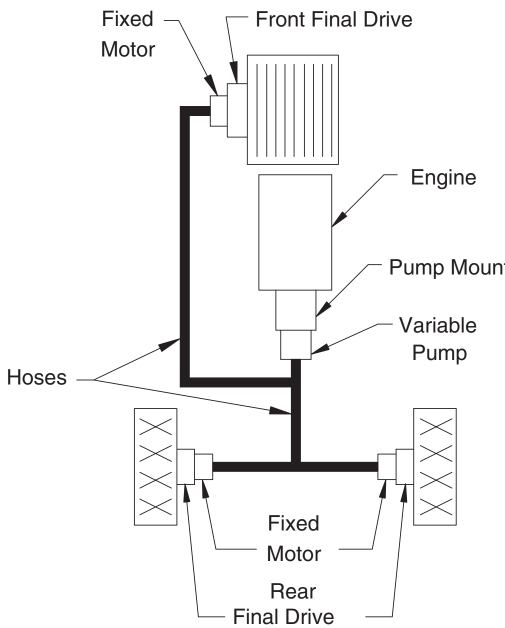


FIGURE 6.17
Hydrostatic drive with three wheel motors.

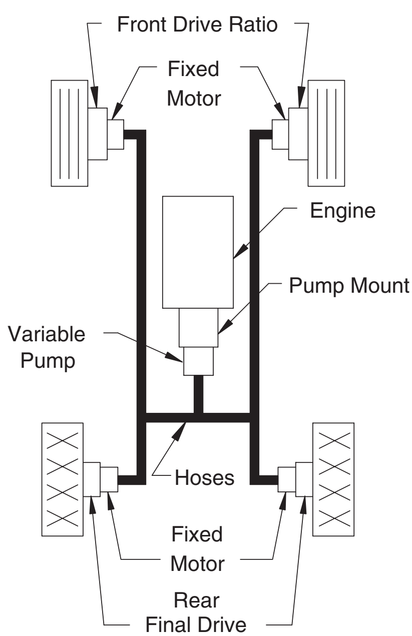


FIGURE 6.18
Hydrostatic drive with four wheel motors.

6.5.7 Summary

The examples described in this section were chosen to give the reader an appreciation of the range of options available to propel a vehicle with a hydrostatic transmission. They are not intended to be a complete list. Also, the examples were somewhat simplified and are not intended as a design guide.

6.6 Classification of Hydrostatic Transmissions

Hydrostatic transmissions can be classified as shown in Fig. 6.19. An open circuit is one where oil is delivered from the reservoir to the motor by the pump, and flow from the motor returns to the reservoir (Fig. 6.20a). In a closed-circuit hydrostatic transmission, fluid flows from the pump to the motor and back to the pump (Fig. 6.20b). Provision for ensuring that the circuit is always filled with fluid will be discussed in the next section.

An open-circuit hydrostatic transmission, like that shown in Fig. 6.20a or 6.20c, would not be used for a vehicle, because it cannot be reversed. Also, it provides no braking. (Mechanical brakes would have to provide all the dissipation of mechanical energy to bring the vehicle to a stop.) This transmission would be used only for an application, perhaps a conveyor drive, where the load is resistive and rotation is always in one direction.

It is useful at this point to consider what would happen if an open-circuit transmission were used for a vehicle. Suppose the vehicle is starting down a hill, and the weight causes the wheel to drive the motor (overrunning load). As the motor speeds up, it tries to "pull" fluid from the line faster than the pump is delivering fluid into the line. Assuming the line does not collapse due to this suction, the motor eventually begins to pull fluid out of the pump,

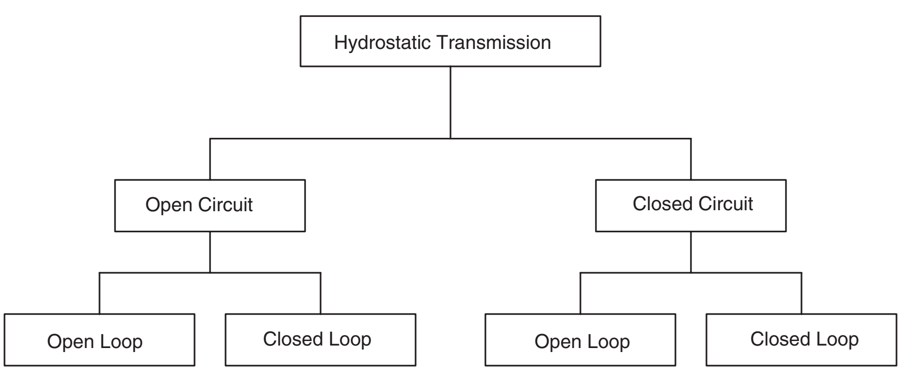


FIGURE 6.19
Classification of hydrostatic transmissions based on circuit type.

and a condition develops in which both the pump and motor are partly filled with air. Both can be damaged at this point.

Closed-loop circuits have a provision for sensing the motor speed and using this signal to adjust the pump displacement to increase or decrease the motor speed until it reaches the set point. An open-loop circuit has no feedback of the motor speed. When the pump displacement is set at a given point, motor speed decreases as pressure (load) increases. Speed then varies with load. Sometimes this is a undesirable. An open-circuit, closed-loop hydrostatic transmission is shown in Fig. 6.20c, and a closed-circuit, closed-loop hydrostatic transmission is shown in Fig. 6.20d.

The closed-circuit transmissions (Figs. 6.20b and 6.20d) have reversible pumps and motors. Notice that the symbols have solid arrowheads pointing in both directions.

The circuit diagrams in Figs. 6.20a–d contain only the key features of the various transmissions. These diagrams are not complete; they are included to illustrate the four classifications.

6.7 Closed-Circuit Hydrostatic Transmissions

6.7.1 Charge Pump

Closed-circuit hydrostatic transmissions are used for vehicles and for other applications in which reversing is required. The technique used to ensure that the main circuit is always filled with fluid is illustrated in Fig. 6.21. The main pump is an axial piston pump, and the motor is an axial piston motor. A charge pump (generally a small, fixed displacement pump) is built into the housing with the main pump and operates off the same input shaft as the main pump. The purpose of the charge pump is twofold:

1. It replaces the fluid that leaks past the pistons into the pump housing. The same leakage flow occurs in the motor. This flow is essential, because it provides lubrication and seals clearances.
2. It provides a flow of cooling fluid through the pump and motor housings. When the high-pressure fluid in the main circuit leaks into the housing, mechanical energy is converted into heat energy. In addition, heat results from friction between the moving parts. A flow of cooling fluid is required to remove this heat.

Fluid flow in the main circuit is quite simple. The fluid flows to the motor and returns. Flow in the charge pump circuit is as follows:

1. The charge pump receives fluid from the reservoir and delivers it to two check valves, one on each side of the main circuit. Main

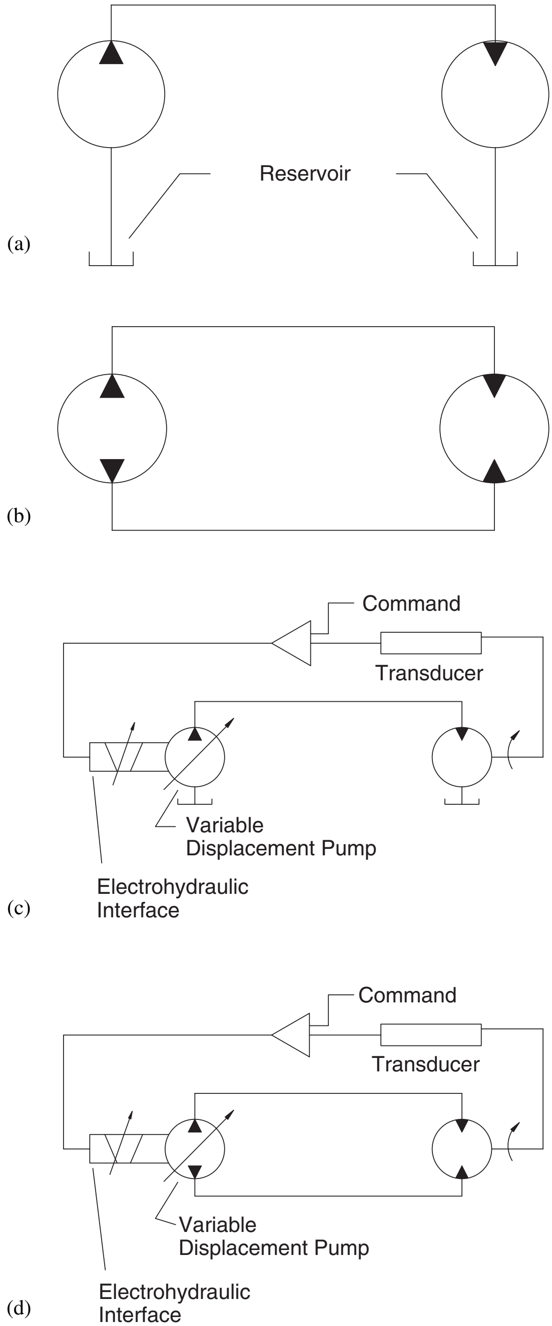


FIGURE 6.20
(a) Open-circuit hydrostatic transmission, (b) closed-circuit hydrostatic transmission, (c) open-circuit, closed-loop hydrostatic transmission, and (d) closed-circuit, closed-loop hydrostatic transmission.

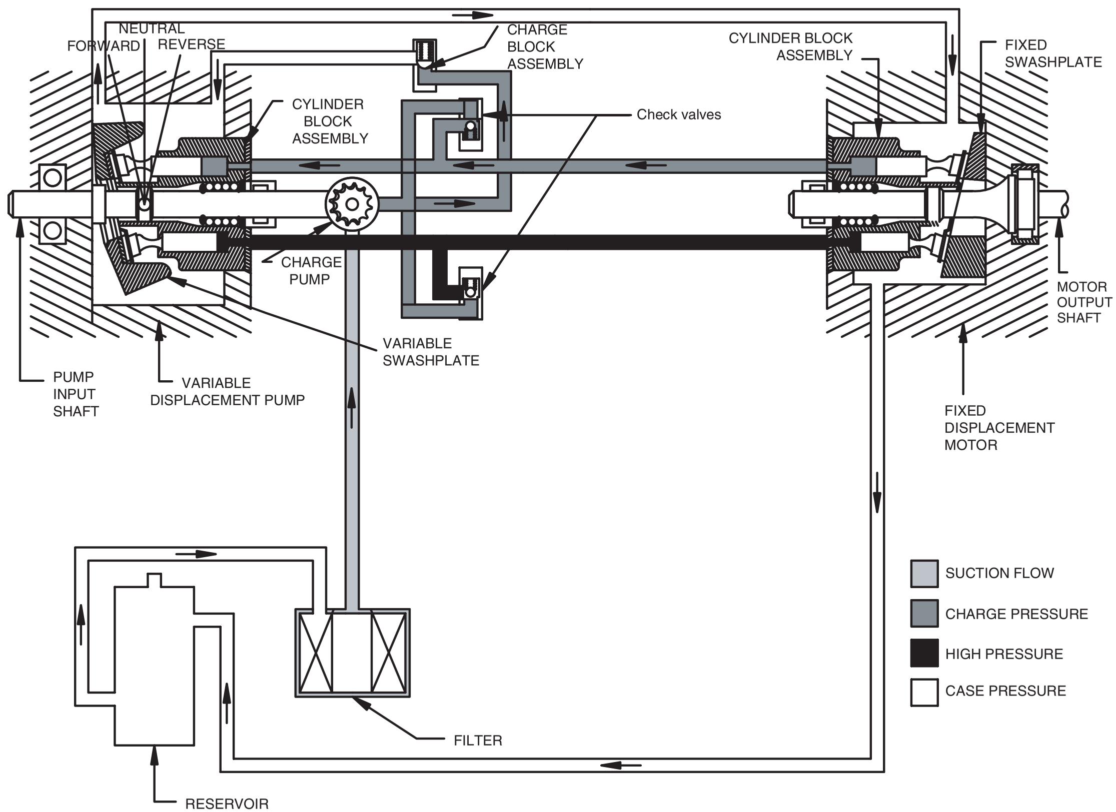


FIGURE 6.21
Typical closed-circuit hydrostatic transmission with charge pump circuit. (Reprinted with permission from Sauer-Danfoss Corp.)

circuit pressure keeps the check valve seated on the high-pressure side. Flow will occur through the check valve on the low-pressure side when the main circuit return pressure is less than the charge pump pressure. Pressure on the return side will be low if fluid has leaked out of the main circuit. A check valve is needed on both sides, because the pump is reversible; either side can be the high-pressure side.

2. Fluid that does not flow over a check valve drops across the charge relief valve into the pump housing. In the example shown in Fig. 6.21, this fluid flows through a drain line to the motor housing, through the motor housing, and into a drain line back to the reservoir.

Because of the fluid that leaks across in the clearances between the pistons and the pump housing, the motor "sees" less fluid than the pump is theoretically pumping. In like manner, the motor has a flow of fluid in the clearances between moving parts. The required charge pump flow is the sum of the fluid lost from the main circuit at both the pump and motor ends. Suppose the volumetric efficiency of the pump is e_{vp}. Volumetric efficiency of the motor is e_{vm}. The required flow of make-up oil is

$$Q_1 = (1 - e_{vp}e_{vm})Q_p \tag{6.14}$$

where Q_p is the theoretical flow from the main pump.

Maximum charge pump flow is required when the main circuit is at maximum operating pressure. Suppose that $e_{vp} = e_{vm} = 0.9$, typical values for an axial piston pump (Chapter 4) and an axial piston motor (Chapter 5).

$$Q_1 = 0.19Q_p \tag{6.15}$$

The charge pump must be sized to provide at least 19% of the main pump flow just to replace the leakage. It may need to be twice as large to provide the needed cooling flow.

6.7.2 Shuttle Valve

For larger hydrostatic transmissions, a shuttle valve is incorporated in the motor end (Fig. 6.22). Pressure on the high-pressure side shifts the spool of the shuttle valve so that the fluid on the low-pressure side has a pathway to a charge relief valve mounted in the end plate of the motor. Fluid drops across this relief valve into the case of the motor where it combines with leakage flow and flows through a drain line to the pump, through the pump housing, and back to the reservoir.

The arrangement shown in Fig. 6.21 has a charge relief valve only at the pump end. Case drain fluid from the pump flows to the motor through a case

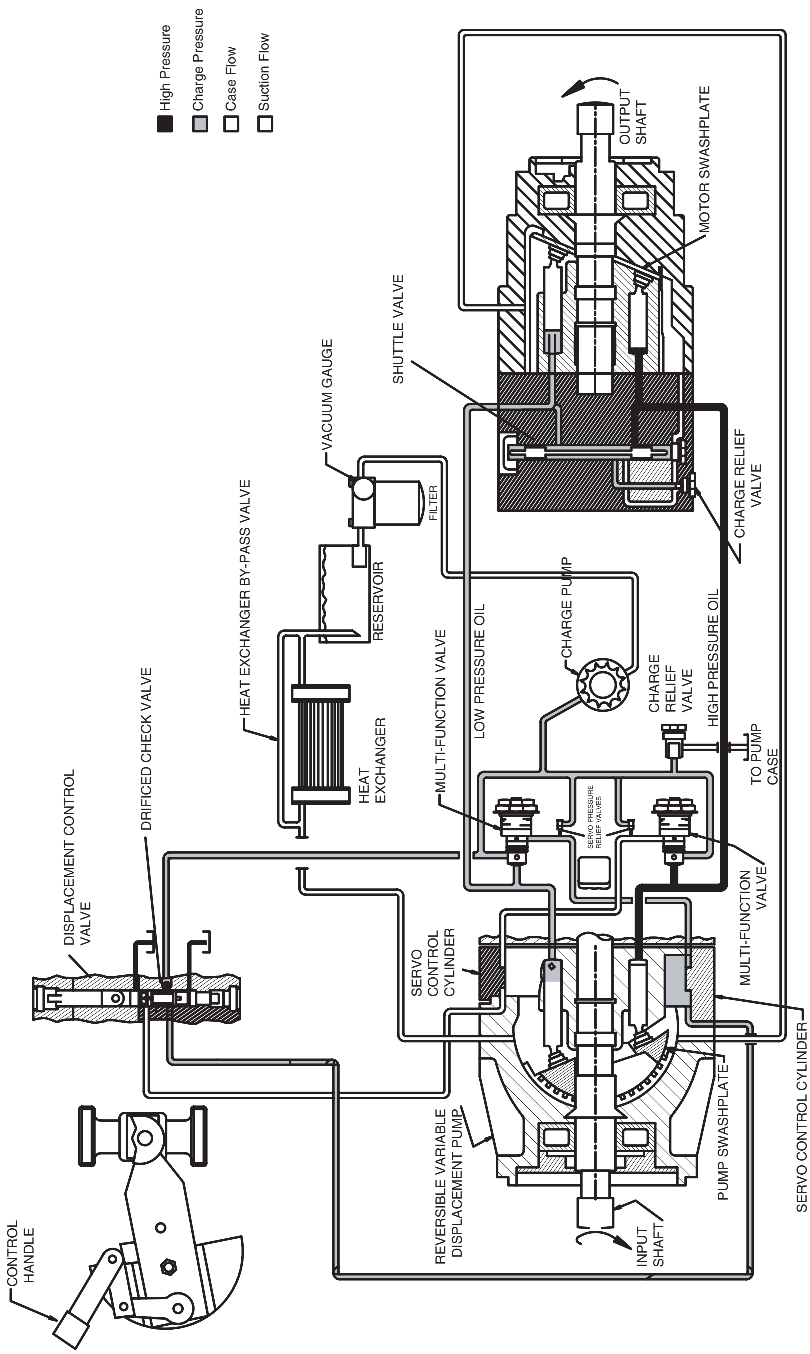


FIGURE 6.22
Shuttle valve in motor end of hydrostatic transmission.

drain line, through the motor housing, through another drain line back to the pump, through the pump housing, and then to the reservoir. A larger transmission requires more cooling flow at the motor; consequently, it is necessary to have the shuttle valve and second charge relief at the motor end, as shown in Fig. 6.22.

The preceding discussion about the second charge relief valve required in the motor end of larger hydrostatic transmissions illustrates a key point about the selection of a hydrostatic transmission. The pump and motor must be engineered to work together as a unit. When the two are supplied by the same manufacturer and sold as a unit, it is understood that a great deal of engineering has gone into the design. Many problems are solved for you by the circuitry built into the unit. If the transmission does not perform satisfactorily in a vehicle, the vehicle designer and transmission designer work together to solve the problem. Excellent technical support is supplied by many transmission manufacturers.

6.7.3 Cross-Port Relief Valves

A circuit with a cross-port relief valve is shown in Fig. 6.23. The cross-port relief valve package has two relief valves, one for each side. (Two are needed, because either side can be the high-pressure side.) Once the pressure reaches a certain level, fluid drops across the relief valve to the other side of the main circuit. The motor then stops turning, and the transmission is stalled.

A vehicle hydrostatic transmission should be designed to achieve wheel slip before the relief valve opens. (A brief review of traction mechanics is given in Appendix 6.1) The slipping wheel is the pressure-limiting device. The main purpose of crossover relief valves is to "shave" the pressure peaks resulting from dynamic maneuvers. These values should pass flow for only short periods. High flows at high pressure drops generate heat at a rapid rate. A condition that opens the cross-port relief must be relieved quickly to prevent the transmission from overheating and being damaged.

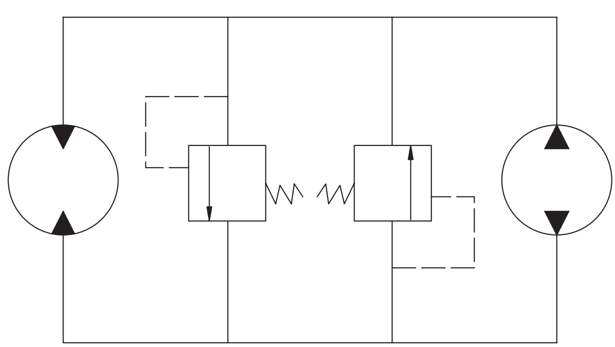


FIGURE 6.23
Circuit with cross-port relief valves.

In applications where we cannot depend on wheel slip, other arrangements must be made. One option is to increase the size of the charge pump and thus provide additional cooling flow. A second option is to incorporate a hot-oil replenishing valve in the circuit.

6.7.4 Multipurpose Valves

Some manufacturers supply a multipurpose valve that incorporates several features in one valve. Typically, two of these valves are installed in the main pump housing, one for each side of the main circuit. One type of multipurpose valve incorporates the following features:

1. High-pressure relief [meets the requirement for a cross-port relief valve (Fig. 6.23)]
2. Check valve [meets the requirement for check valve (Fig. 6.21)]
3. Bypass valve [If the vehicle will not start, the bypass valve allows the vehicle to be towed. The transmission will "free wheel."]
4. Pressure limiter [This feature destrokes the pump (reduces flow to zero) in response to excessive pressure. It provides the "remain stalled without damage" feature of a hydrostatic transmission.]

6.7.5 Summary

Using the broadest definition, a hydrostatic transmission is simply a pump and motor connected together. For some simple applications, it is possible to select a pump from one manufacturer and motor (or motors) from another manufacturer, connect them together with hoses or tubing, and get acceptable performance. An example of a simple application would be a conveyor drive on a mobile machine. The load varies over a relatively narrow range, and speed control is not critical. Dramatic shock loads are infrequent. Reversing is not required. The pump and motor can be sized to operate at less than 1,500 psi. In this case, a fixed displacement pump and fixed displacement motor can be used satisfactorily with very little engineering design required.

Considerable circuit design effort is required to get a pump and motor to function satisfactorily as a hydrostatic transmission for a vehicle, particularly a heavy vehicle performing dynamic maneuvers. These units have the following built-in features:

1. Charge pump, charge relief valve, check valves and related circuitry
2. Shuttle valve and motor-end charge relief valve
3. Cross-port relief valve
4. Bypass valve

Several of these features can be incorporated into one valve known as a *multipurpose valve.*

6.8 Closed-Circuit, Closed-Loop Hydrostatic Transmissions

6.8.1 Review of Pump and Motor Operating Characteristics

As explained in Chapters 4 and 5, both pump and motor volumetric efficiencies decrease as pressure increases. A simple example illustrates how hydrostatic transmission performance is governed by the characteristics of the pump *and* motor. Suppose, a pump has a displacement of 1.925 in^3/rev and is driven at a speed of 2400 rpm. Theoretical flow is

$$\frac{1.925\ \text{in}^3/\text{rev} \times 2400\ \text{rev/min}}{231\ \text{in}^3/\text{gal}} = 20\ \text{GPM}$$

The motor used in this hydrostatic transmission also has a displacement of 1.925 in^3/rev. When the pump flow is delivered to the motor, the motor speed is

$$\frac{20\ \text{gal/min} \times 231\ \text{in}^3/\text{gal}}{1.925\ \text{in}^3} = 2400\ \text{rpm}$$

Now, consider what happens when operating pressure increases to 1000 psi. The pump volumetric efficiency at this operating pressure is 0.90, so the actual pump output is

$$20 \times 0.90 = 18\ \text{GPM}$$

Flow to the motor is no longer 20 GPM but 18 GPM. If there is no leakage in the motor, the motor speed will be

$$\frac{18\ \text{gal/min} \times 231\ \text{in}^3/\text{gal}}{1.925\ \text{in}^3/\text{rev}} = 2160\ \text{rpm}$$

However, the 1000 psi operating pressure also causes leakage in the motor, resulting in a motor volumetric efficiency of 0.90. Actual motor output speed is

$$\frac{18\ \text{gal/min} \times 231\ \text{in}^3/\text{gal}}{1.925\ \text{in}^3/\text{rev}} \times 0.90 = 1944\ \text{rpm}$$

Input speed to the pump is 2400 rpm, and the achieved output speed from the motor is 1944 rpm. Overall volumetric efficiency of the HST is

$$\frac{1944}{2400} = 0.81$$

Another way of calculating this efficiency is to multiply the volumetric efficiency of the pump times the volumetric efficiency of the motor.

$$0.9 \times 0.9 = 0.81$$

Motor speed varies with pressure in a hydrostatic transmission.

To reinforce the concept of speed variation with pressure variation, it is helpful to plot the set of pump and motor performance curves on the same graph (Fig. 6.24). (The curves in Fig. 6.24 are for illustration purposes only. They were chosen to give an exaggerated view of pump and motor interaction. Manufacturer's data for a specific pump and motor must be used for a design.) The pump and motor have the same displacement, so the theoretical performance of both is shown as a heavy dark line in the middle of the family of curves. The pump performance curves are shown below the theoretical line, and the motor curves above the line.

Suppose, the pump is driven at 2400 rpm. Proceeding vertically from 2400 rpm on the horizontal axis up to the theoretical line and then across to the vertical axis, the theoretical pump flow is 20 GPM. If this flow is delivered to the motor, the theoretical speed is 2400 rpm. (Find 20 GPM on the vertical axis, move horizontally to the theoretical line, and down to the horizontal scale to read a motor speed of 2400 rpm.) With this instruction on how to use the graph, we will proceed to work an example.

The pump is driven at 2400 rpm and develops 2000 psi pressure. What flow is delivered to the motor? Follow the dotted line up to the 2000 psi curve and then move horizontally to the vertical axis to read a pump flow of 16 GPM. What is the motor speed when this 16 GPM is delivered to the motor? Follow the dotted curve from 16 GPM horizontally to the 2000 psi motor curve and then down to the horizontal axis to read a motor speed of 1530 rpm. If no pressure was developed, the motor speed would have been 2400 rpm. Actual speed is 1530 rpm, or 36% less.

With this review of pump and motor operating characteristics, it should be evident how motor speed varies with pressure. If precise control of the motor speed is required, some means must be provided to sense the actual motor speed, and adjust pump output until this speed is achieved.

It is possible to use the curves in Fig. 6.24 to work a different problem. Suppose the motor must turn at 1800 rpm when operating pressure is 2000 psi. What pump flow is required? To avoid confusion, the curves are replotted in Fig. 6.25. Find 1800 rpm on the horizontal scale and follow the dotted line up to the 2000 psi motor curve, then across to the vertical axis to read 18.7 GPM.

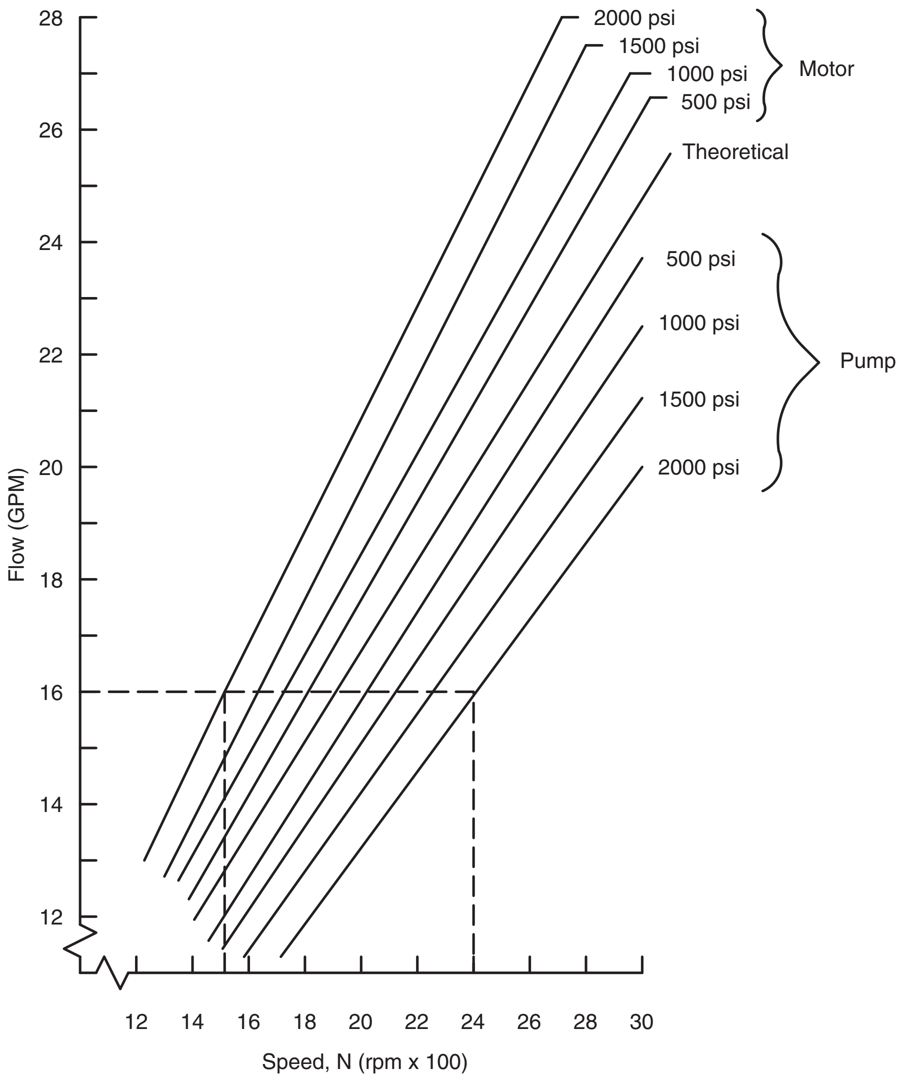


FIGURE 6.24
Defined performance curves for a pump and motor in a hydrostatic transmission.

What speed must the pump be driven to deliver 18.7 GPM at 2000 psi? To answer this question, follow the 18.7 GPM line horizontally to the 2000 psi pump curve. Then, drop down to the horizontal axis to read 2800 rpm. The pump must turn at 2800 rpm for the motor to turn at 1800 rpm.

It must be emphasized that many (most) HST designs will have much better performance than that indicated by the defined curves in Fig. 6.24. These curves were chosen to help the reader understand the influence of pump-motor interaction on output speed.

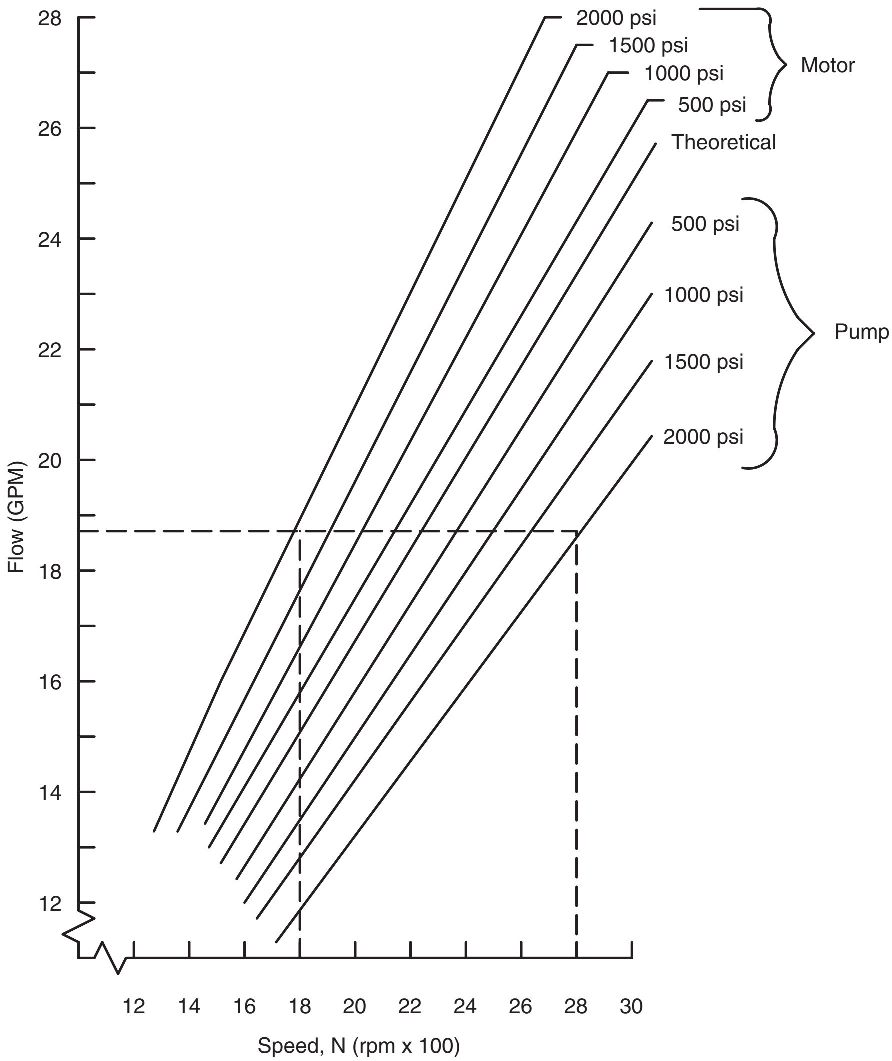


FIGURE 6.25
Defined performance curves for a pump and motor in a hydrostatic transmission, replotted to work example problem.

6.8.2 Servo-Controlled Pump

Before beginning our discussion of closed-loop hydrostatic transmissions, it is necessary to first learn how a servo-controlled pump operates. A variable displacement axial piston pump will be used as an illustration. When configured for servo control, this pump will have a control piston mounted in the pump housing. When the control piston extends, it moves the swashplate to increase the amount of fluid pumped by the pistons. (Displacement of the pump is increased.) Flow of fluid to the control piston, and thus its position, is controlled with a servo valve. A pump with these features is called a *servo-controlled pump.*

A servo valve operates like a directional control valve. The spool shifts in one direction to direct pressurized fluid to Port *A*, and in the other direction to direct pressurized fluid to Port *B*. The spool in a servo valve is precisely machined; consequently, the cost is higher than the cost of a standard directional control valve.

It is helpful to first consider a manually controlled servo pump (Fig. 6.26). Suppose the manual control lever *L* is moved to the right (rotated in the clockwise direction). The spool of the servo valve is shifted to the left. High-pressure fluid is directed to the top control piston, causing it to extend. The bottom control piston is connected to the case drain; thus it retracts when the top control piston extends. As the two control pistons move, the swashplate is rotated counterclockwise, thus reducing the amount of fluid pumped.

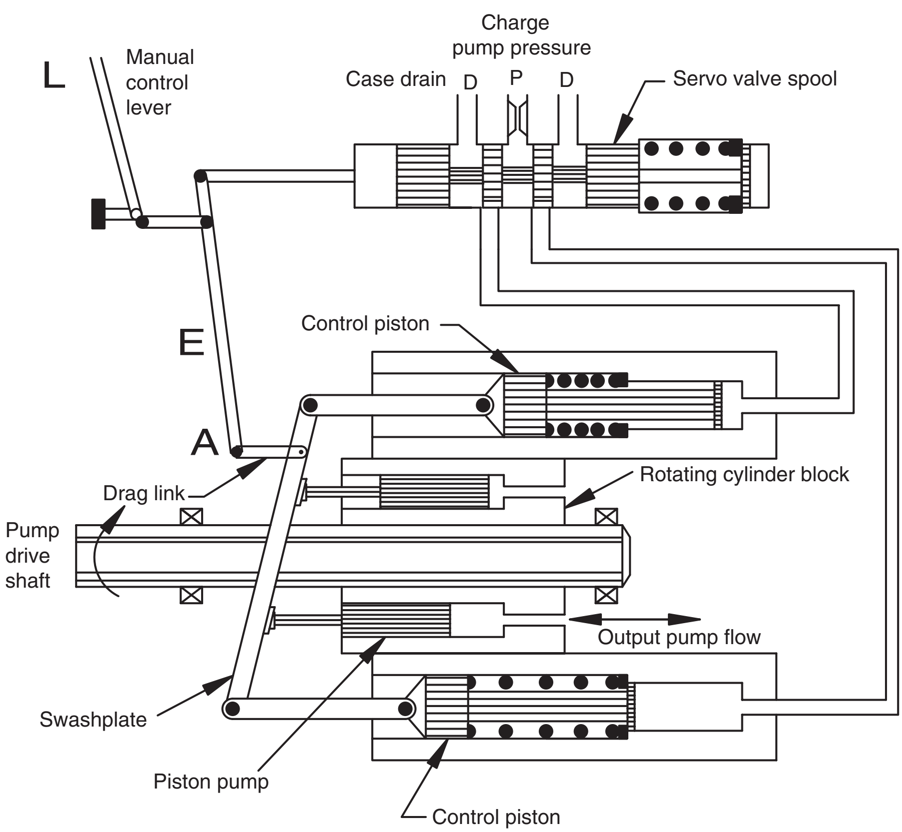


FIGURE 6.26
Manually operated servo pump.

When the swashplate moves, it pushes Point *A* to the left. The link *E* pivots and pushes the spool of the servo valve to the right. This spool movement closes Ports A and B of the servo valve, thus locking the control pistons in a position that corresponds to the new position of the manual control lever. This position of the swashplate is held until the manual control lever is moved to a new position.

The obvious question is, why not connect the manual control lever directly to the swashplate? Then, when the lever is moved, the swashplate is rotated. Smaller hydrostatic transmissions are operated in this manner. With larger transmissions (>50 hp), the force required to move the swashplate becomes large enough that operator fatigue becomes an issue. The servo valve and control pistons make the swashplate control lever much easier to operate.

The servo pump shown in Fig. 6.27 is like that shown in Fig. 6.26, except a torque motor is used to position the spool of the servo valve. A torque motor rotates through several degrees of rotation, when a current is passed through the winding. The torque motor shown has a *flapper* attached to the armature. This flapper is centered in the nozzle such that the pressure drop on both sides is equal. Pressure on both ends of the servo valve spool is equal. This design is called a *flapper nozzle* torque motor.

When the servo valve spool is centered, both Ports *A* and *B* are blocked, and the control pistons are locked into position. The swashplate is then locked into position.

The following sequence of events occurs when a current is delivered to the torque motor.

1. The armature rotates and moves the flapper to partly block one side of the nozzle.
2. Pressure builds on the side of the partly blocked nozzle, and this pressure shifts the servo valve spool. Pressurized fluid is directed to Port *A*, and Port *B* is connected to the case drain.
3. Pressurized fluid from Port *A* causes the top control piston to extend. Fluid from the bottom control piston returns to the reservoir through Port *B*, and it retracts.
4. The swashplate is moved to a new position.

As the swashplate moves to a new position, some *feedback* is needed to move the servo valve spool back to the center position. Otherwise, the spool will stay shifted, Ports *A* and *B* will stay open, and the swashplate will continue to move until it reaches its full displacement in one direction. To accomplish this feedback, a feedback lever is connected to the swashplate. This lever rotates as the swashplate moves. When the lever rotates, it displaces the swashplate feedback spring, which moves the armature until the flapper is recentered. The nozzle now has equal pressure drop on each side. Equal pressure on both sides causes the servo valve spool to move to the center, Ports *A* and *B* are blocked, and the control pistons are locked into position. The

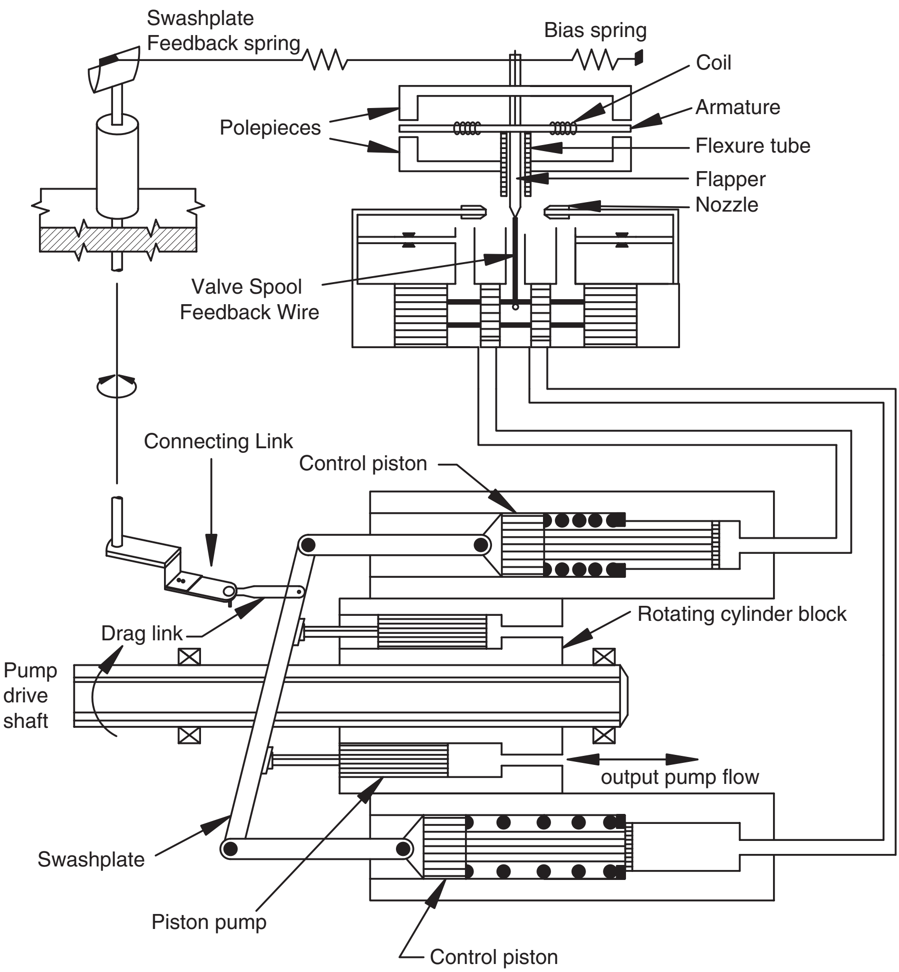


FIGURE 6.27
Servo pump operated with flapper nozzle torque motor.

swashplate stays in this position until another current is delivered to the torque motor winding.

6.8.3 Servo Valve Circuit

Some explanation of the electrical circuit required for a servo valve is presented here to help readers understand how the servo valve in the servo pump operates when it is used in a closed-loop transmission. A more detailed explanation will be given in the Chapter 11. The key features of the electrical circuit for a closed-circuit, closed-loop hydrostatic transmission are shown above the dotted line in Fig. 6.28. The motor output shaft drives a

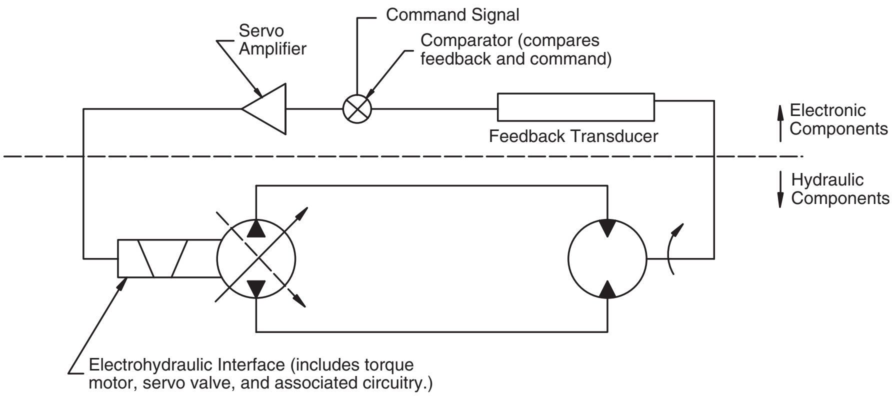


FIGURE 6.28
Electrical circuit for a servo valve.

transducer, typically a tachometer generator. Voltage output from the tachometer generator is directly proportional to shaft speed. As shaft speed increases, voltage increases, and vice versa.

The comparator compares the tachometer generator voltage with the command voltage. The difference between the two is the error voltage. If the tachometer generator voltage is equal to the command voltage, meaning that the motor is turning at the desired speed, then the error voltage is zero.

The error voltage is fed to a servo amplifier. The servo amplifier produces an output current proportional to the input voltage. This current is fed directly to the coil of the torque motor. (Coil is another name for the windings on the armature.) The armature rotates, initiating the sequence of events described in the previous section.

6.8.4 Response Time for Closed-Loop Circuit

Response time is the time required for motor speed to reach a new set point. Suppose the command signal is changed with a step input; the command signal is 5 V, and it is stepped to 10 V. The time required to reach the new motor speed depends on the natural frequency of the circuit. Neglecting energy dissipation (damping), natural frequency is given by

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{6.16}$$

where f = natural frequency (Hz)
k = elasticity (force required to produce a unit deflection) (lb_f/ft)
m = mass ($lb_f \cdot s^2/ft$)

If k is large, the system is said to be *stiff*, meaning that little deformation occurs when a large force is applied. A stiff system has a high natural frequency. If m is large, meaning that a heavy load is being moved, then the natural frequency will be low.

Elasticity, k, is a function of the quantity of fluid under compression. (For this discussion, we ignore the swelling of the lines when pressure is applied.) If the lines are long, k will be small, meaning that a small force will cause a deformation. (The deformation is the change in volume of fluid in the lines.) Conversely, if the lines are short, k will be large.

Suppose a heavy load is attached to the motor (m is large), and the lines between the pump and motor are long (k is small). The natural frequency will be low, and it will take a relatively long time for the motor to reach a new speed. Systems with a high natural frequency have a fast response time. To reduce response time, the line length must be reduced as much as possible. Reducing length increases k and thus increases natural frequency.

Fast response time is often important in a closed-circuit design. Components must be sized correctly and positioned to minimize the fluid under compression. After doing all that can be done with the hydraulic circuit design, there are methods for reducing response time that can be incorporated in the design of the electronic circuit.

6.8.5 Operation of Closed-Circuit, Closed-Loop Hydrostatic Transmission

The hydrostatic transmission, shown in Fig. 6.28, is operating under a constant load. Speed is set by the command voltage at 1000 rpm. The feedback transducer is a tachometer generator, which produces a voltage proportional to motor rpm. The load starts to increase, pressure increases, and the pump and motor volumetric efficiencies decrease. The motor begins to slow, and the following sequence of events is initiated.

1. The tachometer generator voltage falls when the motor speed decreases.
2. When the new tachometer generator voltage is compared to the command voltage, a negative error voltage is produced.
3. The negative error voltage causes the servo amplifier to produce a current.
4. Current from the servo amplifier causes the torque motor armature to rotate.
5. Rotation of the armature moves the flapper to create a pressure imbalance on the servo valve spool.
6. The spool shifts to direct high-pressure fluid to the bottom control piston, causing it to extend. The top control piston simultaneously retracts.

7. When the control pistons move, the swashplate rotates clockwise to increase pump displacement.
8. More pump flow is delivered to the motor and motor speed increases.
9. Tachometer generator speed increases as motor speed increases.
10. When the tachometer generator voltage equals the command voltage, the error voltage is zero.
11. Zero error voltage produces zero current.
12. Zero current means the armature returns to the neutral position.
13. With the armature centered, the flapper is centered, and equal pressure exists on both ends of the servo valve spool, and the spool is centered.
14. When the servo valve spool is centered, Ports *A* and *B* are blocked, and the control pistons are locked into position.
15. The swashplate is locked into position, and the pump operates at this new displacement.

If the load decreases and the motor speeds up, the swashplate angle is reduced, the pump displacement is less, and the motor slows until its speed equals the speed set by the command voltage. Each time the load changes, the control compensates to keep speed constant.

The feedback lever, shown in Fig. 6.27, is not needed when a servo pump is operated in the closed-loop configuration shown in Fig. 6.28. When the tachometer generator voltage equals the command voltage, the error voltage is zero. With zero error voltage, current from the servo amplifier drops to zero, and the torque motor armature returns to its centered position.

6.9 Hydrostatic Transmission Design

The design of a hydrostatic transmission is a classic design problem. The various variables are interrelated in a very interesting way. Typically, the designer has to loop back several times to restart the design at a different point.

The most efficient way to illustrate the design procedure is to work an example. The example given here is the design of a hydrostatic transmission (HST) for a sweet sorghum harvester.

Hydrostatic transmissions are used for propel applications (mobile machines like the sweet sorghum harvester) and nonpropel applications (for example, a winch). The manufacturers of the different components (pump, hydraulic motor, pump mount, and planetary final drive) may list different ratings for propel vs. nonpropel applications.

In general, the block diagram for a propel application is shown in Fig. 6.29. The pump mount is a gearbox (may have a built-in clutch) that bolts directly to the engine flywheel housing. The splined shaft of the HST pump slides in a matching coupling in the pump mount, and the pump is bolted in place. The final drive is generally a planetary gear set. The hydraulic motor splined shaft slides in a matching coupling and drives the sun gear. The wheel is bolted to the planetary gear carrier. The final drive converts a high-speed, low-torque input into a low-speed, high-torque output.

The notation used for the design example is given in Appendix 6.1, Table A6.1. It is helpful to review several basic concepts. The total force to move the vehicle is given by

$$F = W_g\left(\frac{R}{1000} + \frac{P_g}{100}\right) + F_d \tag{6.17}$$

where W_g = gross vehicle weight (lb_f)
R = rolling resistance (lb_f/1000 lb_f gross vehicle weight)
P_g = maximum grade vehicle must climb (%)
F_d = required drawbar pull (lb_f)

Rolling resistance for a range of surface conditions is given in Table 6.1. The total wheel torque to move the vehicle is

$$T_w = Fr \tag{6.18}$$

where r = wheel rolling radius (in)

Total wheel torque is divided by the number of driving wheels to determine the torque that must be developed at each wheel.

Torque required to spin the wheels is given by

$$T_s = W_d \mu r \tag{6.19}$$

where W_d = weight on driving wheels (lb_f)
μ = coefficient of friction between wheel and surface (decimal)
r = wheel rolling radius (in)

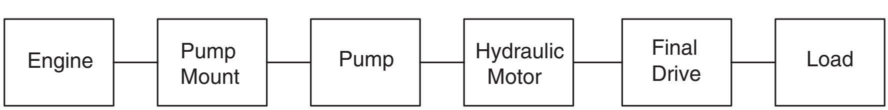


FIGURE 6.29
Block diagram for vehicle HST.

TABLE 6.1

Rolling Resistance Expressed as lb_f per 1000 lb_f Gross Vehicle Weight

Surface	Rolling resistance, *R*
Concrete, excellent	10
Concrete, good	15
Concrete, poor	20
Asphalt, good	12
Asphalt, fair	17
Asphalt, poor	22
Macadam, good	15
Macadam, fair	22
Macadam, poor	37
Cobbles, ordinary	55
Cobbles, poor	85
Snow, two-inch	25
Snow, four-inch	37
Dirt, smooth	25
Dirt, sandy	37
Mud	37–150
Sand, level and soft	60–150
Sand, dune	160–300

If $T_s > T_w$, the vehicle can meet the functional objective, i.e., it can overcome the rolling resistance, climb the grade, and provide the required drawbar pull.

This very brief review does not consider the weight transfer that occurs when a vehicle is traveling up a slope, pulling a load. The purpose of the design problem is to focus on the HST design, not solve the entire vehicle propel problem. Typically, the vehicle designer will partner with a HST designer and work as a team to solve the total problem.

6.9.1 Hydrostatic Drive for Sweet Sorghum Harvester

An experimental vehicle to harvest whole-stalk sweet sorghum was designed by a university research team. At present, the harvester is a pull-type machine, meaning it is towed behind a tractor and powered via a universal joint driveline.

The decision has been made to convert the harvester to a self-propelled machine. The configuration chosen is a single caster wheel at the rear and two drive wheels at the front. Your assignment is to design a hydrostatic transmission to drive the two front wheels. You may use high-torque, low-speed motors to drive the wheels directly, or high-speed motors mounted in planetary gear final drives.

Some catalog data for various pumps, motors, and planetary final drives are given in Appendix 6.2. These data are provided for solution of this example problem and the problems given at the end of the chapter.

Total weight on the two front wheels is 3310 lb_f, and weight on the rear caster wheel is 3750 lb_f. It is estimated that the addition of an engine, pump mount, pump, and associated accessories will add 1110 lb_f to the front wheels. Additional weight due to wheel motors can be neglected.

Constraints:

1. You will specify the engine power required. The hydraulic circuits on the harvester (other than the HST) require a total power of 34 hp.
2. Initially, assume that the pump mount is a direct drive, meaning that the pump turns at the same rpm as the engine. You will want to operate the engine in the range 1800 to 2000 rpm when it is developing maximum torque for field operations. Do not plan to continuously operate the engine at a speed higher than 2000 rpm when it is under full load. It can be operated at 2200 rpm for road travel, as it will be developing only the power required for the HST. If needed, you may choose a two-speed pump mount with a ratio in the range 1:1.2 to 1:1.5.
3. As a starting point for your design, assume that the harvester has 42-in. diameter front wheels. You may choose larger or smaller wheels if needed to satisfy a design criterion, but do not go outside the range 28–48 in.
4. Assume a pump volumetric efficiency of e_{vp} = 0.85 and a motor volumetric efficiency of e_{vm} = 0.85 at design pressure.
5. Do not design for a relief valve setting greater than 3500 psi.
6. If you use planetary final drives, assume they are 98% efficient. It is also appropriate to assume that the pump mount is 98% efficient.

Design for the following field conditions:

1. Rolling resistance R = 50.
2. Maximum grade in the field 8%.
3. The machine must be able to develop a maximum drawbar pull of 250 lb_f when traveling at the desired harvesting speed of 2.2 mph. Assume a coefficient of friction, μ = 0.4 for field conditions.

Design for the following transport conditions:

1. When fields are less than two miles apart, the machine will be driven between fields. Design for a maximum road speed of 15 mph.

2. When fields are more than two miles apart, the machine must be loaded onto a tilt-bed trailer for transport. Design the HST so that the harvester can load itself onto a tilt-bed trailer at a 27% incline. Assume a coefficient of friction μ = 0.6.

Requirements:

1. Select a hydrostatic drive [variable displacement pump, fixed displacement motors (planetary final drive)] for the harvester. Specify:
   a. Maximum pump flow
   b. System operating pressure
   c. Show that the HST can supply enough torque to spin the wheels.
2. Specify required engine horsepower for the harvester. Show that all power requirements are met.
3. What is the maximum drawbar pull with μ = 0.4? μ = 0.6?
   What drawbar pull can be maintained at a 2.2 mph field speed?
4. Show that the harvester can load itself onto the tilt-bed trailer.

6.9.2 Example Solution for Design of Hydrostatic Drive for Sweet Sorghum Harvester

This solution is *a* solution, not *the* solution. Several other designs will also meet the functional requirements. Notation used for presentation of the design is summarized in Table 6.2.

6.9.2.1 Maximum Torque Requirement

The maximum torque occurs at wheel slip.

$$T_s = W_d \mu r$$

where W_d = weight on drive wheel (lb_f)
μ = coefficient of friction (decimal)
r = rolling (or loaded) radius of drive wheel (in)

We assume μ = 0.4 for a sweet sorghum field. The smallest diameter wheel is initially selected. Weight on each drive wheel, after installation of the engine, is

$$W_d = (3310 + 1110)/2$$

$$= 2210 \text{ lb}_f$$

TABLE 6.2

Notation Used for HS Design Problem

Variable Name	Description	Units
D_w	wheel diameter	in
ev_m	motor volumetric efficiency	decimal
ev_p	pump volumetric efficiency	decimal
G_r	gear ratio of final drive	decimal
F_d	drawbar pull	lb_f
F_s	drawbar pull at wheel slip	lb_f
μ	coefficient of friction between tire surface and ground	decimal
n_d	number of drive wheels	
N_m	motor speed	rpm
N_p	pump speed	rpm
N_w	wheel speed	rpm
P_g	maximum grade	%
Q_m	required flow to motor	GPM
Q_p	pump flow	GPM
R	rolling resistance	lb_f/lb_f
r	loaded wheel radius	in
TE	tractive effort	lb_f
T_m	motor output torque	lb_f–in
T_s	torque at wheel slip	lb_f–in
T_w	wheel torque	lb_f–in
v	vehicle speed	mph
V_m	motor displacement	in^3/rev
V_p	pump displacement	in^3/rev
W_d	weight on drive wheels	lb_f
W_g	gross vehicle weight	lb_f

$$T_s = 2210(0.4)(28/2)$$

$$= 12380 \text{ lb}_f\text{-in}$$

The vehicle must develop a drawbar pull of 250 lb_f when traveling up an 8% slope. Rolling resistance is $R = 50$. Gross vehicle weight is $W_g = 8170$ lb_f. What is the required wheel torque at each of the two drive wheels.?

$$F_s = \frac{n_d T_w}{r} - W_g\left(\frac{R}{1000} + \frac{R_g}{100}\right)$$

or

$$
\begin{aligned}
T_w &= \frac{r}{n_d}\left[F_s + W_g\left(\frac{R}{1000} + \frac{P_g}{100}\right)\right] \\
&= \frac{14}{2}\left[250 + 8170\left(\frac{50}{1000} + \frac{8}{100}\right)\right] \\
&= 9180 \text{ lb}_f\text{-in}
\end{aligned}
$$

Since $T_s = 12{,}380 > T_w = 9180$, the machine will develop sufficient traction before wheel slip to climb the grade and develop 250 lb_f of drawbar pull.

At wheel slip ($T_w = T_s$), the maximum drawbar pull is

$$
\begin{aligned}
F_s &= \frac{2T_w}{r} - W_g\left(\frac{R}{1000} + \frac{P_g}{100}\right) \\
&= \frac{2(12380)}{14} - 8170\left(\frac{50}{1000} + \frac{8}{100}\right) \\
&= 710 \text{ lb}_f
\end{aligned}
$$

In like manner, it can be shown that if $\mu = 0.6$ the maximum drawbar pull at wheel slip is $F_s = 1590$ lb_f.

6.9.2.2 Maximum Power (Preliminary Calculation)

The tractive effort at wheel slip for a two-wheel-drive vehicle is

$$
\begin{aligned}
TE &= (T_s/r)n_d \\
&= (12380/14)(2) \\
&= 1770 \text{ lb}_f
\end{aligned}
$$

At a ground speed of 2.2 mph = 194 ft/min,

$$
\mathcal{P} = \frac{TE \times 194}{33000} = 10.4 \text{ hp}
$$

The final drives are 98% efficient, so the hydraulic motors must actually deliver

$$\frac{10.4}{0.98} = 10.6$$

The motors are 85% efficient, therefore the hydraulic power that must be delivered to the motors is

$$\frac{10.6}{0.85} = 12.5 \text{ hp}$$

The pump is 85% efficient, therefore the mechanical power that must be delivered to the pump is

$$\frac{12.5}{0.85} = 14.7 \text{ hp}$$

The pump mount is 98% efficient, therefore the engine must develop

$$\frac{14.7}{0.98} = 15 \text{ hp}$$

6.9.2.3 Required Engine Power (Preliminary)

Total engine power required is

$$\begin{aligned}\mathcal{P}_{tot} &= \mathcal{P}_{HST} + \mathcal{P}_{aux} \\ &= 15 + 34 \\ &= 49 \approx 50 \text{ hp}\end{aligned}$$

6.9.2.4 Wheel Motor (Initial Selection)

If the relief valve is set on 3000 psi, design pressure should not be more than 2500 psi at maximum tractive effort.

$$T_m = \Delta P V_m / 2\pi$$

At wheel slip, $T_s = T_w = 12{,}380$ lb_f-in. Solving for V_m,

$$\begin{aligned}V_m &= 2\pi T_m / \Delta P \\ &= 2\pi(12380)/2500 \\ &= 31.1 \text{ in}^3/\text{rev}\end{aligned}$$

Motors with displacements up to 113 in^3/rev are given in Appendix 6.2, so we can consider driving the wheel directly with a motor. The Vickers MHT-50 motor is rated for up to 200 rpm at 3000 psi. The displacement is 38 in^3/rev. Using this motor, the ΔP at wheel slip is

$$\begin{aligned}\Delta P &= 2\pi T_m/V_m \\ &= 2\pi(12380)/38 \\ &= 2050 \text{ psi}\end{aligned}$$

If the vehicle travels at 15 mph on the road, the wheel speed is

$$N_w = \frac{15(5280)/60}{2\pi(14/12)} = 180 \text{ rpm}$$

which is below the 200 rpm rating for this motor.

Required wheel speed for field operations at 2.2 mph is

$$N_w = \frac{2.2(5280)/60}{2\pi(14/12)} = 26.4 \approx 27 \text{ rpm}$$

Therefore, the flow is

$$\begin{aligned}Q_m &= \frac{N_w V_m}{231 e_{vm}} = \frac{27(38)}{231(0.85)} \\ &= 5.2 \text{ GPM}\end{aligned}$$

6.9.2.5 Pump Selection (Initial)

If pump volumetric efficiency is 85%, total flow for two wheel motors is

$$Q_p = \frac{2Q_m}{e_{vp}} = \frac{2(5.2)}{0.85} = 12.3 \text{ GPM}$$

If the pump speed is 1800 rpm, the required displacement is

$$\begin{aligned}V_p &= 231 Q_p/N_p \\ &= 231(12.3)/1800 \\ &= 1.58 \text{ in}^3/\text{rev}\end{aligned}$$

The closest selection from the catalog summary data is a Sauer-Danfoss Series 20 at 1.79 in^3/rev maximum displacement. This pump, with the swash-plate set for 1.58 in^3/rev, will provide adequate flow for field operations.

6.9.2.6 Maximum Power—Field Operations

Maximum hydraulic power is

$$\mathcal{P} = \frac{\Delta P Q}{1714} = \frac{2050(12.3)}{1714}$$

$$= 14.7 \text{ hp}$$

If the pump mount is 98% efficient, required engine power is

$$\frac{14.7}{0.98} = 15 \text{ hp}$$

which is quite close to the preliminary calculation.

6.9.2.7 Maximum Speed—Road Travel

Suppose the pump is set for maximum displacement, $V_p = 1.79$ in^3/rev, and it is driven at 2200 rpm. Maximum theoretical flow is

$$Q_p = \frac{1.79(2200)}{231} = 17 \text{ GPM}$$

The pressure requirement is less for road travel. Suppose the pump volumetric efficiency is 90% and the motor volumetric efficiency is 90%. Total flow to an individual wheel motor is

$$Q_m = (Q_p e_{vp})/2$$

$$= (17)(0.90)/2 = 7.67 \text{ GPM}$$

Maximum wheel speed is

$$N_w = 231 Q_m e_{vm}/V_m$$

$$= 231(7.67)(0.90)/38 = 42 \text{ rpm}$$

Maximum road speed is

$$V = 2\pi(14/12)N_w \; 60/5280$$
$$= 3.5 \text{ mph}$$

The vehicle has a maximum speed of 3.5 mph on the road. Suppose valving is installed such that only one motor is used to drive the machine. Total flow then goes to this motor.

$$Q_m = Q_p e_{vp} = 17(0.90)$$
$$= 15.3 \text{ GPM}$$
$$N_w = 231 Q_m e_{vm} / V_m$$
$$= 231(15.3)(0.90)/38$$
$$= 83.7 \text{ rpm}$$
$$V = 2\pi(14/12)83.7(60)/5280$$
$$= 7 \text{ mph}$$

The road speed is still less than half the desired 15 mph. There are several choices for a redesign.

1. Choose a larger pump
2. Choose smaller wheel motors
3. Choose larger wheels

We will first examine the potential for using a high-speed motor mounted in a planetary final drive.

6.9.2.8 Planetary Final Drive

The torque developed at wheel slip is 12,380 lb_f-in. In the catalog data, the Fairfield Model WIA torque hub is rated for a maximum intermittent torque output of 30,000 lb_f-in. (Torque hub is the name used by Fairfield for their planetary gear sets.) It is good design practice to design for half this rating, or 15,000 lb_f-in for continuous output. Since 12,380 < 15,000, the model WIA is acceptable.

It is helpful to construct a table of the required motor torque (T_m) to generate $T_w = T_s$ using the various gear ratios (G_r) of the Model WIA. Remember that the efficiency of the torque hub is 98%; therefore, the torque that must be delivered by the wheel motor is

$$T_m = (T_w/G_r)/0.98$$

Example:

$$T_m = (12380/18.25)/0.98$$

$$= 692 \text{ lb}_f\text{-in}$$

Required motor displacement to supply this torque at 2500 psi pressure drop (relief valve set on 3000 psi) is

$$V_m = 2\pi T/\Delta P$$

$$= 2\pi(692)/2500$$

$$= 1.74 \text{ in}^3/\text{rev}$$

Required motor displacement for each gear ratio is as shown in the following table.

Gear ratio (G_r)	**Motor torque (T_m), lb_f-in**	**Motor displacement (V_m), in^3/rev**
18.25	692	1.74
24.85	508	1.28
30.06	420	1.06
35.13	359	0.90
68.00	186	0.47

Required wheel speed for a 15-mph road speed is $N_w = 180$. Maximum required motor speed occurs for highest gear ratio.

$$N_m = N_w G_r = 180(68) = (12240) \text{ RPM}$$

Hydraulic motors are not rated this high. Referring to the catalog summary data, the Sauer-Danfoss motors are rated up to 4000 rpm. What motor speed is required for $G_r = 18.25$?

$$N_m = N_w G_r = 180(18.25) = 3285 \text{ rpm}$$

The Model WIA torque hub is rated for a maximum input speed of 4000 rpm, so it is possible to use one of the Sauer-Danfoss motors and the WIA torque hub.

The maximum flow required to achieve N_m = 3285 rpm is

$$Q_m = N_m V_m / e_{vm}$$

From the catalog summary data, the closest motor displacement greater than V_m = 1.74 in^3/rev is the Sauer-Danfoss Model 20 with a displacement of 2.03 in^3/rev.

$$Q_m = 3285(2.03)/0.9$$

$$= 7410 \text{ in}^3/\text{min}$$

$$\text{or } 32 \text{ GPM}$$

If valving is installed to deliver all the flow to one motor for road travel, the pump must develop a maximum flow of 32 GPM. Suppose the pump is driven at 2200 rpm for road travel. Required pump displacement is

$$V_p = 231 Q_m / N_p e_{vp}$$

$$= 231(32)/(2200 \times 0.9)$$

$$= 3.73 \text{ in}^3/\text{rev}$$

We must now make a choice between the Sauer-Danfoss Model 22 pumps. There are two available: 3.76 and 4.26 in^3/rev.

6.9.2.9 Redesign with Larger Diameter Wheels

Is it possible to choose a wheel size that will allow the use of a Series 20 Sauer-Danfoss pump, 2.03 in^3/rev displacement? When driven at 2200 rpm, this pump will deliver

$$Q_p = (V_p N_p / 231) e_{vp}$$

$$= [2.03(2200)/231]0.9$$

$$= 21.5 \text{ GPM}$$

This flow to a single Series 20 wheel motor will produce the following wheel rpm:

$$\begin{aligned} N_w &= 231 Q_m e_{vm} / V_m / G_r \\ &= 231(21.5)(0.9)/2.03/18.25 \\ &= 120 \text{ rpm} \end{aligned}$$

The required wheel diameter for this wheel rpm to produce a 15-mph road speed is

$$\begin{aligned} D_w &= 5280v(12)/(60\pi N_w) \\ &= 5280(15)(12)/[60\pi(120)] \\ &= 42 \text{ in} \end{aligned}$$

What is the consequence of choosing a 42-in. wheel?

Torque at wheel slip is

$$\begin{aligned} T_s &= W_d \mu r \\ &= 2210(0.4)(42/2) \\ &= 18,560 \text{ lb}_f\text{-in} \end{aligned}$$

Since the torque hub is rated for 30,000 lb_f-in intermittent, it is not necessary to choose a new torque hub.

Maximum drawbar pull at wheel slip ($T_w = T_s$) is unaffected by selection of larger wheel.

$$\begin{aligned} F_s &= \frac{2T_w}{r} - W_g\left(\frac{R}{1000} + \frac{P_g}{100}\right) \\ &= \frac{2(18,560)}{21} - 8170\left(\frac{50}{1000} + \frac{8}{100}\right) \\ &= 706 \text{ lb}_f \end{aligned}$$

The operating pressure is affected because of the higher torque that must be developed at the wheel.

$$\begin{aligned} T_m &= (T_w/G_r)/0.98 \\ &= (18560/18.25)/0.98 \\ &= 1040 \text{ lb}_f\text{-in} \end{aligned}$$

The pressure drop across the motor required to develop this torque is

$$\Delta P = 2\pi T_m / V_m$$

$$= 2\pi(1040)/2.03$$

$$= 3220 \text{ psi}$$

We can design for a relief valve setting of 3500 psi, but if pressure builds to 3220 psi, some loss will probably occur across the relief valve. Selection of 42-in. wheels may not be the best choice.

6.9.2.10 Compromise Solution

As a final iteration of the design process, let us determine whether a Sauer-Danfoss Series 21 pump and Series 18 motor combination will work.

Pump:	2.78 in^3/rev
Motors:	2.3 in^3/rev
Torque hub:	18.25:1

Maximum flow from the pump for road travel is

$$Q_p = N_p V_p e_{vp}/231$$

$$= 2200(2.78)(0.9)/231$$

$$= 23.8 \text{ GPM}$$

Maximum speed of the wheel motor if all flow is directed to one motor is

$$N_m = 231 Q_m e_{vm} / V_m$$

$$= 231(23.8)(0.9)/2.3$$

$$= 2150 \text{ rpm}$$

Maximum speed of the wheel is

$$N_w = N_m / G_r$$

$$= 2150/18.25$$

$$= 118 \text{ rpm}$$

Required wheel diameter is

$$
\begin{aligned}
D_w &= 5280(15)(12)/[60\pi(118)] \\
&= 42.7 \text{ in}
\end{aligned}
$$

The choice of a 42-in. wheel is close enough.

Required torque to spin the wheels is 1040 lb_f-in. Pressure drop across the motor to develop this torque is

$$
\begin{aligned}
\Delta P &= 2\pi T_m / V_m \\
&= 2\pi(1040)/2.3 \\
&= 2840 \text{ psi}
\end{aligned}
$$

This pressure can be developed if the relief valve setting is 3500 psi.

6.9.2.11 Requirements

1. Select hydrostatic transmission

Pump:	Sauer-Danfoss Series 21
	2.78 in^3/rev
Wheel motors:	Sauer-Danfoss Series 18
	2.3 in^3/rev
Planetary final drive:	Fairfield Model WIA
	18:25:1 Gear Ratio

   a. Maximum pump flow: 24 GPM
   b. System operating pressure at wheel slip: 2840 psi
      Relief valve setting: 3500 psi
   c. Show HST can spin wheels
      ΔP = 2850 psi will generate enough torque to spin wheels

2. Select an engine for the harvester.

   As the following analysis shows, total power required for field operations is approximately 49 hp. Assume that the wheels are just beginning to slip.

$$F_s = \frac{2T_w}{r} - W_g\left(\frac{R}{1000} + \frac{P_g}{100}\right)$$

$$= \frac{2(18,560)}{21} - 8170\left(\frac{50}{1000} + \frac{8}{100}\right)$$

$$= 706 \text{ lb}_\text{f}$$

Required ΔP = 2850 psi.

Required flow:

$$N_w = 5280v(12)/[60\pi D_w]$$

$$= 5280(2.2)(12)/[60\pi(42)]$$

$$= 17.6 \text{ rpm}$$

$$N_m = N_w G_r = 17.6(18.25)$$

$$= 321 \text{ rpm}$$

$$Q_m = (N_m V_m/e_{vm})/231$$

$$= 3.76 \text{ GPM}$$

Since there are two wheel motors, pump flow is obtained by

$$Q_p = 2Q_m/e_{vp}$$

$$= 2(3.76)/0.85$$

$$= 8.85 \text{ GPM}$$

The total hydraulic power the pump must develop for field operations when wheels are just beginning to slip is

$$\mathcal{P} = \Delta PQ/1714$$

$$= 2850(8.85)/1714$$

$$= 14.7 \text{ hp}$$

The pump mount is 98% efficient; therefore, the engine must develop 14.7/0.98 = 15 hp. An engine that will develop 15 + 34 =

49 hp at 1800 rpm will meet the total calculated power requirement for the harvester.

It is good design practice to select an engine that will develop maximum calculated hp at 75% of the engine's rated maximum power. We choose an engine rated for

$$\text{hp} = 49/0.75 = 65$$

3. What is the maximum drawbar pull with $\mu = 0.4$?

   Assume that the wheels are just beginning to slip, $T_w = T_s$.

$$\begin{aligned} F_s &= \frac{2T_w}{r} - W_g\left(\frac{R}{1000} + \frac{P_g}{100}\right) \\ &= \frac{2(18,560)}{21} - 8170\left(\frac{50}{1000} + \frac{8}{100}\right) \\ &= 706 \text{ lb}_f \end{aligned}$$

On level ground, $P_g = 0$, the maximum drawbar pull is $F_s = 1360$ lb_f.

What is the maximum drawbar pull with $\mu = 0.6$?

$$\begin{aligned} T_s &= W_d \mu r = 2210(0.6)(21) \\ &= 27,860 \text{ lb}_f\text{-in} \\ F_s &= \frac{2(27,850)}{21} - 8170\left(\frac{50}{1000} + \frac{8}{100}\right) \\ &= 1590 \text{ lb}_f \end{aligned}$$

What drawbar pull can be maintained at a 2.2-mph field speed?

As shown for Requirement #2, an engine that can develop 50 hp at 1800 rpm has sufficient power to operate the HST and the other hydraulic circuits. Full power is available to produce wheel slip. At wheel slip, $F_s = 706$ lb_f.

4. Show that the harvester can load itself onto the tilt-bed trailer. Coefficient of friction $\mu = 0.6$.

   Neglect weight transfer. We cannot calculate weight transfer, as we are not given the geometry and center of gravity of the machine.

$$
\begin{aligned}
T_s &= W_d \mu r \\
&= 2210(0.6)(21) \\
&= 27,846 \text{ lb}_f\text{-in}
\end{aligned}
$$

Assume that the rolling resistance is R = 15, equivalent to good macadam (Table 6.1). Slope of trailer is 27%. Drawbar pull is F_s = 0; therefore, the torque required is

$$
\begin{aligned}
T_w &= \frac{r}{2} W_g \left(\frac{R}{1000} + \frac{P_g}{100} \right) \\
&= \frac{21}{2}(8170)\left(\frac{15}{1000} + \frac{27}{100} \right) \\
&= 24488 \text{ lb}_f\text{-in}
\end{aligned}
$$

Since $T_s > T_w$, the harvester will have enough traction to load itself, assuming no weight transfer. Can the HST develop T_w = 24,448 lb_f-in?

Assume that the maximum ΔP across the motor with a 3500 psi relief valve setting is 3500 – 200 = 3300 psi.

Maximum motor torque is

$$
\begin{aligned}
T_m &= \Delta P V_m / 2\pi \\
&= 330(2.3)/2\pi \\
&= 1208 \text{ lb}_f\text{-in}
\end{aligned}
$$

Maximum wheel torque is

$$
\begin{aligned}
T_w &= T_m(G_r) \\
&= 1208(18.25) = 22,045 \text{ lb}_f\text{-in}
\end{aligned}
$$

The machine will probably need some assistance to load itself. You could choose smaller wheels and reduce the ΔP requirement. The machine then will not meet the maximum road speed requirement. It probably will be satisfactory to use a light winch to assist in loading the machine.

6.10 Summary

A hydrostatic transmission is simply a pump and motor connected in a circuit. Other components are included in the circuit design to ensure that the functional objective is achieved. The pump and motor can either be included in the same housing or separate components connected with hoses or tubing.

Typically, mechanical transmissions have efficiencies of 95% or greater, whereas hydrostatic transmissions have an efficiency of around 80%. Some well designed units have an efficiency of 85% over a certain operating range.

Hydrostatic transmissions are used to increase vehicle maneuverability. They also provide continuous speed control from a slow creep up to maximum speed. Before a hydrostatic transmission is chosen over a mechanical transmission, a study is done to ensure that the advantages yield an increase in vehicle productivity (tons handled per operating hour, etc.) to offset the lower efficiency.

If the pump receives fluid from the reservoir, passes it through the motor, and returns it to the reservoir, the design is identified as an *open-circuit* transmission. Transmissions that circulate fluid in a continuous loop between the pump and motor are identified as *closed-circuit* transmissions.

As the load increases, pressure increases, and leakage from the pump and motor increases. Motor output speed decreases due to this leakage. If constant speed is important, a transducer is used to sense motor speed, and this signal is used to increase (or decrease) pump displacement the amount required to maintain motor speed. Sensing motor speed and using this information to control pump displacement is known as *feedback*. Feedback can be used for both open- and closed-circuit transmissions. When feedback is included, the transmission is identified as a *closed-loop* transmission.

A variable displacement axial piston pump can have a control piston mounted in the pump housing. This control piston is used to control the position of the swashplate and thus the displacement of the pump. A servo valve controls the flow, which extends (or retracts) the control piston. The servo valve shifts when current is delivered to a torque motor mounted on the valve. A typical closed-loop transmission operates as follows. The voltage signal from the transducer that senses motor speed is fed to a servo amplifier to obtain a current. This current causes the torque motor to rotate, which opens the servo valve and ultimately increases (or decreases) the pump displacement. Using a feedback design of this type, transmission output speed can be held constant as load varies.

For some simple applications, it is possible to select a pump from one manufacturer and motor (or motors) from another manufacturer, connect them together with hoses or tubing, and achieve acceptable performance. In general, however, a hydrostatic transmission is purchased as a package with needed valving already installed in the pump and motor. Considerable engineering design goes into the circuitry built into the unit. When the transmission is used for a vehicle, the vehicle designer and transmission designer work as a team to ensure that the desired performance is achieved.

APPENDIX 6.1

Basic Concepts in Traction

The ability of a vehicle to develop traction is a function of the weight on the drive wheels and the coefficient of friction between the wheel and surface. Torque at wheel slip is given by

$$T_s = W_d \mu r \tag{A6.1}$$

where T_s = torque at wheel slip (lb_f-in)
W_d = weight on drive wheel (lb_f)
μ = coefficient of friction (decimal)
r = rolling radius (in)

The rolling radius is the distance from the center of the axle to the ground. Because the weight of the vehicle causes the tire to deform, the rolling radius is less than the tire radius.

Required wheel torque is the torque that must be supplied to the wheel to move the vehicle.

$$T_w = Fr \tag{A6.1}$$

where T_w = required torque (lb_f-in)
F = total force to move vehicle (lb_f)
r = rolling radius (in)

The total force to move the vehicle is given by

$$F = W_g \frac{R}{1000} + W_g \frac{P_g}{100} + F_d$$

where F = total force (lb_f)
W_g = gross vehicle weight (GVW) (lb_f)
R = rolling resistance (lb_f/1000 lb_f GVW)
P_g = grade (%)
F_d = drawbar pull (lb_f)

Gross vehicle weight is the total weight of the machine, not just the weight on the drive wheels. Rolling resistance is a function of the surface. Typical values are shown in the following table.

Surface	**R**
Concrete	15
Packed soil	25
Sandy soil	37
Mud	37–150

Maximum grade the vehicle must climb is given in percent, not degree.

Drawbar pull is the force the vehicle must develop above the force just to move the vehicle. In the case of a load being towed by the vehicle, drawbar pull is simply the force at the hitch point to move the load forward. Drawbar pull can also be the force required to push a load. An example would be a front-end loader pushing forward to fill the bucket with dirt.

A simple example will reinforce the traction concept. A self-propelled windrower weighs 7785 lb_f. It moves over a sod surface with rolling resistance, R = 25. The maximum slope the vehicle must climb is 12%, and the drawbar pull is $F_d = 500\ lb_f$. Total force to move the vehicle is

$$F = 7785\left(\frac{25}{1000} + \frac{12}{100}\right) + 500 = 1630\ \text{lb}_\text{f}$$

Required wheel torque at each driving wheel is

$$T_w = \frac{Fr}{n_w}$$

where n_w = number of driving wheels. Both front wheels drive and the rolling radius is 23 in. The required torque at each front wheel is

$$T_w = \frac{1630(23)}{2}$$

$$= 18,745\ \text{lb}_\text{f}\text{-in}$$

If the coefficient of friction on a sod surface is μ = 0.4, will the torque at wheel slip be greater than the required torque? Weight on the two drive wheels is 70% of the total vehicle weight.

$$W_d = 0.7\frac{(7785)}{2} = 2725\ \text{lb}_\text{f}$$

$$T_s = W_d \mu r$$

$$= 2725(0.4)(23)$$

$$= 25,070 \text{ lb}_f\text{-in}$$

Since $T_s > T_w$, the vehicle can meet the functional objective.

Peak torque will be developed when the vehicle is operating on a surface with a high coefficient of friction. Torque at wheel slip is a maximum for this condition. The hydrostatic transmission and other driveline components should be designed to deliver this maximum torque.

More detail on traction is available from an ASAE standard, ASAE S296.3,* and other sources. Before designing a hydrostatic transmission for an off-road vehicle, traction should be investigated in more detail.

* ASAE 2000. ASAE Standards 2000. Am. Soc. Agric. Eng., 2950 Niles Rd., St. Joseph, MI 49085-9659.

APPENDIX 6.2

Selected Catalog Data for Hydrostatic Transmission Design Problems

Available Variable Displacement Pumps

Sauer-Danfoss model no.	Displacement (in^3/rev)	Rated rpm
Series 15	0.913	up to 4000
Series 18	2.3	"
Series 20	1.79, 2.03	"
Series 21	2.78, 3.15	"
Series 22	3.76, 4.26	"
Series 23	4.79, 5.43	"
Series 24	6.39, 7.24	"

Available Fixed Displacement Motors

Mfg. model no.	Displacement (in^3/rev)	Rated rpm
Sauer-Danfoss 15	0.913	up to 4000
Sauer-Danfoss 18	2.3	"
Sauer-Danfoss 20	2.03	"
Sauer-Danfoss 21	3.15	"
Sauer-Danfoss 22	4.26	"
Sauer-Danfoss 23	5.43	"
Vickers M2-200-25	1.62	2700
Vickers M2-200-35	2.3	"
Vickers 25M42	2.68	"
Vickers 25M55	3.52	"
Vickers 25M65	4.19	"
Vickers 35M80	5.10	"
Vickers 35M95	6.12	"
Vickers 35M115	7.44	"
Vickers45M130	8.42	"
Vickers 45M155	9.96	"
Vickers 45M185	11.79	"
Vickers MFB5	0.643	2400
Vickers MFB10	1.29	"
Vickers MFB20	2.61	"
Vickers MFB29	3.76	"
Vickers MFB4S	5.76	"

Available Fixed Displacement, High-Torque, Low-Speed Motors

Mfg. model no.	Displacement (in^3/rev)	Rated peak rpm
Charlynn "H" Series	3.0	2250
	4.5	2100
	6.2	1950
	10.3	1800
	11.9	1650
	14.9	1500
	17.9	1350
	23.8	1200
Charlynn "S" Series	4.2	2400
	5.8	2400
	9.7	2200
	11.2	2000
	14.0	1800
	16.9	1600
	22.5	1300
Vickers MHT-32	24.0	275 @ 3000 psi continuous
		400 @ 2000 psi continuous
Vickers MHT-50	38.0	200 @ 3000 psi
		350 @ 2000 psi
Vickers MHT-70	52.8	150 @ 3000 psi continuous
		300 @ 2000 psi continuous
Vickers MHT-90	67.9	150 @ 3000 psi continuous
		300 @ 2000 psi continuous
Vickers MHT-130	98.0	150 @ 3000 psi
Vickers MHT-150	113.0	250 @ 2000 psi

Selected Torque Hub Specifications

Fairfield Model W1A

Propel applications		Nonpropel applications	
Max. intermittent torque output	30,000 in-lb_f	Max. torque output	15,000 in-lb_f
Max. input speed	4000 rpm		
Reduction ratios	18.75:1 24.43:1 30.04:1 34.49:1 42.50:1 53.58:1		
Output speed	$\dfrac{\text{Input speed}}{\text{Reduction ratio}}$		
Application example	Self-propelled over-the-row sprayer		

Fairfield Model W3B

Propel applications		Nonpropel applications	
Max. intermittent torque output	75,000 in-lb$_f$	Max. torque output	37,500 in-lb$_f$
Max. input speed	4000 rpm		
Reduction ratios	18.75:1 24.43:1 30.04:1 34.49:1 42.50:1 53.58:1		
Application example	Wheel assist drive on steering axle of combine		

Fairfield Model W7C

Propel applications		Nonpropel applications	
Max. intermittent torque output	150,000 in-lb$_f$	Max. torque output	75,000 in-lb$_f$
Max. input speed	3500 rpm		
Reduction ratios	26.4:1 44.2:1 57.8:1 71.5:1 93.7:1		

Fairfield Model W10D

Propel applications		Nonpropel applications	
Max. intermittent torque output	250,000 in-lb$_f$	Max. torque output	125,000 in-lb$_f$
Max. input speed	3500 rpm		
Reduction ratios	43.8:1 57.4:1 69.7:1 80.3:1 98.0:1 123.2:1		

Fairfield Model W20D

Propel applications		Nonpropel applications	
Max. intermittent torque output	500,000 in-lb$_f$	Max. torque output	250,000 in-lb$_f$
Application example	Bucyrus-Erie crane track drive		

Auburn Gear Model 6

Propel applications		Nonpropel applications	
Max. intermittent torque output	50,000 in-lb$_f$	Max. continuous torque output	1/3 to 1/2 max. intermittent
Max. input speed	5000 rpm		
Reduction ratios	28.37:1 24.53:1 21.74:1 19.62:1 15.88:1		
Application example	Rotary combine		

Auburn Gear Model 8

Propel applications		Nonpropel applications	
Max. intermittent torque output	1800,000 in-lb$_f$	Max. continuous torque output	1/3 to 1/2 max. intermittent
Max. input speed	5000 rpm		
Reduction ratios	36.96:1 30.89:1 22.04:1		
Application example	Wheel tractor-scraper		

Auburn Gear Model 10

Propel applications		Nonpropel applications	
Max. intermittent torque output	180,000 in-lb$_f$	Max. continuous torque output	1/3 to 1/2 max. intermittent
Max. input speed	5000 rpm		
Reduction ratios	36.8:1		
Application example	Mobile traveling beam hoist		

Problems

6.1 Design a hydrostatic transmission for a forage plot harvester.

A small machine is being built to harvest forage plots for the project, "Perennial Species for Optimum Production of Herbaceous Biomass in the Piedmont." This machine cuts a 1.5 m (5 ft) wide swath of forage with a cutterbar, elevates it with a wide rubber belt, and dumps it into a holding bin on the back. After weighing, the forage is dumped into a pile on the edge of the field.

Because the plots are small, maneuverability is very important. A hydrostatic drive is needed to allow the operator to quickly change from forward to reverse.

Power Required for Hydraulic Systems other than HST

The cutterbar, reel, and conveyor are all powered with hydraulic motors. Hydraulic cylinders are used for power steering, header lift, and bin dumping. Without going through all the details, it is sufficient to state that plans are to supply flow for all these requirements using a tandem gear pump. The front pump has a displacement of 0.5 in^3/rev and supplies flow for the cutterbar motor only. The cutterbar is assumed to operate continuously when harvesting. If the tandem pump is driven at 2500 rpm, maximum flow delivered by the front pump is

$$Q_f = \frac{V_{pf}N}{231} = \frac{0.5(2500)}{2310} = 5.4 \text{ GPM}$$

The relief valve is set at 1500 psi. Maximum power required is,

$$\mathcal{P}_f = \frac{Q_f \Delta P}{1714} = \frac{5.4(1500)}{1714} = 4.75 \text{ hp}$$

The rear pump has a displacement of 0.75 in^3/rev, and is, of course, driven at 2500 rpm. It supplies the remaining circuits, which are protected by a relief valve set at 2000 psi. Maximum flow delivered by the rear pump is

$$Q_r = \frac{V_{pr}N}{231} = \frac{0.75(2500)}{231} = 8.1 \text{ GPM}$$

and maximum power is

$$\mathcal{P}_r = \frac{Q_r \Delta P}{1714} = \frac{8.1(2000)}{1714} = 9.5 \text{ hp}$$

It is unlikely that the cutterbar circuit and the other circuits will all require their maximum pressure simultaneously; consequently, an addition of the two power requirements is conservative. At this point in the design procedure, the conservative choice is made.

$$\mathcal{P}_{aux} = \mathcal{P}_f + \mathcal{P}_r = 5.4 + 9.5 = 14.9 \approx 15 \text{ hp}$$

Suppose the tandem pump is driven from a pump mount on the engine. The efficiency of the input mechanical drive will be quite high. Assume $e_{mi} = 0.98$. Engine power for the auxiliary circuits is

$$\mathcal{P}_{e(aux)} = \mathcal{P}_{aux}/e_{mi} = 15.3 \text{ hp}$$

The assumed weight distribution for the forage plot harvester is shown in Fig. 6.30. Only the rear wheels drive; consequently, the weight on the drive wheels is 1760 lb_f (798 kg). The bin is 3 ft × 4 ft × 3 ft, and the harvested material weighs 25 lb/ft^3.

Drawbar Pull Required

The forage plot harvester will not be used to pull a load. It will, however, have to push the cutterbar into some rather dense herbaceous crops. The resistance is estimated to be 250 lb_f (1100 *N*) (50 lb per ft cutterbar width), and this is equivalent to a drawbar pull requirement.

Traction Parameters

A grass field is relatively slick; therefore, assume $\mu = 0.4$. The field surface is judged to be equivalent to smooth dirt; therefore, use $R = 25$ (Table 6.1).

Speed Requirement

Maximum design field speed is 4.2 mph (7 km/h). The machine will be hauled from one location to the next on a trailer; consequently, there is no requirement for a road travel speed.

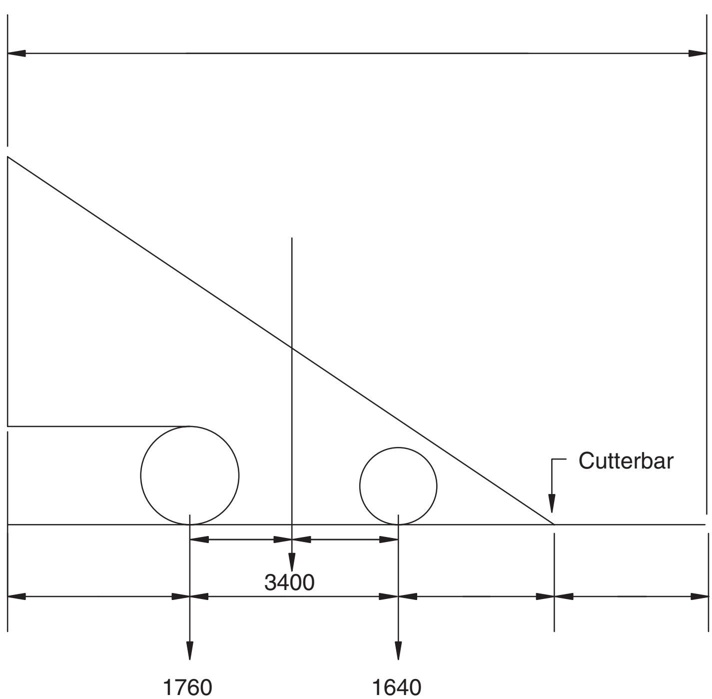


FIGURE 6.30
Estimated weight distribution of forage plot harvester (Problem 6.1).

Wheel Size

The diameter of the rear wheels has tentatively been chosen at 20 in. (508 mm), but this may be changed if needed to improve your overall design. However, do not choose a wheel diameter greater than 32 in. (813 mm).

Maximum Pressure

Because of the cost of high-pressure components, do not choose a relief valve pressure higher than 3500 psi.

Requirements

1. Design the hydrostatic drive so the harvester can cut up a 10% slope at 4.2 mph (7 km/h).
2. Find the maximum slope that the machine can climb while harvesting. If all the power of the engine you select is used, what speed can be maintained while harvesting this slope?
3. Ensure that the harvester has enough power to pull itself onto a tilt-bed trailer at a 27% incline.
4. Show all your work. Lead the reader through your thought process in completing the design.
5. List the components you select for your final design.

6.2 Design a hydrostatic transmission for a compost windrow turner.

The requirement is to design a hydrostatic drive for a machine to aerate windrows of compost. The front wheels are caster wheels. The rear wheels power and steer the machine (Fig. 6.31).

Each rear wheel is powered by a split hydrostatic transmission (variable displacement pump and fixed displacement motor). The configuration is similar to that shown in Fig. 6.15. The operator has a lever to control the swashplate on each pump. To turn the machine, the operator moves one lever forward and one rearward to increase forward speed of one wheel and reduce forward speed of the other. For tight turns, the operator can put one rear wheel in forward and one in reverse to spin the machine around in a tight arc.

The total weight of the machine is 6800 lb_f with 2700 lb_f on the front wheels and 4100 lb_f on the rear wheels. Total clearance to straddle a windrow is 6.5 ft. The machine travels at 4 mph when turning the windrow.

To bound the problem, a series of constraints are given.

Constraints

1. The engine has a double pump mount. One mount is used for the two HST pumps, and one mount has a tandem pump,

FIGURE 6.31
Self-propelled machine for turning compost windrows (Problem 6.2)

which provides hydraulic power for the other functions on the machine. The front pump of the tandem provides power to the agitator, which aerates and compost, and the rear pump provides power for all cylinders on the machine.

2. The pump mount is a direct 1:1 drive, meaning that the pump turns at the same speed as the engine.
3. Maximum torque is delivered by the diesel engine at 1800 rpm, and maximum recommended operating speed is 2200 rpm.
4. The rear wheels are 46 in. diameter, and the front wheels are 30 in. diameter.
5. For this design problem, do not try to look up the volumetric and overall efficiencies of the pump and motor. Assume all these values to be 0.85.

$$e_{vp} = e_{vm} = e_{op} = e_{om} = 0.85$$

(For a final design in the "real world," you do need to use actual efficiencies.

6. Assume that the pressure drop in the line from the pump to the motor is 40 psi, and the pressure drop in the line from the motor back to the pump is 40 psi at the flow corresponding to operating speed.
7. Do not design for a relief valve setting greater than 4000 psi.

8. Assume that the efficiency of the pump mount is 98%.
9. Assume that the efficiency of the planetary drive for the rear wheel is 98%.

Design Parameters

The machine will operate on a hardened surface (compacted gravel), but it will be fairly rough. It will not be driven over the road between jobs, so no road speed is needed.

The rotating drum that turns the compost tends to pull the material into the machine. The HST does not have to "push" the roller under the material as a front-end loader has to push the bucket into a pile of gravel. There is nothing pulled behind the machine, so no pull force is required.

1. Design for rolling resistance $R = 50$, and coefficient of friction $\mu = 0.4$.
2. Maximum grade is 5%.
3. Maximum forward speed is 7 mph. Speed when turning a windrow is 4 mph.
4. The engine must supply a maximum of 27 hp for functions other than the HST. The rotating drum requires most of this power. A small amount is required for the cylinders to raise and lower the drum.
5. No pull or push force must be developed by this machine.

Requirements

1. Select a hydrostatic drive (variable displacement pump, fixed displacement motor, planetary gear final drive) for each rear wheel. Specify the following:
   a. System design pressure
   b. Maximum pump flow
   c. Pump displacement, working pressure, rated rpm
   d. Motor displacement, working pressure, rated rpm
   e. Planetary gear final drive ratio, intermittent torque output rating, continuous torque output rating
2. Specify the total engine power required for the HST and the auxiliary functions. (Remember to increase the calculated engine power for the HST by 10% to provide power for the charge pump.)
3. Calculate the overall efficiency of the HST.

$$\text{Overall Efficiency} = \frac{\text{Power delivered to ground}}{\text{Power from engine}} \times 100$$

4. Show that the functional requirements are met.
   a. Will the HST spin the wheels before the relief valve opens?
   b. Does the engine have enough power to supply 27 hp for the auxiliary functions while traveling at 4 mph up a 5% slope over a surface with rolling resistance $R = 50$?

6.3 Design a hydrostatic transmission for a single-wheel-drive container lift.

A containerized handling system is being considered for the peanut industry. Containers will be loaded in the field, taken to a drying shed for curing, and then taken to market. If the grade indicates that the peanuts are of superior quality, a cover will be installed on the container, and it will be stored outside until shipment to a shelling plant. Production-run peanuts will be dumped and the container returned to the field to be reloaded.

The current commercial version is a container 8 ft wide × 24 ft long × 5.5 ft deep. It is mounted on a hook-lift skid. A tandem-axle truck lifts the container onto the bed for transport. The hook-lift design also provides a dump provision.

A machine is needed to move the containers at the buying point. The storage yard has a hardened gravel surface. The machine must lift the container (no more than 15 in. off the ground) and move it at a maximum speed of 5 mph.

The concept for the container mover is shown in Fig. 6.32. It will drive over the container, and lift arms will extend and lock into slots in the frame under the container. In the back, a third lift arm will engage the A-frame that is part of the hook-lift system. When

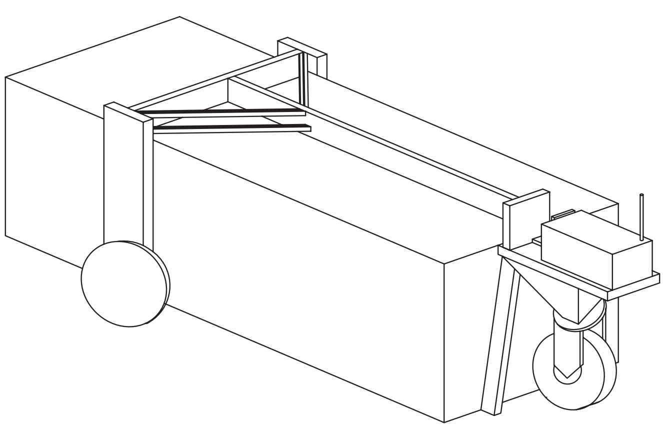


FIGURE 6.32
Container lift for moving 24-ft containers of peanuts.

lifted, the container itself will provide the needed structural strength. The frame for the mover must only provide the structural strength needed when the mover is maneuvering empty.

The total weight of peanuts in the container is 9.2 tons. The empty weight of the container is 2 tons. It is estimated that the mover itself will weigh 4500 lb_f with 1500 lb_f on the two front wheels and 3000 lb_f on the single rear wheel.

Estimated weight distribution for the loaded container is shown in Fig. 6.33. It is assumed that the weight distribution will be uniform along the length of the container. Reaction forces R_f and R_r are the loads on the front and rear wheels, respectively.

Constraints

1. The engine will have a dual pump mount with the HST pump mounted on one side and pump supplying flow to the cylinders (extend, lift, and steering) on the other side. Maximum power required for the cylinder circuits is 12 hp.
2. Assume that the pump mount is a direct drive unit, meaning that the pump turns at the same rpm as the engine. Plan to operate the engine at 1800 rpm when maneuvering around the buying station and storage yard. The engine can be operated at 2200 rpm when the mover is traveling empty.
3. Assume that the mover has 42 in. diameter high-flotation tires on the front. You may choose any tire diameter in the range of 28 to 48 in. for the rear wheel.
4. Choose $e_{vp} = 0.85$ and $e_{vm} = 0.85$ at design pressure for the pump and motor volumetric efficiencies, respectively.
5. Assume that hydraulic line losses between the pump and motor are 2% of the maximum hydraulic power transmitted.
6. Do not design for a relief valve setting greater than 3500 psi.
7. You will probably use a planetary gear set for the final drive. Assume it to be 98% efficient. Also, assume that the pump mount is 98% efficient.

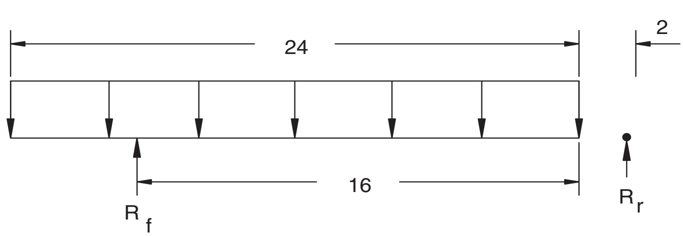


FIGURE 6.33
Estimated weight distribution for the loaded container (all dimensions in ft).

Design Requirements

1. The surface of the storage yard will be fairly rough. Design for rolling resistance $R = 50$. Assume a coefficient of friction, $\mu = 0.4$.
2. Maximum grade is 5%.
3. Maximum speed when loaded is 5 mph.

Requirements

1. Select a hydrostatic drive (variable displacement pump, fixed displacement motor, planetary gear final drive) for the mover. Specify the following:
   a. Maximum pump flow
   b. System design pressure
   c. Pump displacement, working pressure, rated rpm
   d. Motor displacement, working pressure, rated rpm
   e. Planetary gear final drive ratio, continuous torque output rating
2. Specify the engine power required to meet the requirement for the HST and cylinder circuits. What is the overall efficiency of HST?

$$\text{Overall efficiency} = \frac{\text{Power delivered to ground}}{\text{Power from engine}} \times 100$$

3. Show that the design will meet all functional requirements. Specific questions are:
   a. Will the HST spin the wheels before the relief valve opens?
   b. What is the maximum travel speed when the mover is traveling empty?

6.4 Design a hydrostatic transmission for a sweet onion harvester.

A prototype machine is being designed to mechanically harvest sweet onions. The onions are undercut with a rotating bar and left sitting in loosened soil. The harvesting machine passes over the bed and captures the tops between converging belts. As the onions are lifted from the soil, the bulbs are suspended beneath the belts. At the back of the machine, a set of disks cut the tap root from the bulb, and, a short Δt interval later, the tops are cut away to allow the bulbs to fall onto a conveyor for ultimate deposit in a pallet box.

The engine on the machine has a pump mount that supplies power to a variable displacement pump for the hydrostatic transmission (HST) and to other pumps that supply hydraulic power for the ancillary operations. The ancillary operations are the gathering belts, root saw, positioning cylinders for the belts, top saw, and the bulb conveyor. The main operator drives the machine and positions the gathering belts. A second operator monitors the operation of all other functions.

Estimated final weight of the machine with filled pallet boxes on the back platform is 9800 lb_f. It is estimated that 60% of this weight will be on the rear wheels and 40% on the front wheels.

Power required for the ancillary operations is estimated to be 18 hp. This design problem does not involve any aspect of the ancillary circuits. For example, you do not have to size a reservoir for the HST and ancillary circuits.

Constraints

1. You must specify the size diesel engine required. Assume that maximum torque is delivered at 1800 rpm, and maximum speed (for road travel) is 2200 rpm.
2. As an initial try, assume the pump mount to be a 1:1.5 drive, meaning that the pump is driven at 2250 rpm when the engine is operated at 1500 rpm. Other pump mount ratios can be tried if your initial assumption does not work.
3. One variable displacement pump will supply flow to two wheel motors. These motors can be fixed displacement or variable displacement (HST can be VP-FM or VP-VM) and can be mounted on the front wheels or rear wheels. Select the pump and motors from the range of displacements provided by your instructor.
4. Choose the drive wheel diameter from the range 32 to 48 in.
5. Do not choose a relief valve setting greater than 3500 psi. Cost of components (hoses, fittings, etc.) increases significantly above 3500 psi.
6. Assume the following efficiencies:

$$e_{vp} = 0.95; e_{op} = 0.85$$

$$e_{vm} = 0.95; e_{om} = 0.85; e_{tm} = 0.90$$

$$e_{pm} = 0.98 (\text{pump mount})$$

$$e_{fd} = 0.98 (\text{final drive})$$

7. It is appropriate to assume a 100 psi back pressure at the motor outlet and a 100 psi ΔP in the line between the pump and the motor.

Design for the following field conditions:

1. Onions are grown in sandy soils. If the soil is saturated, no harvesting will be attempted. Design for a maximum rolling resistance $R = 37$.
2. Maximum grade in the field will be 3%.
3. The machine is not required to develop a drawbar pull. It must be able to move itself when fully loaded.
4. Coefficient of friction under field conditions is $\mu = 0.4$. Maximum coefficient of friction is $\mu_{max} = 0.6$.
5. The desired harvesting speed is 3 mph.

Design for the following transport conditions:

1. The machine will be driven from field to field. Desired road speed is 15 mph.
2. When the machine is shipped, it will be winched onto the transport vehicle. No requirement is specified for driving onto the transport vehicle.

Requirements

Design the HST for the onion harvester. Show that all constraints and functional requirements are met. Present a summary of your design containing the following information:

1. Drive wheel diameter
2. Engine rated power
3. Hydrostatic transmission:
    a. Pump:
    displacement
    maximum rated speed
    maximum operating speed
    maximum flow
    maximum pressure (continuous operation)
    b. Motor:
    displacement
    maximum rated speed
    maximum operating speed for continuous operation
    c. Relief valve setting
4. Final drive:
    maximum intermittent torque rating
    gear ratio
5. Vehicle performance characteristics:
    a. maximum road speed
    b. torque at wheel slip
    c. Will the vehicle spin its wheels before the relief valve opens?

7

Linear Actuators

7.1 Introduction

Linear actuators are ubiquitous in modern manufacturing plants. Examples are a hydraulic cylinder to tilt a ladle of molten metal, a pneumatic cylinder to install a rivet, and a set of cylinders to close a box of frozen chicken. Mobile industrial and agricultural machines also require cylinders to lift, dig, dump, and otherwise position loads. Cranes, backhoe loaders, and combines are a few examples.

As shown in Chapter 1, the concept is very simple; fluid is pumped into the cylinder, and it pushes against the piston, causing it to extend. The velocity of extension is a function of the flow, and the force developed is a function of the pressure. As with most fluid power circuits, a linear actuator circuit has a few "quirks" that can trip up an engineer. It is necessary to think through several simple circuits to have the understanding of the nonintuitive features of linear actuators.

7.2 Analysis of Cylinders in Parallel and Series

Sometimes there is a need for two cylinders in different locations to extend at the same time. If these cylinders are connected in parallel (Fig. 7.1), the cylinder having the lowest pressure requirements will extend first. The two cylinders shown in Fig. 7.1 are analyzed below.

Cylinder 1

No-load extension of Cylinder 1 requires 80 psi pressure. The back pressure is 20 psi. Using a force balance to calculate the friction force,

$$P_{c1}A_{c1} = F_{f1} + P_{r1}A_{r1}$$

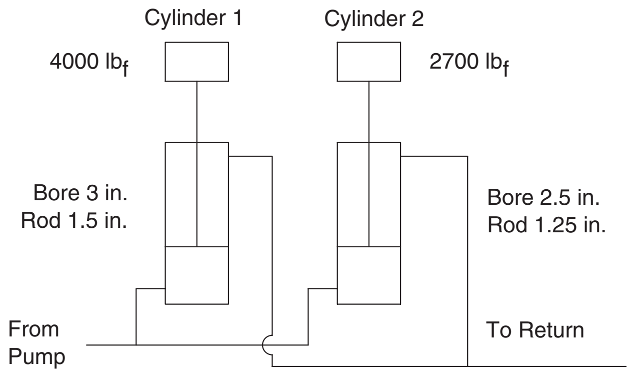


FIGURE 7.1
Cylinders in parallel.

$$A_{c1} = \frac{3^2\pi}{4} = 7.07 \text{ in}^2$$

$$A_{r1} = \frac{3^2\pi}{4} - \frac{1.5^2\pi}{4} = 5.3 \text{ in}^2$$

$$F_{f1} = 80(7.07) - 20(5.3)$$

$$= 459.6 \text{ lb}_f$$

Total pressure at the cap end to extend the load is

$$P_{c1}A_{c1} = F_{f1} + F_{L1} - P_{r1}A_{r1}$$

$$P_{c1} = \frac{459.6 + 4000 + 20(5.3)}{7.07}$$

$$= 646 \text{ psi}$$

Cylinder 2

No-load extension of Cylinder 2 requires 65 psi, and the back pressure is 15 psi. Using a force balance to calculate the friction force,

$$P_{c2}A_{c2} = F_{f2} + P_{r2}A_{r2}$$

$$A_{r2} = \frac{2.5^2\pi}{4} - \frac{1.25^2\pi}{4} = 3.68 \text{ in}^2$$

$$A_{c2} = \frac{2.5^2\pi}{4} = 4.91 \text{ in}^2$$

$$F_{f2} = 65(4.91) - 15(3.58)$$

$$= 263 \text{ lb}_f$$

Total pressure to extend the load is

$$P_{c2}A_{c2} = F_{f2} + F_{L2} - P_{r2}A_{r2}$$

$$P_{c2} = \frac{264 + 2700 + 15(3.68)}{4.91}$$

$$= 615 \text{ psi}$$

Will the two cylinders extend simultaneously? Before answering this question, we need some information on the rest of the circuit. Suppose the pump is a fixed displacement pump, and the complete circuit is as shown in Fig. 7.2. The relief valve is set at 2000 psi. When the directional control valve (DCV) is shifted, the pump builds pressure to 615 psi. At this point, fluid flows to Cylinder 2, causing it to extend fully. Once it reaches full extension, the pressure builds to 646 psi, and now Cylinder 1 extends. When Cylinder 1 reaches full extension, the pressure builds to the relief valve setting, 2000 psi. The two cylinders do not move simultaneously; thus, the problem is not solved.

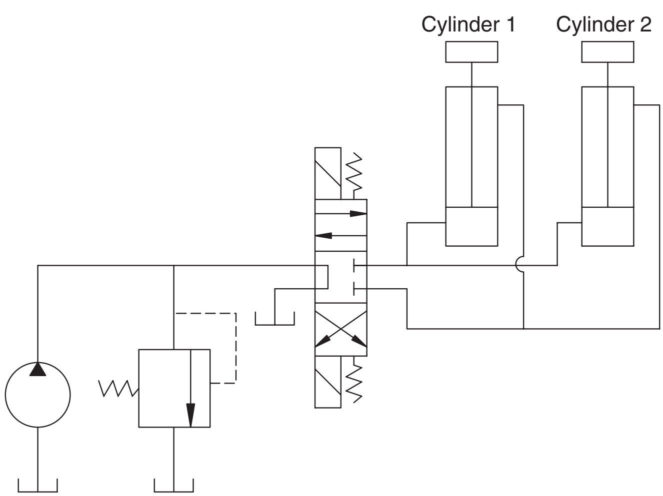


FIGURE 7.2
Fixed displacement pump supplying flow to cylinders in parallel.

A parallel circuit supplied by a pressure-compensated pump is shown in Fig. 7.3. In this case, the pump is set to maintain 2000 psi; thus, 2000 psi is available at the instant the DCV is shifted. What will happen?

Maximum pressure required is 646 psi for Cylinder 1 and 615 psi for Cylinder 2. Both cylinders will start to move. The pressure will quickly drop to the load pressure, in this case 646 psi. Because fluid always seeks the path of least resistance, flow will increase to Cylinder 2 and decrease to Cylinder 1, and pressure will continue to decrease until it matches the 615 psi required to extend Cylinder 2. Cylinder 1 will stop (after moving only a small distance), and Cylinder 2 will extend completely. After Cylinder 2 reaches the end of its stroke, Cylinder 1 will extend completely. The problem of getting both cylinders to extend simultaneously is not solved by using a pressure-compensated pump.

We now consider these two cylinders connected in series (Fig. 7.4). The pressure required to extend Cylinder 2 (615 psi) is the back pressure (rod-end pressure) on Cylinder 1. Total pressure to extend Cylinder 1 now is

$$P_{c1}A_{c1} = F_{f1} + F_{L1} - P_{r1}A_{r1}$$

$$P_{c1} = \frac{459.6 + 4000 + 615(5.3)}{7.07}$$

$$= 1092 \text{ psi}$$

When the pressure reaches 1092 psi, both cylinders move simultaneously.

Cylinder 2 stops when Cylinder 1 stops. If the cylinders are sized such that $A_{c2} = A_{r1}$, both cylinders will extend the same distance. (We are neglecting

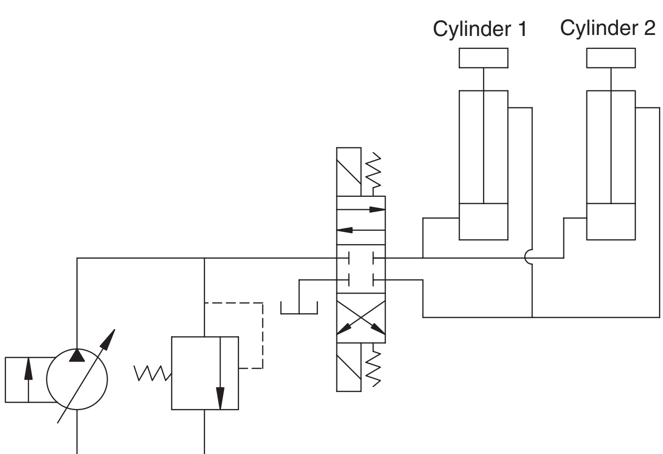


FIGURE 7.3
Pressure-compensated pump supplying flow to cylinders in parallel.

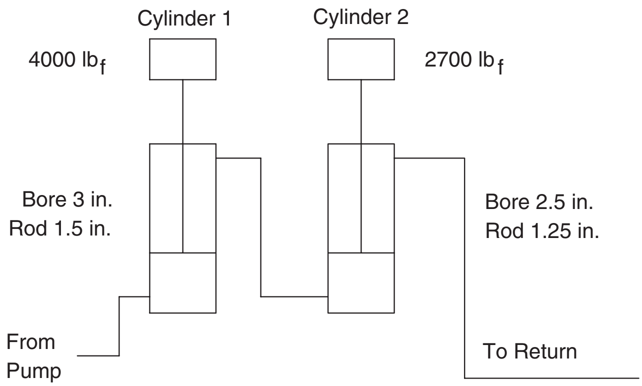


FIGURE 7.4
Cylinders in series.

leakage at this point. In actual practice, the extensions will never be exactly equal).

To illustrate the difference in extension times and extension rates, suppose the circuit in Fig. 7.4 is specified as follows:

Pump:	Flow 8 GPM (at pressure below 1200 psi, leakage is negligible)
Cylinder 1:	Stroke, x_1 = 20 in.
Cylinder 2:	Stroke, x_2 = 36 in.

Rate of extension of Cylinder 1

$$Q_1 = \frac{8 \text{ gal/min} \times 231 \text{ in}^3\text{/gal}}{60 \text{ s/min}} = 30.8 \text{ in}^3/\text{s}$$

$$\dot{x}_i = \frac{Q_1}{A_{c1}} = \frac{30.8}{7.07} = 4.35 \text{ in/s}$$

Note: A single dot above the variable indicates a first differentiation with respect to time, $\dot{x} = dx/dt$.

Rate of extension of Cylinder 2

The only flow that reaches Cylinder 2 is the flow out the rod end of Cylinder 1.

$$Q_2 = \dot{x}_1 A_{r1}$$

$$= (4.35 \text{ in/s})(5.3 \text{ in}^2) = 23 \text{ in}^3\text{/s}$$

$$\dot{x}_2 = \frac{Q_2}{A_{c2}} = \frac{23}{4.91} = 4.68 \text{ in/s}$$

Cylinder 2 extends faster than Cylinder 1, because $A_{c2} < A_{r1}$. Only when $A_{c2} = A_{r1}$ are the extension rates equal.

The distance the cylinder extends is also a key performance parameter. How far does Cylinder 2 extend? Total flow from Cylinder 1 is

$$\bar{Q}_1 = A_{r1}x_1$$

$$= 5.3(20) = 106 \text{ in}^3$$

$$x_2 = \frac{\bar{Q}_1}{A_{c2}}$$

$$= \frac{106}{4.91} = 21.5 \text{ in}$$

Cylinder 2 has a stroke of 40 in. but never extends beyond 21.5 in.

It is interesting to compute the maximum pressure required to extend the first cylinder, when the two are reversed. A "prime" will be used to denote this new configuration.

Cylinder 1′

$$\text{Bore 2.5 inch} \quad A_{c1} = 4.91 \text{ in}^2$$

$$\text{Rod 1.25 inch} \quad A_{r1} = 3.68 \text{ in}^2$$

$$\text{Stroke 40 inch} \quad F_{f1} = 264 \text{ lb}_f$$

$$F_{L1} = 2700 \text{ lb}_f$$

Cylinder 2′

$$\text{Bore 3.0 inch} \quad A_{c1} = 4.91 \text{ in}^2$$

$$\text{Rod 1.5 inch} \quad A_{r2} = 5.3 \text{ in}^2$$

$$\text{Stroke 20 inch} \quad F_{f2} = 459.6 \text{ lb}_f$$

$$F_{L2} = 4000 \text{ lb}_f$$

Pressure to cause Cylinder 1′ to extend is

$$P_{c1}A_{c1} = F_{f1} + F_{L1} + P_{r1}A_{r1}$$

$$P_{c1} = \frac{264 + 2700 + 646(3.68)}{4.91}$$

$$= 1088 \text{ psi}$$

The pressure required is approximately the same, 1088 psi as compared to 1092 psi. What happens when Cylinder 2′ extends completely and stops? When Cylinder 2′ stops, no more flow can leave Cylinder 1′, so it stops also. The distance moved by Cylinder 1′ is calculated as follows:

$$\bar{Q}_2 = x_2 A_{c2}$$

$$= 20(7.07) = 141.4 \text{ in}^3$$

$$x_1 = \frac{\bar{Q}_2}{A_{r1}} = \frac{141.4}{3.68} = 38.4 \text{ in.}$$

After the cylinders are reversed in position, Cylinder 1′ stops at 38.4 in., or 1.6 in. from full extension.

The rate of extension is

$$\dot{x}_1 = \frac{Q_1}{A_{c1}} = \frac{30.8}{4.91} = 6.27 \text{ in/s}$$

$$Q_2 = \dot{x}_1 A_{r1} = 6.27(3.68) = 23 \text{ in/s}$$

$$\dot{x}_2 = \frac{Q_2}{A_{c2}} = \frac{23}{7.07} = 3.25 \text{ in/s}$$

The speed of Cylinder 1′ is 44% faster, and the speed of Cylinder 2′ is 30% slower than the original series configuration.

7.3 Synchronization of Cylinders

There are instances when a large mass must be moved, and it is not feasible to move it with just one cylinder. It is often not a question of being able to generate enough force with one cylinder. As we will see in a later section, cylinders are available with 36-in. bores and larger. If the load to be moved is several feet in length, two or more cylinders are used to prevent a moment, or moments, that might distort and damage the load.

Presses are used in manufacturing for molding and shearing parts. The platen is the heavy member that the cylinder (or cylinders) forces down against the workpiece. If the platen is several feet wide, it has to be of very heavy construction to prevent damage when it is pressed down by a single cylinder in the middle. It can be designed with less material if it is pressed down with two (or more) cylinders. These cylinders must be synchronized.

There are three techniques that can be used to synchronize two cylinders.

1. Orifice-type flow divider
2. Gear-type flow divider
3. Mechanical coupling

These three techniques are illustrated schematically in Fig. 7.5.

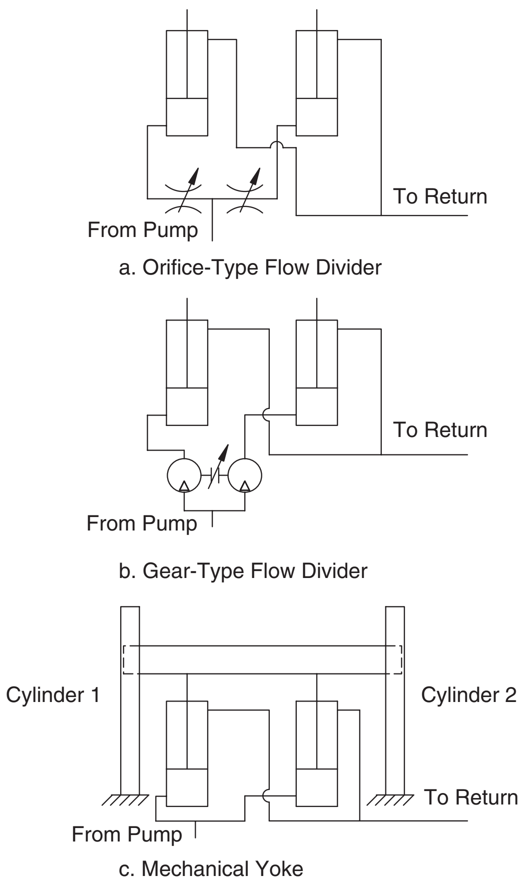


FIGURE 7.5
Techniques for synchronizing two cylinders.

7.3.1 Orifice-Type Flow Divider

Theoretically, the orifice on both sides of the flow divider can be adjusted so that the ΔP across the orifice plus the load ΔP is equal on both sides. Then, the flow from the pump will divide equally, and both cylinders will extend at the same time. If the cylinders are the same size, they will extend at the same rate.

This solution works until the load ΔP changes. Now, the flow goes to the lower-pressure side. The effect of some load fluctuation can be handled by a pressure-compensated flow control. Eventually, the cylinders get out of synchronization and have to be resynchronized.

7.3.2 Gear-Type Flow Divider

The gear-type flow divider functions like two gear motors with their shafts rigidly attached. We will consider first the case in which both motors have the same displacement. Since their shafts are attached, they both have to turn at the same speed; consequently, the same flow goes through both sides. Except for leakage, which is always a little different for the two sides, the flow is equally divided, and the cylinders extend simultaneously.

Pressure intensification is a potential problem with a gear-type flow divider. In Fig. 7.1, suppose that the load is removed from one cylinder, say Cylinder 2. The no-load pressure to extend Cylinder 2 is 65 psi, and the pressure to extend Cylinder 1 is 10 times higher at 646 psi. Suppose the pressure drop due to flow restriction through the flow divider is $\Delta P = 9$ psi. The pump will build pressure to

$$P = 646 + 9 = 655 \text{ psi}$$

The pressure drop across the Cylinder 2 side of the divider is 655 – 65 = 590 psi. The side-2 motor torque due to this pressure drop is delivered through the coupled shafts to side 1. The side-1 motor has a torque delivered "into" its shaft; thus, it functions like a pump and builds pressure on side 1. This pressure is higher than the 646 psi needed to move load at the desired velocity. Under certain circumstances, it could get to a level that causes damage. When using a gear-type flow divider, the designer must always analyze the circuit to ensure that pressure intensification will not cause a safety problem.

7.3.3 Mechanical Coupling

The most reliable way to ensure that two cylinders stay synchronized is to mechanically couple them together. One technique is shown in Fig. 7.5c. The beam (load) slides in a track on both sides. If Cylinder 1 gets a little ahead of Cylinder 2, the beam will bind in the track on that side, causing the pressure requirement on Cylinder 1 to increase. More flow goes to Cylinder 2, it catches up, and the bind is relieved. The two cylinders adjust back and forth to stay in reasonable synchronization.

The mechanical yoke technique illustrates a key principle in design. *Sometimes, the problem can be better solved with a modification to the functional requirement rather than a modification to the hydraulic circuit.*

7.4 Cushioning

When cylinders reach the end of their stroke, the pressure rises quickly, creating a shock wave in the hydraulic circuit. Cushioning is done to reduce this stock. The concept, shown in Fig. 7.6, is quite simple. First, we consider the case in which the cylinder is retracting. The spear closes off the large opening where the fluid is exiting the cap end of the cylinder. Fluid must now flow out the small opening past a needle valve. This valve adjusts the orifice and sets the back pressure that develops in the cap end. The resultant force slows the piston so that it "coasts" to a stop. The resultant pressure shock in the main circuit is significantly reduced.

The same technique is used to cushion the cylinder when it is extending. In this case, a sleeve is mounted on the rod to close the main opening so that flow goes through the orifice.

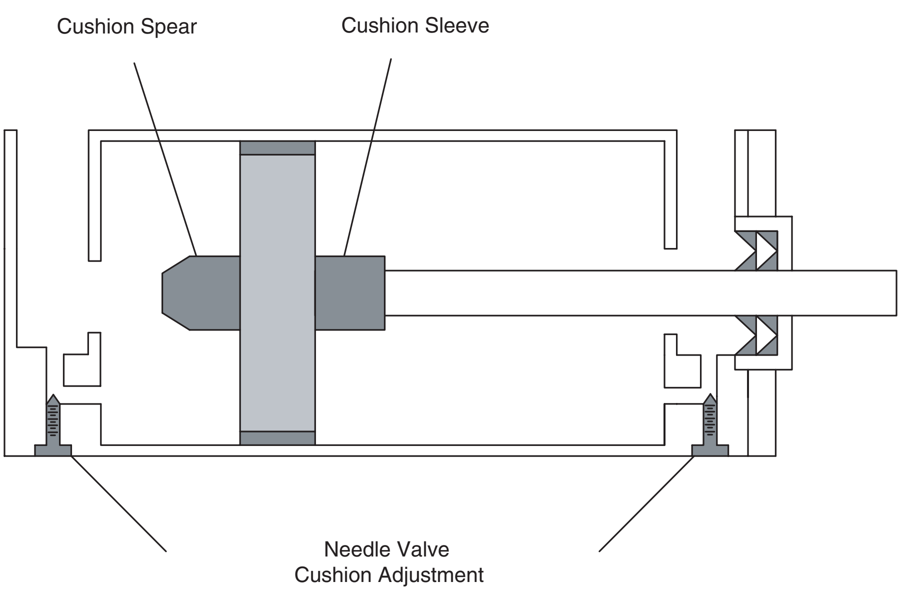


FIGURE 7.6
Schematic showing a technique for cushioning a hydraulic cylinder.

To understand cushioning, it is appropriate to review a basic principle of fluid power. Always ask the question, what is happening at the relief valve? We will consider the case where a fixed displacement pump is supplying the flow to extend the cylinder.

When the spear closes the large opening, the fluid must flow past the needle valve. The resultant pressure drop increases the back pressure on the rod end. The pump must build a higher pressure on the cap end. This cap end pressure must build to the point where the relief valve cracks open before the cylinder will slow. Remember, a fixed displacement pump puts out a given volume of oil for each revolution. Neglecting pump leakage, this oil either goes to the cylinder or through the relief valve. The adjustments made to the needle valve on the cylinder interact with the characteristics of the relief valve (and to a lesser degree with other components in the circuit) to produce a given deceleration rate.

A number of techniques have been developed to cushion cylinders. Large cylinders are cushioned with some type of stepped procedure that decelerates the piston in increments. Manufacturer's literature can be referenced for the back-pressure curves generated by these techniques.

The ANSI symbol for a cylinder cushioned on both ends is given in Fig. 7.7. The arrow through the cushion block indicates that the cushioning is adjustable.

7.5 Rephasing of Cylinders

When cylinders are connected in series, it is necessary to provide a feature for *rephasing* these cylinders when they are fully retracted. Otherwise, leakage will cause the downstream cylinder to not fully extend.

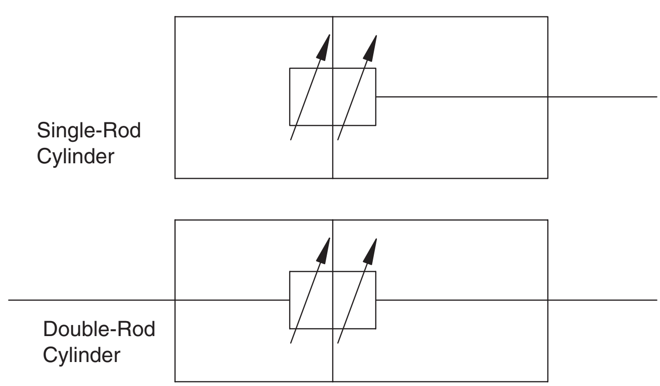


FIGURE 7.7
ANSI symbols for single- and double-rod cushioned cylinder.

An example of the need for rephasing is an agricultural implement (planter, cultivator, disk harrow, grain drill, etc.) that must be folded to an 8-ft width for road travel and unfolded to 24-ft width (or wider) for field operation. After several cycles, the downstream cylinders might not extend completely so that the sections make proper ground contact.

One technique used for rephasing is shown in Fig. 7.8. The cylinder is designed with a passageway for oil to flow from the cap end to the rod end when the piston reaches full extension. This passageway has a small diameter, as it is not expected to pass a large flow. Basically, it passes the flow required to make up leakage from the cap end of the cylinder immediately downstream so this cylinder will then extend completely

In some cases, three cylinders are connected in series. All three need the rephasing feature, not just the first and second cylinders. The third cylinder must have rephasing, because it may reach full extension before the two upstream cylinders both reach their full extension. It will then stop and block flow so that the other cylinders cannot fully extend.

7.6 Presses

Presses are used for molding, shaping, shearing, and many other operations. Some of the older manufacturing plants have lines of presses connected in parallel as shown in Fig. 7.9.

To give perspective to the problem, let us assume that the press cylinder has a 30-in. bore and 10-in. stroke. It needs to close in 30 s to achieve the desired cycle time.

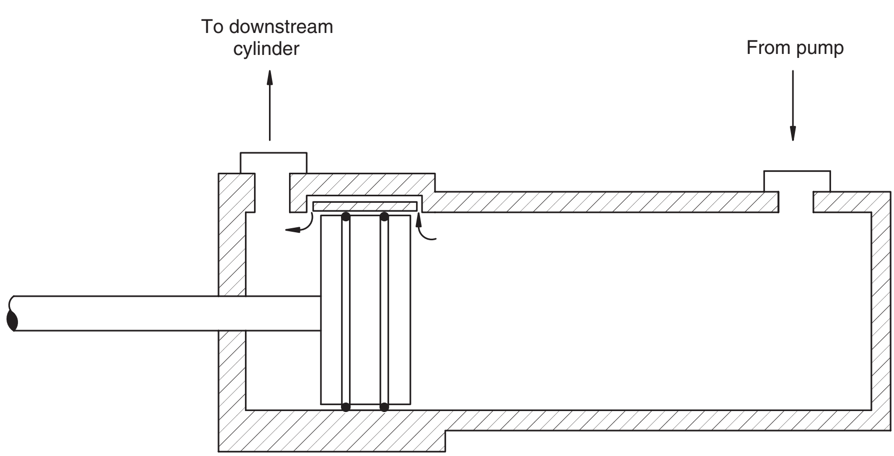


FIGURE 7.8
Cylinder construction used to provide rephasing for cylinders connected in series.

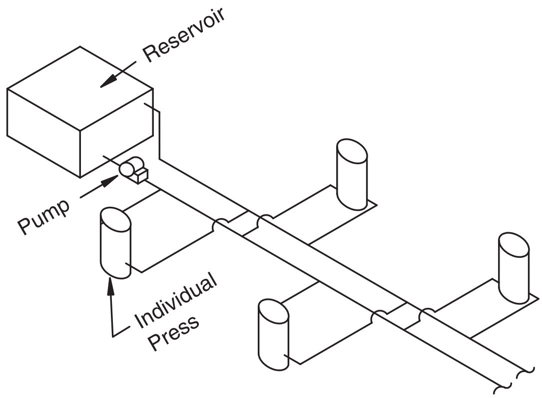


FIGURE 7.9
Presses connected in parallel in an older manufacturing plant.

$$Q = \frac{A_c x}{t}$$

$$= \frac{\pi(30)^2(10)}{4(30)}$$

$$= \frac{236 \text{ in}^3}{s} = 61 \text{ GPM}$$

The flow rate required to close an individual press is 61 GPM; thus, a high-capacity line is required. Presses along the line are closing under the control of individual operators, meaning that several presses can be closing simultaneously. (Some of the older press lines have 10, 20, or maybe more presses on a single circuit.) The reader can readily visualize that the flow dynamics in the main supply and return lines are hopelessly complex. Press operation is erratic; sometimes a press closes in 30 s, and sometimes it takes 60 s. When two or more presses are closing simultaneously, flow takes the path of least resistance, so it goes to the press with the smallest pressure drop first. After this press is closed, the flow completes the closing of the other presses.

Another significant disadvantage of the parallel circuit design is the volume of fluid that must be moved from the reservoir to the individual presses. This flow requires a high pump capacity and thus a high energy input.

A design that avoids the pumping of fluid back and forth from the reservoir is shown in Fig. 7.10. A manual DCV is shown, but typically the press is controlled by a solenoid-actuated DCV.

The main press cylinder is the large center cylinder in Fig. 7.10. The two other cylinders will be referred to as *side* cylinders. They are sometimes referred to as *kicker* cylinders. Their primary function is to raise and lower the

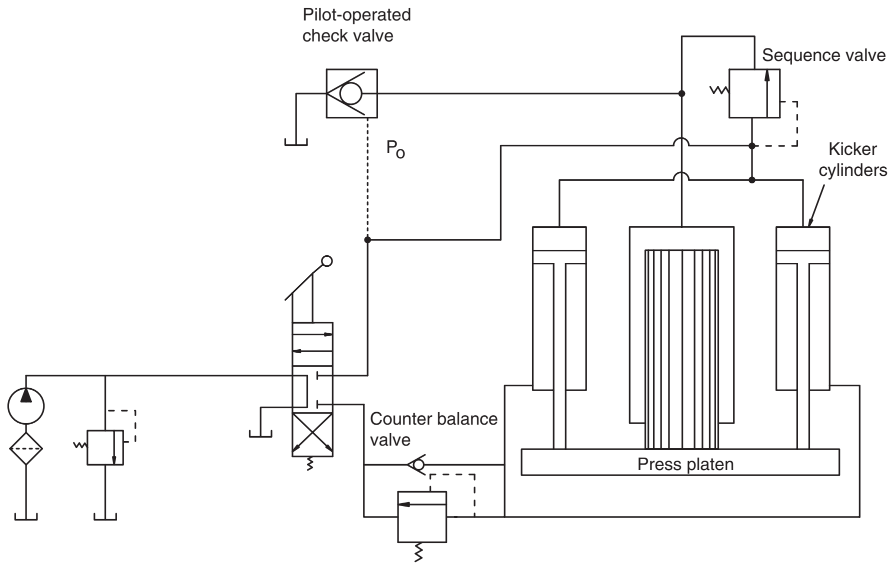


FIGURE 7.10
Circuit showing side cylinders to raise and lower the platen on a large press.

platen. The main press cylinder supplies most of the force needed once the platen contacts the work piece.

The circuit in Fig. 7.10 works as follows. When the operator shifts the DCV, flow goes to the two side cylinders, and they extend. Flow does not go to the press cylinder, because the sequence valve remains closed. As the press cylinder descends, the resulting negative pressure in the cap end pulls fluid from the reservoir into the press cylinder. (Often, the reservoir is above the press so that gravity helps to fill the large cylinder.) When the platen contacts the workpiece, the pressure builds, and the sequence valve opens. Now system pressure is applied to the press cylinder, and full force (side cylinders + press cylinder) is applied to the work piece.

When the DCV is shifted for retraction, the line to the sequence valve is connected to the reservoir. There is no pressure to hold the sequence valve open; consequently, it closes. Flow from the press cylinder cannot go back through the sequence valve; it must go through the check valve into the reservoir. The key to operation of this circuit is the *pilot-operated* check valve.

7.6.1 Pilot-Operated Check Valve

The pilot-operated check valve is shown in Fig. 7.11. For flow in the forward direction, this valve operates just like a normal check valve. Pilot line pressure holds the valve open for fluid in the reverse direction.

If the pilot line is pressurized (for example, if there is pressure at P_0 in Fig. 7.10), the valve is held open for fluid flow from the press cylinder back into

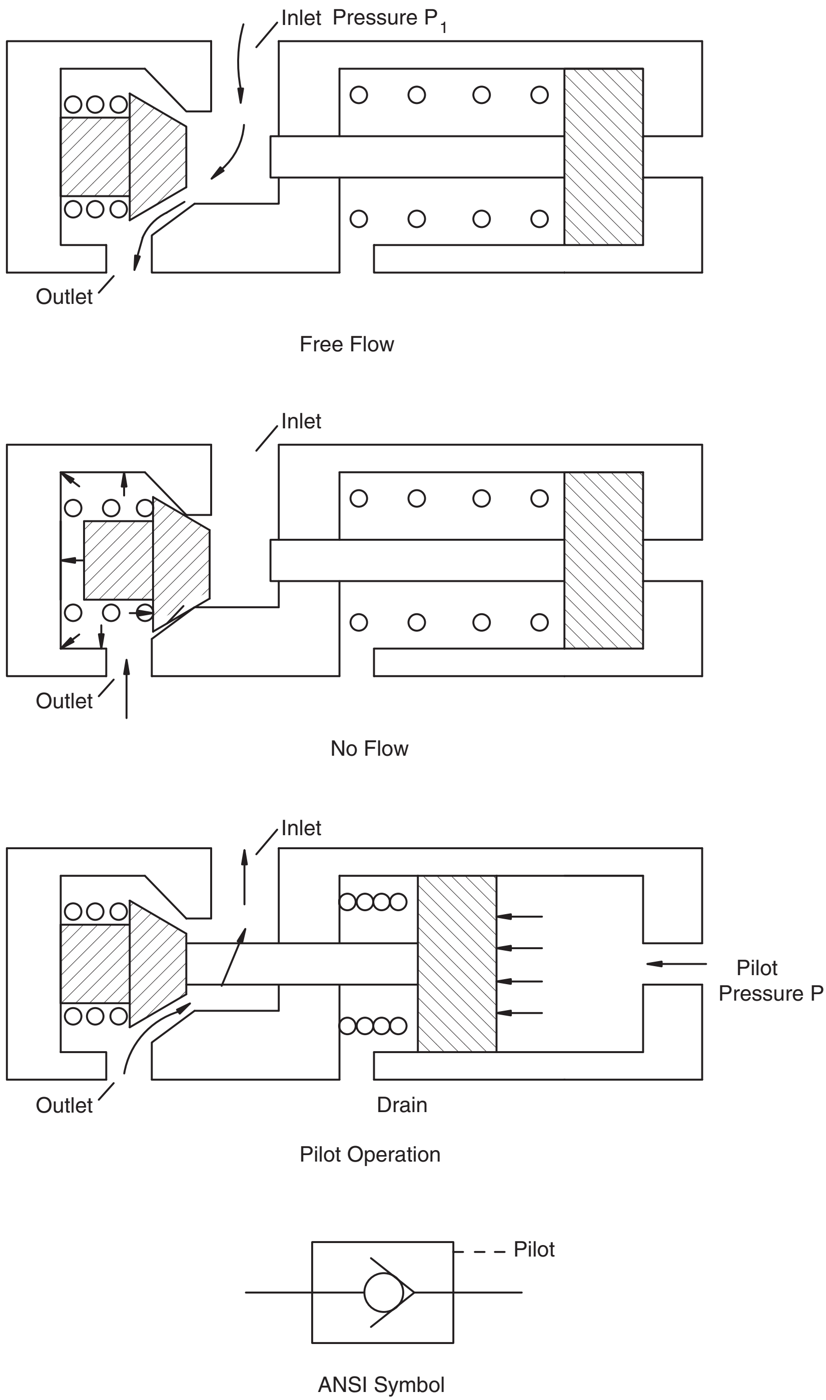


FIGURE 7.11
Pilot-operated check valve. (Reprinted with permission from Parker Hannifin Corp.)

the reservoir. Some specific pilot pressure is needed to open the check. If the required pilot pressure is 33% of load pressure, the valve is designated a 3:1 pilot-operated check valve. When a pilot-operated check valve is used for the application shown in Fig. 7.10, it is typical to have it open with as small a pilot pressure as possible—say a 5:1 valve.

7.6.2 Load-Locking Circuit

The counterbalance valve is used in the pressure circuit (Fig. 7.10) to prevent the platen from falling. A certain pressure must be developed in the rod end of the side cylinders to open the counterbalance valve. This requirement means that the pump has to be operating, and the DCV shifted, before the platen will descend. It must be "pumped" down; it cannot fall uncontrollably.

The same safety feature is provided in the load-locking circuit shown in Fig. 7.12. The cylinder is prevented from moving in either direction until pressure is applied from the pump. The pump supplies the pilot line pressure to open the check valve. A load force (F_L) in either direction will not cause the cylinder to move, except for some small leakage past the piston seals.

There are many applications where load locking is needed, and the extra pressure drop across the pilot-operated check valve in the return line is justified. Cylinders used to open large gate valves are an example.

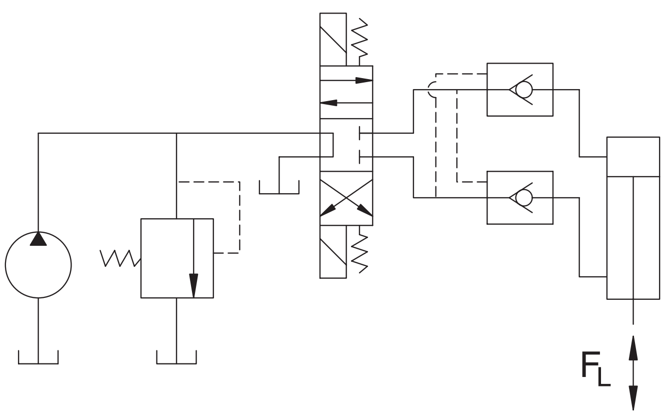


FIGURE 7.12
Load-locking circuit using pilot-operated check valves.

7.7 Load Analysis

There are two general classifications for loads.

Resistive load (opposite to the direction of motion)

Overrunning load (in the same direction as the motion)

Both types of load can be applied to the circuit in one cylinder cycle. In Fig. 7.12, a cylinder is lifting a weight during retraction and lowering this weight during extension. The extension load is overrunning, and the retraction load is resistive.

Many of us have activated a cylinder with a manual DCV, and we have an intuitive *feel* for this very simple operation. The DCV opening can be continuously adjusted to create the pressure needed to dump a variable amount of fluid across the relief valve. Cylinder speed is controlled by varying the position of the DCV handle.

The force versus time function required to move a load can be divided into several categories.

1. *Breakaway.* If the load is resting on a surface, the cylinder must develop the force required to overcome the *static* friction.
2. *Inertial.* Force must be developed to accelerate the load.
3. *Constant velocity.* If the load slides along a surface, the cylinder must supply the force required to overcome the *dynamic* friction.

Since the load in Fig. 7.13 is resting on frictionless rollers, it has no breakaway force and no constant velocity force (neglecting windage). It is interesting to consider an analysis of the inertial force.

7.7.1 Analysis of Acceleration of a Load with a Cylinder

The cylinder in Fig. 7.13 has a 3-in. bore, 1.5-in. rod diameter and 24-in. stroke. The fixed displacement pump has a theoretical output of 12 GPM and relief valve is set on 1500 psi. The load is 4,000 lb_f. During no-load extension,

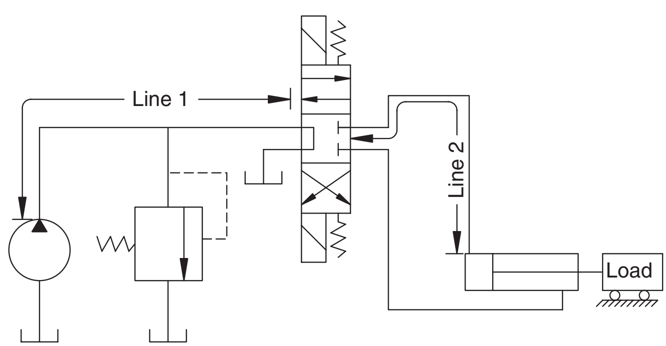


FIGURE 7.13
Resistive and overrunning load.

pressure drop between the relief valve and cap end of the cylinder was 40 psi. The rod end pressure was 15 psi. A force balance was done and the friction force was found to be $F_f = 330 \text{ lb}_f$.

Pressure during extension

$$P_c = \frac{F_f + F_L + P_r A_r}{A_C}$$

$$= \frac{330 + 4000 + 15(5.3)}{7.07}$$

$$= 624 \text{ psi}$$

$$P_{rve} = \text{pressure at relief valve during extension}$$

$$= P_c + 40$$

$$= 624 + 40 = 664 \text{ psi}$$

Cylinder velocity during extension (assuming volumetric efficiency is 92%)

$$\dot{x}_a = \frac{Q}{A_c} = \frac{12(0.92)(231)/60}{7.07} = 6.0 \text{ in/s}$$

The velocity of the load was carefully measured, and it was determined that it took 0.0628 s from the time the DCV was activated for the load to reach a constant velocity of 6.0 in/s. Is this time interval what is expected?

The maximum force that can be exerted is the force when the pressure equals the relief valve setting. Assuming ideal operation of the relief valve (no pressure overshoot),

$$P_{c\,\max} = 1500 - 40 = 1460 \text{ psi}$$

$$F_{\max} = P_{c\,\max} A_c - F_f - P_r A_r$$

$$= 1460(7.07) - 330 - 15(5.3)$$

$$= 9912 \text{ lb}_f$$

Theoretical acceleration of the load is given by

$$\ddot{x}_t = F_{\max}/m$$

where m = mass = 4000/32.2 = 124.2 lb_f-s^2/ft

Note: A double dot above the variable indicates a second differentiation with respect to time, $\ddot{x}_t = d^2x/dt^2$.

The subscript "*t*" is used to denote "theoretical" acceleration, and the "*a*" subscript is used to denote "actual" acceleration.

$$\begin{aligned}\ddot{x}_t &= 9912/124.2 \\ &= 79.8 \text{ ft/s}^2 \\ &= 957.6 \text{ in/s}^2\end{aligned}$$

Using the measured Δt, the actual acceleration is

$$\ddot{x}_a = \frac{\Delta \dot{x}_a}{\Delta t} = \frac{6.0 - 0}{0.0628} = 95.5 \text{ in/s}^2$$

Expected acceleration was 957.6 in/s^2 and achieved acceleration was 95.5 in/s^2, or 10% of expected. What happened?

A number of factors influence the acceleration of a mass with a fluid power circuit. These factors are:

1. Time for valve to open (valve dynamics)
2. Compressibility of oil
3. Compliance of lines (volume change due to pressure increase)
4. Characteristics of relief valve
5. Characteristics of pump
6. Leakage in DCV
7. Leakage in cylinder

These factors interact. Therefore, their influence cannot be analyzed or measured separately without a great deal of effort. The following analysis is simplified and is presented to give a basic understanding of an acceleration event without overloading the reader with pages of analysis.

7.7.1.1 Time to Open Valve

The moving part of a valve has mass, and some finite time is required to move this mass. To say that it opens instantaneously would imply that an infinite force is available, and this is not the case. For this example, we assume that the time to open the valve is Δt = 0.008 s. Flow that occurs during the opening of the valve is neglected.

7.7.1.2 Characteristics of Pump

As we learned in Chapter 4, the volumetric efficiency of a pump is a function of pressure. In this case, the pump volumetric efficiency is $e_{vp} = 0.80$ at 1500 psi, and $e_{vp} = 0.85$ at 1000 psi. The resulting linear equation, adequate for this example, gives e_{vp} as a function of pressure drop.

$$e_{vp} = -1 \times 10^{-4}\Delta P + 0.95$$

When the pressure at the relief valve is 1500 psi, the pressure where it is fully open, the pump volumetric efficiency is $e_{vp} = 0.80$. At 1350 psi, the relief valve cracking pressure, the pump volumetric efficiency is

$$\begin{aligned} e_{vp} &= -1 \times 10^{-4}(1350) + 0.95 \\ &= 0.815 \end{aligned}$$

We can determine an "effective" pump output during the acceleration event by using an average volumetric efficiency during the event.

$$e_{vpavg} = \frac{0.815 + 0.80}{2} = 0.8075$$

Effective pump flow is

$$\begin{aligned} Q_{eff} &= e_{vpavg} Q_t \\ &= 0.8075[12(231)/60] \\ &= 37.3 \text{ in}^3/\text{s} \end{aligned}$$

7.7.1.3 Leakage

For purposes of this example, it is sufficient to combine the DCV leakage and the cylinder leakage. These leakages are approximated using the orifice equation.

$$Q_t = CA\sqrt{2g\Delta P/\gamma}$$

or

$$Q_l = k\sqrt{\Delta P}$$

where $k = CA\sqrt{2g/\gamma}$

We assume that $k = 0.06$ and that the pressure drop between the leakage point and the reservoir is 30 psi. The pressure in the line is assumed to be

$$P_{line} = \frac{1500 + 1350}{2} = 1425 \text{ psi}$$

Pressure drop across the orifice where leakage occurs is then

$$\Delta P = P_{line} - 30$$
$$= 1425 - 30 = 1395 \text{ psi}$$

Resultant leakage flow is

$$Q_l = 0.06\sqrt{1395}$$
$$= 2.24 \text{ in}^3/\text{s}$$

7.7.1.4 Compressibility of Oil

The bulk modulus of oil [Chapter 2, Eq. (2.19)] is given by

$$\beta = V\frac{\Delta P}{\Delta V}$$

where V = total oil being compressed (in^3)
ΔV = change in volume (in^3)
ΔP = change in pressure (psi)

Suppose the total length of 0.5-in. diameter line from the pump to the DCV is 55.5 in. Volume of oil under compression in this line is

$$V_1 = \frac{0.5^2\pi}{4}(55.5) = 10.9 \text{ in}^3$$

Oil in the line between the DCV and cap-end port of the cylinder plus oil in the cap end of the cylinder is $V_2 = 4.9$ in^3.

The pressure the pump is supplying at the instant the DCV begins to open is the pressure drop through the DCV back to the reservoir. This pressure drop is 20 psi; thus, the pressure increase in line 1 (Fig. 7.13) is

$$\Delta P_1 = 1500 - 20 = 1480 \text{ psi}$$

(Remember, 1500 psi is the relief valve setting.)

The pressure in line 2 (Fig. 7.13) at the instant the DCV opens is the pressure required to hold the load. In this case, since the load is not being held against gravity, we can assume that the pressure in line 2 is zero.

When pressure reaches the relief valve setting, the ΔP in line 2 is

$$\Delta P_1 = 1500 - 0 = 1500 \text{ psi}$$

A typical bulk modulus for hydraulic oil is $\beta = 250{,}000$ psi. Total volume change due to oil compressibility is

$$\Delta V_{oc} = \frac{V_1 \Delta P_1}{\beta} + \frac{V_2 \Delta P_2}{\beta}$$

$$= [10.9(1480) + 4.9(1500)]/250{,}000$$

$$= 0.094 \text{ in}^3$$

7.7.1.5 Compliance of Lines

Hoses swell when the pressure increases. This change in cross-sectional area can be felt with low-pressure hose by grasping the hose with your hand. The pressure bulge of high-pressure hose typically cannot be felt but always occurs. Steel tubing also swells when pressurized. The volume change of the hose used in this problem is

$$\Delta V_{lc} = 0.001 \Delta P$$

where ΔV_{lc} = volume change (cm^3/ft)
ΔP = pressure change (psi).

For a $\Delta P = 1480$ psi, total volume change for a 55.5-in. hose is

$$\Delta V_{lc} = 0.001\ (1480)\ (55.5/12) = 6.84 \text{ cm}^3 = 0.418 \text{ in}^3$$

We neglect the compliance of the line between the DCV and cap-end port.

7.7.1.6 Characteristics of Relief Valve

The relief valve begins to open at 1350 psi and is fully open at 1500 psi. All flow from the pump is bypassed across the relief valve when the pressure reaches 1500 psi, and no flow goes to the cylinder. When the cylinder reaches 6.0 in/s, the force to move the load is just the force, $F = F_L + F_f$. As previously calculated, the resulting pressure at the relief valve is 664 psi, well below the 1350 psi at which the relief valve cracks open.

There is leakage across the relief valve during the first part of the time interval required to get the mass up to 6.0 in/s velocity. We might pose the question in this way, what is the time interval required to accelerate the mass to 6.0 in/s when the available pressure at the relief valve is an average pressure for the transition event?

$$P_{rvavg} = \frac{1500 + 1350}{2} = 1425 \text{ psi}$$

The corresponding pressure at the cap end of the cylinder is

$$P_{cavg} = 1425 - 40 = 1385 \text{ psi}$$

Note: Actual flow during the event is less than full flow; thus, the actual pressure drop is less than 40 psi. These details are omitted from this analysis.

It is instructive to first do the analysis without considering any of the factors that influence the acceleration of the mass. The resultant force to accelerate the mass is

$$\begin{aligned} F_{avg} &= P_{cavg}A_c - F_f - P_rA_r \\ &= 1385(7.07) - 330 - 15(5.3) \\ &= 9382 \text{ lb}_f \end{aligned}$$

A force of this magnitude will produce the following acceleration

$$\begin{aligned} \ddot{x}_{avg} &= F_{avg}/m \\ &= 9382/124.2 \\ &= 75.5 \text{ ft/s}^2 = 906 \text{ in/s}^2 \end{aligned}$$

If the acceleration is 906 in/s^2, the time to get the load up to speed is

$$\begin{aligned} \Delta t_t &= \frac{\Delta \dot{x}_a}{\ddot{x}_{avg}} \\ &= \frac{6.0 - 0}{906} = 6.6 \times 10^{-3} \text{s} \end{aligned}$$

How much flow is delivered during this Δt interval? The actual flow is the pump effective flow adjusted for leakage.

$$Q_a = Q_{eff} - Q_l$$
$$= 37.3 - 2.24$$
$$= 35.06 \text{ in}^3/\text{s}$$

Volume delivered by the pump during the transition event is

$$V_p = Q_a \Delta t_t$$
$$= 35.06(6.6 \times 10^{-3}) = 0.231 \text{ in}^3$$

The displacement of the cylinder is

$$x = \frac{\ddot{x}_{avg} t^2}{2}$$
$$= \frac{906(6.6 \times 10^{-3})^2}{2} = 0.02 \text{ in}$$

Total fluid delivered to the cylinder is

$$V_c = A_c x$$
$$= 7.07(0.02) = 0.141 \text{ in}^3$$

Total volume lost across the relief valve during the acceleration event is

$$\Delta V_{rv} = V_p - V_c$$
$$= 0.231 - 0.141 = 0.090 \text{ in}^3$$

We know that this answer is not correct, because some volume is required due to oil compressibility, and some is required due to line compliance. More importantly, the measured acceleration event lasts 0.0628 s, not the $\Delta t_t = 6.6 \times 10^{-3}$ s used to calculate this flow across the relief valve. We will now repeat the analysis and incorporate the various factors.

7.7.2 One Method for Incorporating the Influence of Various Factors during Acceleration Event

The time for the actual acceleration event is the measured time minus the time to open the valve.

$$\Delta t_a = 0.0628 - \Delta t_v$$
$$= 0.0628 - 0.008$$
$$= 0.0548\text{s}$$

The volume of fluid delivered by the pump is

$$V_p = Q_a \Delta t_a$$
$$= 35.06(0.0548) = 1.921 \text{ in}^3$$

Actual acceleration is

$$\ddot{x}_a = \frac{\Delta \dot{x}_a}{\Delta t_a} = \frac{6.0 - 0}{0.0548} = 109.5 \text{ in/s}^2$$

The displacement of the cylinder is

$$x = \frac{\ddot{x}_a \Delta t_a^2}{2}$$
$$= \frac{109.5(0.0548)^2}{2} = 0.164 \text{ in}$$

Total fluid delivered to cylinder to produce this displacement is

$$V_c = A_c x$$
$$= 7.07(0.164) = 1.159 \text{ in}^3$$

The volume of fluid delivered by the pump, minus leakage, was 1.921 in^3, and only 1.159 in^3 made it to the cylinder. Total volume from the pump is partitioned as follows:

$$V_p = V_c + \Delta V_{oc} + V_{lc} + \Delta V_{rv}$$

where V_p = volume delivered by pump (adjusted for leakage)
V_c = volume that produced cylinder displacement
ΔV_{oc} = volume due to oil compressibility
V_{lc} = volume due to line compliance
ΔV_{rv} = volume loss across relief valve

We can now get a better estimate of the volume loss across the relief valve during the acceleration event.

$$\begin{aligned}\Delta V_{rv} &= V_p - V_c - \Delta V_{oc} - V_{lc} \\ &= 1.921 - 1.159 - 0.094 - 0.418 \\ &= 0.025 \text{ in}^3\end{aligned}$$

In this case, we estimate that only 13% of the pump output was lost across the relief valve during the acceleration event.

Time to actually accelerate the load, $\Delta t = 0.0628$ s, was 9.5 times greater than the calculated acceleration time, $\Delta t_a = 0.0066$ s. For many applications, the additional time for acceleration is not critical. In cases where precise control of high-speed events is needed, the concepts presented here will be of greater importance. More on dynamic analysis will be presented in Chapter 11.

7.8 Types of Cylinders

The types of cylinders are:

- Double-acting (Fig. 7.14a)
- Single-acting, spring return (Fig. 7.14b)
- Double-rod (Fig. 7.14c)
- Tandem (Fig. 7.14d)
- Telescoping (Fig. 7.14e)

The double-rod cylinder has the same annular area on both sides, so it develops the same maximum force in both directions for a given relief valve pressure. It also extends and retracts at the same velocity when a given flow is supplied.

The tandem cylinder provides a means for increasing the force that can be generated with a given pressure. For extension, the total force is

$$F = (A_c + A_r)P$$

Telescoping cylinders are used when a long stroke is needed and the space available to mount the cylinder is limited. A typical example is the telescoping cylinder on a dump truck.

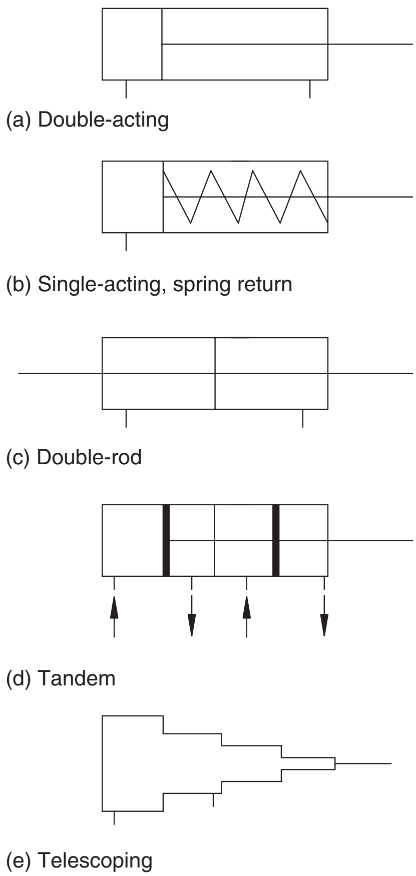


FIGURE 7.14
Types of cylinders.

7.8.1 Cylinder Selection

Cylinder manufacturers typically classify their products as heavy duty, medium duty, and light duty. Pressure ratings up to 6000 psi are available. Some manufacturers build an *agricultural-grade* cylinder. These cylinders are satisfactory for applications where annual use is limited. A manufacturing application, where the cylinder cycles thousands of times per year, requires an industrial-grade cylinder.

7.8.2 Cylinder Failure

Standard cylinders are not designed to take a side load. Care must be taken to ensure that binding does not occur during extension or retraction. A designer will use one of the mounting methods shown in Fig. 7.15 to prevent binding. Also, it is often necessary to provide guides to ensure that the load follows the prescribed pathway. As with any design, it is important to plan for a disturbance from an atypical direction, particularly for cylinders mounted on mobile machines. It is generally less expensive to protect from a side load than to replace a damaged cylinder.

Long cylinders supported at one end become a slender column as they extend under load. Manufacturers typically provide a nomograph that can be used to select the rod size for a given stroke length. Sometimes, a longer cylinder bore and rod size must be selected to provide the needed structural strength. Fig. 7.16 gives the ANSI symbol for a cylinder where the diameter of the rod compared to the bore is critical to circuit function.

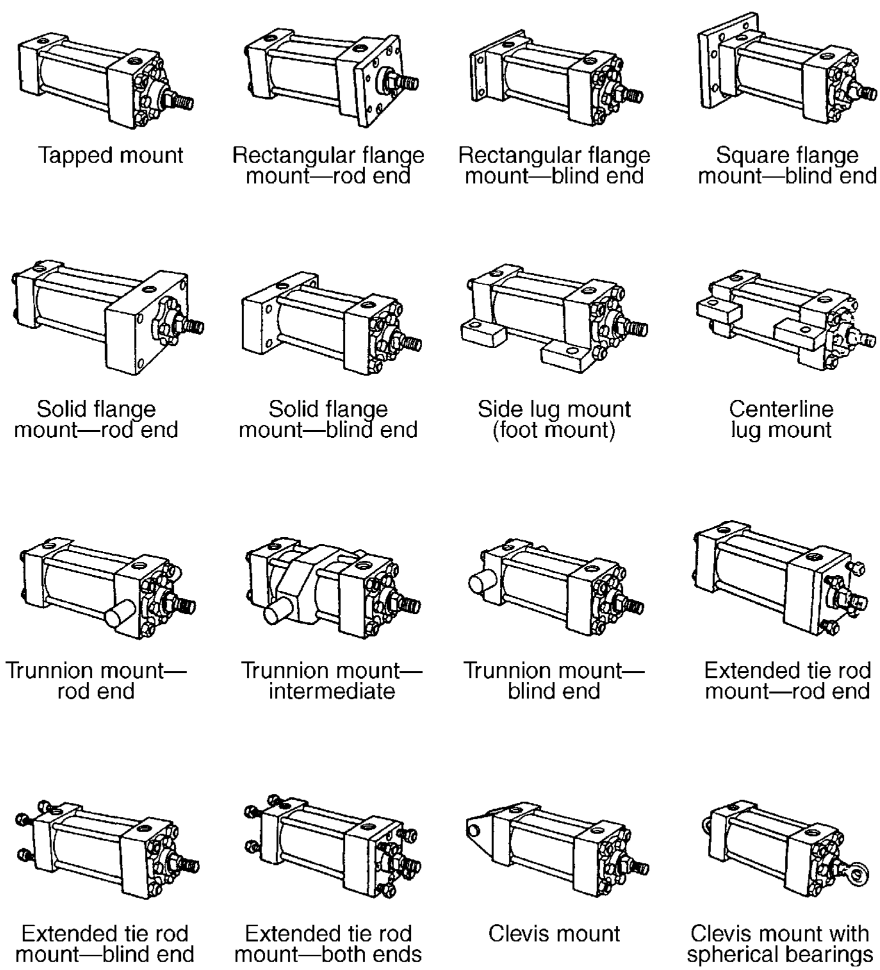


FIGURE 7.15
Partial listing of cylinder mounting methods.

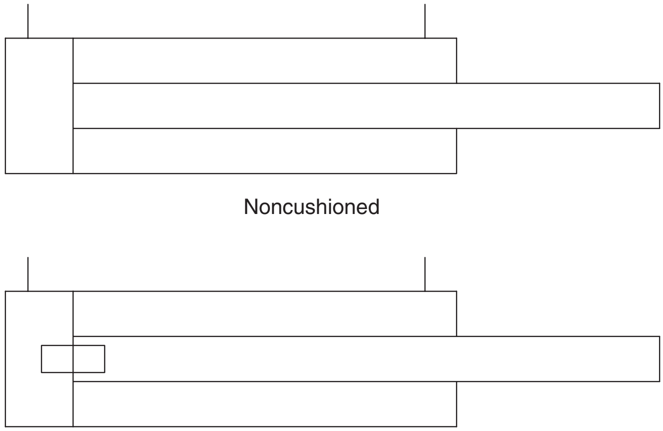


FIGURE 7.16
ANSI symbol for cylinder where rod diameter is critical to circuit function.

From mechanics of materials, the maximum load that can be carried by a column is

$$F = \frac{\pi^2 EI}{L^2} \tag{7.1}$$

where F = maximum longitudinal load (lb_f)
E = modules of elasticity of material (psi)
I = moment of inertia of cross-section (in^4)
L = length (in)

The body of a cylinder filled with oil does not function like a solid column or a hollow column, thus Eq. (7.1) can be used for an approximation, but it is not a replacement for the information given in the design nomograph. It is always a good idea to get specific design help from the manufacturer's technical support staff.

Example Problem 7.1

A cylinder with a 48-in. stroke is supported with a pinned connection at the cap end and at the rod end. The basic length is given by

$$L = \text{stroke} \times \text{stroke factor}$$

For this mounting configuration, the stroke factor is 2.00. Therefore,

$$L = 48 \times 2 = 96 \text{ in.}$$

The solid steel rod is 1.375 in. dia.

$$I = \frac{\pi r^4}{4}$$

$$= \frac{\pi(1.375/2)^4}{4}$$

$$= 0.175 \text{ in}^4$$

The modulus of elasticity for steel is

$$E = 30 \times 10^6 \text{ psi}$$

The longitudinal force that will cause buckling is given by Eq. (7.1).

$$F = \frac{\pi^2 EI}{L^2}$$

$$= \pi^2 \frac{(30 \times 10^6)0.175}{(96)^2}$$

$$= 5622 \text{ lb}_f$$

No safety factor is included in this computation, and no allowance has been made for the different cross-sectional area of the body of the cylinder. (For example, a 4-in. bore cylinder has an outside diameter of approximately 4.5 in. The cross-sectional area is annular and is filled with oil.) One manufacturer's nomograph (Fig. 7.17) gives an allowable thrust force of 1000 lb_f. If the cylinder has a 4-in. bore, the allowable pressure is

$$P = \frac{F}{A_c}$$

$$= \frac{1000}{12.5} = 80 \text{ psi}$$

This is a very low pressure, but a higher pressure may cause the cylinder to buckle when it is fully extended.

Increasing the rod diameter to 2.5 in. increases the allowable thrust force to 11,000 lb_f based on Fig. 7.17. The allowed pressure for this 4-in. bore cylinder is now 880 psi.

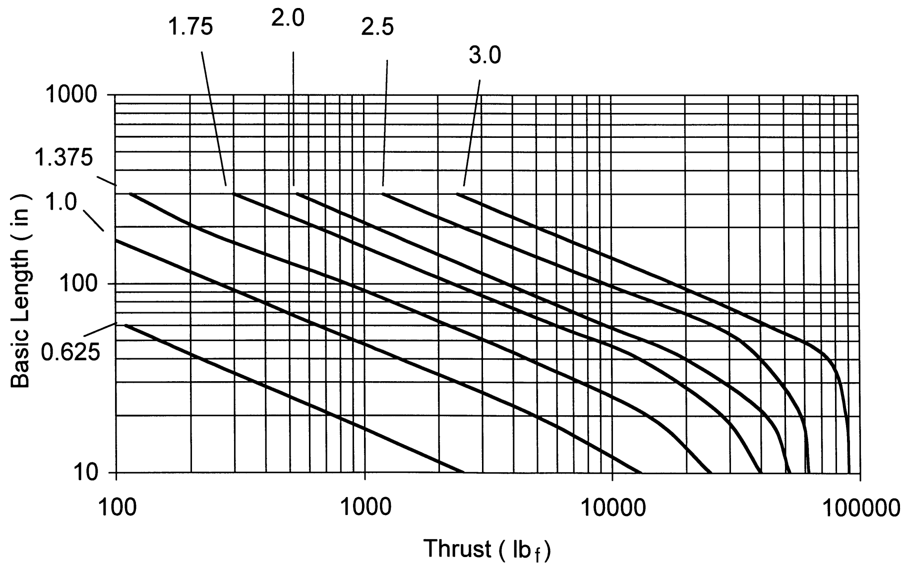


FIGURE 7.17
Nomograph for selection of cylinder to develop a given thrust load. (Reprinted with permission from Parker Hannifin Corp.)

Another possibility is to support the cylinder with a pinned connection at the rod end. (This is the trunnion mount-rod end configuration shown in Fig. 7.14.) Now the stroke factor is 1.00, and L = 48 in. The allowable thrust force is 34,000 lb_f, and the resultant allowable pressure is 2720 psi.

7.9 Cylinder Construction

Typical construction for cylinders used in industrial applications is shown in Fig. 7.18. The seals are a key feature, as is the rod wiper. Dirt from the environment settles on the rod and will ingress into the hydraulic system if it is not removed. Some small particles do escape the wiper, and these must be removed by the filtration system.

Industrial cylinders, because they are designed for a large number of cycles during their design life, will typically have the multiple o-ring seals shown in Fig. 7.18. They also have a rod bearing to support the rod when the load is not a pure axial load. Lower-cost cylinders, designed for fewer cycles, often do not have these features. The lower-cost cylinders can be used for agricultural equipment, some of which is only used 200 hours per year. Industrial cylinders are used on equipment that operates 2000+ hours per year.

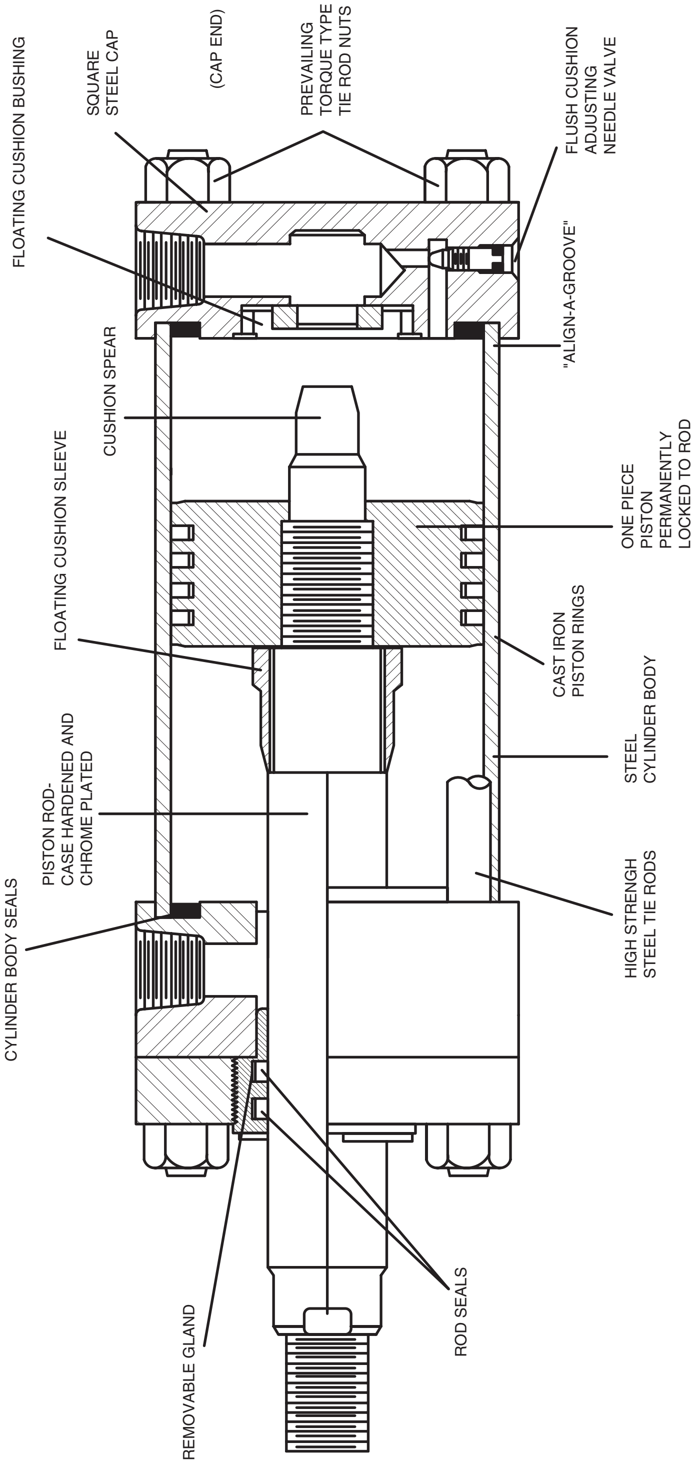


FIGURE 7.18
Typical cylinder construction for cylinders used for industrial applications. (Reprinted with permission from Parker Hannifin Corp.)

7.10 Summary

Cylinders are used in all types of manufacturing and on many mobile machines. In most cases, it is not good design to connect cylinders in series. A stacked directional control valve should be used to provide a separate circuit for each cylinder.

When two or more cylinders are connected in parallel, it is very difficult to get synchronized extension and retraction. Of the three methods commonly used (orifice-type flow divider, gear-type flow divider, and mechanical coupling), the mechanical coupling is the most satisfactory.

To fill the large cylinder on a press, oil must be pumped from the reservoir to the cylinder each time the cylinder extends. To avoid the cost for moving this oil back and forth, a tank is mounted above the press, and oil is returned to this tank through a pilot-operated check valve when the cylinder retracts. This oil then flows into the press cylinder when it is extended.

Load analysis is a key requirement for the proper selection of a cylinder to develop a given force and achieve a given cycle time. The force versus time function will typically have three sections: breakaway, inertial, and constant velocity. Several factors interact to produce a given load acceleration. These factors are:

1. Valve dynamics
2. Oil compressibility
3. Line compliance
4. Relief valve performance
5. Pump volumetric efficiency
6. Leakage (in DCV and past cylinder seals)

Actual acceleration is always less than the calculated acceleration based on the maximum force at relief valve pressure.

References

Henke, R. W. 1983. *Fluid Power Systems and Circuits.* Penton/IPC, Cleveland, Ohio 44114.

Problems

7.1 The circuit shown in Fig. 7.19 is used to lift and clamp cast iron pipe for a cutting operation. Maximum clamping force needs to be

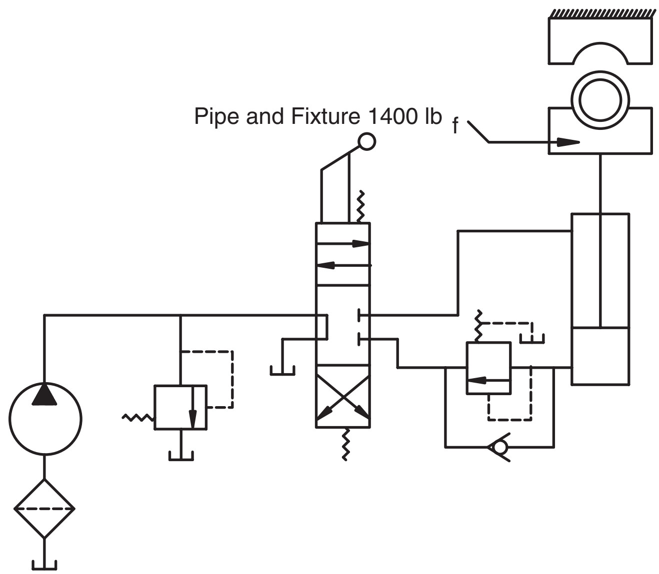


FIGURE 7.19
Circuit for clamping pipe (Problem 7.1).

10 tons, and the time allotted to close 30 in. is 3 s. The weight of the pipe and support structure is 1400 lb_f.

1. Determine the required cylinder bore and stroke based on your choice of a relief valve setting. Use the data in Fig. 7.20 to determine the rod diameter required.

Bore	Piston rod dia.	Operating pressure 3:1 safety factor on yield in psi	Maximum shock service 2:1 safety factor based on yield in psi
1-1/8	5/8	3600	5400
1-1/2	All	2250	3375
2	5/8	1400	2100
	1, 1-3/8	2450	3675
2-1/2	5/8, 1	900	1350
	1-3/8, 1-3/4	1550	2325
3-1/4	All	1400	2100
4	All	925	1390
5	1, 1-3/8, 1-3/4	675	1000
	Balance	1075	1600
6	All	800	1200

FIGURE 7.20
Reference data supplied by a manufacturer of medium-duty cylinders. (Reprinted with permission from Sheffer Corporation, Cincinnati, Ohio, 45242.)

2. Calculate the maximum power required (hp).
3. Suppose the estimated pressure drop from the cylinder rod end to the reservoir is 110 psi and a no-load extension test on your cylinder gives P_c = 130 psi. What is the maximum pressure at the relief valve when 10 tons of clamping force is generated.

7.2 A company is manufacturing bladders for truck brakes. The bladders are formed from pieces of fabric-reinforced rubber by placing the pieces of rubber on a form and closing a press. Heat and pressure mold the rubber into the correct shape.

The ram has a 30-in. bore and 20-in. stroke. Oil is stored in a reservoir over the press. This oil flows down and fills the ram as it is closed by the side cylinders. When the ram closes, a sequence valve opens to apply high pressure to the ram and develop the required force.

The press must close in 30 s. This requirement establishes the flow from the reservoir into the cap end of the ram. Sufficient negative pressure (suction) must be generated to produce the required flow across the pilot operated check valves.

The characteristic curve for the pilot-operated check valve (ΔP vs. Q) is given in Fig. 7.21. Total force to close the platen is 2400 lb_f. If the supply pressure is 2000 psi, what size cylinders are required for the side cylinders?

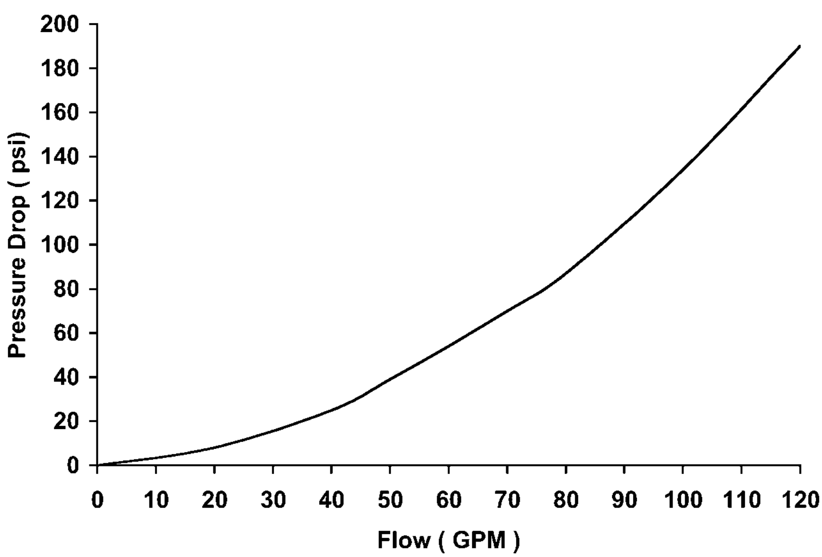


FIGURE 7.21
Characteristic curve for pilot-operated check valve (Problem 7.2).

7.3 Two cylinders are used to position a platform on an assembly line. The platform is moved into position, clamped parts are welded by robots, and the platform is cycled back to the original position.

The desired velocity of movement of the platform during extension is given in Fig. 7.22. (This problem does not deal with retraction.) Total distance moved is 30 in., and total weight of the platform is 8300 lb_f. The time allowed to lift the platform vertically a distance of 30 in. is 4 s.

Two cylinders support the platform (Fig. 7.23). Total force that must be developed by these cylinders is

$$F_L = F_I + F_{fp} + F_w$$

where F_L = total force (lb_f)
F_I = inertial force due to platform acceleration (lb_f)
F_{fp} = force due to friction of platform sliding against the guides (lb_f)
F_w = force due to weight of platform (lb_f)

The friction forces were measured and found to be

F_{fc} = 93 lb_f for each individual cylinder
F_{fp} = 580 lb_f total due to collars sliding along guides as platform is lifted

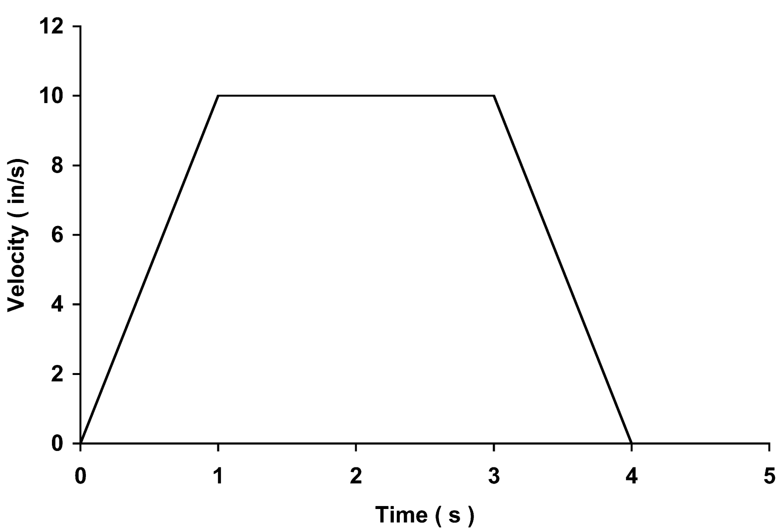


FIGURE 7.22
Desired velocity of platform in assembly line (Problem 7.3).

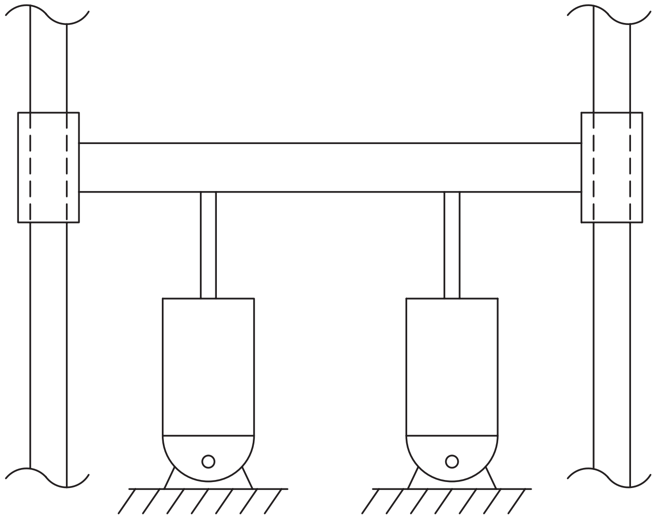


FIGURE 7.23
Two cylinders support assembly line platform (Problem 7.3).

Each cylinder bore is 2.5 in., and the rod diameter is 1.75 in. Back pressure measured at the rod-end port of each cylinder is 85 psi.

1. Calculate the maximum pressure required to extend the platform.
2. Suppose the cylinder circuit is supplied with a fixed displacement pump (3.37 in^3/rev) and the DCV is open center. The pump is driven at 1750 rpm. The relief valve is set on 1200 psi.

Estimate the amount of fluid lost across the relief valve when the DCV is shifted to initiate extension of the cylinder.

1. Neglect the time for DCV to open.
2. Neglect compressibility of oil.
3. Neglect compliance of lines.
4. Characteristics of pump: assume zero leakage as pressure increases.
5. Neglect leakage in DCV.
6. Neglect leakage in cylinder.
7. Characteristics of the relief valve are given in Fig. 7.24 and described by the following equation:

$$Q = 0.655P - 687.7$$

where Q = flow through relief valve (in^3/s)
P = supply pressure (psi)

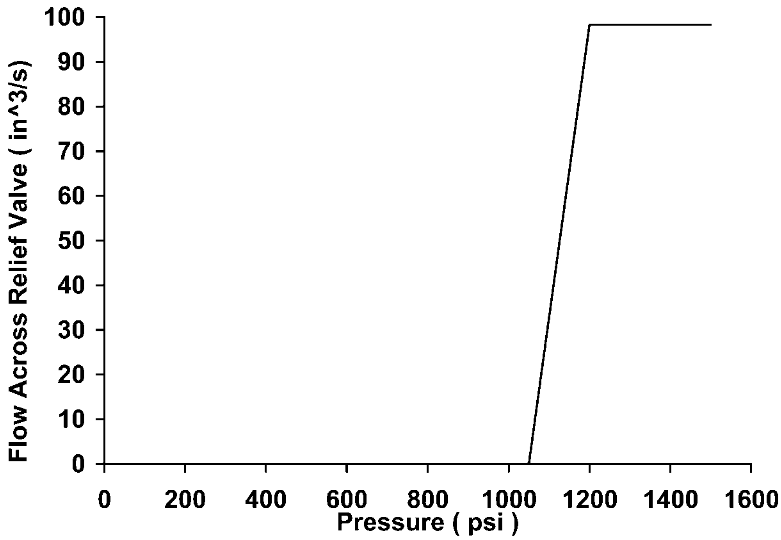


FIGURE 7.24
Characteristics of relief valve (Problem 7.3).

7.4 Two cylinders are connected in series as shown in Fig. 7.25. The load on cylinder 2 is 7360 lb_f and the relief valve setting is 2500 psi.

The following no-load data was collected on the individual cylinders.

Cylinder 1	Cylinder 2
P_{c1} = 37 psi	P_{c2} = 38 psi
P_{r1} = 35 psi	P_{r2} = 40 psi

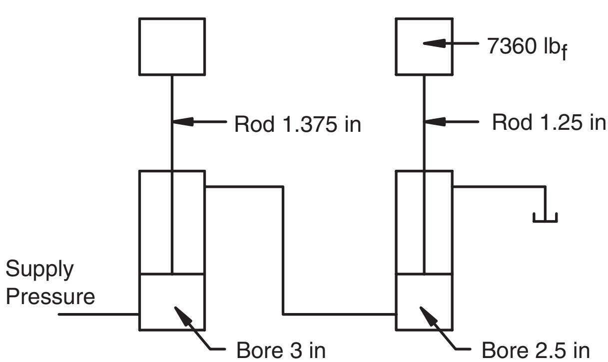


FIGURE 7.25
Cylinders in series (Problem 7.4).

Pressure drop between the relief valve and the cap-end port of Cylinder 1 is $\Delta P_{in} = 25$ psi, and the pressure drop in the line between Cylinder 1 and Cylinder 2 is $\Delta P_{line} = 10$ psi.

If the pressure drop from the Cylinder 2 rod-end port and the reservoir is 40 psi, what is the maximum load that can be lifted by Cylinder 1?

7.5 The cylinder in the circuit shown in Fig. 7.26 has a resistive load for the first 20 in. of the extension stroke and an overrunning load for the second 20 in. of the extension stroke. The cylinder bore is 3.25 in., the rod diameter is 2 in., and the friction force is 93 lb_f. Maximum overrunning load is estimated to be 4380 lb_f. The counterbalance valve is set such that the pump must develop at least 25 psi of cap-end pressure before the cylinder will extend the final 20 in. and lower the overrunning load to a safe stop. The relief valve is set on 3000 psi. For this problem, neglect all pressure drops in the lines.

1. Find the pressure setting needed for the counterbalance valve.
2. Calculate the maximum resistive load when the cylinder is extending.

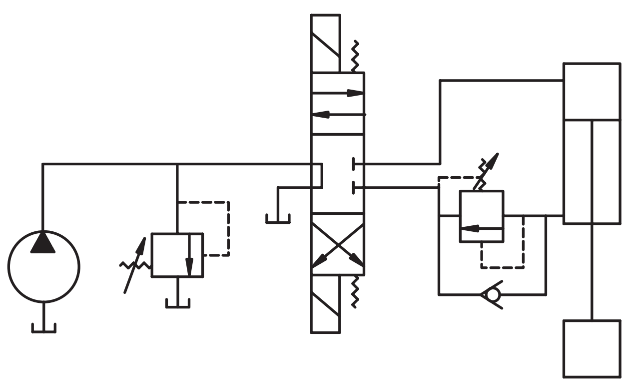


FIGURE 7.26
Cylinder with load that changes from resistive to overrunning at the midpoint of the extension cycle (Problem 7.5).

8

Temperature and Contamination Control

8.1 Introduction

In Chapter 2, we learned that the four functions of a hydraulic fluid are to

1. transmit power
2. lubricate
3. seal clearances
4. provide cooling

Most of the discussion in the intervening chapters has focused on the transmission of power. Mechanical energy is converted to hydraulic energy; this hydraulic energy is flowed to a new location and there reconverted to mechanical energy. This chapter focuses on the lubrication, sealing, and cooling functions.

Hydraulic oil is a lubricant. A hydraulic pump pumps a lubricant, and a hydraulic motor runs in a lubricant. If the oil viscosity is maintained in the correct range and the contamination level is controlled within the design range, hydraulic components last a long, long time. Cost per operating hour is a direct function of the life of the component, and component life is a direct function of the properties of the fluid; thus, temperature control and contamination directly affect operating cost.

8.2 Temperature Control

Operating temperatures consistently above 160°F promote chemical reactions that change the properties of the oil. Effects of high temperature are listed below:

1. Oxidation of the oil

2. Formation of insoluble gums, varnishes, and acids
3. Deterioration of seals (they harden and leakage begins)
4. Loss of lubricity
5. Changes in viscosity

The gums and varnishes clog orifices and cause valves, particularly valves that shift infrequently, to stick. The acids attack the metal surfaces of the components and cause corrosion.

The most significant effect of high temperature is the reduction in viscosity and subsequent reduction in lubricity. Loosely defined, lubricity is the ability of the fluid to maintain a film between moving parts. As viscosity decreases, this film thins. At some point, metal-to-metal contact occurs, and damage results. Metal particles from the damaged surface circulate with the fluid and erode other surfaces as they impact these surfaces. This damage cycle begins with high temperature. It is now readily apparent how the two subjects of temperature and contamination control are interconnected.

It is recommended that a hydraulic system be designed to operate at less than 140°F under worst-case ambient conditions. Over time, components wear, leakage increases, and more heat is generated. It is typical for the maximum operating temperature to slowly increase over the life of the system. Oil temperature at the return to the reservoir (or inlet to the heat exchanger if a heat exchanger is used in the return line) is easy to measure, and it is a valuable indicator of the overall condition of the system. If this temperature reaches 160°F, corrective action is needed.

If the oil has been heat damaged, it will have a darker color and an odor of scorched oil. Both of these characteristics indicate a problem that should have been solved earlier and would have been solved if oil temperature was being monitored.

Another indicator of high oil temperature is heat-peeled paint on the surface of the components or reservoir. If this is observed, it indicates a poorly designed and/or poorly maintained system. Considerable damage has been done by the time heat peeling of paint occurs.

Hydraulic energy is converted to heat energy whenever there is a pressure drop, and no mechanical work is done. Heat generation is unavoidable in fluid power circuits. As discussed in all previous chapters, pressure drops can (and should) be minimized.

Another source of heat generation is the compression of air bubbles in the oil. These bubbles are compressed as pressure is developed by the pump. When gas is compressed, the temperature increases; thus, compression of air bubbles introduces heat into the fluid. (Review the discussion in Section 2.4.4.1) Solving a pump cavitation problem not only reduces the damage caused by shock waves as the bubbles burst, it also reduces the problems caused by heat generation.

When heat is generated within a component, say a DCV, part of it flows into the oil, and part of it is conducted through the housing to an outer surface where it is exchanged into the surroundings by convection and radiation. Some of the heat energy in the oil is exchanged into the area around the lines (hoses and tubing) through which the oil flows. Again, this heat is exchanged by convection and radiation. On mobile machines, the length of the lines and the amount of air flowing around the components as the machine moves will often provide enough heat exchange that a separate heat exchanger, referred to as an *oil cooler,* is not needed. On the other hand, a mobile machine in direct sunlight is subjected to a high radiant energy input from the sun. Temperature of an exposed surface can reach 140° when the machine is sitting still on a clear day. Obviously, the potential for heat exchange from the oil is a function of the ambient conditions. A good design will maintain fluid temperature in the desired range for *worst-case* ambient conditions.

8.2.1 Methods for Cooling Hydraulic Oil

Two types of heat exchanger are used to cool hydraulic oil: (1) shell-and-tube and (2) finned tube. The shell-and-tube (Fig. 8.1) has a series of tubes inside a closed cylinder. The oil flows through the small tubes, and the fluid receiving the heat (typically water) flows around the small tubes. Routing of the oil can be done to produce a single pass (oil enters one end and exits the other end) or a double pass (oil enters one end, makes a u-turn at the other end, and travels back to exit at the same end it entered).

The finned tube exchanger (Fig. 8.2) is used for oil-to-air exchange. The air may be forced through the exchanger with a fan or may flow naturally. If an oil cooler is used on a mobile machine, it is the finned tube type.

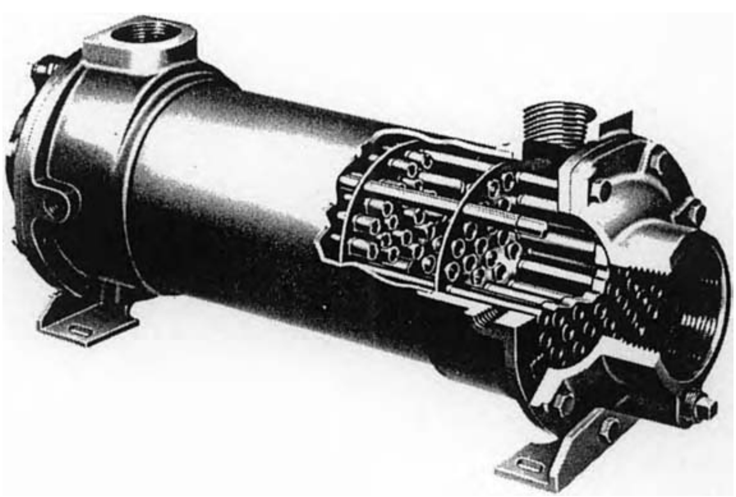


FIGURE 8.1
Shell-and-tube exchanger used to cool hydraulic oil. (Courtesy of Honeywell, Corona, CA.)

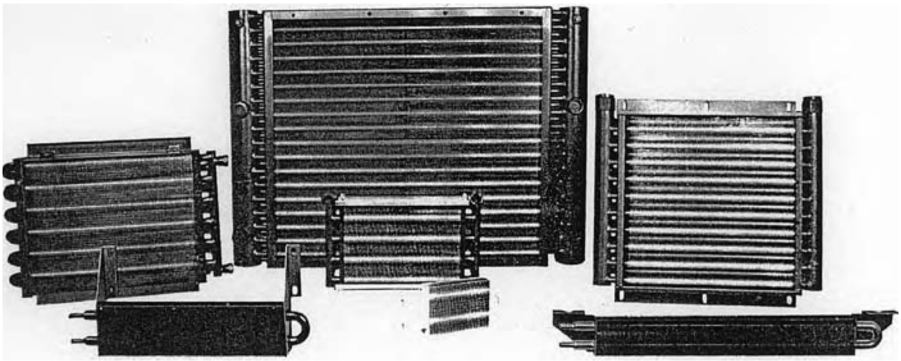


FIGURE 8.2
Finned tube heat exchangers used to cool hydraulic oil. (Courtesy of Honeywell, Corona, CA.)

Oil coolers are not built to withstand pressure; they are mounted in the return line in an off-line loop. The two options used are shown in Figs. 8.3a and 8.3b. In Fig. 8.3a, the system pump flows oil through the heat exchanger in the return line. This arrangement works well for many circuits. The exchanger is sized to give only a small pressure drop at rated flow. The circuit shown in Fig. 8.3b has a separate low-pressure pump to flow oil through the heat exchanger.

More complex circuits can have significant pressure pulses in the return line. These pulses hammer the heat exchanger and, over time, the joints fracture and begin to leak. If significant (greater than 10 psi) pulses are measured in the return line, the circuit shown in Fig. 8.3b should be used. Here, a separate pump is used to circulate oil from the reservoir through the heat exchanger and back to the reservoir. This circulating pump does not have to build pressure (only the 15 psi or so is required to flow fluid through the exchanger); therefore, it can be an inexpensive design. Any kind of pump is satisfactory if it is rated for the needed flow rate and has seals that are compatible with the fluid properties.

8.2.2 Heat Transfer from Reservoir

One of the functions of the reservoir is to exchange heat. Heat is transferred from the reservoir by conduction and convection. Transfer by radiation is ignored for this analysis. Heat transfer is given by

$$q = UA(T_f - T_a) \tag{8.1}$$

where q = heat transfer (Btu/h)
U = overall heat transfer coefficient (Btu/h·ft²·°F)
A = surface area (ft²)
T_f = temperature of fluid (°F)
T_a = ambient temperature (°F)

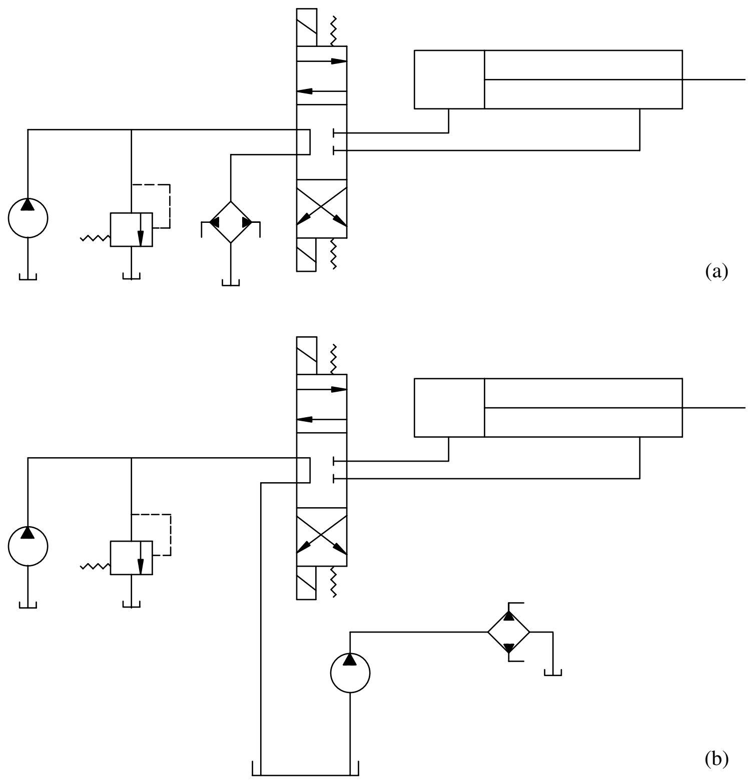


FIGURE 8.3
(a) Location of heat exchanger in a return line and (b) separate pump used to flow oil through heat exchanger.

The overall heat transfer coefficient is given by

$$U = \frac{1}{1/h + L/k} \tag{8.2}$$

where h = convective heat transfer coefficient (Btu/h·ft^2·°F)
L = thickness of reservoir walls (ft)
k = thermal conductivity of reservoir wall (Btu/h·ft^2·°F)

For natural convection of air, the convective heat transfer coefficient generally falls in the range of 0.5 to 3.0 Btu/h·ft^2·°F. If the reservoir on a mobile machine is placed such that air can flow freely around it, it is acceptable to use h = 3.0. For a stationary reservoir in a well ventilated location, h = 1.5 is a good choice.

Generally, hydraulic reservoirs are constructed from mild steel or aluminum. The thermal conductivities for these materials are given in Table 8.1.

TABLE 8.1

Thermal Conductivities of Materials Used to Construct Hydraulic Reservoirs

Material	Thermal conductivity, Btu/h·ft·°F
Aluminum	115
Mild Steel	27

Heat lost by the oil as it passes through the reservoir is

$$q = \dot{m}C(T_i - T_o) \tag{8.3}$$

where q = heat loss (Btu/h)
$\dot{m}$ = mass flow of oil (lb_m/h)
C = specific heat of oil (Btu/lb_m·°F)
T_i = temperature of oil entering reservoir (°F)
T_o = temperature of oil exiting reservoir (°F)

At steady state, the heat lost by the oil equals the heat transferred through the walls of the reservoir. The temperature of the oil decreases as it passes through the reservoir; therefore, we use an average temperature in Eq. (8.1), $T_f = (T_i + T_o)/2$. Substituting this value for T_f, equating Eqs. (8.1) and (8.3), and solving for the exiting temperature,

$$T_o = \frac{T_i(2\dot{m}C - UA) + 2UAT_a}{2\dot{m}C + UA} \tag{8.4}$$

8.2.3 Heat Generated by the System

Merritt (1967) gives the following empirical expression for estimating the heat generated by a hydraulic system.

$$q = 0.3857\,PQ \tag{8.5}$$

where q = heat generated (Btu/h)
P = pressure (psi)
Q = flow (GPM)

In Chapter 1, we learned that hydraulic power is given by

$$P_{hyd} = \frac{PQ}{1714}$$

Since 1 hp = 2547 Btu/h, the heat equivalent of this hydraulic power is

$$q = \frac{PQ}{1714}(2547) = 1.486PQ \text{ Btu/h}$$

The empirical formula then says that (0.3857/1.486)100 = 26% of the hydraulic energy is converted to heat energy in a typical hydraulic circuit.

It is appropriate to relate this 26% estimate to an estimate calculated using the overall efficiencies presented in Chapters 4 (pumps) and 5 (motors). Suppose a circuit has a gear pump with an overall efficiency e_{op} = 0.875 at 1,000 psi and a gear motor with overall efficiency e_{om} = 0.875. Flow delivered by the pump is 8 GPM. Hydraulic power delivered by the pump is

$$P_{hyd} = \frac{PQ}{1714} = \frac{1000(8)}{1714} = 4.67 \text{ hp}$$

Total input power is

$$P_{in} = P_{hyd}/e_{op} = 4.67/0.875 = 5.33 \text{ hp}$$

The input power converted to heat at the pump is

$$P_{qp} = P_n - P_{hyd} = 5.33 - 4.67 = 0.66 \text{ hp}$$

Neglecting line losses, the hydraulic power reaching the motor is $\mathcal{P}_{hyd}$ = 4.67 hp. Mechanical power delivered by the motor is $\mathcal{P}_{out} = \mathcal{P}_{hyd} \times e_{om}$ = 4.67(0.875) = 4.09 hp. Hydraulic power converted to heat at the motor is $\mathcal{P}_{qm}$ = $\mathcal{P}_{hyd} - \mathcal{P}_{out}$ = 4.67 − 4.09 = 0.58 hp.

Total heat generated is

$$q = (P_{qp} + P_{qm})2547\frac{\text{Btu/h}}{hp} = (0.66 + 0.58)2547 = 3158 \text{ Btu/h}$$

Equivalent thermal energy in the hydraulic flow is calculated

$$q_{tot} = 2547\frac{PQ}{1714} = \frac{2547(1000)(8)}{1714} = 11890 \text{ Btu/h}$$

The amount of heat energy generated as a percentage of the equivalent energy in the hydraulic flow is

$$\frac{3158}{11890} \times 100 = 26.5\%$$

We can derive an empirical constant like that given in Eq. (8.5) as follows:

$$q = KPQ$$

where K = constant

$$K = \frac{q}{PQ} = \frac{3158}{(1000)(8)} = 0.3948$$

which compares with the 0.3857 in Eq. (8.5). It is hoped that this simple analysis gives the reader some understanding of how an empirical formula like Eq. (8.5) is developed.

8.2.4 Design Example

The key question is whether an oil cooler is needed. If the answer is yes, a cooler can then be selected using performance data supplied by the manufacturer. To select a finned tube cooler, the three most important pieces of information are

1. Mass flow rate of oil
2. Rate of heat exchange (Btu/h)
3. Difference between entering oil temperature and surrounding air temperature

This design example will illustrate how the required rate of heat exchange is determined.

A mobile machine has three hydraulic pumps. Pump A is a piston pump, and it supplies flow to a closed-circuit hydrostatic transmission. This pump has a displacement of 2.3 in^3/rev and is driven at 1885 rpm. The charge pump displacement is 0.33 in^3/rev, and the charge relief valve is set on 150 psi. Pump B is a gear pump mounted on the auxiliary pump mount of pump A; thus it is also driven at 1885 rpm. Displacement of Pump B is 1.25 in^3/rev. Pump C is driven with a belt drive at 2000 rpm. Its displacement is 1.21 in^3/rev. Both pumps B and C supply open circuits.

The reservoir is constructed from 0.125-in. thick mild steel and is rectangular, $6 \times 30 \times 36$ in. The specific gravity of the hydraulic fluid is 0.85, and the specific heat is 1.5 Btu/lb_m·°F. Maximum fluid temperature is 140°F, and maximum ambient temperature is 95°F.

The various subsystems on the machine are widely separated; thus, there are some long runs of hydraulic lines that provide ample opportunity for heat exchange from the Pump B and Pump C circuits to the surroundings.

The first step in solving the problem is to find the total heat generated by the system. (Since all three pumps use oil from the same reservoir, the system is defined as the circuits supplied by all three pumps.) We can use the empirical equation [Eq. (8.5)] for the open circuits supplied by Pumps B and C if we know, or assume, the operating pressure for these circuits. Pump A supplies a closed circuit. The oil from the main pump circulates to the motor and returns to the pump without passing through the reservoir.

The charge pump in Pump A draws oil from the reservoir and replaces any leakage from the main circuit. Extra oil, not needed to replace leakage, drops across the charge relief valve and returns to the reservoir. At this point in the analysis, we will assume that all the charge pump flow drops across the relief valve (set at 150 psi) without doing mechanical work; thus, all this flow is converted to heat. Later in the analysis, this assumption will be revisited.

The functional requirement of the Pump B circuit was analyzed, and the operating pressure was calculated to be 400 psi. It is estimated that the peak pressure requirement will be perhaps 2.5 times the normal pressure, or 1000 psi. This peak is expected to occur only 10% of the total operating time.

Analysis of the Pump C circuit revealed that normal operating pressure would be 650 psi. Peak pressures up to 1500 psi are expected, and it is estimated that these peaks will occur 20% of the operating time.

8.2.4.1 Calculation of Heat Generation

Pump A (Charge Pump)

Flow: $$\frac{0.33\,\text{in}^3/\text{rev} \times 1885\text{ rpm}}{231\text{ in}^3/\text{gal}} = 2.69\text{ GPM}$$

Pressure: 150 psi

Pump B

Flow: $$\frac{1.25\,\text{in}^3/\text{rev} \times 1885\text{ rpm}}{231\text{ in}^3/\text{gal}} = 10.2\text{ GPM}$$

Pressure: 400 psi (normal)
1000 psi (peak)

$$P_{Bavg} = 0.9(400) \times 0.1(1000) = 460\text{ psi}$$

Pump C

Flow: $$\frac{1.21\text{ in}^3/\text{rev} \times 2000\text{ rpm}}{231\text{ in}^3/\text{gal}} = 10.5\text{ GPM}$$

Pressure: 650 psi (normal)
1500 psi (peak)

$$P_{Cavg} = 0.8(650) \times 0.2(1500) = 820 \text{ psi}$$

1. Heat generated by Pump A circuit (all charge pump hydraulic energy is assumed to be converted to heat energy):

$$q_{Achrg} = \frac{2547PQ}{1714} = \frac{2547(150)(2.69)}{1714} = 600 \text{ Btu/h}$$

2. Heat generated by Pump B circuit:

$$q_B = 0.3857PQ = 0.3857(460)(10.2) = 1810 \text{ Btu/h}$$

3. Heat generated by Pump C circuit:

$$q_c = 0.3857PQ = 0.3857(820)(10.5) = 3320 \text{ Btu/h}$$

4. Required heat dissipation by reservoir:
   Total heat generation is

$$q_{tot} = q_{Achrg} + q_B + q_c = 600 + 1810 + 3320 = 5730 \text{ Btu/h}$$

We estimate that one-half of the q_B and q_c heat will be dissipated from the lines and surface of the components; therefore, the required heat dissipation by the reservoir is

$$q_{gen} = q_{Achrg} + (q_B + q_c)/2 = 600 + (1810 + 3320)/2 = 3165 \text{ Btu/h}$$

5. Heat dissipated by reservoir:
   Find area of reservoir:

$$A = 2\left(\frac{6 \times 30}{144}\right) + 2\left(\frac{6 \times 36}{144}\right) + 2\left(\frac{30 \times 36}{144}\right) = 20.5 \text{ ft}^2$$

   Find overall heat transfer coefficient:

$$U = \frac{1}{1/h + L/k}$$

   Being a mobile machine, there is ample opportunity for airflow across the reservoir.
   We will use:

$$h = \frac{3.0 \text{ Btu}}{h \cdot \text{ft}^2 \cdot {}^\circ\text{F}}$$

for the convective heat transfer coefficient. From Table 8.1,

$$k = \frac{27 \text{ Btu}}{h \cdot ft \cdot {}^\circ\text{F}}$$

for mild steel. The wall thickness is $L = 0.125$ in $= 0.01$ ft.

$$U = \frac{1}{\frac{1}{3.0} + \frac{0.01}{27}} = 3.0 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

Calculate mass flow of oil.

The specific gravity is 0.85; thus, the weight density of the oil is

$$\text{wt. density} = 0.85(62.5 \text{ lb}_m/\text{ft}^3) = 53.1 \text{ lb}_m/\text{ft}^3$$

Total flow is

$$Q_{tot} = Q_{Achrg} + Q_B + Q_c = 2.69 + 10.2 + 10.5 = 23.4 \text{ GPM}$$

$$\dot{m} = 23.4 \text{ GPM}\left(\frac{1 \text{ ft}^3}{7.48 \text{ gal}}\right)\left(53.1\frac{\text{lb}_m}{\text{ft}^3}\right)\left(\frac{60 \text{ min}}{\text{h}}\right)$$

$$= 9967 \text{ lb}_m/\text{h}$$

Equation (8.4) is used to calculate the temperature of the oil exiting the reservoir.

$$T_o = \frac{(2\dot{m}C - UA)T_i + 2UAT_a}{2\dot{m}C + UA}$$

$C = 1.5$

$U = 3.0$

$A = 20.5$

$T_i = 140^\circ$F maximum fluid temperature

$T_a = 95^\circ$F maximum ambient temperature

$$T_o = \frac{[2(9967)(1.5) - 3.0(20.5)]140 + 2(3.0)(20.5)(95)}{2(9967)(1.5) + 3.0(20.5)}$$

$$= 139.79°\text{F}$$

Heat dissipated by the reservoir is

$$q_{res} = UA\left[\left(\frac{T_i + T_o}{2}\right) - T_a\right] = 3.0(20.5)\left[\left(\frac{140 + 139.79}{2}\right) - 95\right]$$

$$= 2760 \text{ Btu/h}$$

6. Required cooling:
   Heat is dissipated by the reservoir at a rate of 2760 Btu/h and is generated at a rate of 3165 Btu/h. Therefore, the cooling required is 3165 – 2760 = 405 Btu/h.

It is not realistic to select an oil cooler that is small enough to dissipate only 405 Btu/h. A better choice is to increase the size of the reservoir to dissipate the extra heat. It is left as an exercise to show that increasing the reservoir from 6 × 30 × 36 in. to 6 × 30 × 42 in. will give $q_{res} = q_{gen}$, and no additional cooling is needed.

8.2.4.2 Re-evaluation of Assumption for Pump A Circuit

Pump A is a variable displacement axial piston pump that delivers flow to an axial piston motor. Pressure in the main circuit will generally be less than 2000 psi, so the overall efficiencies of both pump and motor will be greater than 90%. The volumetric efficiency of both pump and motor is 95%. The displacements of both pump and motor are 2.3 in^3/rev.

We will now recalculate the heat generated by the Pump A closed circuit. We want a maximum heat generation estimate, so we assume that the pump is developing 2000 psi continuously. Total flow at maximum displacement is

$$Q_a = \frac{2.3 \text{ in}^3\text{/rev} \times 1885 \text{ rpm}}{231 \text{ in}^3\text{/gal}}$$

Total input power is

$$P_{hyd} = P_{hyd}/e_{op} = 21.9/0.90 = 24.3 \text{ hp}$$

Input power converted to heat is

$$P_{qp} = P_{in} - P_{hyd} = 24.3 - 21.9 = 2.4 \text{ hp}$$

Neglecting line losses, the hydraulic power reaching the motor is $\mathcal{P}_{hyd}$ = 21.9 hp. Mechanical power delivered by the motor is

$$P_{out} = P_{hyd} \times e_{om} = 21.9(0.9) = 19.7 \text{ hp}$$

Hydraulic power converted to heat at the motor end is

$$P_{qm} = P_{hyd} - P_{out} = 21.9 - 19.7 \text{ hp} = 2.2 \text{ hp}$$

Total heat generated by the pump and motor is

$$q_{Amain} = (P_{qp} + P_{qm})2547\frac{\text{Btu/h}}{hp} = (2.4 + 2.2)2547 = 11716\text{Btu/h}$$

The charge pump replaces the leakage at the pump and motor ends and then dumps the remaining flow over the charge relief valve. How much of the 2.69 GPM charge pump flow goes across the charge relief valve?

Leakage at the pump is

$$Q_{lp} = Q_t(1 - e_{vp}) = \frac{2.3(1885)}{231}(1 - 0.95) = 0.938 \text{ GPM}$$

Leakage at the motor is

$$Q_{lm} = Q_t(1 - e_{vm})$$

Here, Q is the flow that actually reaches the motor, therefore $Q = Q_t - Q_{\ell p}$

$$Q_{lm} = (Q_t - Q_{lp})(1 - e_{vm}) = (18.77 - 0.938)(1 - 0.95) = 0.891 \text{ GPM}$$

Total leakage flow replaced by the charge pump is

$$Q_l = Q_{lp} + Q_{lm} = 0.938 + 0.891 = 1.829 \text{ GPM}$$

Flow across the charge relief valve is

$$Q_{chrg} = 2.69 - 1.83 = 0.86 \text{ GPM}$$

Heat generated by flow across the charge relief valve is

$$Q_{Achrg} = \frac{P_{chrg}Q_{chrg}}{1714}2547 = \frac{150(0.86)}{1714}2547 = 191.7 \text{ Btu/h}$$

This heat generation is small compared to the 11,716 Btu/h generated within the Pump A closed circuit. Total heat generated by the Pump A circuit is

$$Q_A = Q_{Amain} + Q_{Achrg} = 11716 + 192 = 11908 \text{ Btu/h}$$

Total heat generated by the system is

$$q_{tot} = q_A + q_B + q_c = 11910 + 1810 + 3320 = 14050 \text{ Btu/h}$$

The Pump A closed circuit is a split hydrostatic transmission, so there is opportunity for heat to dissipate from the lines between the pump and motor. If we estimate that 25% of the Pump A circuit heat is dissipated from the lines and the surfaces of the pump and motor, and again use the assumption that 50% of the heat generated by the Pump B and Pump C circuits is dissipated from the lines, the required heat exchange is

$$q_{gen} = (1 - 0.25)q_A + (1 - 0.50)(q_B + q_c) = 8930 + 2566 = 11495 \text{ Btu/h}$$

Using the heat dissipated by the larger reservoir ($6 \times 30 \times 42$ in.), total cooling required is

$$q_{cool} = q_{gen} - q_{res} = 11495 - 3165 = 8330 \text{ Btu/h}$$

An oil cooler rated for 24 GPM and an exchange rate of 8,330 Btu/h with inlet temperature of 140°F and 95°F ambient temperature will be satisfactory. The design $\Delta T = 140 - 95 = 45$°F.

To give a frame of reference for the size exchanger needed, one manufacturer can supply a finned tube exchanger rated for 9,500 Btu/h heat rejection at 24 GPM flow rate with a 45°F ΔT. The finned area is 12×13.25 in., and the thickness is 1.5 in. The fitting size is 1 in. NPT.

8.2.5 Temperature Control Summary

Heat generation is an unavoidable consequence of fluid power circuit operation. Operating temperatures consistently above 160°F promote chemical reactions that change the properties of the oil. It is recommended that a hydraulic system be designed such that maximum oil temperature under worst-case ambient conditions be less than 140°F.

Some agricultural equipment is designed to operate at temperatures up to 150°F. This equipment is generally used for fewer annual operating hours than industrial equipment, and its design life is normally less; consequently, a higher operating temperature is allowed.

The most significant effect of a high operating temperature is the reduction in oil viscosity and subsequent reduction in lubricity. If the oil film between moving parts thins to the point where metal-to-metal contact occurs, the parts are damaged, and the life of the component is shortened.

Heat is dissipated from hydraulic lines and the surfaces of the components. When possible, the hydraulic reservoir is shaped to give maximum surface area and thus maximum heat dissipation. If the system is generating heat at a faster rate than it is being dissipated, an oil cooler is needed. Generally, a finned tube cooler is used for mobile applications. When water is used to cool the oil, a shell-and-tube exchanger is typically used. Oil coolers are very cost effective. Their use maintains the oil temperature within design specifications and thus extends the operating life of the components and oil.

8.3 Contamination Control

There are three types of failure in a fluid power system.

1. *Degradation*. The performance of the component degrades over time as surfaces wear, clearances increase, and leakage increases.
2. *Intermittent*. Valves stick and then break loose such that operation is intermittent.
3. *Catastrophic*. Catastrophic failure occurs when a major component breaks apart. Often, debris causes the failure of other components, and a total replacement of the circuit is required.

Proper selection, placement, and servicing of contamination control devices will eliminate an estimated 80% of all system failures. Maintaining system cleanliness is a key issue in the operation of all fluid power systems, and particularly high-pressure oil circuits.

8.3.1 Sources of Contamination

There are four sources of contamination in hydraulic fluid.

1. *Built-in contamination.* This is contaminate that was left in the system when it was assembled. It can range from a piece of teflon tape to a piece of welding slag.
2. *Contaminated new oil*. Contaminate is introduced during the manufacture and subsequent handling of oil. If this is not removed, it enters the system.
3. *Ingressed contamination.* This contaminate can enter with air flowing into the reservoir through the breather cap, or it can ride in on a cylinder rod. No rod seal can totally prevent the entrance of par-

ticles. Also, whenever a line is disconnected or the system is opened in any way, there is the potential for contamination to ingress.

4. *Internally generated contaminate.* Particles removed from the interior surface of the components will circulate in the system until they are removed. Each impact of one of these particles with a surface causes more damage. This phenomenon is known as the *wear regeneration cycle.*

Internally generated contamination causes damage in the following ways:

Abrasive wear. The particles that break loose from the hardened surfaces of the components are very hard. These particles bridge across the clearance between two moving surfaces and abrade one or both surfaces.

Adhesive wear. As discussed in the previous section, high temperature reduces oil viscosity and thins the film between moving parts. These parts then adhere, or stick together, and damage results.

Fatigue wear. Particles that bridge a clearance can cause a stress concentration. This stress concentration eventually causes a crack to form (Fig. 8.4), and this crack spreads until part of the surface breaks away. This type of surface failure is called a *spall.*

Erosive wear. Particles in a stream of moving fluid erode away the surface of the metering edge (Fig. 8.5).

Cavitation wear. When the pump compresses fluid that contains air bubbles, the bubbles implode. The resulting shock wave impacts the surface and, over time, these impacts cause damage. Cavitation wear and aeration wear are sometimes discussed as separate wear phenomena, but they are quite similar.

Corrosive wear. Chemical attack of the surface causes a loss of material. A simple example is water condensation in the reservoir that causes the metal to rust. More complex reactions than the oxidation of iron occur, and all these reactions are grouped under corrosive wear.

8.3.2 Quantifying Fluid Cleanliness

The current international standard for cleanliness of a hydraulic or lubricating fluid is ISO 4406. The standard specifies a laboratory particle counting procedure to determine number and size (in microns) of solid particles in a milliliter of fluid. These particle counts are referenced to a cleanliness code chart to determine the cleanliness code, a set of numbers separated by slashes. The procedure is best illustrated with an example taken from Vickers (1992). Typical data obtained with the approved particle counting procedure is given in Table 8.2. These data are plotted in Fig. 8.6. The data points at 15 and 5 μm are refer-

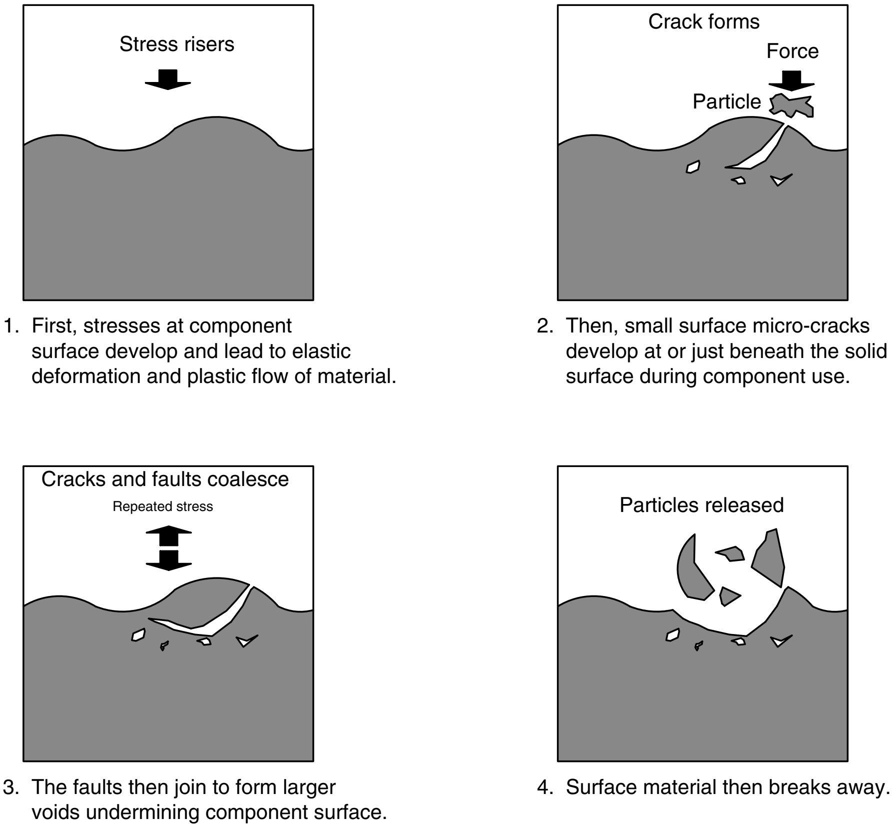


FIGURE 8.4
Illustration showing damage produced by fatigue wear. (Reprinted with permission from Eaton Hydraulics, Training Div.)

TABLE 8.2
Typical Data Obtained by Analyzing a Sample of Hydraulic Fluid with an Automatic Particle Counter

Particle size X (μm)	Number of particles greater than X in 1 mL test fluid
2	5120
5	89
10	43
25	3
50	0.4[a]

a. Particle counts are normally run on larger samples, 10 to 100 ml, and then divided to report the results per ml. This procedure explains the reporting of a fraction of a particle.

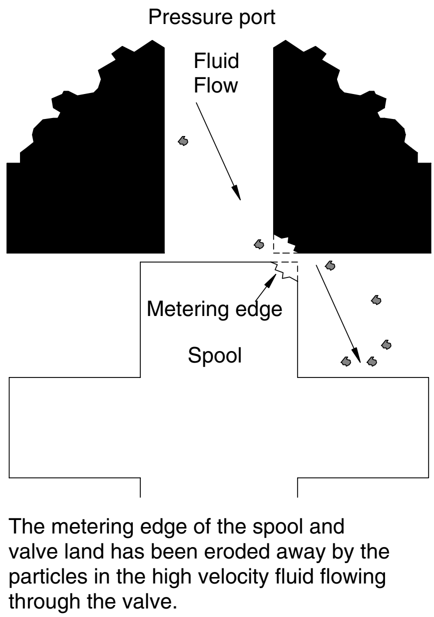


FIGURE 8.5
Erosive wear caused by particles in fluid flowing past a metering edge. (Reprinted with permission from Eaton Hydraulics, Training Div.)

enced to the range code shown on the left vertical axis. A third data point at 2 μm is also shown. Vickers has adopted, and ISO is considering, expanding the code to three ranges corresponding to 2, 5, and 15 μm.

The cleanliness code corresponding to the data in Table 8.2 is 20/**14/2**. In this text, the cleanliness code will be shown with three ranges with the last two in bold type to signify that they are the current ISO standard.

8.3.3 Effects of Contamination on Various Components

8.3.3.1 Pumps

The heart of a hydraulic system is the pump. It is an expensive component with many surfaces that can be scarred, eroded, or chemically altered by contamination. The operating life of most pumps is determined by removal of a very small quantity of material from a few surfaces. Leakage increases through the enlarged clearances, operating temperature goes up, chemical reactions between the metal and fluid increase (which leads to more material loss), and the pump performance quickly degrades.

Lower-pressure units, for example gear pumps, have relatively large clearances, and typically only large (10 μm or larger) particles have a significant damaging effect. Also, at lower pressure, there is less force to drive the particles into the clearances.

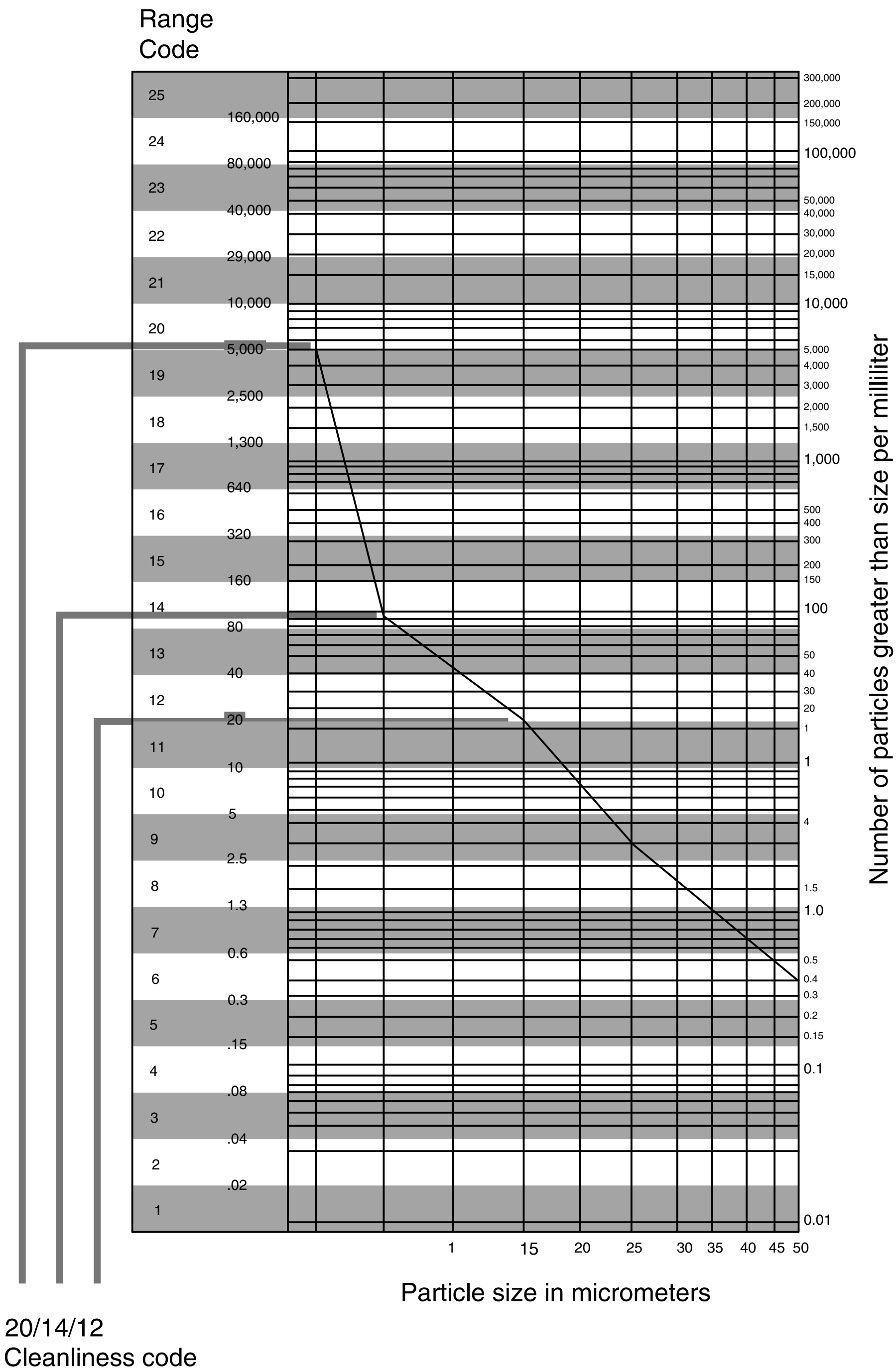


FIGURE 8.6
Cleanliness code chart showing particle distribution data plotted to obtain the cleanliness code.

Gear pumps are more tolerant to contamination because of the larger clearances, but they require careful temperature control. Higher fluid viscosity is needed for the fluid to seal the clearances. If fluid temperature goes above the operating range, gear pump leakage will increase dramatically.

The areas of a vane pump subject to contamination are shown in Fig. 8.7.

- Vane tip to cam ring
- Vane and vane slot both wear as the vane slides in the vane slot
- Side plate (stationary) and rotor (rotating) both wear as particles get in the film between these parts

Vane pump manufacturers generally recommend a higher cleanliness code than that recommended for gear pumps.

The areas of an axial piston pump most subject to contamination are shown in Fig. 8.8.

- *Shoe to swashplate.* As might be expected, there are a number of factors associated with the maintenance of an oil film on the swashplate. The shoes slide across and down this plate as the cylinder block turns.

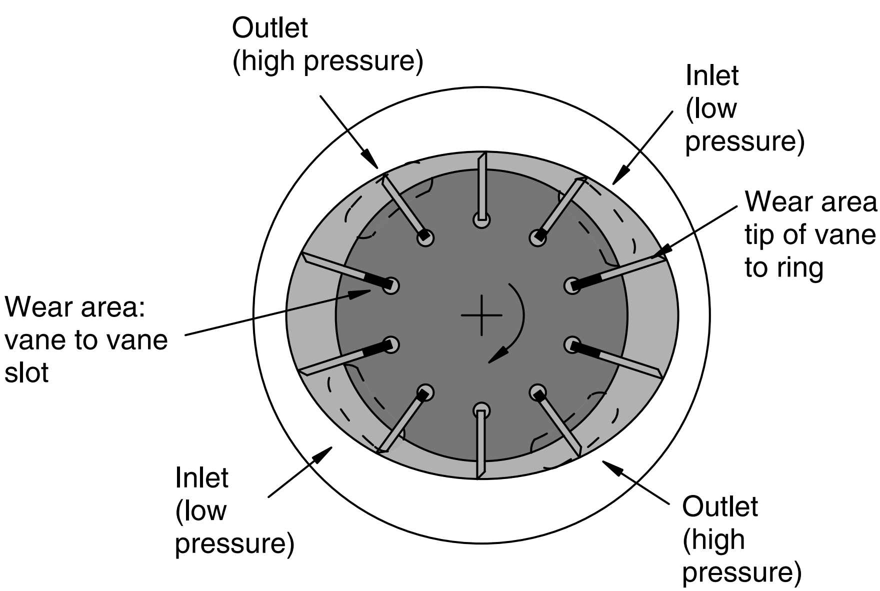


FIGURE 8.7
Critical areas in a vane pump where contamination produces excessive wear. (Reprinted with permission from Eaton Hydraulics, Training Div.)

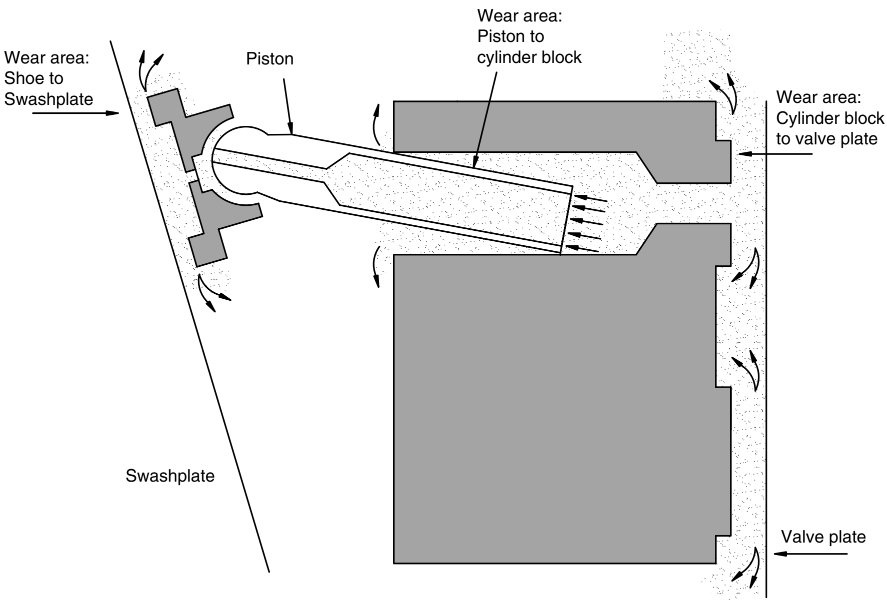


FIGURE 8.8
Critical areas in an axial piston pump where contamination produces excessive wear. (Rotation of piston in the cylinder is greatly exaggerated in this illustration.) (Reprinted with permission from Eaton Hydraulics, Training Div.)

- *Piston to cylinder block.* As shown in Fig. 8.8, the piston tends to rotate slightly as it moves in the cylinder. The film can increase and decrease in thickness due to this action and pump particles through the clearance.
- *Cylinder block to valve plate.* Opportunities for wear here are similar to the opportunity for wear between the side plate and rotor in a vane pump.

8.3.3.2 Motors

Motors are constructed like pumps and have the same critical areas.

8.3.3.3 Directional Control Valves

In spool-type valves, the specified clearance between the bore and spool is typically in the 4- to 13-μm range. Because it is so difficult to produce perfectly round and straight bores, the spool is never exactly centered in the bore. Minimum clearances less than 2.5 μm are found in commercial valves. A single large particle can bridge the clearance and cause the valve to stick.

Another problem with spool-type valves is a phenomenon known as *silting*. Very small particles are forced into clearances by the pressure. When the valve is activated infrequently, the accumulation of this silt will stick the valve, and considerable force is required to break it free. This failure is a good example of an intermittent failure. It is difficult to remove the particles that cause silting using filtration, because they are so small. There are cases in which a preventive maintenance plan will include periodic cycling of valves to interrupt the silting phenomena. Poppet valves are less subject to silting and can be used in applications where spool valves have a history of sticking.

8.3.3.4 Pressure Controls

When a relief valve cracks open, the high-pressure oil escapes through a small opening at high velocity. If this oil has a high population of contaminant particles, these particles erode the spool and opening. All pressure controls are affected by contaminant in much the same way.

8.3.3.5 Flow Controls

The contamination tolerance of flow control valves depends on the orifice configuration. Consider the two orifices show in Fig. 8.9. Both orifices have the same area, but the groove orifice (a) will tolerate a higher contamination level when it cracks open than the flat orifice (b). Actually, the flat-type orifice is more prone to silting at all settings, not just when it cracks open.

8.3.3.6 Summary

High performance components are designed to protect themselves from contaminants to the maximum degree possible. Special materials, surface preparations, and flow paths are used. Pressure pulsation and dynamic loading can cause parts to deflect and allow contaminant to enter clearances. Sometimes unique measures are needed to operate a system at the target cleanliness level.

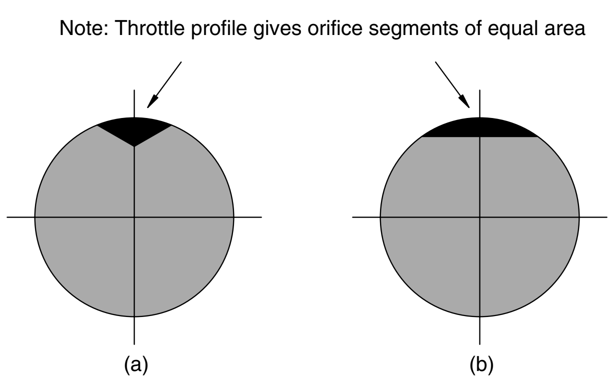


FIGURE 8.9
Orifice types used to throttle flow in a flow-control valve. (Reprinted with permission from Eaton Hydraulics, Training Div.)

8.3.4 Setting a Target Cleanliness Level

A number of factors are considered in setting a target cleanliness level.

1. Components in a system
2. Fluid
3. Start-up temperature
4. Duty cycle
5. System design life
6. Cost of production interruption
7. Safety

The target cleanliness level is assumed to be specified for the return line upstream of the return line filter unless otherwise stated.

Vickers has published a recommended cleanliness code chart, part of which is reproduced in Table 8.3 (Vickers, 1992). This chart is a starting point

TABLE 8.3

Recommended Cleanliness Code Chart

	Pressure (psi)		
Component	**<2000**	**2000–3000**	**>3000**
Pumps			
Gear	20/**18/15**	19/**17/15**	
Fixed Vane	20/**18/15**	19/**17/14**	18/**16/13**
Fixed Piston	19/**17/15**	18/**16/14**	17/**15/13**
Variable Vane	18/**16/14**	17/**15/13**	
Variable Piston	18/1**6/14**	17/**15/13**	16/**14/12**
Valves			
Directional (solenoid actuated)		20/**18/15**	19/**17/14**
Pressure Controls (modulating)		19/**17/14**	19/**17/14**
Flow Controls		19/**17/14**	19/**17/14**
Check		20/**18/15**	20/**18/15**
Proportional, directional		17/**15/12**	16/**14/11**[a]
Proportional, pressure		17/**15/12**	16/**14/11**[a]
Proportional, flow		17/**15/13**	17/**15/13**
Servo		16/**14/11**[a]	15/**13/10**[a]
Actuators			
Cylinders	20/**18/15**	20/18/15	20/**18/15**
Gear Motors	21/**19/17**	20/**18/15**	19/**17/14**
Vane Motors	20/**18/15**	19/**17/14**	18/**16/13**
Axial Piston Motors	19/**17/14**	18/**16/13**	17/**15/12**
Radial Piston Motors	20/**18/14**	19/**17/13**	18/**16/13**

a. Requires precise sampling practicers to verify cleanliness levels.

for setting a target cleanliness level for a specific system. For closed-circuit hydrostatic transmissions, the target cleanliness level is given for the in-loop fluid (Table 8.4).

TABLE 8.4
Recommended Cleanliness Code Chart for Closed-Circuit Hydrostatic Transmissions

Component	Pressure (psi)		
	<2000	2000–3000	>3000
In-loop Fluid	17/**15/13**	16/**14/12**	16/**14/11**[a]

a. Requires precise sampling practices to verify cleanliness levels.

The following steps are used to set a target cleanliness level.

8.3.4.1 Step One

Use the cleanliness code chart (Tables 8.3 and 8.4) to determine the cleanest fluid required for any component in the system. The pressure rating used is the maximum pressure that can be achieved in any circuit that draws from the same reservoir. If several circuits draw from the same reservoir, they are all considered to be part of the same system.

8.3.4.2 Step Two

For any system where the fluid is not 100% petroleum oil, set the target one range code cleaner for each particle size. Example: if the cleanest code required for petroleum oil is 17/**15/13**, and water glycol is the system fluid, the target becomes 16/**14/12.**

8.3.4.3 Step Three

If two or more of the following conditions are experienced by the system, set the target cleanliness code one level lower for each particle size.

- Frequent cold starts at less than 0°F
- Intermittent operation with fluid temperatures over 160°F
- High shocks to the system
- Critical dependence on the system as part of a manufacturing sequence
- Malfunction will endanger operator or others in area

If Step Two is used to reduce the code one cleanliness level, Step Three will lower it one *additional* cleanliness level. The use of a non-petroleum fluid lowered the level from 17/**15/13** to 16/**14/12**, and if two of the Step Three conditions are satisfied, the level will be lowered to 15/**13/11**.

8.3.5 Achieving a Target Cleanliness Level

There are four major factors in positioning a contamination control device in a hydraulic system:

- Initial filter efficiency
- Filter efficiency under system operating conditions
- Location and sizing of filters in the system
- Filter element service life

8.3.5.1 Filter Efficiency

The function of a filter is to remove contaminant from the fluid. The international standard for rating the efficiency of this removal is the Multipass Filter Performance Beta Test (ISO 4572) (Fig. 8.10). The results are reported as a ratio of the number of particles greater than a certain micron diameter upstream of the test filter compared to the number of these particles downstream of the test filter. This ratio is called the *Beta ratio*.

A listing of removal efficiencies for various Beta ratios is given in Table 8.5. A Beta ratio of 2 means that 50% of the particles of a certain size are removed by the filter, and a Beta ratio of 10 means that 90% are removed. A Beta ratio of 100 is often used as a reference point; this ratio means that 99% of the particles of a given size are removed.

Often, the particle size is shown as a subscript in the Beta rating. For example, $B_{10} = 100$ means that 99% of the particles larger than 10 μm were removed. $B_5 = 20$ means that 95% of the particles larger than 5 μm were removed.

There are a number of filter manufacturers and a number of different designs. Generally, the construction of a filter includes two layers of stainless

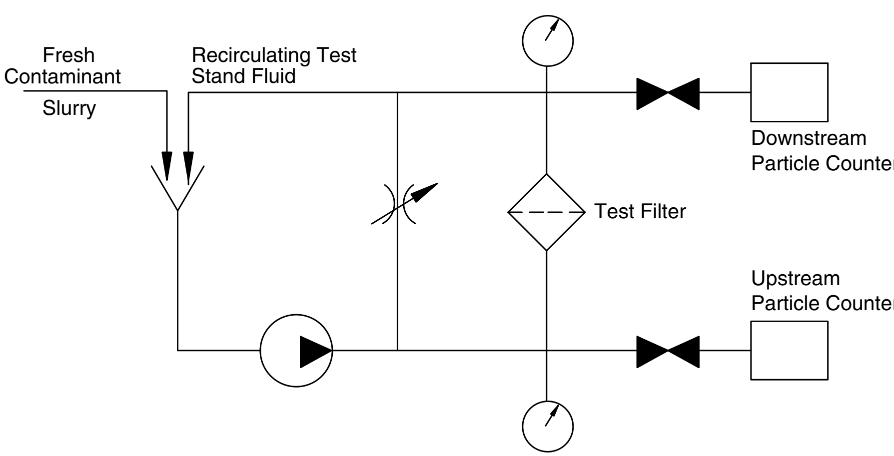


FIGURE 8.10
Schematic for *Multipass Filter Performance Test.*

TABLE 8.5

Filter Efficiencies for Various Beta Ratios Determined with the "Multipass Filter Performance Beta Test (ISO 4572)

Beta ratio	Efficiency percentage
1	0
2	50
5	80
10	90
20	95
75	98.7
100	99
200	99.5
1000	99.9
5000	99.98

steel mesh with multiple layers of specially treated material captured between these layers. This "sandwich" of layers is folded in a given pattern, and the oil is channeled to flow through the layers along a given pathway. Proper support of the layers is critical in filter construction. If the folds collapse together, the pressure drop across the filter will increase. Depending on filter construction, filter efficiency may increase or may decrease. Also, when the folds flex due to pressure pulses or vibration, fatigue of the materials is accelerated. The downstream mesh layer is critical for containment of debris if the layers break down.

Canister-type filters fit in a steel housing. The housing is removed, and the filter element is replaced. With screw-on filters, like the engine oil filter on most cars, the entire unit is replaced.

8.3.5.2 Filter Efficiency under System Operating Conditions

The difference between test conditions and field conditions is dramatic, as shown in Table 8.6. The selection of a filter with the correct Beta ratio is an important first step. This step must be followed with a fluid testing program that monitors filter performance over time.

8.3.5.3 Location and Sizing of Filters

The first goal of filtration is to prevent the ingression of contaminant. All air entering the reservoir needs to be filtered with a filter designed to remove particles 3 μm and larger. It is much easier to remove contaminant from the air than from the fluid. All fluid entering the reservoir must be filtered. The best way to do this is with a transfer cart. This cart has a pump, filter, and supply of fluid. The pump pumps fluid through the filter into the reservoir; thus, any contaminant in the replacement fluid is removed prior to it entering the reservoir.

TABLE 8.6

Comparison of Operating Parameters "Seen" by Filter during Test and Field Conditions

Parameter	Test	Field
Pressure change	One gradual rise	Thousands of cycles
Fatigue cycles on materials	One	Millions
Element aging	Minutes	Months
Contaminant	AC fine test dust	Range of solid, liquid, and gaseous materials
Challenge rate	Constant	Variable
Fluid used	MIL 5606	Wide variety
Temperature	100°F	–20 to 200°F
Flow	Steady	Variable

There are three locations for placement of a filter.

- pressure line
- return line
- offline

A pressure line filter (Fig. 8.11) traps the wear particles from the pump and thus protects all downstream components. Pressure line filters are recommended for

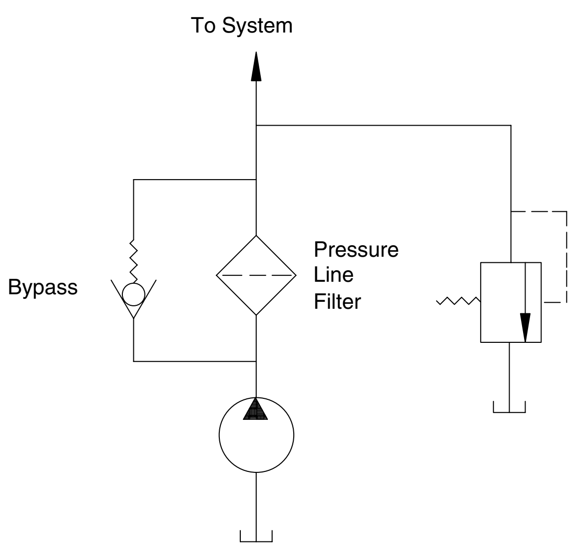


FIGURE 8.11
Circuit with pressure line filter.

- All circuits operating at 2250 psi or above
- All circuits supplied with variable displacement pumps and operating at greater than 1500 psi
- All circuits with servo or proportional valves

Pressure line filters are more expensive, because the canister must be designed for maximum operating pressure.

The return-line filter (Fig. 8.12) is a low-pressure filter and thus does not need to be mounted in a pressure vessel. The return line is an acceptable location for the main system contamination control filter as long as it "sees" at least 20% of system flow during each minute of operation.

Fluid power is filled with little quirks that trip up the designer. One of these is illustrated in Fig. 8.12. Suppose the cylinder has a 2:1 area ratio. Now the return-line filter must be sized to handle twice the pump flow. During retraction, each unit of displacement produces twice the pump flow rate out the cap end of the cylinder. This flow passes through the return-line filter; thus, it must be sized for twice the pump flow rate.

Off-line filtration is needed when there are long periods of time when the pump is in compensation, meaning that the displacement has been shifted back to a minimum, and the pump is maintaining system pressure at very low flow. With off-line filtration, a separate pump continuously flows fluid from the reservoir, through the filter, and back to the reservoir. As shown in Fig. 8.13, cooling can be accomplished simultaneously with off-line filtration.

Component isolation devices are used to protect high-value components. A strainer is relatively inexpensive and can protect a pump from larger particles that inadvertently get into the reservoir. In addition, strainers can be used to protect downstream valves from the larger pieces of debris that result when a pump fails.

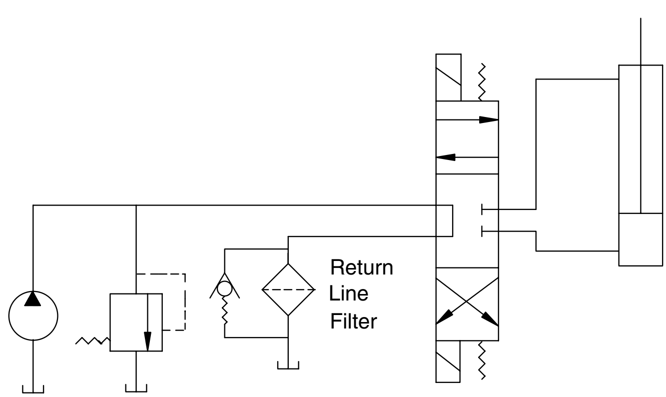


FIGURE 8.12
Circuit with return-line filter.

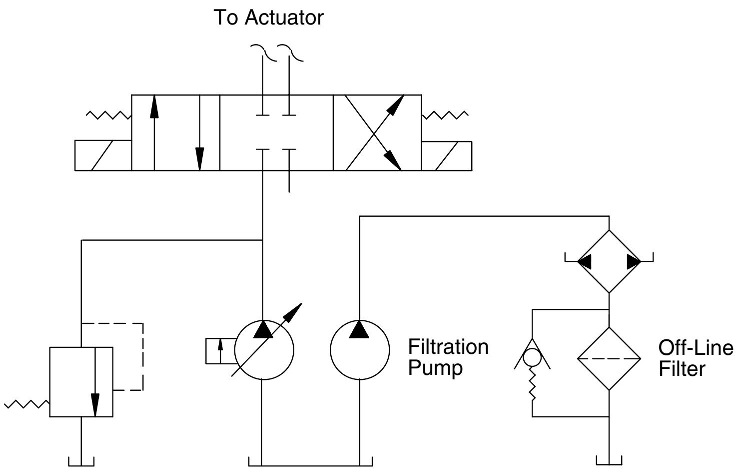


FIGURE 8.13
Circuit with off-line filtration.

Servo and proportional valves all have close-tolerance spools that must be protected from silt. Non-bypass filters are used immediately upstream of the valve (Fig. 8.14). Note that a return-line filter is used as the main system clean-up filter. The component isolation filters should not have a finer rating than the clean-up filter. If it does, it will have the main job of cleaning the system, and its life will be shortened.

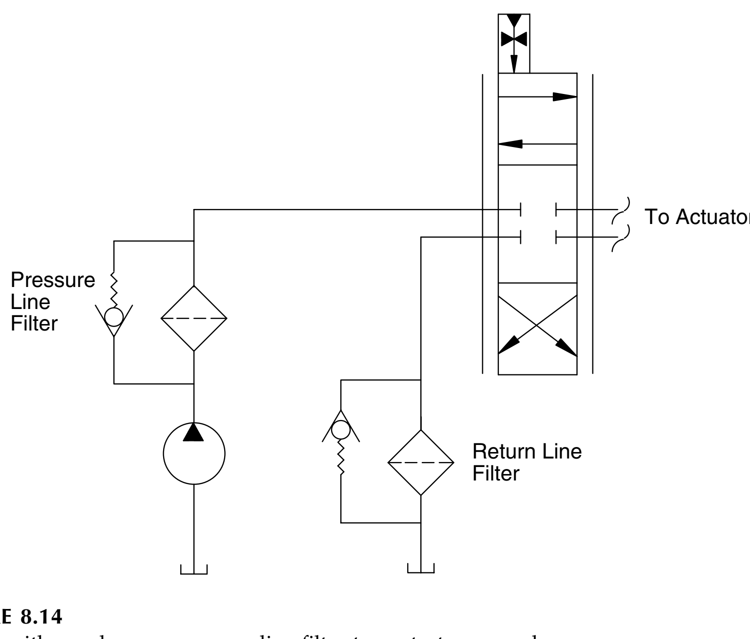


FIGURE 8.14
Circuit with non-bypass pressure-line filter to protect servo valve.

It is not recommended that a return-line filter be used to filter the case drain flow from a pump. (In the closed-circuit hydrostatic transmission, the pump case drain flows through the motor housing and then back to the return.) The pump shaft seals experience accelerated wear when additional back pressure is applied to the pump case because of the pressure drop across the filter.

The key issue in a closed-circuit hydrostatic transmission is the *in-loop* fluid cleanliness. The circuit shown in Fig. 8.15 has a strainer to prevent large particles from entering the charge pump, a low-pressure filter to ensure that only clean fluid enters the closed circuit, and a pressure-line filter to clean in-loop fluid during operation. For bidirectional operation, where approximately 50% of the duty cycle is in each direction, two pressure-line filters should be used, one on each side of the loop.

In many hydrostatic transmissions, the charge pump, charge relief valve, and check valves are all internal. Provision for the filter in the charge pump outlet must be provided by the manufacturer; the end user has no way to plumb this filter into the circuit.

8.3.5.4 Filter Condition Indicators

How will the user know when to change the filter? One option is to replace filters after a certain number of operating hours. This option is not a good choice because of the wide range of operating conditions a system can experience as compared to a test condition (Table 8.5). ANSI standard T2.24.1-1991 recommends that all filters be fitted with a differential pressure indicator.

A typical pressure drop vs. time curve for a filter is given in Fig. 8.16. This indicator should change at a ΔP corresponding to 95% operating time. In other words, the indicator change shows the operator that 5% of the total operating time remains before the bypass valve cracks open.

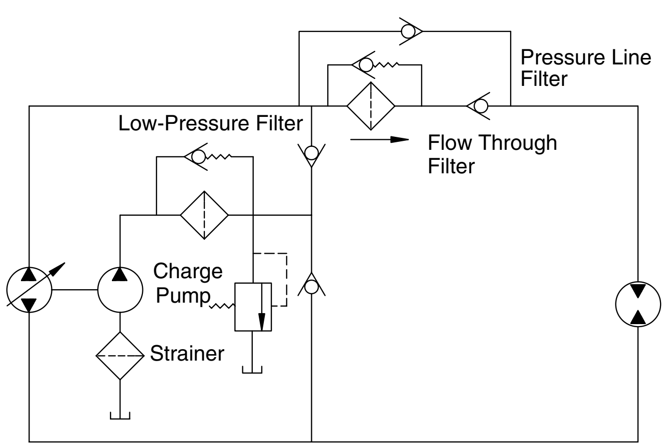


FIGURE 8.15
Recommended filter placement for closed-circuit hydrostatic transmission with a high percentage of the operation in one direction.

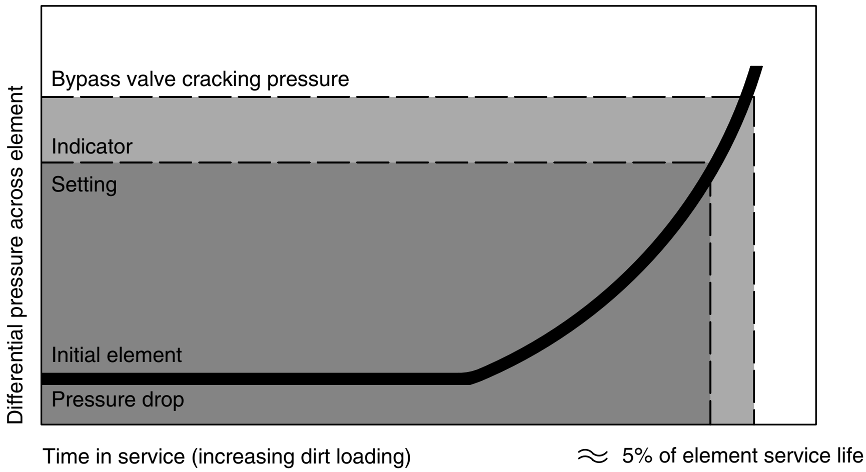


FIGURE 8.16
Typical change in pressure drop across a filter as it fills with contaminant over time. (Reprinted with permission from Eaton Hydraulics, Training Div.)

Maintenance records should record the average life of each filter. If filter replacement is required at frequent intervals, steps should be taken to reduce ingression. Is the reservoir vent properly filtered? Can something be done to shield cylinder rods from dust and dirt? (There are flexible shields that can be placed over the rod. These shields accordion back and forth as the rod extends and retracts.) Are access ports kept sealed? If ingression is reduced as much as possible, and the filter replacement interval is still too short, use a larger filter with greater dirt-holding capacity. Maintenance labor cost is reduced by extending the replacement interval. Larger elements are often more cost effective as opposed to smaller elements that are replaced more frequently.

It is very important to keep the fluid cleanliness below the target level. If the fluid gets dirty, wear on all component surfaces will increase, and the resulting particulates will further damage component surfaces. *Remember the wear regeneration cycle.*

8.3.5.5 System Flushing

The most critical time in the life of a hydraulic system is the initial *run-in* period. Debris is left in components by the manufacturing process (even though a cleaning procedure is done before the components are shipped), and other debris is added during assembly. It is critical that this contaminant be removed before the system is put under load.

New system flushing should accomplish three tasks: (1) dislodge the contaminant and transport it to the filter, (2) activate all circuits to flow fluid through all lines and components, and (3) capture the contaminant with a high-efficiency filter. Flushing is best accomplished with a low-viscosity fluid pumped at high velocity. It is best to use a flushing cart with its own pump and fluid reservoir. Be sure to remove the system pump and any contaminant intolerant components such as servo valves or proportional valves before flushing.

In cases where it is not practical to use special flushing fluid, the actual system hydraulic fluid can be used. Some means is needed to increase its temperature (and thus reduce its viscosity) to a range that will give good flushing. It may be necessary to route a line around a component to get the necessary fluid velocity through that line.

It is recommended that the flushing target cleanliness level be two ISO codes below the target cleanliness level for system operation. When clean oil is added (using a filter cart) to a system that has been properly flushed, it takes less time for the system to reach a cleanliness equilibrium.

8.3.6 Monitoring the System Cleanliness Level

Once the target cleanliness level has been set, and the filters have been selected and located in the system, the last and ongoing step is to confirm and monitor the target level. After the system has operated for a period of time (maybe 50 h) in the load environment, collect a sample of fluid from the return line ahead of the filter, and send it to a laboratory that reports particle counting in accordance with ISO 4406. After verifying that the contamination control plan is working, samples should be taken at periodic intervals (maybe more frequently than 500 h, or maybe less frequently, depending on the management objective for component life) and these samples analyzed to ensure that the target cleanliness level is being maintained. Remember that filters *are being replaced* when flagged by the filter indicators during the intervals between collection of oil samples.

It is good design practice to install a sampling port in the return line immediately upstream from the filter. This will facilitate collection of the sample. An alternative sampling location is the reservoir. Use a vacuum pump to pull fluid from a point about halfway down in the fluid, otherwise stratification within the reservoir can cause the sample to be nonrepresentative.

Some fluid power companies publish a guide for the establishment and maintenance of a given cleanliness level. These guides more fully develop the concepts presented in this chapter and can be used with confidence. There is continuing emphasis on contamination control by all manufacturers, particularly now that fluid power is being used for more and more sophisticated motion control functions.

8.4 Summary

Hydraulic fluid has four functions: (1) transmit power, (2) lubricate, (3) seal clearances, and (4) provide cooling. Most of the material in previous chapters dealt with the power transmission function. In this chapter, we discussed how to maintain the fluid properties needed for the other three functions.

Since viscosity is a function of temperature, temperature control provides viscosity control. Maintenance of an oil film between moving parts is important for all machinery, and it is critical for pumps and motors operating at high speeds and pressures. Fluid viscosity must be maintained within a given operating range to ensure that fluid will flow into clearances and establish the needed lubricating film and then seal these clearances to prevent excessive leakage.

It is recommended that a hydraulic system be designed for a maximum operating temperature of 140°F. Heat is exchanged to the surrounding atmosphere by conduction, convection, and radiation from the lines and components. If additional heat exchange is needed, the first step is to increase the surface area of the reservoir. If the additional heat exchange is not sufficient, an oil cooler is added. Oil-to-water exchangers are used for stationary applications and oil-to-air exchangers for mobile applications.

Proper selection and placement of contamination control devices will eliminate up to 80% of all system failures. Sources of contamination are built-in, contaminated new oil, ingressed, and internally generated. Particles removed from the interior surface of components will circulate in the system until they are removed. Each impact of one of these particles produces more particles, a phenomena known as the *wear regeneration cycle.*

A laboratory particle counting procedure that defines the size distribution of particles in a fluid sample is defined by ISO Standard 4406. These data are the basis for establishing a target cleanliness code. Filters are placed in the hydraulic system to achieve and maintain the desired cleanliness level.

Normally, filters are placed in the return line. Components with low contamination tolerance, like servo valves and proportional valves, require a pressure-line filter immediately upstream of the component. All filters should have a pressure-drop indicator to indicate when the filter is at 95% of its dirt holding capacity.

The international standard for rating the efficiency of contaminate removal is the Multipass Filter Performance Beta Test (ISO 4572). The ratio of the number of particles greater than a certain micron diameter upstream of the test filter compared to the number downstream is known as the *Beta ratio.*

The most critical time in the life of a hydraulic system is the initial run-in period. System flushing removes debris remaining from manufacture and assembly and reduces the time for a system to reach cleanliness equilibrium.

Once the target cleanliness level has been set and the filters have been selected and located in the system, a monitoring program is put in place to ensure the target cleanliness level is maintained. Samples are typically collected from the return line immediately upstream from the filter and sent to a laboratory that reports particle counting data in accordance with ISO 4406.

Control of fluid properties provides a number of economic advantages. Extension of component life and reduction of unscheduled downtime are the two most important. Also, the extension of oil life reduces both new oil cost and disposal cost for used oil.

References

Merritt, H.E. 1967. *Hydraulic Control Systems.* John Wiley & Sons. New York.

Vickers. 1992. *Vickers Guide to Systemic Contamination Control.* Vickers, Inc., PO Box 302, Troy, MI 48007-0302.

Problems

8.1 A system has a 2.44 in^3/rev displacement pump driven at 1730 rpm. Average system pressure is 930 psi. It is estimated that the system is 65% efficient, meaning that 35% of the hydraulic energy delivered by the pump is converted to heat energy.

The overall heat transfer coefficient for the reservoir is estimated to be $U = 5$ Btu/ft^2·h·F. It is desired that the oil temperature not exceed 140°F on a day when the ambient temperature is 95°F. If half the total heat generated is dissipated from the lines and the surface of the components, what size reservoir is required to dissipate the remaining heat?

8.2 A mobile machine has 47 ft of 0.75 OD steel tubing that connects the pump and actuators. The pump is a tandem unit with a variable displacement front section with 1.83 in^3/rev maximum displacement and a fixed displacement gear pump rear section. The gear pump has a displacement of 3.16 in^3/rev. The variable displacement pump is identified as Pump A and the gear pump is identified as Pump B.

The charge pump displacement in Pump A is 0.69 in^3/rev, and the charge relief is set on 260 psi. Case pressure in Pump A is 40 psi. Pumps A and B are driven at 2000 rpm. Average operating pressure of the Pump A circuit (Pump A and motor operating as a closed circuit HST) is estimated to be 1150 psi. Average operating pressure of the Pump B circuit is estimated to be about 600 psi.

Total surface area of the housings of the various components are given in Table 8.7.

For this problem, use the empirical equation given in Eq. (8.5) for the Pump B circuit. You may assume that 25% of the Pump A charge pump flow drops across the charge relief valve, thus the pressure drop is

TABLE 8.7

Surface Areas

Component	Surface area (ft^2)
Pump A	2.478
Pump B	1.784
Motor	1.08
Cylinder 1	1.854
Cylinder 2	2.71
Cylinder 3	6.497
Directional control valve	1.261

$$\Delta P_{chrg1} = \text{charge relief pressure–case pressure}$$

$$= 260 - 40$$

$$= 220 \text{ psi}$$

The remaining charge pump flow replaces leakage from the main circuit, thus the pressure drop is

$$\Delta P_{chrg2} = \text{main circuit pressure–case pressure}$$

$$= 1150(\text{avg}) - 40$$

$$= 1110 \text{ psi}$$

It is appropriate to use an overall heat transfer coefficient U = 3 Btu/(h·ft^2·F) for all surfaces on the machine. Ambient temperature is 95°F, and we desire for the maximum fluid temperature to be 140°F.

1. Calculate the total rate of heat generation by the Pump A and Pump B circuits.
2. Calculate the total heat dissipated by the lines and components.
3. If the reservoir area is 28 ft^2, determine the capacity of an air cooled heat exchanger to dissipate the heat not dissipated by the lines, components, and reservoir.

8.3 A mobile machine has two hydrostatic transmissions. One hydrostatic transmission propels the machine, and the other transmission powers a rotary mechanism on the machine. A diesel engine drives a twin pump mount at 2000 rpm. Pump A (propel transmission) is mounted on the right side of the pump mount, and Pump B (rotary mechanism transmission) is mounted on the left side. These transmissions are specified as follows:

Pump A: Variable displacement pump
4.57 in^3/rev maximum displacement
Fixed displacement motor
4.57 in^3/rev displacement
Charge pump displacement 1.03 in^3/rev
Charge pump relief 350 psi
Case pressure 40 psi

Pump B: Fixed displacement pump
2.56 in^3/rev displacement
Fixed displacement motor
2.56 in^3/rev displacement
Charge pump displacement 0.86 in^3/rev
Charge pump relief 290 psi
Case pressure 40 psi

Average operating pressure for the Pump A circuit is estimated to be 1600 psi. Pump and motor overall efficiencies are estimated to be 93.5%, and volumetric efficiencies are estimated to be 97.5%. The pump will operate most of the time at 75% of maximum displacement.

Average operating pressure for the Pump B circuit is estimated to be 3600 psi. Pump and motor overall efficiencies are estimated to be 92% and volumetric efficiencies are estimated to be 95.5%.

Maximum ambient temperature is 95°F, and we must design for a maximum fluid temperature of 140°F. You may use the following assumptions:

1. The overall heat transfer coefficient for all heat transfer surfaces is $U = 30$ Btu/h·ft²·F.
2. 25% of the total heat generated is dissipated from the surface of the lines and components:
   a. Calculate total heat generated by the two hydrostatic transmissions.
   b. Design the reservoir with a capacity equal to four times the total reservoir flow (both circuits). You may use any shape you like. Calculate the total heat dissipated by the system (q_{sys}) and reservoir (q_{res}). If additional heat exchange is needed, write the specifications for a finned-tube oil cooler for the machine.

8.4 An axial piston pump, $V_p = 0.933$ in^3/rev, was tested using AC Fine Test Dust (ACFTD) under controlled laboratory conditions. The pump was operated at 1750 rpm and developed 3000 psi for all tests. Output flow was measured with a turbine-type flow meter. Fluid temperature was maintained at 125°F.

Test 1

First, the pump was operated for one hour as a break-in period. Clean fluid was used and the measured flow was Q_o = 6.57 GPM.

Test 2

Fluid was prepared by adding ACFTD with a 0–5 micron size range to a level of 300 mg/L. The pump was challenged with this fluid and operated for 30 min. Measured flow was Q_5 = 6.56 GPM.

Test 3

Same contaminant loading rate with 0–10 micron ACFTD. Q_{10} = 6.55 GPM.

Test 4

0–20 micron ACFTD. Q_{20} = 6.50 GPM.

The remaining tests are summarized in the table below:

Contaminant (ACFTD 0-*x* micron)	**Delivered flow (GPM)**
0	6.57
5	6.56
10	6.55
20	6.50
30	6.45
40	6.40
50	6.15
60	5.75
70	5.25
80	4.60

For each test, the pump was run until the flow rate stabilized, indicating that the clearances had been eroded, and thus enlarged sufficiently for the contaminant to pass through. Erosion continues due to the wear regeneration cycle but at a much slower rate.

1. Calculate the volumetric efficiency and plot it as a function of contaminant size.
2. Calculate the orifice coefficient (k) for the leakage pathways in the pump.

$$Q_l = k\sqrt{\Delta P}$$

where Q_ℓ = leakage flow (GPM)
ΔP = pressure rise across pump (psi)

Report your results in a table with columns for volumetric efficiency and the orifice coefficient. Think about the design of an axial piston pump and explain why the pump performance degrades as a nonlinear function of contaminant particle size.

8.5 Design a circuit with correct contamination control for the machine in Problem 8.3.

Pump A Circuit

Variable displacement axial piston pump

Axial piston motor

Operating pressure (average) 1600 psi

Circuit is used to propel the machine in the forward and reverse directions.

Pump B Circuit

Fixed displacement axial piston pump

Axial piston motor

Operating pressure (average) 3600 psi

The Pump B circuit is used to power a rotary mechanism that turns in one direction. Petroleum oil is the fluid for the system, and both pumps pull fluid from the same reservoir. The temperature control ensures that the fluid temperature does not exceed 140°F. The machine will not be operated where cold starts are required, and it will not be subjected to dynamic maneuvers where high shocks are required.

1. Select the cleanliness code required for both circuits.
2. Draw circuit diagrams for both circuits showing the completed circuit with filters in place.

9

Auxiliary Components

9.1 Introduction

The objective of this chapter is to give the reader some practical understanding of the hardware needed to connect the various components (pumps, actuators, valves, accumulators, filters, oil coolers) in a circuit. Just as the two key parameters in fluid power are P (pressure) and Q (flow), the two key issues in the design of the lines between components are sizing for the recommended maximum fluid velocity (related to Q) and selection of the pressure rating (P).

There are two pressure ratings of interest to the designer: working pressure and burst pressure. Conductors are chosen that have a working pressure rating greater than the expected maximum pressure in the system. (Typically, this maximum pressure is the relief valve setting.) Burst pressure is the pressure at which the conductor is expected to rupture.

The only essential component not previously discussed is the reservoir. It is expedient to present the key design issues relative to the reservoir before beginning our discussion of conductors and fittings.

9.2 Reservoir

In addition to holding the supply of fluid needed to ensure that all lines and components are completely filled with fluid at all times, the reservoir has four other functions:

1. *It separates entrained air.* Problems caused by entrained air were discussed in Chapters 2 and 8. Dwell time in the reservoir provides opportunity for air bubbles in the fluid to rise to the top and burst on the surface.

2. *It dissipates pressure pulses.* Circuits with several actuators and random actuation of these actuators can have significant pressure pulses in the return line. As discussed in Chapter 8, these pulses can degrade filter performance. Off-line filtration eliminates the influence of these pulses on the filter, because they are dissipated by the reservoir.
3. *It provides cooling.* Heat exchange from the reservoir to the surroundings was discussed in Chapter 8.
4. *It traps contaminant.* If the reservoir has to trap contaminant, the filtration is not working correctly. The reservoir does trap contaminant when the filter bypasses.

9.2.1 Reservoir Construction

It is recommended that the reservoir capacity be one to three times the pump output. For example, if total fluid withdrawal (one or more pumps) is 10 GPM, then the capacity should be $3 \times 10 = 30$ gal. On mobile machines, the size is often less than three times the pump flow, but it should never be less that one times the total pump flow.

The reservoir must be sized to hold all fluid from the cylinders when they are fully retracted. When all cylinders are fully extended, the level of fluid in the reservoir must still be above the suction line to the pump.

A reservoir with maximum surface area per unit volume gives the best heat exchange. If space permits, a tall thin reservoir (Fig. 9.1a) gives better heat exchange than a rectangular reservoir (Fig. 9.1b). On mobile machines, there are cases in which a structural member, for example a long piece of rectangular tubing, can be used as the reservoir. The ends are closed, and suitable fittings are welded in place. The member then serves its structural function and also serves as a reservoir for the hydraulic system.

The reservoir location should be chosen such that the distance h between the fluid level and pump inlet is as great as possible (Fig. 9.2). A larger h reduces the potential for cavitation. Some modular units (Fig. 9.3) have the pump mounted on top of the reservoir. In this case, the pump must provide enough suction to lift the fluid into the pump.

Since maximum heat exchange from the reservoir is an objective, it should be mounted such that there is good air circulation across all exterior surfaces. Try not to install a reservoir where it will receive radiant or convected heat from other equipment. Paint the exterior with a light colored paint to minimize absorption of radiant energy when the reservoir has to be exposed to sunlight, as on a mobile machine.

Both inlet and return lines should be submerged. If the return flow jets across the surface of the fluid as shown in Fig. 9.4, air will be entrained in the fluid. In extreme cases, enough foam will be produced that flecks of foam will exit the vent cap.

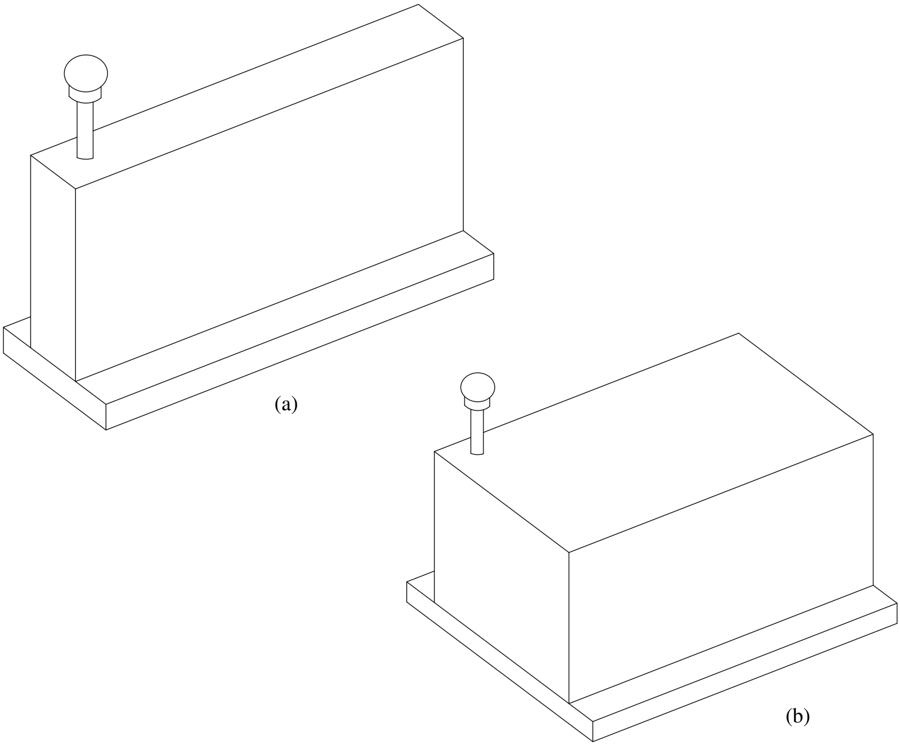


FIGURE 9.1
(a) Flat-type and (b) rectangular reservoirs.

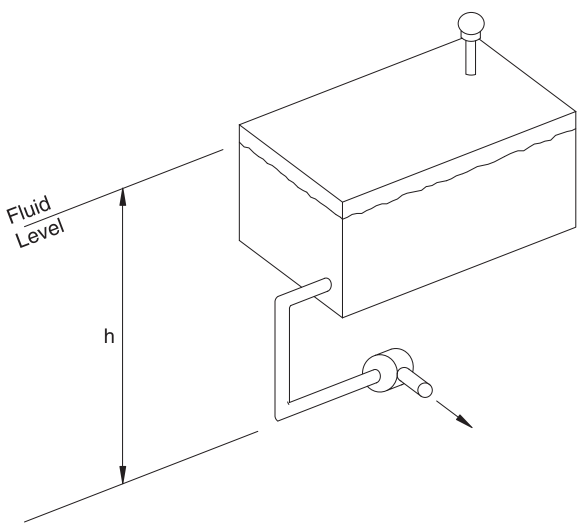


FIGURE 9.2
Preferred location of reservoir relative to pump inlet.

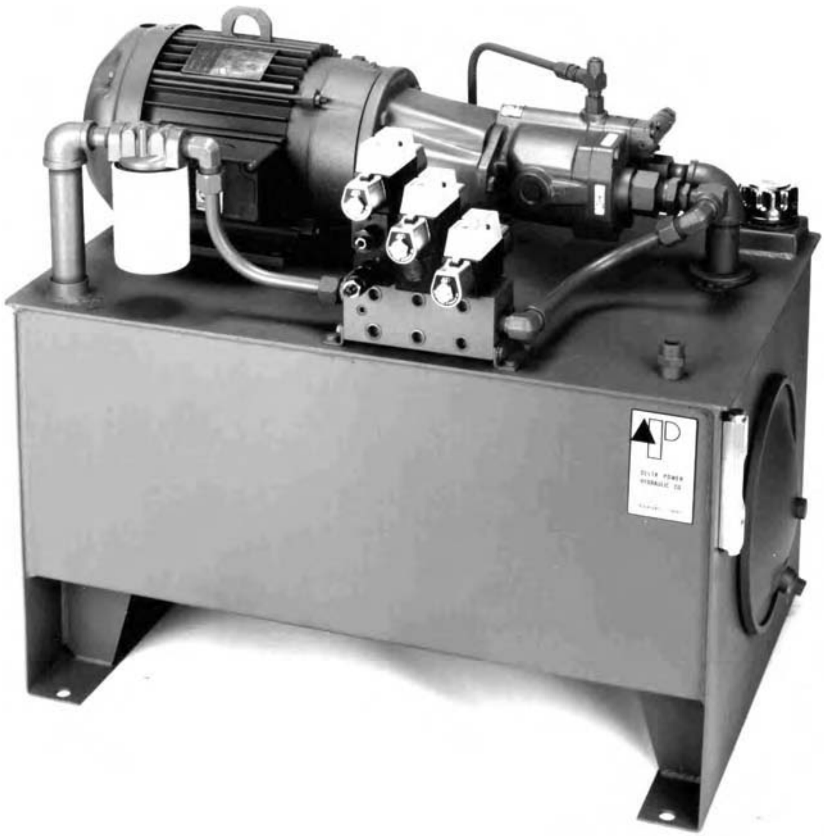


FIGURE 9.3
Power unit with pump, reservoir, filter, and valving manufactured as a complete assembly. (Courtesy of Delta Power Hydraulic Co., Rockford, IL.)

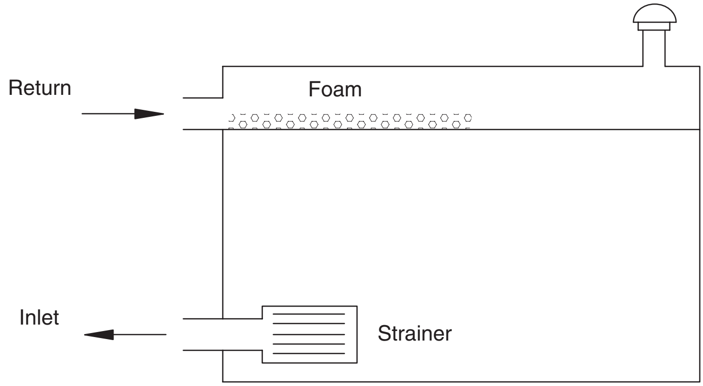


FIGURE 9.4
Potential problem with foaming of the fluid when the return line is not submerged.

The inlet should be some distance above the bottom of the reservoir to minimize the potential for ingestion of contaminants that have settled on the bottom. It is recommended that a strainer always be used at the inlet. The strainer is a 25-mesh screen that protects the pump from large pieces of debris. It is not a filter and typically is not cleaned or replaced except when the system fluid is replaced. A drain plug should be located such that all oil can be drained from the reservoir.

Thorough mixing of the oil in larger reservoirs prevents temperature gradients and improves heat exchange. Two baffle patterns are shown in Fig. 9.5.

If it is not possible to provide a large enough *h* to completely fill the pump at design operating speed, it may be necessary to pressurize the reservoir using a bladder (Fig. 9.6). Here, fluid from retracted cylinders compresses the gas in the bladder. When the cylinders extend, the bladder expands. The bladder is sized to provide the needed change in fluid volume in the reservoir. Pressure change may range from 25 to 50 psi. These are low pressures, but they produce a significant load on the sides of the reservoir; thus, it must be designed for these loads.

The pressurized reservoir eliminates the exchange of air into the reservoir. This eliminates a pathway for dirt ingress, and it also eliminates the entrance of water vapor into the reservoir, which subsequently condenses on the interior surface and drops water into the oil. These advantages are offset by the higher cost of the pressurized reservoir. In certain applications, a cost comparison must also include the option of using a booster pump, as described

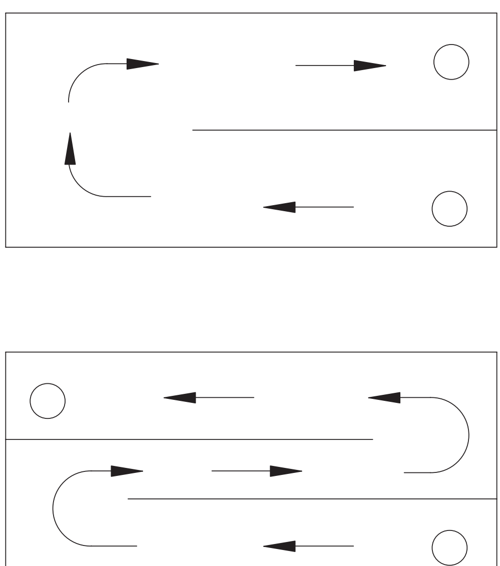


FIGURE 9.5
Baffles used to prevent stratification in a reservoir.

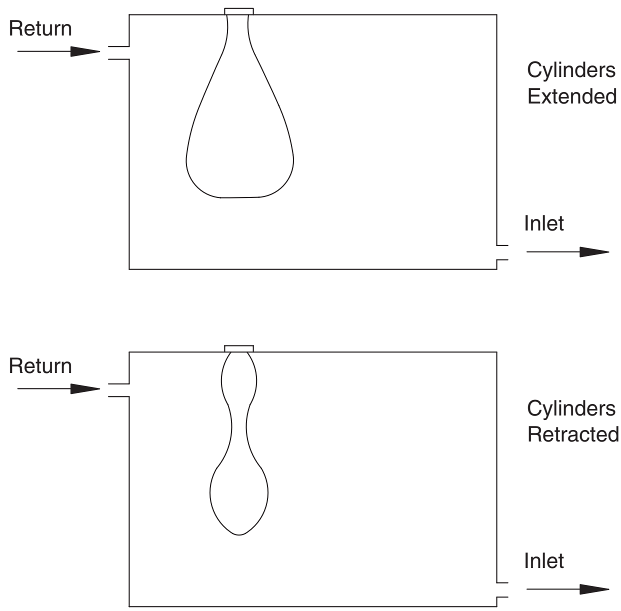


FIGURE 9.6
Pressurized reservoir.

in Chapter 5, rather than a pressurized reservoir to ensure that the system pump is always filled with oil.

The fluid mechanics equations presented in Chapter 2 were used to investigate the filling of a gear pump as speed is increased. The model calculated the absolute pressure in a gear tooth space of a gear pump at different speeds (Table 9.1). At higher speeds, the pump is pumping more fluid and the flow velocity causes a higher pressure drop, which reduces the pressure at the pump inlet. When the pump speed exceeds 3800 rpm, the absolute pressure in the tooth space is only 0.65 psia. At a slightly higher speed, it becomes zero, and thereafter the pump begins to cavitate. Air is compressed along with the fluid with the resultant problems discussed in Chapters 2 and 8.

One possible solution for this particular cavitation problem is to mount the reservoir such that the fluid level is above the pump inlet. Even though this helps to fill the pump, there will still be an upper bound on speed where the pump will not fill completely. The designer must then consider a larger displacement pump driven at a slower speed.

9.3 Hydraulic Lines

There are three types of lines used for pressurized fluid: (1) pipe, (2) seamless tubing, and (3) hose.

TABLE 9.1

Calculated Absolute Pressure in Gear Tooth Space of a Gear Pump Operated at Increasing Speed[a]

Pump Speed rpm	Inlet Line Velocity ft/s	Inlet Line Pressure Drop psi/ft	Absolute Pressure in Tooth Space psia[b]
2000	7.60	0.12	10.28
2200	8.36	0.14	9.48
2400	9.12	0.16	8.61
2600	9.88	0.18	7.68
2800	10.64	0.21	6.67
3000	11.40	0.24	5.60
3200	12.16	0.27	4.46
3400	12.92	0.30	3.26
3600	13.67	0.33	1.99
3800	14.43	0.36	0.65

a. Appreciation is expressed to Charles Throckmorton, Sauer-Danfoss, Inc., for the data used in this table.

b. Calculation assumes the pump inlet is 1 ft above the reservoir fluid level and that there is one 90° elbow in the inlet line.

9.3.1 Pipe

Pipe and tubing are both rigid conductors. Nominal sizes of American standard pipe and pipe fittings are defined by ANSI standard B36.10, 1970. There are four schedules: 40, 80, 160, and double extra heavy. The abbreviation (Sch) is normally used for all schedules. The nominal sizes do not actually exist as an inside diameter (ID) or outside diameter (OD). The outside diameter is held constant for all schedules of a given nominal size, because the threads cut into the OD must always fit those tapped into a mating port or fitting. The wall thickness increases to provide a higher pressure rating (Fig. 9.7).

Pressure ratings for steel pipe are given in Table 9.2. Considering first the Sch 40 pipe, note that the working pressure rating declines for pipe sizes above 1 in. For sizes from 1/4 to 1 in., Sch 40 pipe has a working pressure rating of around 2000 psi. The burst to working pressure ratio declines from 9.3 for 1/4 in. to 5.2 for 2 in.

The same trend is noted for Sch 80 pipe. The working pressure declines for sizes above 1 in. For sizes from 1/4 to 1 in., Sch 80 pipe has a working pressure rating of 3500 psi. At 2 in., the working pressure is 2500 psi. The ratio range (burst-to-working) is lower, 6.0 to 4.4, as compared to a range of 9.3 to 5.2 for Sch 40 pipe.

For Sch 160 pipe, the working pressure declines from 7300 psi to 4500 psi as pipe size increases from 1/2 to 2 in. The burst-to-working ratio is approximately constant at slightly less than 4:1.

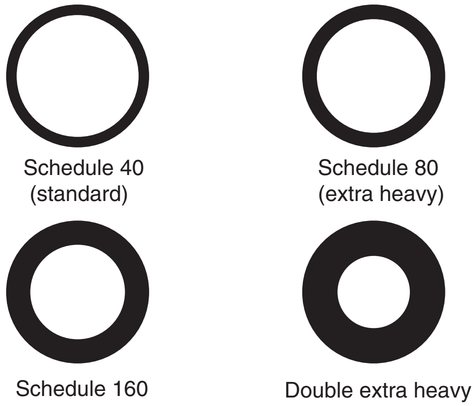


FIGURE 9.7
Four standard pipe schedules.

TABLE 9.2
Pressure Ratings for ASTM A53 Grade B or ASTM A106 Grade B Seamless Steel Pipe

		Pressure (psi)		Ratio
Nominal size	**Schedule no.**	**Working**	**Burst**	**Burst: working**
1/4	40	2100	19,500	9.3
1/2	40	2300	15,600	6.8
3/4	40	2000	12,900	6.4
1	40	2100	12,100	5.7
11/2	40	1700	9,100	5.3
2	40	1500	7,800	5.2
1/4	80	4350	26,400	6.0
1/2	80	4100	21,000	5.1
3/4	80	3500	17,600	5.0
1	80	3500	15,900	4.5
11/2	80	2800	12,600	4.5
2	80	2500	11,000	4.4
1/4	160	–	–	–
1/2	160	7300	26,700	3.6
3/4	160	6500	25,000	3.8
1	160	5700	22,300	3.9
11/2	160	4500	17,700	3.9
2	160	4600	17,500	3.8

Pipe fittings are manufactured as Sch 40, 80, 160 and double extra heavy. The working pressure rating of the pipe fittings *must* match the working pressure rating of the pipe. For example, never use Sch 40 fittings with Sch 80 pipe.

Pipe and pipe fittings have tapered threads. These threads seal by a metal-to-metal contact between the threads in the mating parts. There are two types of tapered threads: NPT and NPTF. The NPT threads engage mating threads on their flanks (Fig. 9.8a). This design leaves a small spiral groove along the thread tips, which must be filled with sealant. There are several types of pipe sealant used to develop this seal. The sealant serves a dual purpose of lubricating the threads, which facilitates disassembly, and of providing the seal. The NPTF threads seal by tightening until the thread crests are crushed enough that the flanks make *full* contact (Figs. 9.8b and 9.8c).

9.3.2 Hydraulic Tubing

Tubing discussed in this chapter is either seamless carbon steel or seamless stainless steel. The stainless steel tubing is for applications in which the external surface is attacked by the surrounding chemical environment, or the fluid itself is corrosive and the inside is subject to attack.

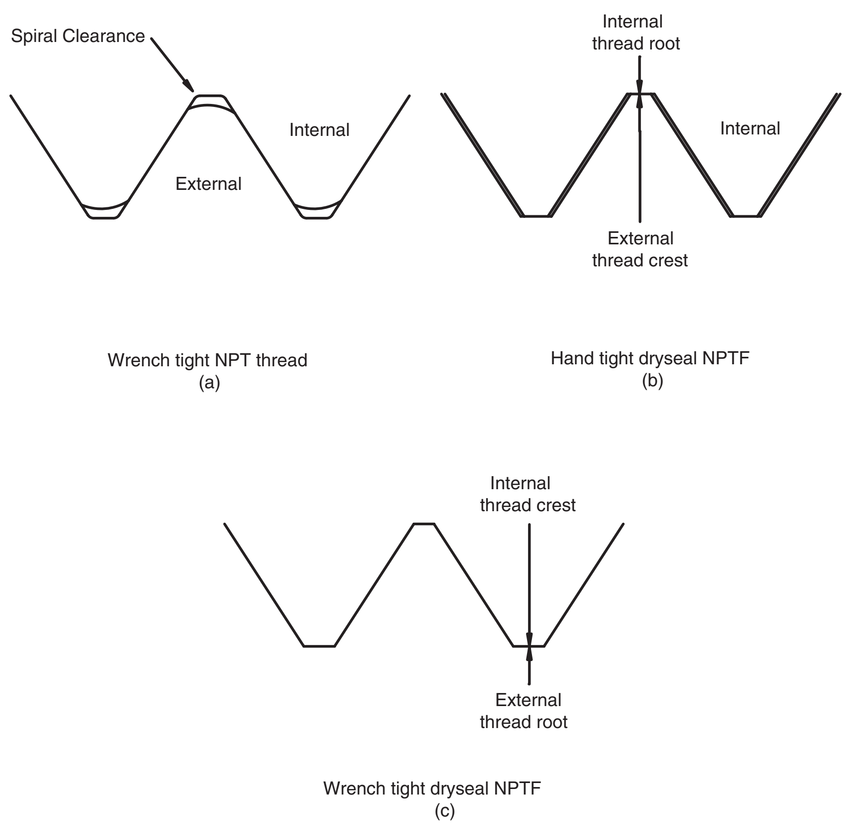


FIGURE 9.8
Examples of sealing with two types of tapered threads.

Hydraulic tubing is specified by its outside diameter (OD). It has a thin wall compared to pipe; thus, methods other than threading have been developed to connect tubing. The main types of fittings are flared, flareless, and welded.

9.3.2.1 Flared Tubing Fittings

Three types of flared fitting designs are shown in Fig. 9.9. The three-piece type is the most widely used for hydraulic circuits. A sleeve is placed on the tubing before it is flared. When the flare nut is tightened, the sleeve absorbs the twisting friction produced by the nut so that only axial forces are exerted against the flared tube. Two-piece fittings, widely used to connect lines for lower-pressure applications, can weaken the tubing by twisting it as the fitting is tightened.

The standard flare angle for hydraulic tubing is 37°. Making the flare too broad or too narrow is a common fabrication error. The correct amount of extension of the flare above the sleeve is shown in Fig. 9.10.

Another common flare angle, used mainly in low pressure automotive and refrigeration applications, is 45° from the center line. Fittings with 37° and 45° flares cannot be mixed. A simple method for determining the flare angle is shown in Fig. 9.11. A business card (any 90° edge will do) is inserted in the end of the tube. If both sides mate with the flare angle, the flare is 45°.

9.3.2.2 Flareless Tubing Fittings

Tubing wall thickness is increased to produce tubing with a higher pressure rating. Flaring becomes more difficult as wall thickness increases, and this problem led to the development of the flareless fittings.

There are a number of different designs, but all of them have some means whereby a ferrule is pressed against the tubing and actually "bites" into the surface. Once this ferrule is seated, it cannot be removed. The ferrule should be seated and inspected for a good seat before assembly of the complete fitting. If a mistake is made, the damaged end of the tubing must be cut off and a new ferrule installed. Flareless fittings will leak if under-tightened or over-

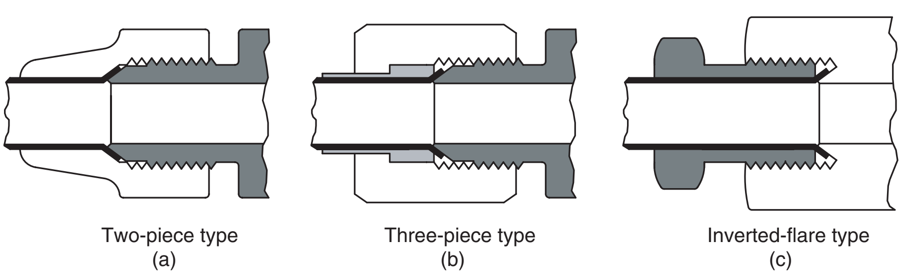


FIGURE 9.9
Three types of flare fitting designs.

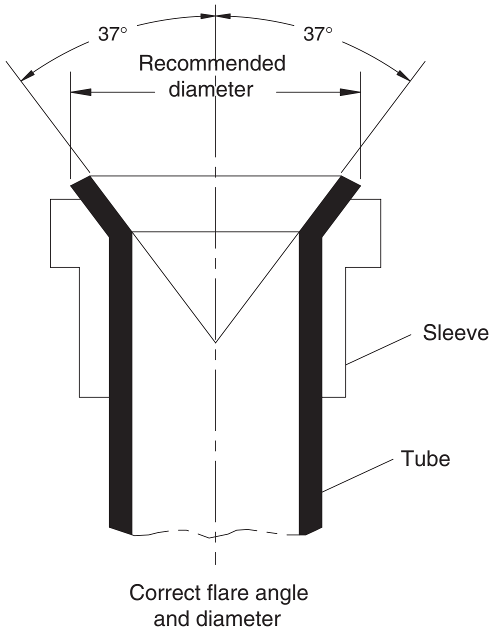


FIGURE 9.10
Recommended flare for hydraulic tubing.

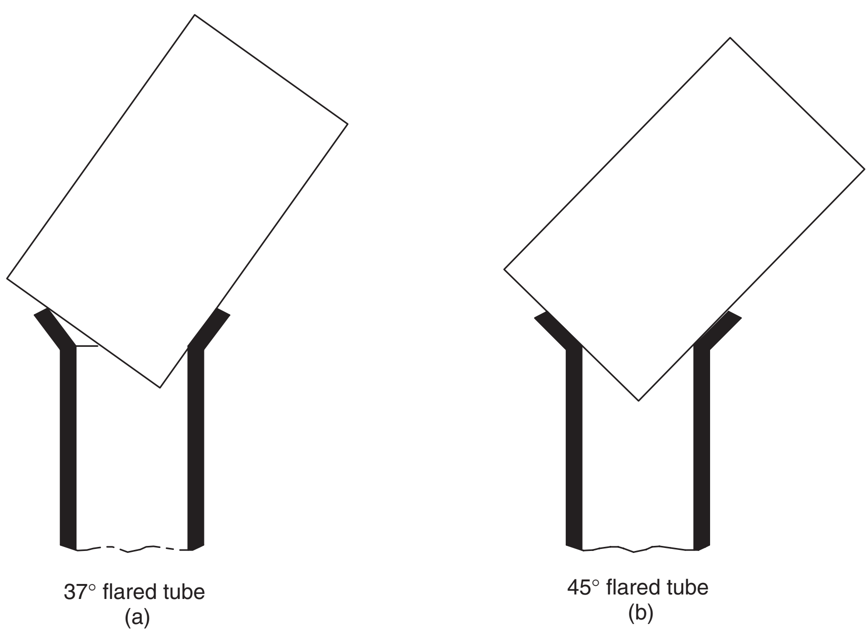


FIGURE 9.11
Business card test to determine if flare is 45° or 37°.

tightened. It is best to moderately tighten, check for leaks, and then tighten in 30° increments until the leak is sealed.

Flareless fittings are not recommend for tubing below a certain wall thickness. Compressive hoop strength relates to the collapse resistance of the tubing when it is subjected to a uniform external pressure in the radial direction at all points on the OD. Tubing must have adequate compressive hoop strength to withstand the stresses exerted by a flareless fitting.

Flared fittings are difficult to use for tubing *above* a given wall thickness, and flareless fittings cannot be used for tubing *below* a given wall thickness. The recommended wall thickness range for the two types of fittings is given in Table 9.3. As an example, flared fittings are not recommended for 3/4-in. tubing with a wall thickness greater than 0.109 in., and flareless fittings are not recommended for tubing with wall thickness less than 0.065 in. Between 0.065 and 0.109 in., both types of fittings can be used.

TABLE 9.3

Recommended Wall Thickness Range for Use of Flared and Flareless Fittings on Steel and Stainless Steel Tubing

Tube Size		Fittings	
OD (in.)	**Dash no.**[a]	**Flared**	**Flareless**
1/4	–4	0.020–0.065	0.028–0.065
1/2	–8	0.028–0.083	0.049–0.120
3/4	–12	0.035–0.109	0.065–0.120
1	–16	0.035–0.120	0.083–0.148
11/4	–20	0.049–0.120	0.095–0.188
11/2	–24	0.049–0.120	0.095–0.203
2	–32	0.058–0.134	0.095–0.220

a. See Sec. 9.5 for an explanation of the dash numbering system.

9.3.2.3 Welded Fittings

Welded fittings are used for the most severe applications. A certified welder must weld the fitting to the tubing; consequently, the cost for welded fittings is the highest of the three types. Some manufacturers supply a welded fitting design that can be welded with an automatic welding machine. Where sufficient assembly is done to justify one of these machines, this option is less expensive.

9.3.2.4 Selection of Hydraulic Tubing

Selection of tubing involves choosing the correct material and then determining the size, OD, and wall thickness. Required ID to limit fluid velocity to a recommended maximum will be discussed later. This section presents a procedure for calculating the correct wall thickness for the system operating pressure.

Pressure ratings for hydraulic tubing and fittings are given in SAE Standard J1065 Jul95. The design pressure data given in Appendix A9, Table A9.1, is based on severity of service rating A. This rating uses a design factor of 4, meaning that the burst pressure to working pressure ratio is 4:l. [A more complete listing of tubing design pressure data can be found in manufacturer's technical data, for example, Parker (1996).] For more severe service, the values in Table A9.1 are derated using the factors given in Table 9.4 (shock derating factor) and Table 9.5 (temperature derating factor). Normal service is service in a typical manufacturing plant where fluid power is used for assembly operations. An example of severe service is an application where large masses are being accelerated and decelerated and pressure pulses, or shocks, are produced routinely. Hazardous service is defined as an application where shocks are produced routinely, and the equipment operates in a hazardous environment. Steel mills and chemical plants are examples of a hazardous environment.

TABLE 9.4

Derating Factors for Hydraulic Tubing Used in Various Severity or Service Applications

Severity of service	Description	Design factor	Derating factor
A (normal)	Moderate mechanical and hydraulic shock	4	1.00
B (severe)	Severe hydraulic shocks and mechanical strain	6	0.67
C (hazardous)	Hazardous application *and* severe service conditions	8	0.50

The use of the derating factors is illustrated in the following design example. A press is used to form bladders for truck brakes. Sheets of reinforced elastomer material are placed in the mold and the press closes. A combination of pressure and temperature is used to form the part. The procedure calls for a "bump" cycle where pressure is cycled for several cycles. Operating temperature is 320°F.

The relief valve for the press circuit is set on 2000 psi, and 0.75 OD carbon steel tubing has been selected. Find the wall thickness required.

The derating factor for B service is 0.67 (Table 9.4) and for 320°F operating temperature, the derating factor is 0.99 (Table 9.5). The combined factor is $0.67 \times 0.99 = 0.663$. Tubing selected must have a design pressure rating of

$$\frac{2000\text{ psi}}{0.663} = 3015\text{ psi}$$

Referencing Table A9.1 (Appendix 9.1), 0.75-in. C-1010 steel tubing with a 0.083-in. wall thickness has a design pressure rating of 3050 psi. This tubing is chosen for the press circuit. It has a burst pressure rating of 4×3050 psi = 12,200 psi; thus, the overall design factor for this application is

TABLE 9.5

Derating Factors for Hydraulic Tubing Used in Applications where Tubing is Exposed to Temperatures >100°F

Maximum operating temperature (°F)	Material: Steel	Material: Stainless steel	
	C-1010 and C-4130	304	316
100	1.00	1.00	1.00
150	1.00	0.91	1.00
200	1.00	0.84	1.00
250	1.00	0.79	1.00
300	1.00	0.75	1.00
350	0.99	0.72	0.99
400	0.98	0.69	0.97
500	0.96	0.65	0.90
600		0.61	0.85
700		0.59	0.82
800		0.57	0.80
900		0.54	0.78
1000		0.52	0.77
1100		0.47	0.62
1200		0.32	0.37

$$\frac{12,200 \text{ psi}}{2,000 \text{ psi}} = 6.1$$

Referencing Table 9.3, the tubing has an 0.083-in. wall thickness; thus, either flared or flareless fittings can be used.

9.3.3 Hydraulic Hose

Hydraulic hose can be divided into two categories: fabric-reinforced and wire-reinforced. The fabric-reinforced hose has a plastic (or rubber) inner tube covered by one or more layers of woven fabric. The outer surface is protected by a rubber or plastic covering (Fig. 9.12). Wire-reinforced hose has a synthetic rubber inner tube, one or more layers of wire reinforcement, and a synthetic rubber outer coating to protect the wire (Fig. 9.13). Two types of wire reinforcement are used: woven and spiral bound. The spiral bound is recommended for applications in which a great deal of flexing is required. With the spiral binding, there is less wear due to rubbing of adjacent wires as the hose flexes.

Pressure ratings for hydraulic hose are given in SAE Standard J517 May 97, a portion of which is reproduced as Table A9.2 (Appendix A9). The design pressure rating decreases as diameter increases. For 100R1 hose, an eight-fold

FIGURE 9.12
Typical fabric-reinforced hydraulic hose.

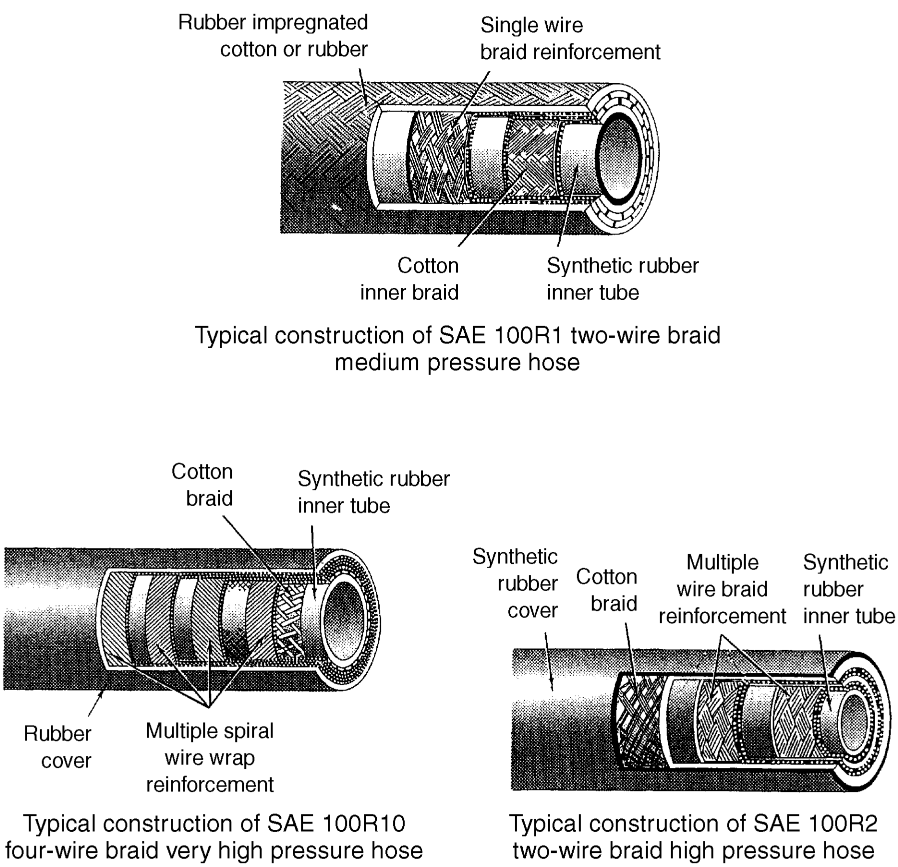


FIGURE 9.13
Typical construction of wire-reinforced hydraulic hose.

increase in diameter (from 0.25 to 2.00) gives a seven-fold decrease in design pressure. The 100R10 hose has a 3.5-fold decrease over the same diameter range.

There are several techniques for attaching the hose end to the hose. A barbed nipple is pushed into the hose and held with a two-bolt exterior clamp (Fig. 9.14) for low-pressure applications. Reusable fittings are attached by screwing a socket (left-handed thread) onto the hose and then screwing a nipple (right-hand thread) into this socket. As the nipple goes into the socket, it clamps the hose between the two parts, nipple and socket. An advantage of the reusable fitting, other than the advantage of being able to salvage the fitting when the hose is replaced, is the potential for changing the nipple to obtain a different fitting on the end of the hose (Fig. 9.15).

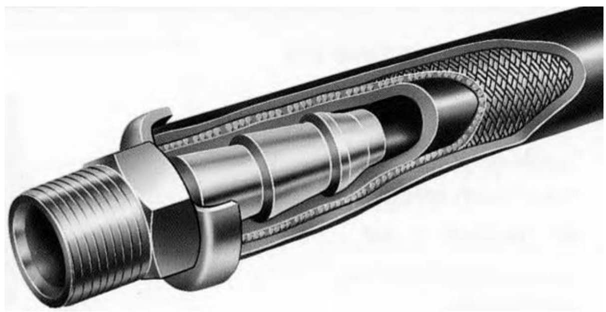


FIGURE 9.14
Barb and clamp arrangement used to attach end on low-pressure hose (clamp not shown).

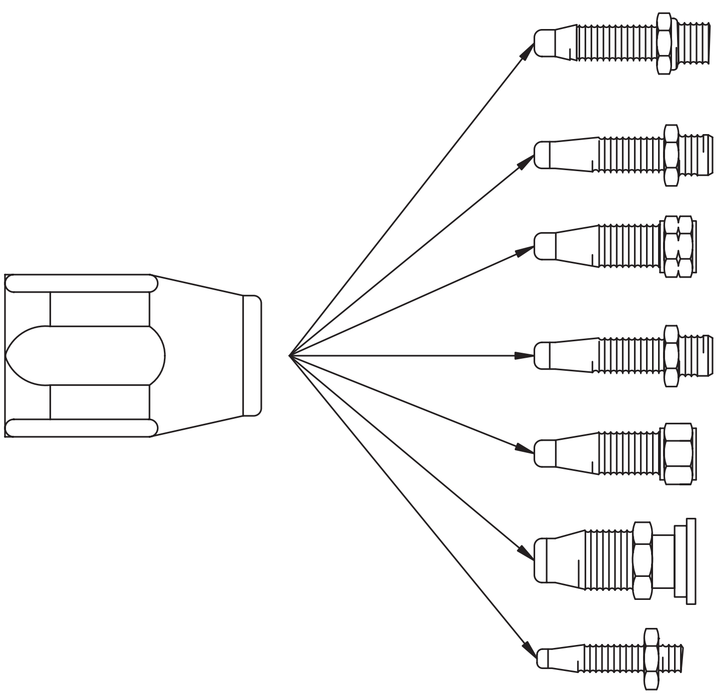


FIGURE 9.15
Variety of nipples available to screw into socket of reusable hose fitting.

Swage-on fittings are permanent. They are discarded with the hose when it is replaced. In this case, the socket is pressed on the hose end as shown in Fig. 9.16.

Large hoses have large fittings, and it is often difficult to get the right size wrench in position to tighten the fitting. Split flange fittings (Fig. 9.17a) were developed so that assembly can be done by tightening four small bolts (Fig. 9.17b).

9.4 Fluid Velocity in Conductors

Pressure drop per foot of tubing is plotted in Fig. 9.18 for fluid having a viscosity of 100 SUS and flowing in a straight 0.5-in. ID tube or hose. The pressure drop in the hose is slightly higher, because the friction factor for the elastomer inner tube is greater than the factor for the steel surface of the tubing. Comparison of the two curves shows the difference is not dramatic. The decision to choose between hose and tubing rests on factors other than pressure drop. These factors will be discussed later.

The importance of maintaining oil temperature, and therefore viscosity, in the desired range is shown in Fig. 9.19. Here, pressure drop is plotted versus fluid velocity for two viscosities, 100 and 400 SUS. (These viscosities were chosen because they represent the viscosity range of typical hydraulic oil where the system is started at 0°F ambient temperature versus starting at 100°F.) At a velocity of 10 ft/s, the 400 SUS fluid has a pressure drop greater than 4 times that of the 100 SUS fluid. At 30 ft/s, the pressure drop is about 1.4 times that of the 100 SUS fluid. Pressure drop in the lines decreases as the system comes up to the operating temperature.

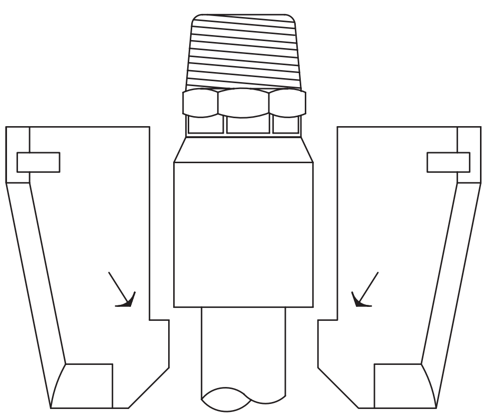


FIGURE 9.16
Pressed-on hose fitting shown in fixture.

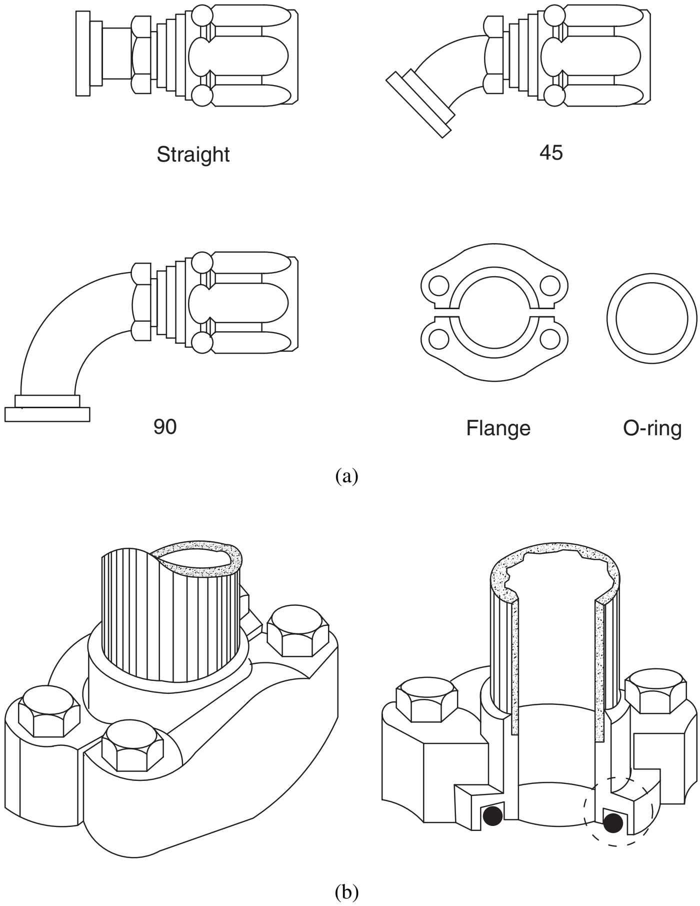


FIGURE 9.17
(a) Examples of hose ends available in split-flange fittings and (b) split-flange fitting (left) assembled and (right) cut away, showing O-ring being compressed to form a seal.

It is recommended that lines be sized such that the following maximum velocities are not exceeded:

1. Pressure line–15 ft/s
2. Return line–10 ft/s
3. Suction line–4 ft/s

The cost for the conductors (pipe, tubing, or hose) increases significantly as the size increases. One-inch fittings cost about eight times more than 0.5-in. fittings. Selection of a maximum velocity for pressure lines of 15 ft/s is a trade-off between fixed cost for the conductors and higher operating costs due to higher pressure drop in the lines. As shown in Fig. 9.18, pressure drop

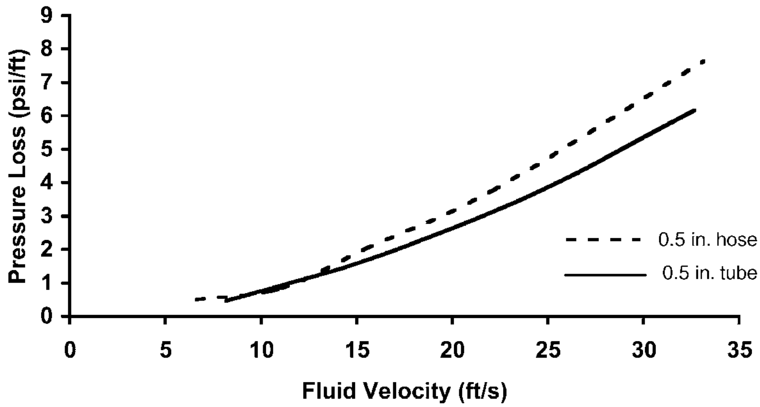


FIGURE 9.18
Pressure drop as a function of fluid velocity for 100 SUS viscosity fluid flowing in an 0.5-in. dia. tubing and hose.

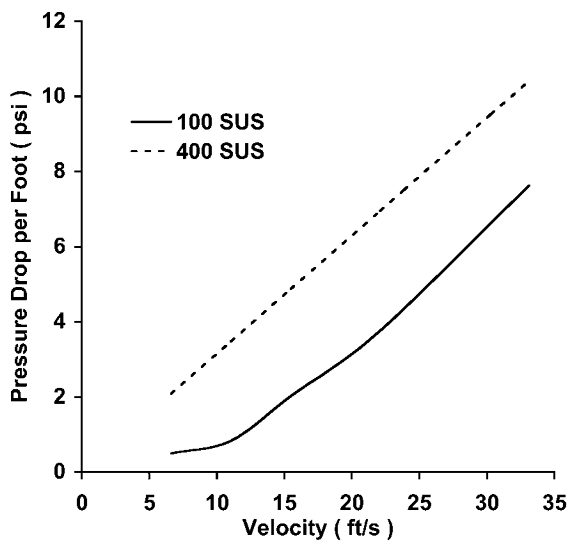


FIGURE 9.19
Pressure drop as a function of fluid velocity for 100 SUS and 400 SUS viscosity fluid flowing in an 0.5-in. dia. tube.

increases more rapidly as velocity increases above 15 ft/s. Pressure drop at 30 ft/s is three times the drop at 15 ft/s (Fig. 9.19). Pressure drop in the lines represents a conversion of hydraulic energy to heat energy. It reduces the overall efficiency of the system and thus increases operating cost. There are times when considerable analysis is done to find the optimal compromise when sizing the components to balance fixed cost against operating cost over the design life of the system. The 15 ft/s design recommendation for pressure lines is a good compromise for cases where the complete analysis is not done.

Return lines are low-pressure lines and thus less expensive. It costs less to increase return line size and achieve a lower velocity. The 10 ft/s recommendation gives a low pressure drop in the return line and is a good choice.

The importance of preventing pump cavitation has been discussed several times previously. Pressure drop in the suction line should be as low as possible, thus the 4 ft/s recommendation. The key requirement for a suction line is that it have enough reinforcement to prevent collapse when the pressure is negative.

9.5 Options for Connecting Components

There are a wide variety of fittings used to connect tubing, hose, and pipe. A dash numbering system has been developed to facilitate the selection of needed fittings. The dash number is the number of sixteenths of an inch in the nominal size. Data for selected tubing and hose are given in Table 9.6. Tubing with a 0.5-in. OD has a –8 number. Nominal 0.5-in. hose has a –8 number. The real advantage comes in the selection of fittings as shown in Fig. 9.20. A –8 tube fitting mates with a –8 adapter, which mates with a –8 hose end. Note that the ID is approximately constant through the connection, which minimizes pressure drop. Adaptors are available to connect a –8 hose to a –6 tube

TABLE 9.6

Dash Numbering System for Hydraulic Lines and Fittings

Dash size[a]	Tube OD	Tube ID[b]	Hose ID[c] (medium pressure)
–4	1/4	0.180	0.188
–5	5/16	0.242	0.250
–6	3/8	0.305	0.313
–8	1/2	0.402	0.406
–10	5/8	0.495	0.500
–12	3/4	0.620	0.625
–16	1	0.870	0.875
–20	11/4	1.120	1.125
–24	11/2	1.370	1.375
–32	2	1.810	1.813

a. Dash size is the number of 16ths of an inch in the nominal size.

b. Tube ID is a function of the wall thickness, thus the ID is different for various tubing with the same OD.

c. As with tubing, hose ID is a function of the hose construction.

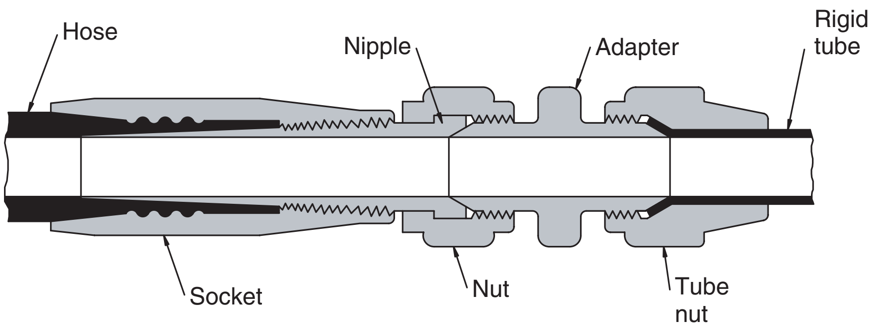


FIGURE 9.20
Connection of –8 hose to –8 tubing using a –8 adaptor.

or –6 tube to a –4 hose. Manufacturers' catalogs should be referenced to determine the many options available.

Many tubing and hose connections for mobile and stationary applications are made with 37° flare fittings. Though manufacturers still supply components with pipe thread ports, it is recommended that pumps, motors, and valves be purchased with straight-thread O-ring ports. An adaptor is screwed into this port as shown in Fig. 9.21. The O-ring fits into a shallow groove machined into the surface of the port. This O-ring is compressed as the adaptor is tightened. The threads do not seal; sealing is done by the O-ring. A fitting with tapered threads (pipe treads) can continue to be tightened. Excessive tightening can crack the port and ruin the component (pump, motor, etc.), so the advantage of O-ring fittings is obvious.

Elbows that adapt from O-ring to 37° flare have a lock nut that screws down against the O-ring (Fig. 9.21). The elbow is screwed in several turns and

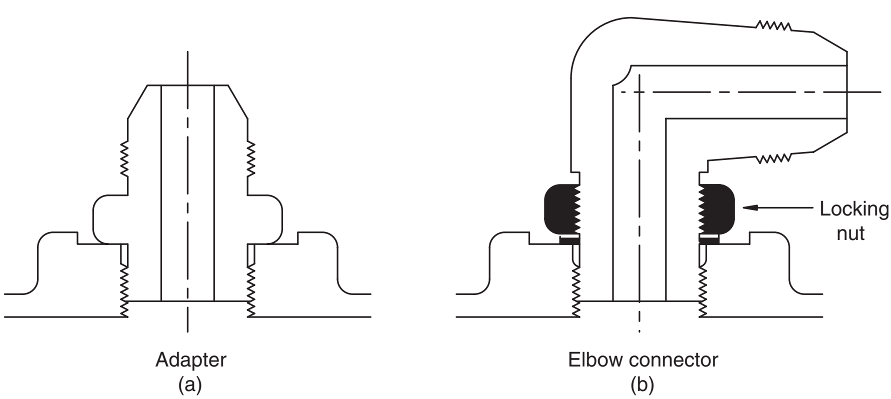


FIGURE 9.21
Adaptor screwed into straight-thread O-ring port to allow connection of 37° flare tube or hose fitting.

rotated to point in the right direction, and then the lock nut is screwed down to seal it in this position.

Fittings are available that can swivel 360° (Fig. 9.22). These fittings are more expensive, but sometimes they are needed to prevent a hose from twisting as it moves to follow an actuator.

9.5.1 Manifolds

Manifolds are simply blocks of metal with drilled passages to connect various ports. Often, a passage is drilled to intersect with the main passage so that a pressure gage can be installed. High-cost valves, like servo and proportional valves, will typically mount on a manifold so that the fluid lines can connect to the more structurally robust manifold rather than directly to the

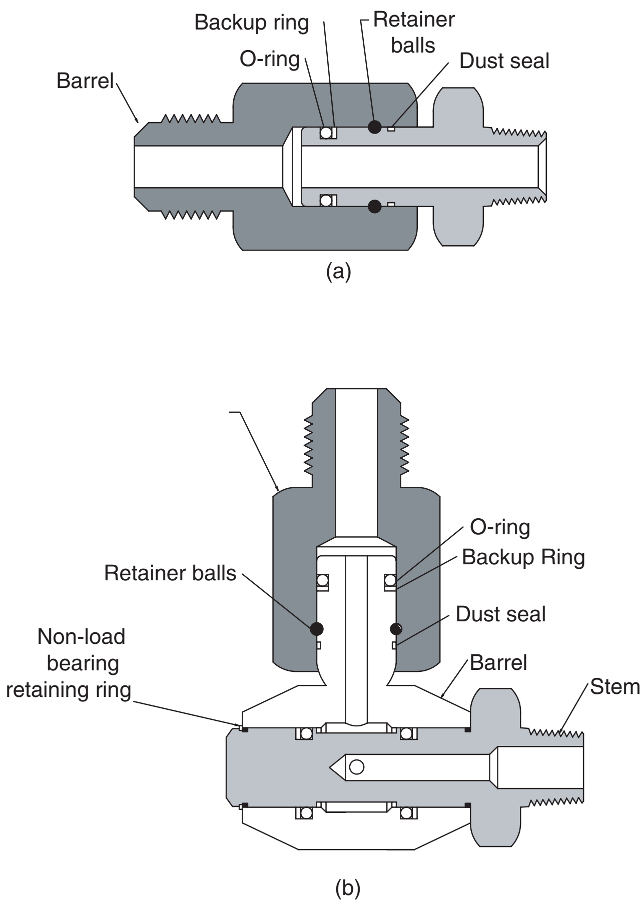


FIGURE 9.22
Swivel fitting that allows 360° rotation of the 37° flare end.

expensive valve. There are also cases where pilot pressures lines connect to the manifold.

Many valves (relief, sequence, counterbalance, check, etc.) are manufactured in a cartridge configuration. These cartridges fit into a machined cavity in a manifold block called a *line body* (Fig. 9.23). The line body has ports for connection of the lines. A number of line body configurations are available and, in some cases, two or more valve cartridges can be installed in one line body. An example is the cross-port relief valve discussed in Chapter 6. Here, a relief valve cartridge is mounted in both sides of the line body (Fig. 9.24).

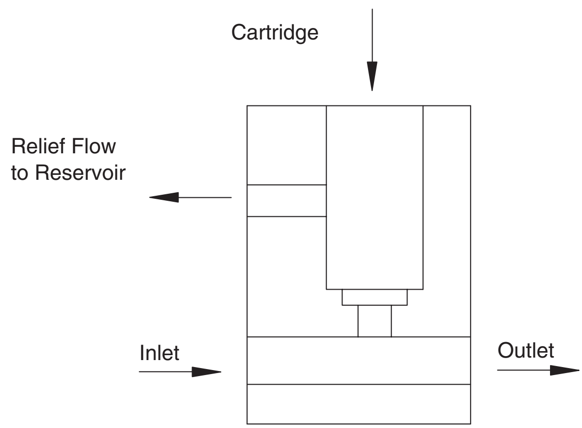


FIGURE 9.23
Cartridge relief valve mounted in "through port" line body.

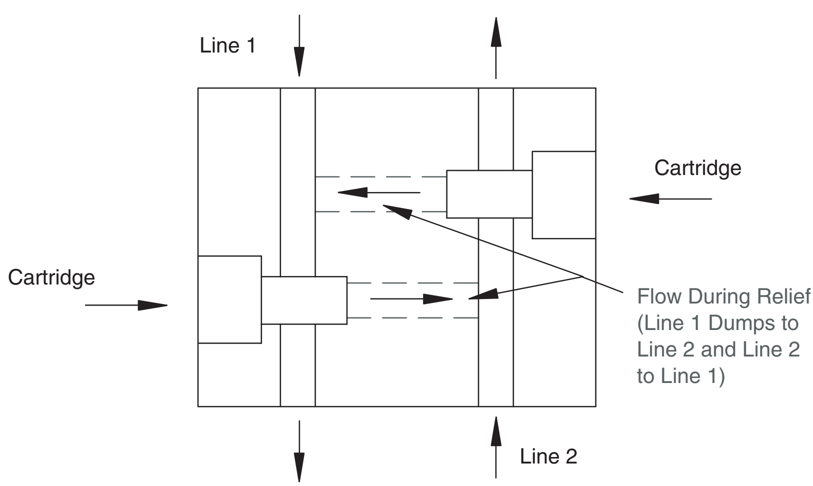


FIGURE 9.24
Line body for cross-port relief valve.

9.5.2 Quick-Disconnect Coupling

Quick-disconnect couplings are used for lines that are connected and disconnected frequently. There are two main types:

- Single shut-off
- Double shut-off

The single shut-off type has a shut-off valve in only one side, and the double shut-off has a valve in both sides. The single shut-off was originally developed to connect portable air tools to compressed-air lines. Normally, the female part has the shut-off valve and is installed on the pressure line. When the male part is removed, the air exhausts from the pneumatic tool and the female part seals the air line.

The double shut-off type is used for hydraulic lines and is available with a pressure rating up to 5,000 psi. Valves on both sides close to prevent the loss of hydraulic fluid when the coupling is disconnected.

A cross section of a double shut-off coupling is shown in Fig. 9.25. A collar on the female port is manually pulled back to release the mechanism holding the parts together. While holding the collar back, the two sides can be pulled apart. The valves in both sides act like, and are similar in construction to, check valves. When the two sides are connected, a projection from one side contacts the valve on the other side and holds it off its seat. Fluid can flow through the coupling. When the two sides are disconnected, the valve is reseated by a spring, and flow is blocked.

The mechanism within a quick-disconnect coupling produces a higher pressure drop than most other fittings. Quick-disconnect couplings should be used only where their convenience offsets the pressure drop disadvantage.

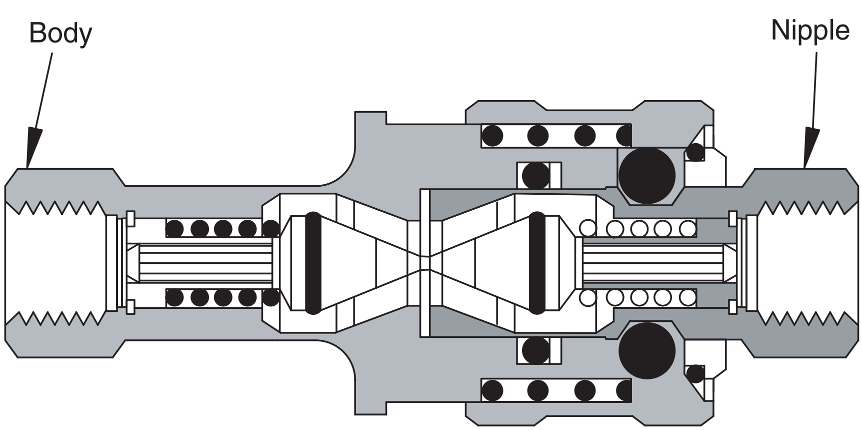


FIGURE 9.25
Cross-sectional view of a double shut-off, quick-disconnect coupling.

9.6 Installation of Lines

Routing of a pressurized fluid can be done with either rigid (tubing) or flexible (hose) conductors. Both are manufactured for the range of design pressures used for modern hydraulic systems. As seen in Fig. 9.18, the difference in friction factor for hose and tubing has little influence on pressure drop. The decision to use one conductor over the other is made based on other factors.

- *Cost.* A hose with end fittings costs more than the same length of tubing with end fittings.
- *Heat exchange.* Approximately four times as much heat is exchanged per unit length of tubing as compared to hose.
- *Compliance.* When hose is subjected to a pressure pulse, its cross-sectional area increases more than the cross-sectional area of tubing. The resulting volume increase helps to damp pressure spikes much like an accumulator damps spikes. For some applications, where response time is critical, hose "swelling" is a disadvantage. The volume change of the hose increases response time. This issue will be discussed in some detail in Chapter 11.

There are, of course, instances in which a flexible line must be used. An example is a cylinder that pivots as it extends and retracts.

All hydraulic lines experience stress cycles as the pressure cycles. They also can be subjected to significant external vibration. *External* vibration is used to define any vibration that does not result from fluid flow. External vibration is always a significant issue on mobile machines.

Tubing is more susceptible to external vibration than hose. It will eventually leak, typically at the fitting, if it is not properly supported. The key issue in the installation of tubing is to ensure that it is *properly supported*.

Hose, because it moves as the actuator moves, can rub against a surface and be damaged. Once the outer rubber covering is damaged and the wire mesh exposed, the hose must be replaced.

There are several common-sense rules for the placement of lines (tubing and hose).

1. Do not place lines where they can be stepped on by an operator or maintenance personnel.
2. Do not place lines where they are directly impacted by a heat source. For example, never place hydraulic lines near a steam line. Shield or insulate the hydraulic line in some way. Remember, heat exchange is one of the functions of a hydraulic line.
3. Always place the lines to minimize the potential for damage by impact. A sharp object can dent thin-wall tubing and cause a stress

concentration which may, over time, lead to a failure. An example is the placement of lines under a vehicle. Shield from impact by debris thrown by the wheels.

9.6.1 Recommended Practices for Hydraulic Hose

The information in this section is extracted from SAE J1273 Oct 96, and this reference should be consulted for more detail.

9.6.1.1 Minimum Bend Radius

Sharp bending at the hose/fitting junction can result in leaking and/or failure (Figs. 9.26a–c). Recommendations given in the hose manufacturer's product literature should not be exceeded.

Hose changes length when the pressure increases (Fig. 9.27). The hose mounting should provide opportunity for this movement.

9.6.1.2 Twisting

Hose is weakened when it is twisted during installation or by machine movement (Figs. 9.28a and b). Pressure cycles tend to untwist the twisted tubing and thus loosen the fittings.

Hydraulic hose should be supported to prevent bending in more than one plane (Figs. 9.29a and b). It is a good procedure to use a visoelastic sleeve around the hose to ensure that the outer cover will not be damaged by rubbing against the clamp. Sometimes the hose must be supported to prevent it from getting in a position when it can be caught between two machine parts and crimped.

9.6.2 Environmental Issues

The outer cover of hydraulic hose must remain intact for the hose to function as designed. Conditions that can cause hose degradation are

1. Ultraviolet light
2. Saltwater
3. Air pollutants
4. Temperature
5. Ozone
6. Electricity

Some very good elastomer materials have been developed; however, when exposure is severe enough and long enough, all these materials eventually degrade. The preventive maintenance plan should include routine examination of the hose outer covering.

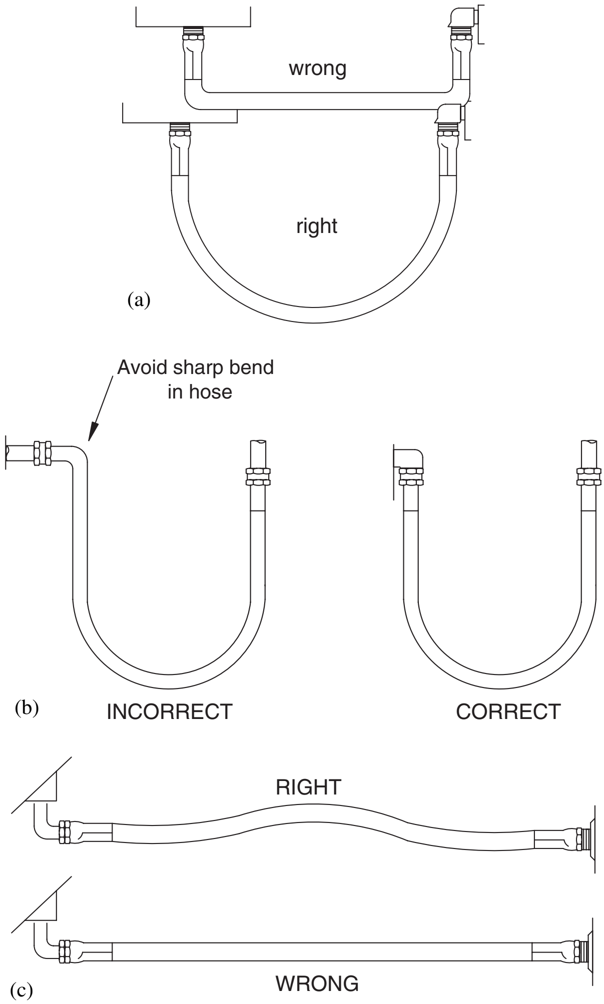


FIGURE 9.26
(a) Minimum bend radius for hydraulic hose, (b) proper use of elbow to reduce strain at hose–fitting juncture, and (c) proper hose length to avoid strain at hose–fitting junction.

9.7 Design Life of Components

No standard that defines the design life of hydraulic lines and fittings has been agreed upon. The work environment of hydraulic components is so diverse that it has been impossible for engineers to agree on a recommended replacement interval. There is a procedure for cumulative damage analysis given in SAE J1927 Jul 93. In severe use environments, where personnel

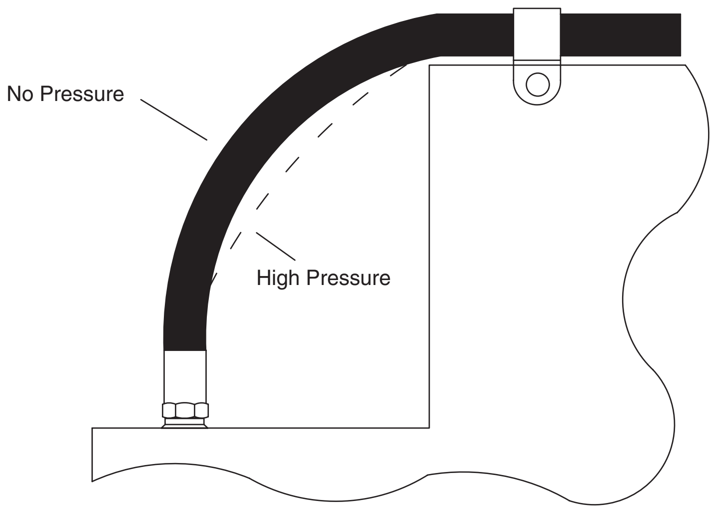


FIGURE 9.27
Illustration of hose movement when pressure is applied.

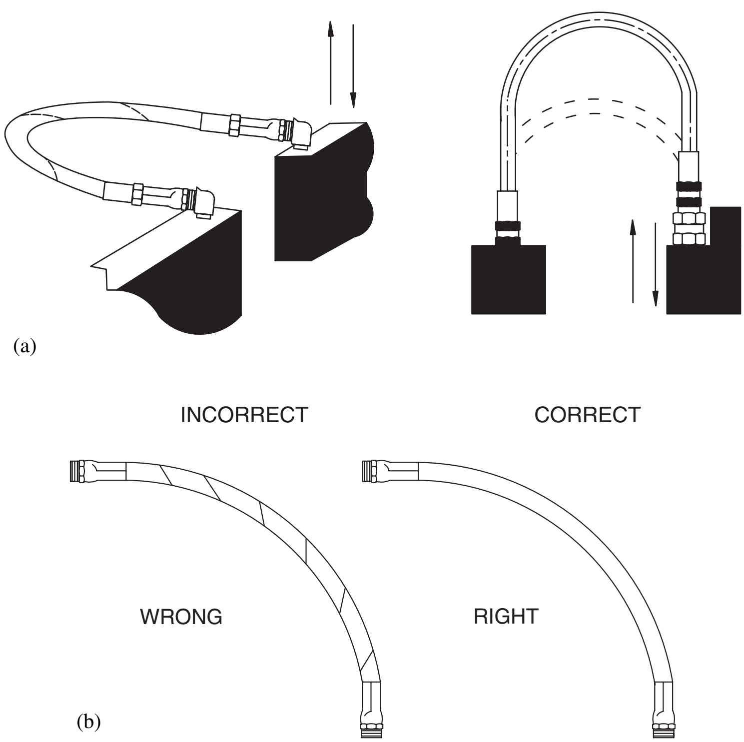


FIGURE 9.28
Hose twisted (a) during installation and (b) by machine movement. (To avoid twisting, mount so that bending occurs in only one plane.)

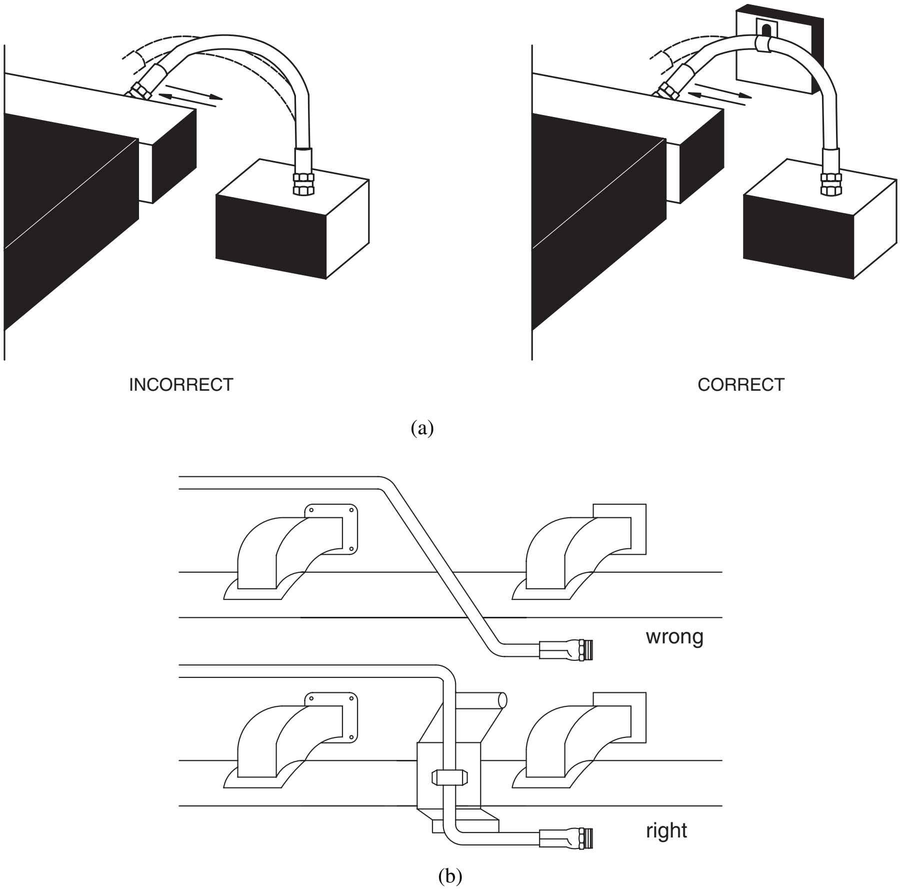


FIGURE 9.29
Hose support (a) to prevent bending in more than one plane and (b) to prevent hose from straying outside assigned pathway.

safety is an issue, cumulative damage analysis should be used to establish a planned replacement interval.

Failures can be arbitrarily defined as

- Routine
- Pinhole
- Burst

Routine Failure

Routine failure is some failure that produces an observable leak. If the materials are stressed when the end fittings are applied, a leak can develop as a service history is accumulated. With hose, a leak begins with a small crack in the inner tube, probably due to a change in the material properties caused by temperature cycles. High-pressure fluid works its way through the wire mesh and eventually through the outer cover. The leak may be observed first as a film of liquid on the surface of the hose.

It the outer cover has cracked due to ultraviolet light or other attack, then the wire mesh can be attacked by the surrounding environment. Water produces rust, which weakens the individual wires. Eventually, a leak will begin.

Pinhole Failure

A pinhole failure is a thin stream of fluid that appears as a mist when it exits the hydraulic line. Sometimes, this mist can be seen and sometimes not. It can penetrate the skin and cause a wound that requires special medical attention. *Never run your hand along a line under pressure if a leak is suspected*. Once a pinhole failure has begun, the thin stream of fluid will begin to erode a larger opening, and loss of fluid will increase. The line must be replaced.

Burst Failure

When a hose is loaded with increasing pressure during a test to determine the burst pressure, it first becomes very rigid. Failure occurs in a fraction of a second and sounds like a gunshot. When the burst occurs, fluid exists as a cloud of fluid droplets. (Probably some liquid also vaporizes into a gas.) Liquid then spews out after the pressure has dropped. A hose typically ruptures within two diameters of the fitting in a burst test.

Fortunately, burst failures are rare. Remember that the burst pressure is four times the design pressure. The relief valve has to fail, and generally the power limiter on the prime mover has to fail, for pressures to build to the burst level. Hoses in particular give ample warning of their condition and their need for replacement, and these warnings should be heeded. A burst failure can send shrapnel at high velocity for long distances. Personnel can be badly injured or killed. *Replace any line that begins to leak!*

9.8 System Integration

The industry commonly uses the following criteria, described with the acronym STAMP, when designing the fluid containment system. This acronym refers to

- *S*ize
- *T*emperature
- *A*pplication
- *M*edia
- *P*ressure

Size

Size refers not only to the inside diameter of the conductor, set by the requirement to limit fluid velocity for a given flow, but also to the port sizes of the various components. It is poor design to have the conductor properly sized and use a fitting at a valve or actuator that has a smaller inside diameter. The resultant pressure drop wastes hydraulic energy and adds to heat load.

Temperature

If a high temperature (above 200°F) or a low temperature (due to cold ambient conditions) is expected, special O-ring seals may be required. Also, thermoplastic hose has an operating temperature range that differs from that of wire-bound hose.

Application

When there is a high degree of vibration, hose must be used. Tubing is used for stationary applications and can be used on mobile machines to convey fluid along a length of the frame. The tubing should be well supported, protected from impact, and placed where it will not be used as a step for the operator or a maintenance worker. In some applications, hoses need to be protected with an abrasion-resistant cover.

Pipe can be an economical choice for a return line. The pressure is much lower, and sealing the tapered threads is easier.

Media

Water-based fluid requires different seals from those of petroleum-based fluid. Also, certain seals are not compatible with certain additives. It is the ultimate responsibility of the system designer to work with the fluid supplier and the component supplier to resolve these issues.

Pressure

As stated before, all components must have a higher pressure rating than the design maximum pressure. This is true of all conductors, hose and tubing, and all fittings.

9.8.1 Port Connections

The various straight-thread connections include SAE straight threads, ISO 6149, metric, and BSPP. Only SAE straight threads were discussed previously. The reader is referred to manufacturer's literature for a description of the other options. With all these connections, an O-ring provides the seal, and the threads hold the parts together and squeeze the O-ring until it seals.

Tapered threads can be BSPT, NPT, or NPTF. For all tapered threads, the threads hold the parts and make the seal. A sealant (teflon tape or pipe sealant) must be applied to get a good seal. A tapered thread works best when the parts are assembled and left undisturbed. It is very difficult to reassemble tapered thread connections and get them to reseal.

9.8.2 Tube or Hose Connections

The three most common seal configurations are the O-ring face seal, O-ring flange seal, and the soft seal with mechanical seal. Mechanical seal options include the 37° flare (JIC), JIS, bite-type, and welded connection. Of these, the 37° flare is the most common in the United States.

9.8.3 Assembly

Using proper procedures to assemble the conductors and components will ensure long, leak-free service. It is generally best to choose a connection system, perhaps one of the O-ring options, and use this system for the whole machine or throughout the manufacturing plant. Workers gain experience with the system and learn the right procedures, right amount of torque, and right amount of sealant (if required) to achieve a clean, leak-free connection. Cleanliness is very important. Remember, any contaminant incorporated during assembly must be removed by the filtration system. Life of the initial filter is reduced by built-in contaminant.

9.9 Summary

A hydraulic reservoir can be pressurized at some low pressure (≈50 psi) but most often is designed to operate at atmospheric pressure. In addition to holding the supply of fluid needed to ensure that all lines and components are completely filled, it separates entrained air, dissipates pressure pulses, provides cooling, and traps contaminants.

Three types of conductors are used to route the fluid in hydraulic circuits: pipe, tubing, and hose. Most modern systems use tubing or hose for the pressure lines. Because of its low cost, pipe is still sometimes used for return lines.

All conductors have a rated working pressure and burst pressure. For normal service, the working pressure is one-fourth of the burst pressure; thus, the design ratio is 4:1. Severe applications require a 6:1 ratio, and severe applications in a hazardous environment require an 8:1 ratio.

A variety of fittings are available to connect the conductors. Most often, 37° flare fittings are used on the ends of the conductors, and these screw onto straight-thread O-ring adaptors at the component (pump, motor, valve, etc.).

Hydraulic hose is used for applications in which the actuator moves and a flexible connection is required. Hose has an elastomer inner tube, one or more layers of wire mesh reinforcement, and an outer cover that typically is an elastomer material. Ends are attached to the hose with reusable fittings or permanent swage-on fittings.

It is recommended that the lines be sized such that the velocity in pressure lines does not exceed 15 ft/s, a compromise choice that balances the fixed cost of the lines (higher for larger diameters) with operating cost over the design life (lower for larger diameters). Recommendations for the return line and suction line are 10 and 4 ft/s, respectively.

Manifolds, blocks of metal with passages drilled in them, can be used to simplify the assembly of a circuit. Cartridge valves for all types of pressure control are available for installation in machined cavities in a manifold.

Both tubing and hose can be used interchangeably except where flexibility is needed. Hose costs more and its heat exchange rate is less. Hose swells and thus dampens pressure spikes (internal vibration). It also is less susceptible to external vibration. The volume change when a hose swells does increase response time, a disadvantage for some high-performance applications.

The design life of hydraulic lines and fittings is undefined, because the work environment is so diverse. Failures can be arbitrarily defined as routine, pinhole, and burst. A routine failure is defined as deterioration over time that produces a leak. With hose, a pinhole failure is a thin stream of fluid that finds its way through the wire mesh. This thin stream initially emerges as a fine mist that becomes more visible as the stream erodes a large hole. Fortunately, burst failures are rare. They are very dangerous and should be avoided by monitoring the other types of failure to replace degraded hose (or tubing) before it reaches a condition where a burst failure can occur.

References

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Parker. 1996. Industrial Tube Fittings, Catalog 4300, Parker Hannifin Corp., 17325 Euclid Ave, Cleveland, OH 44112

SAE. 1997 SAE standards recommended practices and information report. SAE International, 400 Commonwealth Dr., Warrendale, PA 15096-0001.

Standard: SAE J517 May 97 Hydraulic Hose.

Standard: SAE 6343 Nov 95 Test and Test Procedures for SAE 100R Series Hydraulic Hose and Hose Assemblies.

Recommended Practice: SAE J1273 Oct 96 Recommended Practices for Hydraulic Hose Assembly.

Information Report: SAE J1065 Jul 95 Pressure Ratings for Hydraulic Tubing and Fittings

Information Report: SAE J1927 Jul 93 Cumulative Damage Analysis for Hydraulic Hose Assemblies.

APPENDIX 9.1

Selected Design Data for Fluid Conductors

Table A9.1 gives recommended design pressures for seamless steel tubing. Corresponding data for hydraulic hose is given in Table A9.2.

TABLE A9.1

Recommended Design Pressures (psi) for Hydraulic Tubing

Tube OD (in)	Wall thickness (in)	Steel C-1010	Alloy steel C-4130	Stainless steel 304 and 316
0.25	0.020	2150	3250	3250
0.25	0.028	3100	4650	4650
0.25	0.035	3950	5950	5950
0.25	0.049	5750	8650	8650
0.25	0.058	6900	10400	10400
0.25	0.065	7800	11750	11750
0.25	0.083	9950	15000	15000
0.50	0.028	1500	2200	2200
0.50	0.035	1850	2800	2800
0.50	0.049	2700	4050	4050
0.50	0.058	3250	4850	4850
0.50	0.065	3650	5500	5500
0.50	0.072	4100	6150	6150
0.50	0.083	4800	7200	7200
0.50	0.095	5550	8350	8350
0.50	0.109	6450	9750	9750
0.50	0.120	7200	10800	10800
0.50	0.134	8050	12150	12150
0.50	0.148	8950	13450	13450
0.50	0.188	11050	16600	16600
0.75	0.035	1200	1850	1850
0.75	0.049	1750	2600	2600
0.75	0.058	2100	3150	3150
0.75	0.065	2350	3550	3550
0.75	0.072	2650	3950	3950
0.75	0.083	3050	4600	4600

TABLE A9.1

Recommended Design Pressures (psi) for Hydraulic Tubing *(continued)*

Tube OD (in)	Wall thickness (in)	Steel C-1010	Alloy steel C-4130	Stainless steel 304 and 316
0.75	0.095	3550	5350	5350
0.75	0.109	4150	6200	6200
0.75	0.120	4600	6900	6900
0.75	0.134	5200	7800	7800
0.75	0.148	5800	8700	8700
0.75	0.188	7500	11300	11300
1.00	0.035	900	1350	1350
1.00	0.049	1300	1950	1950
1.00	0.058	1550	2300	2300
1.00	0.065	1750	2600	2600
1.00	0.072	1950	2900	2900
1.00	0.083	2250	3400	3400
1.00	0.095	2600	3900	3900
1.00	0.109	3000	4550	4550
1.00	0.120	3350	5050	5050
1.00	0.134	3800	5700	5700
1.00	0.148	4200	6350	6350
1.00	0.156	4450	6700	6700
1.00	0.188	5500	8250	8250
1.00	0.220	6550	9800	9800

TABLE A9.2

Design Pressure (psi), 100R-Series Hydraulic Hose[a]

Nominal hose ID (in)	Hose resignation			
	100R1	100R2	100R10	100R11
0.25	2750	5000	8750	11250
0.50	2000	3500	6250	7500
0.75	1250	2250	5000	6250
1.00	1000	2000	4000	5000
1.25	625	1625	3000	3500
1.50	500	1250	2500	3000
2.00	375	1125	2500	3000

a. Data extracted from SAE Standard J517 May 97.

100R1 2-wire braid medium pressure hose

100R2 2-wire braid high pressure hose

100R10 4-wire braid very high pressure hose

100R11 4-and 6-wire braid super high pressure hose

Problems

9.1 An 8 GPM flow of petroleum oil is being delivered through a 0.5-in. hose to a hydraulic motor. The hose length is 16 ft, and no elbows or other fittings need be considered.

The following handbook data are available:

			Pressure drop (psi/ft)	
Flow (GPM)	**Hose ID (in.)**	**Velocity (ft/s)**	**20.6 cS**	**87.7 cS**
7	0.5	11.4	0.98	2.66
10	0.5	16.3	1.83	3.80
15	0.5	24.5	3.73	4.55

The handbook data is for fluid with a specific gravity of 1.0 and the specific gravity of the oil is 0.865. The table data must be multiplied by a correction factor, defined by the following equation, to obtain the projected pressure drop per foot of hose length.

$$C_f = 0.73^{S_g} + 0.268$$

where C_f = correction factor (dec)
S_g = specific gravity (dec)

1. Compare the expected pressure drop in the line if the fluid viscosity is ν = 20.6 cS, and 87.7 cS.
2. Calculate the hydraulic power loss (hp) for both viscosities.

9.2 The following handbook data are available for hydraulic tubing. The fluid has a specific gravity S_g = 1.0, and viscosity is ν = 20.6 cS.

Flow (GPM)	Tube ID (in)	Velocity (ft/s)	Pressure drop (psi/ft)
3	0.319	12.0	1.63
5	0.402	12.6	1.08
7	0.495	11.7	1.03
10	0.584	12.0	0.88

The fluid velocity is approximately equal because size (ID) increases as flow increases. The pressure drop decreases as ID increases.

1. Derive the table values using the equations given in Chapter 2. You may use a friction factor f = 0.045 for seamless hydraulic tubing.

2. Explain why pressure drop decreases as tubing ID increases when velocity is relatively constant.

9.3 The following handbook data are available for hydraulic hose. The fluid has a specific gravity S_g= 0.868 and viscosity ν = 20.6 cS.

Flow (GPM)	Hose ID (in)	Velocity (ft/s)	Pressure drop (psi/ft)
10	0.5	16.3	1.65
15	0.625	15.7	1.17
30	0.875	16.0	0.79
40	1.0	16.3	0.70

Note: The velocities are slightly higher than the 15 ft/s recommended for pressure lines (Sec. 9.4).

The fluid velocity is approximately equal, because size (ID) increases as flow increases. However, the pressure drop decreases as ID increases.

1. Derive the table values using the equations given in Chapter 2. You may use a friction factor given in Fig. 9.30 for the elastomer inner tube of the hydraulic hose.
2. Suppose viscosity doubles to ν = 43.3 cS. What influence will this have on pressure drop? Report your result for the 10 GPM flow only.

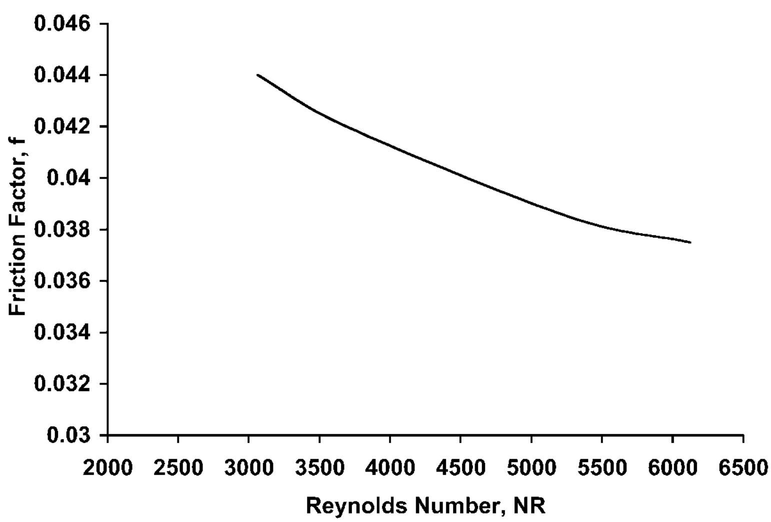


FIGURE 9.30
Friction factor (f) vs. Reynolds number (N_R) for elastomer inner tube of hydraulic hose (Problem 9.3).

9.4 Accurate positioning of a heavy load is required for a manufacturing process. Total length of 0.5-in. ID hose under pressure between the DCV and cap-end port of the cylinder is 74 in.

The cylinder has a 2.5-in. bore and a 1.25-in. rod. It holds the load at a position 6 in. above full retraction. During the manufacturing operation, the load on the cap end of the cylinder varies from 1340 to 9800 lb_f. Neglecting leakage in the cylinder and DCV, calculate the change in position of the cylinder during the operation. Bulk modulus for the oil is β = 250,000 psi, and the hose volumetric expansion is given by

$$\Delta V_l = 0.001\ \Delta P$$

where ΔV_1 = hose volumetric expansion (cm^3/ft)
ΔP = pressure change (psi)

9.5 Test data were collected on a split-configuration hydrostatic transmission (Fig. 9.31). The hose and fittings connecting the pump and motor are all –8 (0.5-in. ID). Load was applied to build test pressures of 1000, 1500, and 2000 psi, and the following data were

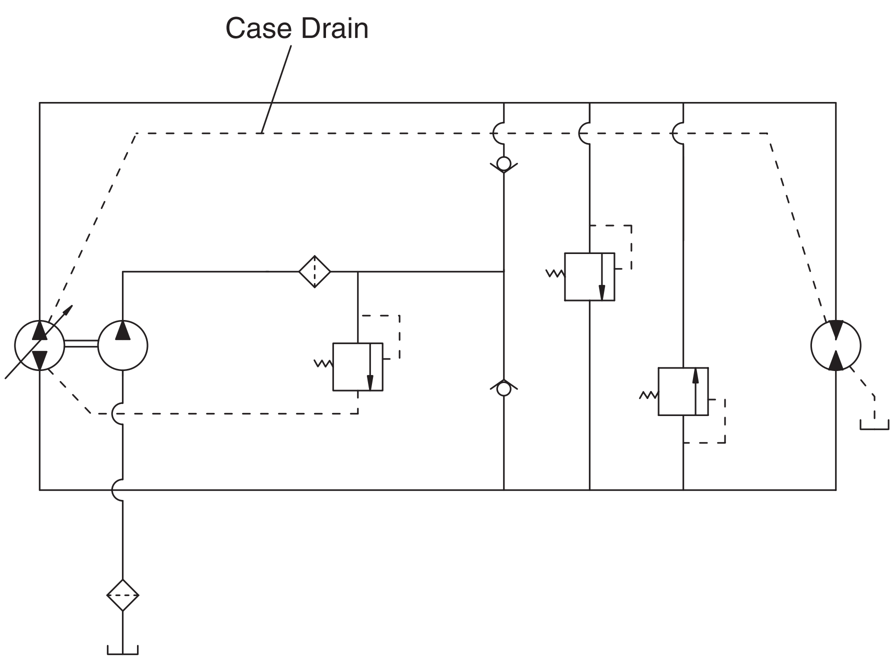


FIGURE 9.31
Circuit diagram for hydrostatic transmission (Problem 9.5).

collected. Temperature of the fluid was 54°C, and corresponding viscosity was ν = 22.7 cS.

	Pressure (psi)				
Test description	P_1	P_2	P_3	P_4	**Flow (GPM)**
1000 psi Test	120	1020	995	140	3.74
1500 psi Test	110	1570	1535	170	3.92
2000 psi Test	115	1950	1910	140	3.96

The following pressure drops are defined:

- ΔP_{pm}: pressure drop between pump and motor (forward side of main circuit)
- ΔP_{mp}: pressure drop between motor and pump (return side of main circuit)

$$\Delta P_{pm} = P_2 - P_3$$

$$\Delta P_{mp} = P_4 - P_1$$

Plot the line pressure drops vs. supply pressure. (P_2 is supply pressure.) Two plots are required—one for the forward side and one for the return side. Discuss your results. Is the measured variability a result of changes in supply pressure, or is it random experimental error?

10

Pneumatics

10.1 Introduction

The use of air as an energy transfer medium can be traced back 2000 years. Practical industrial application of pneumatics dates to around 1950, when the demand for automation in industrial production lines increased.

Some desirable features of pneumatics are

1. *Stability under temperature changes.* Pneumatic actuators can be used in manufacturing plants where high and low temperatures are required.
2. *Cleanliness.* Leaks do not cause contamination.
3. *High speed.* Cylinders have a working speed of three to seven ft/s (fps).
4. *Overload safety.* Tools can be loaded until they stall and are therefore overload safe.

It is also important to understand the undesirable features.

1. *Compressibility.* Constant actuator velocity is not possible.
2. *Force limitation.* At a normal working pressure, 100 psi, the limit is between 4500 and 6700 lb_f.
3. *Fluid preparation.* Dirt and humidity must be cleaned from the air the prevent wear of pneumatic components.
4. *Cost.* Compressed air is a relatively expensive means of conveying power.

Part of the cost disadvantage is offset by the fact that pneumatic components are inexpensive and can operate at a high number of cycles per workday. Operating costs can be controlled by taking care to ensure that leaks are minimized.

10.2 Orifice Equation

Flow of air in a pneumatic system is expressed as standard cubic feet per minute (scfm). Standard conditions are defined by the air absolute temperature and pressure

$$T_s = 520°\mathrm{R}(60°\mathrm{F})$$

$$P_s = 14.7 \text{ psia (atmospheric pressure at sea level)}$$

Conversion to standard temperature and pressure is done using Boyle's law

$$P_1V_1 = P_2V_2 \tag{10.1}$$

and Charles' law

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \tag{10.2}$$

(See example in Appendix 10.1.)

A third law, Gay-Lussac's law, can also be written

$$\frac{P_1}{P_2} = \frac{T_1}{T_2} \tag{10.3}$$

These three laws [Eqs. (10.1) through (10.3)] can be combined the give the equation of state for an ideal gas.

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \tag{10.4}$$

Most pneumatic systems operate at around 100 psi and, at this pressure, air can be treated as a ideal gas with negligible error.

Pneumatic valves, like hydraulic valves, are variable orifices that are used to throttle the flow of fluid. The orifice equation for a pneumatic valve is

$$Q = 22.7C_v\sqrt{\frac{(P_1 - P_2)P_2}{T}} \tag{10.5}$$

where Q = flow (scfm)
C_v = flow constant (decimal)
P_1 = upstream pressure (psia)
P_2 = downstream pressure (psia)
T = absolute temperature (°R)

The notation *scfm* refers to standard cubic feet per minute.

Equation (10.5) is valid when the downstream pressure, P_2, is more than $0.53P_1$. At $P_2 = 0.53P_1$, the exiting velocity equals the speed of sound. If P_2 is reduced further, the flow through the orifice is *choked,* and Eq. (10.5) is no longer valid.

Good design practice requires that the pressure drop across a valve be around 10% of line (upstream) pressure. If P_1 = 100 psi, then $P_2 = 100(1 - 0.1) = 90$ psi $= 0.9P_1$. Equation (10.5) can be used for the analysis of most commercial pneumatic systems.

Example Problem 10.1

Upstream pressure at a valve is 100 psi, and the desired pressure drop through the valve is 10 psi. Flow is 45 scfm, and air temperature is 80°F. Find the required C_v for the valve.

Solving Eq. (10.5) for C_v,

$$C_v = \frac{Q}{22.7}\sqrt{\frac{T}{(P_1 - P_2)P_2}}$$

$$T = 80 + 460 = 540°R$$

$$P_1 = 100 + 14.7 = 114.7 \text{ psia}$$

$$P_2 = (100 - 10) + 14.7 = 104.7 \text{ psia}$$

$$C_v = \frac{45}{22.7}\sqrt{\frac{540}{(114.7 - 104.7)(104.7)}}$$

$$C_v = 1.42$$

Any valve with a C_v value greater than 1.42 will give a pressure drop less than 10 psi.

Manufacturers of pneumatic valves typically publish the C_v factor for their valves. These data greatly facilitate the selection of components for a pneumatic system design. Choosing a C_v that is too small gives a larger pressure drop across the valve, and since this pressure drop does no work, the system efficiency decreases. Choosing a C_v that is too large means choosing a larger, more expensive valve.

10.3 Compressors

Typically, a compressor is located in a separate location away from the actuators. Compressed air is delivered through a piping system in the building.

A compressor operates differently from a pump in that it builds and maintains pressure in a tank called a *receiver*. When air pressure in the system drops below a set point, often referred to as the *cut-in* pressure, the compressor comes on and runs until it builds air pressure to a setting known as the *cut-out* pressure. Hydraulic pumps, other than pressure-compensated pumps, build pressure when load is applied. The compressor-receiver is somewhat analogous to a pump-accumulator, except that the pump runs continuously, and the compressor starts and stops.

10.3.1 Reciprocating Piston Compressor

The reciprocating piston compressor is the most widely used compressor. It operates very much like an internal combustion engine. Air enters through an intake valve, is compressed, and exits through an exhaust valve. The optimal range for reciprocating piston compressors are

Single stage:	up to 60 psi
Two-stage:	up to 220 psi
Triple-stage:	greater than 220 psi

10.3.2 Diaphragm Compressor

In the diaphragm compressor, the piston pushes against a diaphragm, so the air does not come in contact with the reciprocating parts (Fig. 10.1). This type compressor is preferred for food preparation, pharmaceutical, and chemical industries, because no effluent from the compressor enters the fluid.

10.3.3 Sliding Vane Rotary Compressor

This type of compressor, shown in Fig. 10.2, looks and functions like a vane-type hydraulic pump. An eccentrically mounted rotor turns in a cylindrical housing having an inlet and outlet. Vanes slide back and forth in grooves in the rotor. Air pressure or spring force keeps the tip of these vanes in contact with the housing. Air is trapped in the compartments formed by the vanes and housing and is compressed as the rotor turns.

10.3.4 Cooling

When air is compressed, heat is generated and must be removed. On smaller compressors, cooling fins around the cylinder are sufficient to remove the

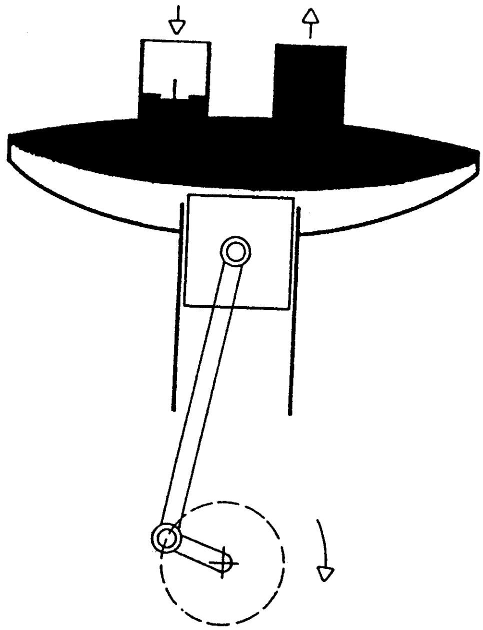


FIGURE 10.1
Diaphragm compressor. Reprinted with permission from Festo Corporation.

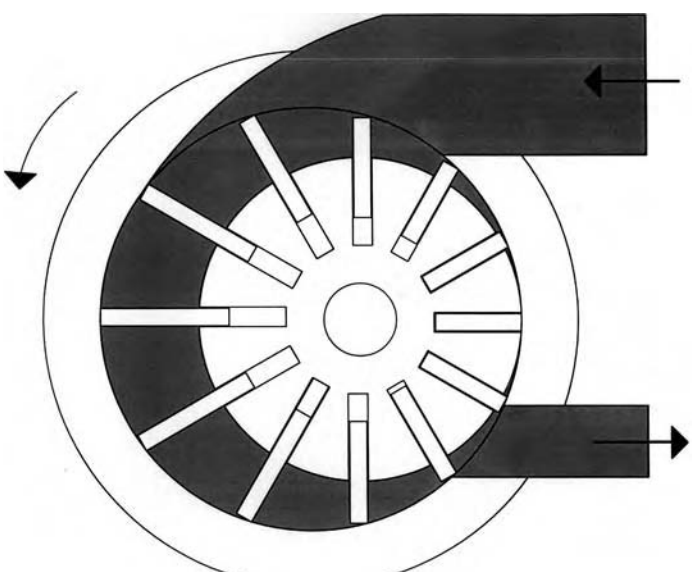


FIGURE 10.2
Sliding vane rotary compressor. Reprinted with permission from Festo Corporation.

heat. On larger compressors, a fan is used to move air over the heated surfaces and increase convective heat transfer.

10.4 Receiver

The compressed air tank is called a *receiver*. Receiver size can be determined using nomographs like that shown in Fig. 10.3. Suppose the delivery volume is 700 cfm, and on-off regulation is used. We desire that the compressor not cycle more than 20 times/h, so $z = 20$. Pressure can drop 15 psi before the compressor comes on, therefore $\Delta P = 15$. Referring to Fig. 10.3, find 700 cfm on the bottom half of the left vertical scale. Travel across to the $\Delta P = 15$ psi curve, up to the $z = 20$ curve, and over to the upper half of the left vertical scale. The size receiver needed is 500 ft^3.

Receivers are pressurized vessels (unlike hydraulic reservoirs); consequently, they must have a rating seal applied by the American Society of Mechanical Engineers (ASME).

10.5 Pipelines

Examples of pipeline installation are shown in Fig. 10.4. Instructions for sizing the pipeline are given in a number of texts. (See, for example, *Introduction to Pneumatics,* 1982). There is no return line, as air is simply dumped to the atmosphere after exiting the actuator.

10.6 Preparation of Compressed Air

Contamination in the compressed air (e.g., dirt, rust particles, excess lubricant, and moisture) cause damage to pneumatic components. Moisture, in particular, is of concern.

10.6.1 Drying of Air

Absorption drying is a chemical process. Water vapor comes into contact with the drying agent in the dryer, and a chemical reaction takes place. The agent dissolves, and a solution is formed that holds the water. After an interval of time (two to four times per year), the solution must be drained and new drying agent added.

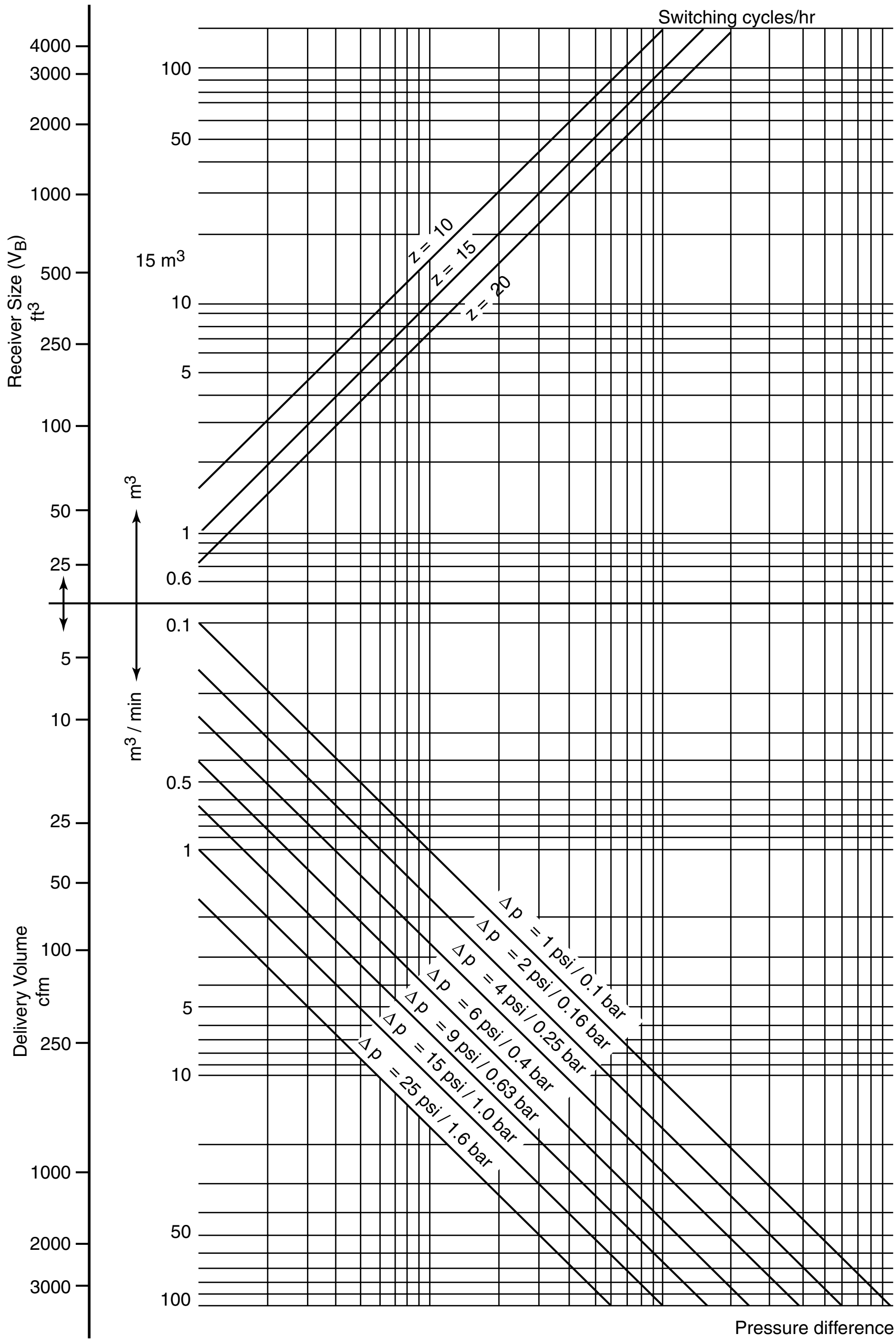


FIGURE 10.3
Nomograph for sizing receiver. Reprinted with permission from Festo Corporation.

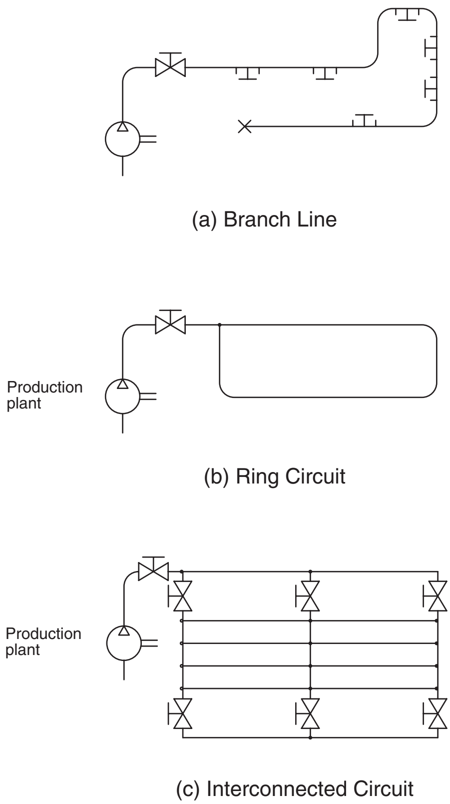


FIGURE 10.4
Examples of pipeline layouts.

Adsorption drying is a physical process. The drying agent, generally silica gel, adsorbs moisture. When the gel becomes saturated, it must be dried by passing warm dry air through the bed. Generally, two adsorption beds are connected in parallel so that one can be dried while the other is in use.

10.6.2 Air Filtration

A compressed air filter (Fig. 10.5) removes contaminants and water droplets from the air. As compressed air enters the filter bowl, it is forced to flow through the guide slots in the baffle plate (2). As it makes the u-turn, the particles are centrifuged out and collect in the lower part of the filter bowl. Condensate that collects in the bowl must be drained periodically. The filter shown in Fig. 10.5 has a manual drain, but filters are available with an automatic drain.

Compressed air microfilters are used in industries that require extra filtration. Microfilters remove particles down to a size of 0.01 micron. (As a frame of reference, the naked eye cannot see particles smaller than 40 microns.)

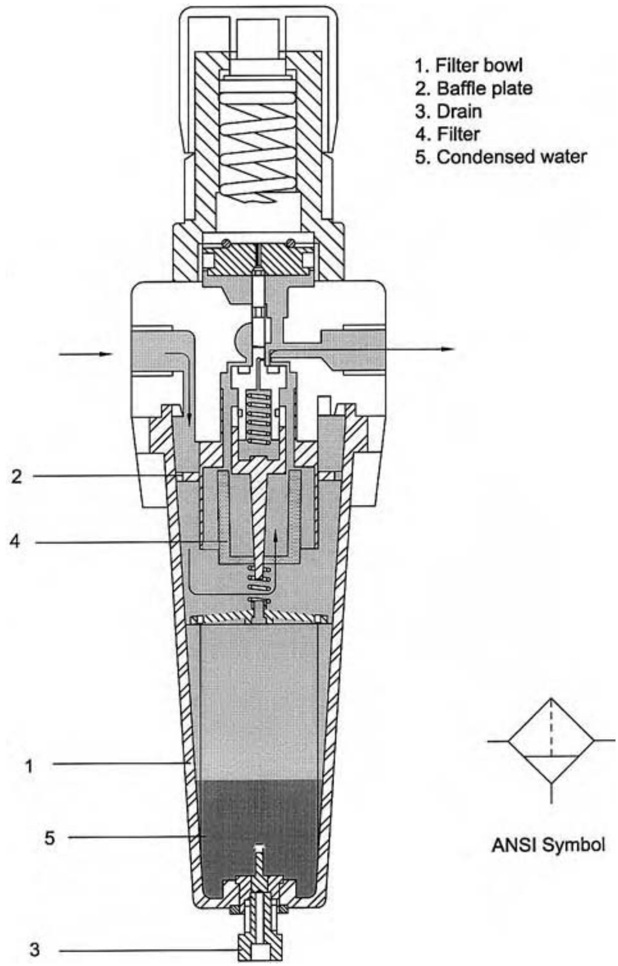


FIGURE 10.5
Air filter with manual drain.

10.6.3 Pressure Regulation

A regulator keeps the operating pressure constant regardless of fluctuations in line pressure or demand flow. As shown in Fig. 10.6, a spring (2) holds a diaphragm (1) in place. When the outlet pressure increases, the diaphragm moves downward to partly close the valve seat (4). Pressure drop through the regulator increases, and the outlet pressure is reduced. When the outlet pressure decreases, the opposite occurs. If the pressure at the outlet increases beyond the regulator set maximum pressure, the diaphragm deforms enough that the center hole is opened, and air flows through the vent holes to atmosphere.

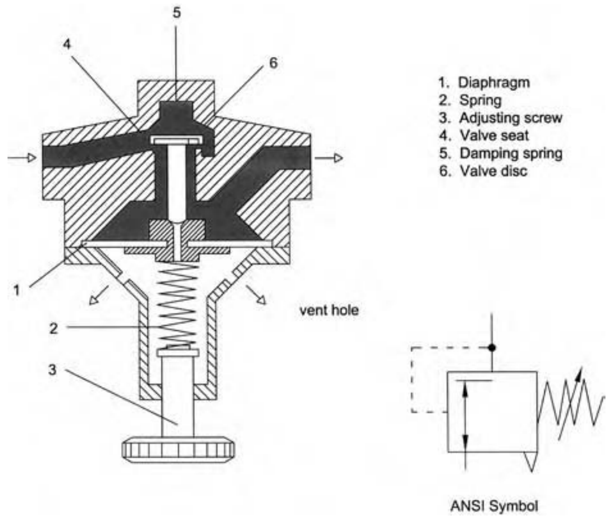


FIGURE 10.6
Pressure regulator with vent holds to release. Reprinted with permission from Festo Corporation.

10.6.4 Compressed Air Lubrication

Lubrication reduces wear in the actuators by reducing friction and protecting from corrosion. A lubricator works in accordance with the Venturi principle (Fig. 10.7). Flow velocity increases in the throat, and pressure decreases. Oil is sucked up into the airstream because of this pressure difference.

10.6.5 Service Units

Several manufacturers supply a filter, regulator, and lubricator assembled in one housing. The complete and abbreviated symbols for this component are shown in Fig. 10.8. The manufacturers supply tables or graphs that give the pressure drop vs. flow data for their units.

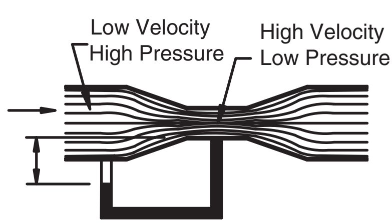


FIGURE 10.7
Venturi principle used for lubricator.

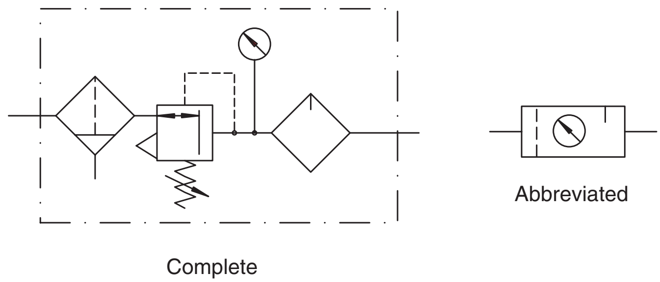


FIGURE 10.8
ANSI symbols used for various air preparation components.

10.7 Cylinders

10.7.1 Single-Acting Cylinders

Compressed air extends a single-acting cylinder, and an internal spring retracts it. Cycle time is often limited by the spring return, which is typically slower than a compressed air return.

10.7.2 Diaphragm Cylinder

A diaphragm cylinder is shown in Fig. 10.9. It has a limited stroke, generally not more than 0.25 in., and typically is used for clamping applications.

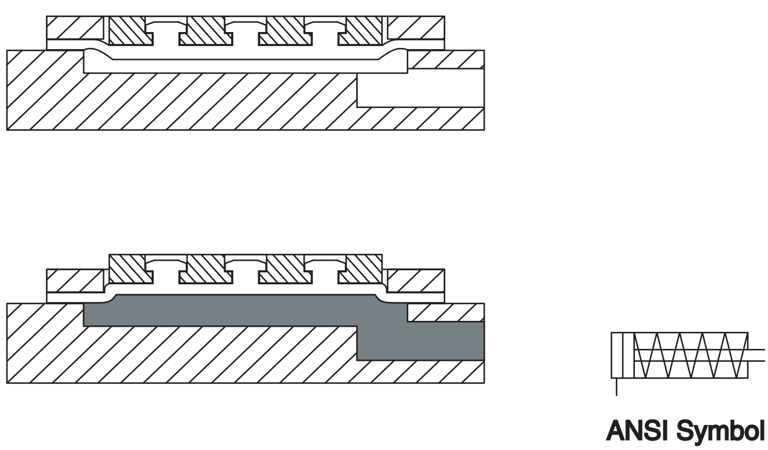


FIGURE 10.9
Diaphragm cylinder. Reprinted with permission from Festo Corporation.

10.7.3 Rolling Diaphragm Cylinders

A rolling diaphragm cylinder (Fig. 10.10) is another type of single-acting cylinder. It may have a stroke of 2 to 3 in.

10.7.4 Double-Acting Cylinders

A double-acting cylinder with cushioning is shown in Fig. 10.11. Like hydraulic cylinders, cushioning of pneumatic cylinders prevents hard impact and damage at the end of the stroke. As the cylinder moves to the end of its stroke, exhaust air is trapped in the cushioning cylinder by the cushioning piston. The trapped air is allowed to escape through the escape apertures. Rate of flow is controlled by adjustment of needle valves. (*Note the adjustment shown on the ANSI symbol.)* Pneumatic double-acting, double-rod cylinders are available. Double rod cylinders can resist a small lateral load, because the rod is supported at both ends.

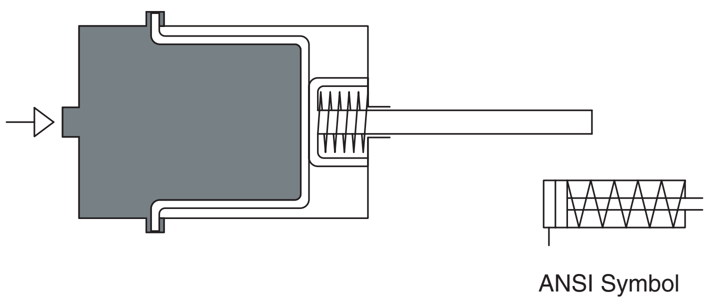


FIGURE 10.10
Rolling diaphragm cylinder. Reprinted with permission from Festo Corporation.

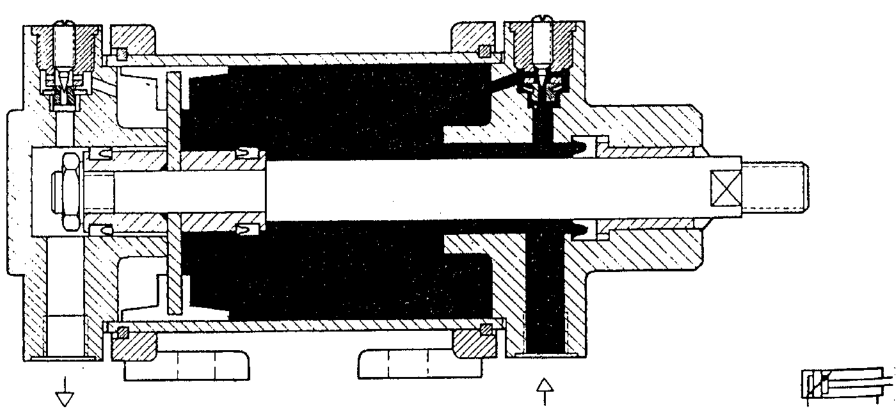


FIGURE 10.11
Double-acting cylinder with cushioning. Reprinted with permission from Festo Corporation.

10.7.5 Impact Cylinder

Forming operations require high kinetic energy, and impact cylinders are designed for this purpose. Normal cylinders have stroke velocities of 3 to 6 ft/s, but impact cylinders develop stroke velocities of 25 to 30 ft/s. Impact cylinders are used for short-stroke pressing, flanging, riveting, punching, and similar operations.

Referencing Fig. 10.12, cylinder space *(A)* is under pressure. When a directional control valve opens, pressure builds up in space *(B)*. The air in space *(A)* is then exhausted. When the pressure falls to the point where the force on area *(C)* is greater than the force on the annular area of the piston due to pressure in space *(A)*, the piston will move in direction *(Z)*. All of the piston surface is then exposed to the pressure in space *(B)*, and the force on the piston increases by a large factor. Compressed air can flow quickly from space *(B)* through the large hole; thus the piston is rapidly accelerated.

10.7.6 Free Air Consumption by Cylinders

Free air consumption is the air that must enter the compressor to produce the desired cylinder cycle rate.

$$Q_{fa} = C_r(A_c + A_r)sn \tag{10.6}$$

where Q_{fa} = air flow (in^3/min)
C_r = compression ratio (decimal)
A_c = cap end area (in^2)
A_r = rod end area (in^2)
s = stroke (in)
n = number cycles per minute

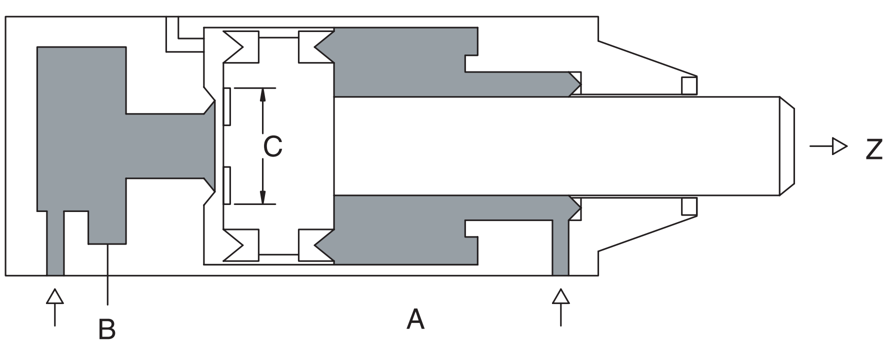


FIGURE 10.12
Impact cylinder. Reprinted with permission from Festo Corporation.

Compression ratio is given by,

$$C_r = \frac{15 + P_s}{15} \tag{10.7}$$

where P_s = supply pressure (psi)

Atmospheric pressure is 14.7 psi at sea level. Ambient pressure is typically rounded off to 15 psi.

The flow of air at line pressure to the cylinder is given by

$$Q = (A_c + A_r)sn \tag{10.8}$$

The compression ratio converts this flow to a flow of free air.

$$Q_{fa} = C_r Q \tag{10.9}$$

Charles' law states

$$P_1 V_1 = P_2 V_2 \quad \text{or}$$

$$V_1 = \frac{P_2 V_2}{P_1} \tag{10.10}$$

where P = absolute pressure
V = volume

In terms of flow, this equation becomes

$$Q_1 = \frac{P_2 Q_2}{P_1} \tag{10.11}$$

Note that $P_2/P_1 = C_r$, thus

$$Q_1 = C_r Q_2$$

which is equivalent to Eq. (10.9). Flow of air through the actuator (linear or rotary) is converted to the flow of free air using Eq. (10.9).

Properly, all airflows should be expressed in standard ft^3/min (scfm). Because ambient conditions are close to standard conditions, the terms *free* and *standard* are both used in commercial literature. In calculations, both terms define air at atmospheric pressure and 60°F.

Example Problem 10.2

What is the free air consumption of a double-acting cylinder with 2-in. bore, 0.5-in. rod diameter, and 4-in. stroke? The cylinder makes 10 cycles per minute, and operating pressure is 90 psi.

$$
\begin{aligned}
Q_{fa} &= C_r(A_c + A_r)sn \\
&= 7\left[\frac{2^2\pi}{4} + \frac{(2^2 - 0.5^2)}{4}\pi\right]4(10) \\
&= 1700 \text{ in}^3/\text{min} \\
&\approx 1.0 \text{ cfm}
\end{aligned}
$$

Generally, airflow in pneumatic systems is reported in ft^3/min or cfm. Since the abbreviation "cfm" is widely used in industry, this notation will be used in this chapter.

10.8 Motors

Pneumatic motors convert compressed air to rotary motion. Several of the common types are discussed below.

10.8.1 Types of Pneumatic Motors

10.8.1.1 Piston Motors

Piston motors are either radial (Fig. 10.13a) or axial (Fig. 10.13b) and are available in the power range of 2 to 25 hp. Maximum speed is around 5000 rpm. The axial piston motor has axial pistons, which act on a swashplate and produce rotary motion just as an axial piston hydraulic motor. The key to the radial piston motor is the valving, which directs compressed air to the correct piston. The radial pistons act on the cam in sequence to produce rotary motion.

10.8.1.2 Sliding-Vane Motor

Sliding-vane motors have simple construction, are lightweight, and are widely used. As might be anticipated, their principle of operation is the opposite of the sliding-vane compressor. Like the sliding-vane compressor, the vanes are forced out against the housing by centrifugal force. Small

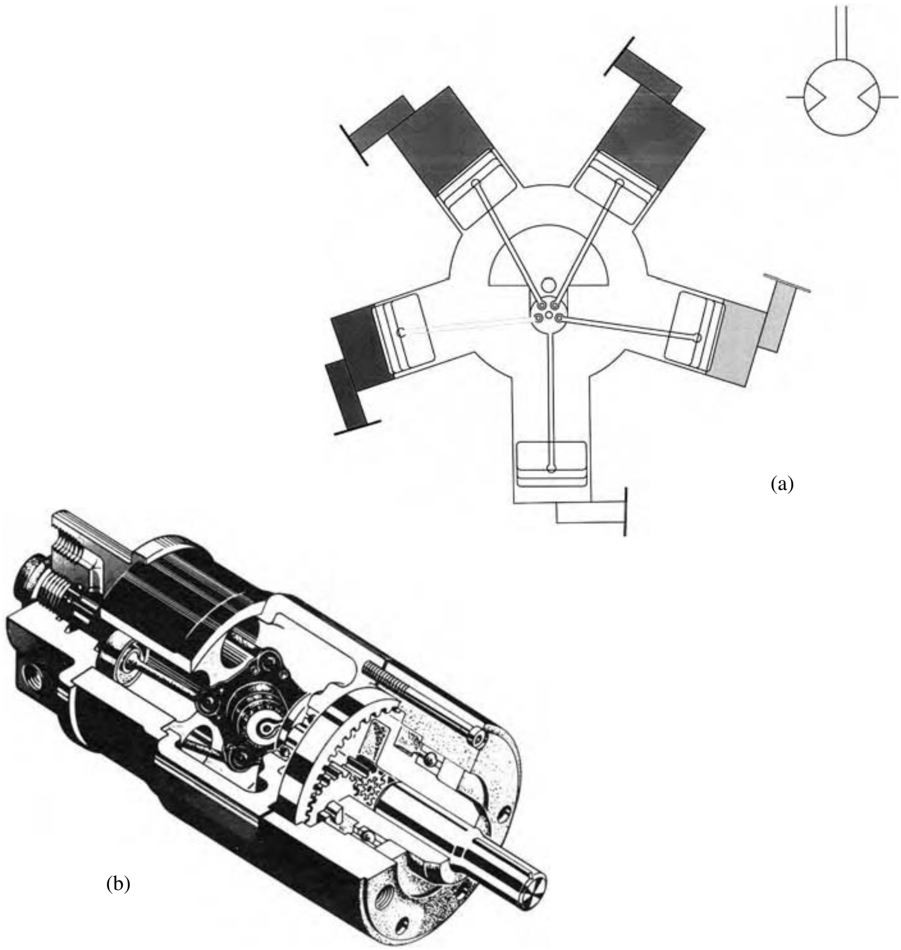


FIGURE 10.13
(a) Radial and (b) axial piston motors. Reprinted with permission from Festo Corporation.

amounts of compressed air are ported to the vane slots to force the vanes against the housing for a good seal. Compressed air flows into the chambers formed by the housing and two successive vanes. (Sliding-vane motors will have 3 to 10 vanes.) This air expands and causes the rotor to turn. Air is exhausted as the leading vane passes the exhaust port. Rotor speeds of 3000 to 8500 rpm are typical, and reversible units are available.

10.8.1.3 Gear Motors

Just as with a gear-type hydraulic motor, torque is generated by pressure against the tooth profiles of two meshed gears, either spur or helical gears. Gear motors have higher power ratings up to 60 hp.

10.8.2 Characteristics of Pneumatic Motors

The speed of pneumatic motors is controlled by throttling the flow of air to the motor. As with hydraulic motors, torque output is a function of available line pressure.

$$T_{mth} = \Delta P V_m / 2\pi \tag{10.12}$$

where T_{mth} = theoretical torque (lb_f-in)
V_m = motor displacement (in^3/rev)
ΔP = pressure drop across the motor (psi)

In this case, $\Delta P = P_s$ the line pressure.

Pneumatic motors slow down as load increases, and the torque increases simultaneously to the point at which it just matches the load. Anyone who has used a pneumatic tool that provides rotary motion (for example, an impact wrench) has observed how this works. The wrench turns slowly (low rpm, high torque) until the nut loosens, and then it speeds up (high rpm, low torque). The motor adjusts to match the load.

If airflow to a pneumatic motor is known, the output speed is given by

$$N_{mth} = 1728 Q / V_m \tag{10.13}$$

where N_{mth} = output speed (rpm)
Q = flow to motor (cfm)
V_m = motor displacement (in^3/rev)

Speed control is generally not an issue with a pneumatic motor application. It is understood that the motor speed changes significantly as load changes. (Speed change as a function of load change is much greater for pneumatic motors than hydraulic motors, because compressibility *and* leakage contribute to the speed change.) Because constant speed is not an issue, the definition and use of volumetric efficiency is generally not meaningful for pneumatic motors.

It is possible to measure airflow in a pneumatic distribution line. This measurement has meaning if it is made with line pressure held constant at the motor inlet and the line pressure is reported with the flow data. This flow, Q, substituted into Eq. (10.13) gives the theoretical motor speed at that line pressure. Now, a volumetric efficiency can be defined

$$e_{vm} = (N / N_{mth}) 100 \tag{10.14}$$

where e_{vm} = volumetric efficiency (%)
N = measured output speed while motor is delivering the torque required by the load (rpm)
N_{mth} = theoretical output speed [Eq. (10.13)] (rpm)

Note that N_{mth} for pneumatic motors is based on a flow measurement at a given pressure and is not the no-load speed used to define volumetric efficiency for a hydraulic motor.

As with hydraulic motors, a torque efficiency can be defined for pneumatic motors.

$$e_{tm} = (T_0/T_{mth})100 \tag{10.15}$$

where e_{tm} = torque efficiency (%)
T_0 = measured torque output (lb_f-in)
T_{mth} = theoretical torque [Eq. (10.12)]

Pneumatic power delivered to a pneumatic motor is

$$\mathcal{P}_{pneu} = 144PQ/33000 \tag{10.16}$$

where $\mathcal{P}_{pneu}$ = pneumatic power (hp)
P = line pressure at motor inlet (psi)
Q = flow at line pressure (cfm)

For Eq. (10.16) to have meaning, the flow must be measured at line pressure. Remember that flow through the motor changes with line pressure, so the two measurements must always be made simultaneously.

Mechanical power delivered by a pneumatic motor is defined the same way that mechanical power delivered by a hydraulic motor is defined [Eq. (5.6)].

$$\mathcal{P}_{out} = TN/63025 \tag{10.17}$$

where $\mathcal{P}_{out}$ = output mechanical power (hp)
T = measured output torque (lb_f-in)
N = measured output speed (rpm)

An overall efficiency can be defined by

$$e_{om} = (P_{out}/P_{pneu})100 \tag{10.18}$$

where e_{om} = overall efficiency (%)

Overall efficiency has more utility for pneumatic motors than either volumetric or torque efficiency. Generally, overall efficiency can be inferred from data supplied by manufacturers.

Example Problem 10.3

A pneumatic motor with 12.8 in^3/rev displacement is operated with 90 psi line pressure. Measured output speed and torque are 3000 rpm and 165 lb_f-in, respectively. Flow in the supply line was measured and found to be 28.5 cfm when the motor inlet pressure was 90 psi. Find the volumetric efficiency and overall efficiency at this operating point.

Theoretical output speed

$$\begin{aligned} N_{mth} &= 1728Q/V_m \\ &= 1728(28.5)/12.8 \\ &= 3847.5 \text{ rpm} \end{aligned}$$

Volumetric efficiency

$$\begin{aligned} e_{vm} &= (N/N_{mth})100 \\ &= (3000/3847.5)100 = 78\% \end{aligned}$$

Input pneumatic power

$$\begin{aligned} \mathcal{P}_{pneu} &= 144PQ/33000 \\ &= 144(90)(28.5)/33000 \\ &= 11.2 \text{ hp} \end{aligned}$$

Output mechanical power

$$\begin{aligned} \mathcal{P}_{out} &= TN/63025 \\ &= 165(3000)/63025 \\ &= 7.85 \text{ hp} \end{aligned}$$

Overall efficiency

$$\begin{aligned} e_{om} &= (\mathcal{P}_{out}/\mathcal{P}_{pneu})100 \\ &= (7.85/11.2) = 70\% \end{aligned}$$

Pneumatic motors are widely used for hand-held tools like drills, impact wrenches, and grinders. The following is a partial list of the advantages of these pneumatic tools:

1. They offer smooth adjustment of speed and torque as the load changes.
2. They are easily reversible.
3. They provide a high power-to-weight-ratio. (This characteristic is particularly important with hand-held tools.)
4. They are overload safe. (The operator can apply too much force and stall the motor without damaging it. Also, the air cushions rapid accelerations and decelerations of the tool and thus protects the moving parts of the motor.)
5. They are explosion proof. (In certain industries, a spark will ignite gases escaping from the process. Pneumatic motors do not emit sparks like electric motors, although the tool being driven may emit sparks.)
6. They work in dirty environment with little maintenance. (If the pressurized air is kept clean, and proper lubrication is added, pneumatic tools will have a long service life. Dirt in the surrounding environment does not enter the tool and cause wear.)

10.9 Additional Actuator Units

10.9.1 Cylinder with Mounted Air Control Block

This unit has a cylinder and DCV combined in the same assembly (Fig. 10.14). The cylinder cycles back and forth until the air supply is turned off. Stroke length is infinitely variable by adjusting the stops on the slide attached to the piston rod. These stops mechanically shift the DCV. The unit works well for piston speeds of 10 to 200 ft/min.

10.9.2 Hydropneumatic Feed Unit

A hydropneumatic feed unit has a pneumatic cylinder, hydraulic check cylinder, and DCV (Fig. 10.15). The purpose of the hydraulic cylinder is to ensure a uniform feed rate (stroke velocity). As the pneumatic cylinder moves, the hydraulic cylinder moves with it. (The two are mechanically yoked.) Oil is forced through a control valve as it flows from the rod end to the cap end. This valve can be adjusted to regulate speed in the range of 0.1

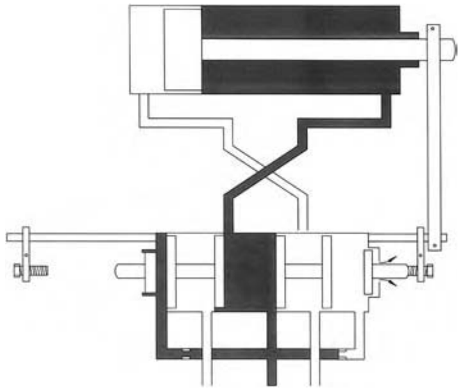


FIGURE 10.14
Cylinder with mounted air control block. Reprinted with permission from Festo Corporation.

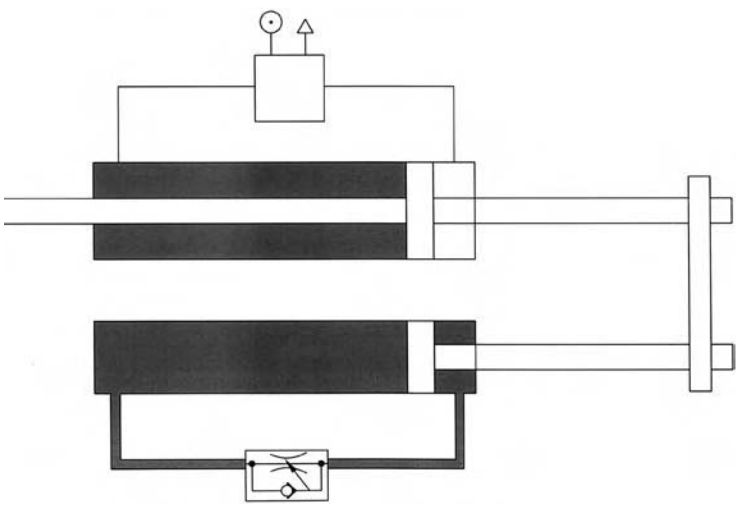


FIGURE 10.15
Hydropneumatic feed unit. Reprinted with permission from Festo Corporation.

to 20 ft/min. On the return stroke, the oil can pass quickly to the other side of the piston; thus, a rapid return is achieved. The DCV can be shifted mechanically, as shown in Fig. 10.14, to achieve continuous cycling.

10.9.3 Feed Unit

The feed unit (Fig. 10.16) can be used to feed flat strips, rods, or tubes, depending on the design of the clamping and feed grippers. The operating sequence is as follows:

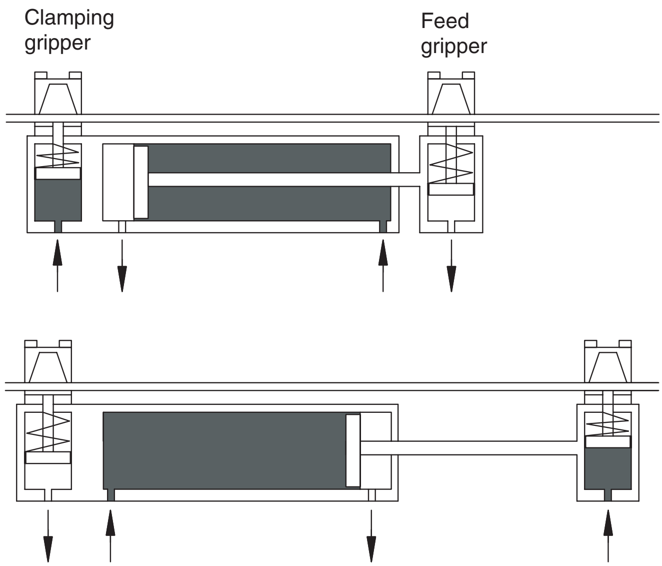


FIGURE 10.16
Feed unit. Reprinted with permission from Festo Corporation.

1. Feed gripper closes
2. Clamping gripper opens
3. Cylinder extends (feeds forward)
4. Clamping gripper closes
5. Feed gripper opens
6. Cylinder retracts
7. Cycle repeats

10.9.4 Rotary Index Table

A rotary index table is widely used in industry to position work pieces for sequential machining operations. At given intervals, compressed air indexes, or turns, the table a given number of degrees.

10.9.5 Air Cushion Sliding Table

Fixtures and heavy workpieces can be easily and accurately located in machine tools by using sliding tables (Fig. 10.17). Compressed air flows out a pattern of nozzles beneath the table, raising it approximately 0.002 to 0.004 in. To move a 300-lb_f fixture, a force of about 70 lb_f would typically be required. With a sliding table, the force required is only 7.0 lb_f.

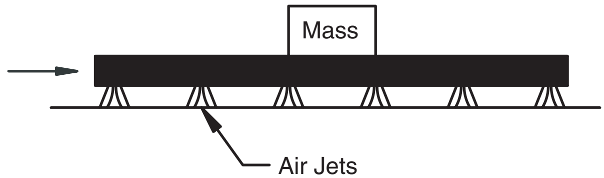


FIGURE 10.17
Air cushion sliding table. Reprinted with permission from Festo Corporation.

10.10 Valves

There are five groups of valves, categorized according to their function:

1. Directional control valves
2. Check valves
3. Pressure control valves
4. Flow control valves
5. Shut-off valves

10.10.1 Valve Symbols

There are several differences between pneumatic and hydraulic valve symbols. A triangle directly on the symbol represents an exhaust path without pipe construction (free exhaust) (Fig. 10.18). A triangle not directly connected to the symbol is an exhaust flow path with threads for a pipe connection (constrained exhaust) (Fig. 10.18). A selection of valve symbols is shown in Fig. 10.19.

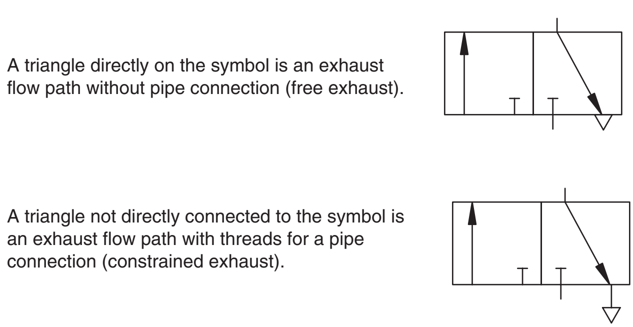


FIGURE 10.18
Valve symbols showing exhaust port.

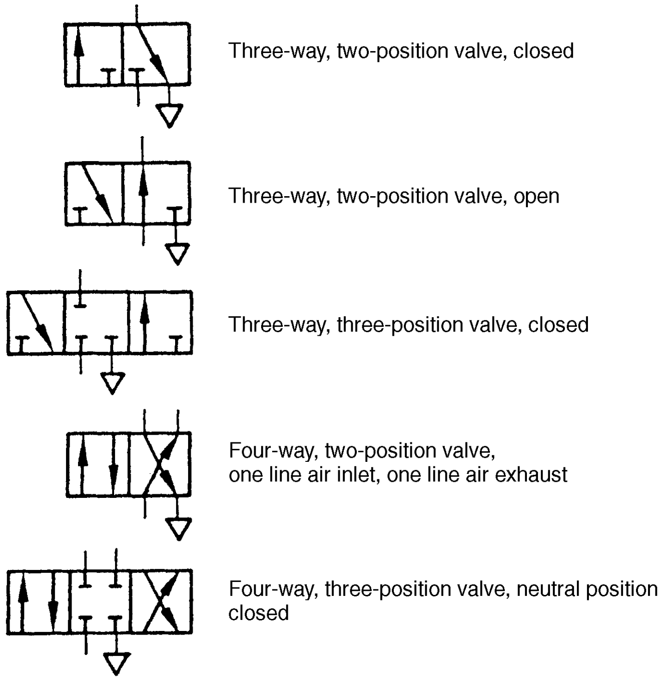


FIGURE 10.19
Valve symbols.

10.10.2 Valve Design

There are two main classifications according to design: poppet and slide. Poppet valves open by moving a ball or disc off the valve seat. Slide valves are spool-type valves similar to the hydraulic DCVs already studied.

10.10.3 Poppet Valves

The ball-type poppet valve (Fig. 10.20) is simple and inexpensive. A spring forces a ball against the valve seat (some type of elastic material). Actuation of the valve plunger causes the ball to be forced away from the seat, thus allowing flow from *(P)* to *(A)*. Like the ball-type valve, disc-type valves (Fig. 10.21) are simple, strong, insensitive to dirt, and have a long life. A small plunger movement opens a large cross-sectional area. Combinations of normally open or normally closed disc-type valves are incorporated in the same valve body to achieve the various designs shown in Fig. 10.19.

To avoid high actuating forces, mechanically controlled directional valves are equipped with an internal air pilot. Referencing Fig. 10.22, as the lever is depressed, a pilot line is opened, and compressed air flows to the diaphragm, which moves the valve disc downward, thus activating the valve. First, the

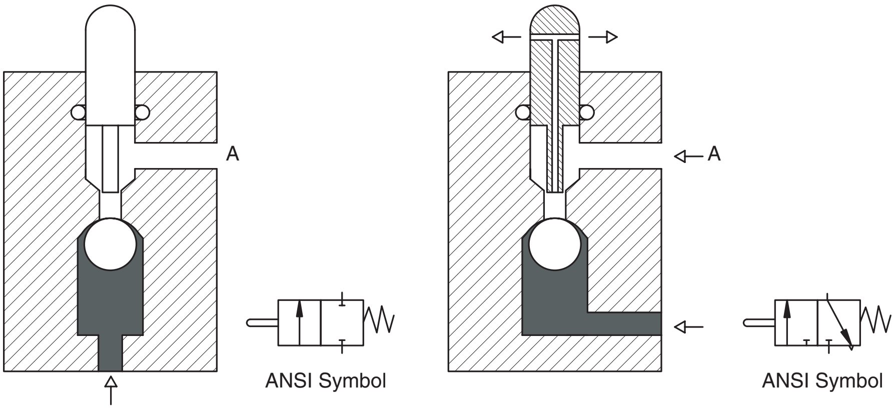


FIGURE 10.20
Ball-type pneumatic valve, normally closed. Reprinted with permission from Festo Corporation.

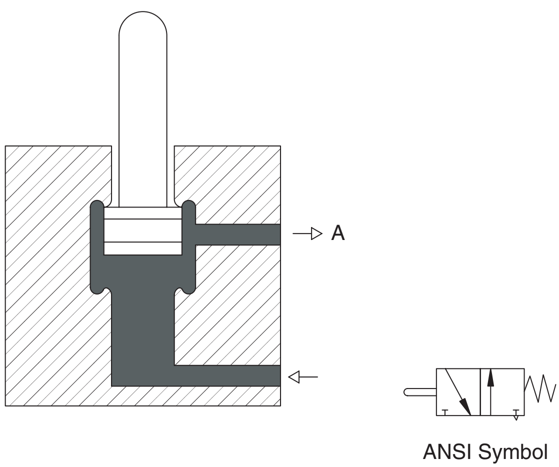


FIGURE 10.21
Disc-type pneumatic valve, normally open. Reprinted with permission from Festo Corporation.

line from A to R is closed, then the line from P to A is opened. P is the pressure line, A is the actuator line, and R is the release or exhaust line. These valves are called *servo-controlled* valves.

10.10.4 Slide Valves

The longitudinal slide valve uses a spool that moves longitudinally to connect or separate air lines. The required actuation force is low, because there

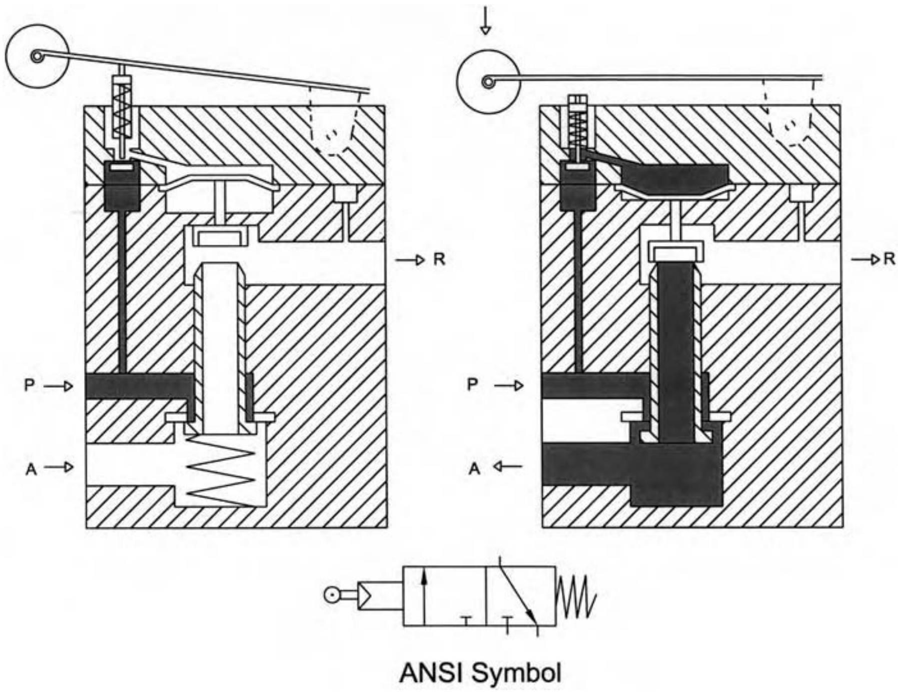


FIGURE 10.22
Servo-controlled three-way, two-position valve (disk seat principle). Reprinted with permission from Festo Corporation.

are no opposing forces due to compressed air or a spring. The actuation travel (distance to open a port) is longer than with a disc-type valve. Sealing is a problem. Clearance between slide and housing bore should not exceed 0.00008 to 0.00015 in., otherwise leakage losses will be too great.

An example of a longitudinal slide valve with pneumatic actuation is shown in Fig. 10.23. When compressed air is applied to the pilot spool through control port *(Y)*, the spool shifts to connect *(P)* to *(B)* and *(A)* to *(R)*. When compressed air is applied to the pilot spool through control port *(Z)*, the spool shifts to connect *(P)* to *(A)* and *(B)* to *(R)*. *Note that there is no center position. When the pilot pressure goes to zero, the valve stays in the position it is in.*

Plate slide valves operate by rotating two discs, generally with a hand lever (Fig. 10.24). When the lever is in the center position, flow to the actuator is blocked.

10.10.5 Other Valves

Other valves—for example, check, shuttle, sequence, pressure-reducing, and flow control—are all available for pneumatic circuits. These valves are all quite similar in function to the hydraulic valves already studied.

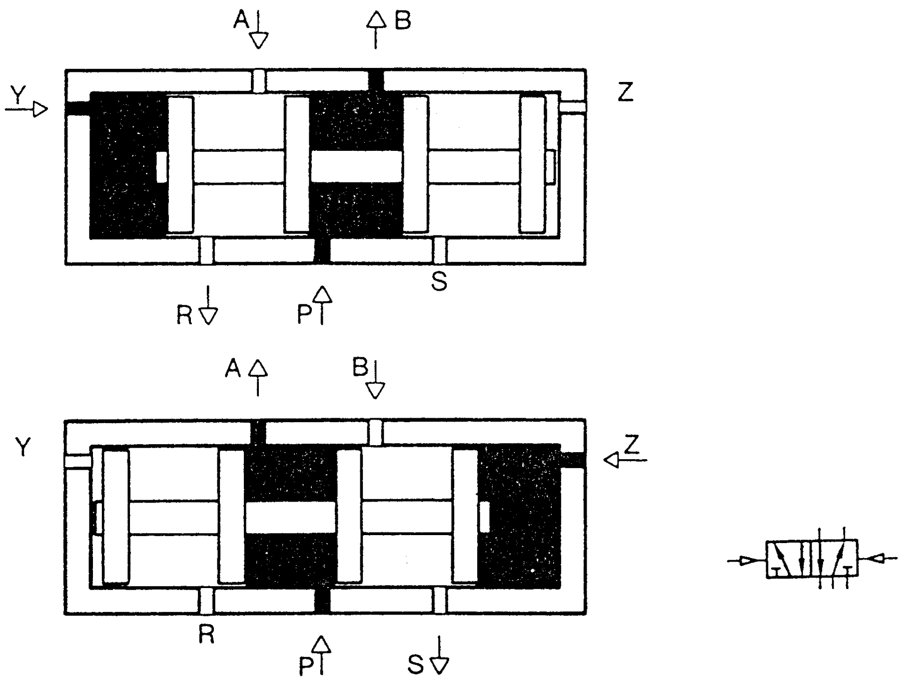


FIGURE 10.23
Longitudinal slide valve: four-way, two-position, double-air piloted.

10.11 Summary

There are many similarities between hydraulic and pneumatic systems. The key difference is the compressibility of the fluid (air) in pneumatic systems. Accurate speed control is not possible. Pneumatic motors slow down as load increases, and the torque simultaneously increases to the point at which it just matches the load.

Desirable features of pneumatics are as follows:

1. They can be used over a wide temperature range.
2. They are clean.
3. High actuator speeds are possible
4. They can be stalled without damage.

The undesirable features, besides compressibility, are the following:

1. Force is limited by operating pressure, which is typically 100 psi.
2. The cost of compressed air is relatively high.

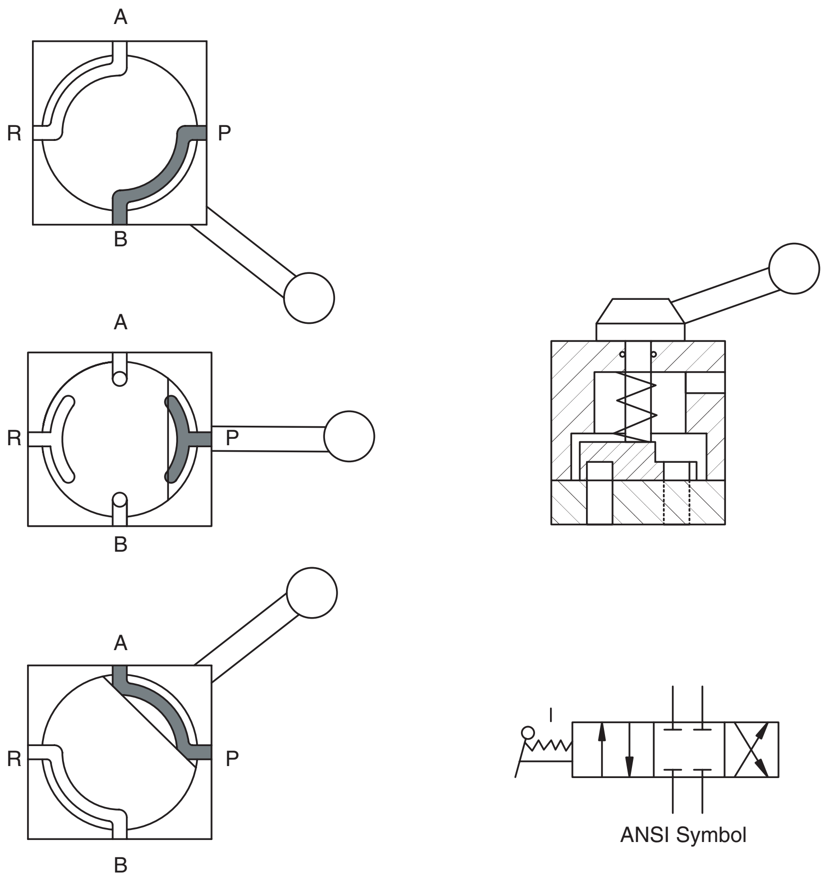


FIGURE 10.24
Plate slide valve (butterfly valve). Reprinted with permission from Festo Corporation.

Pneumatic systems have a compressor, which maintains a supply of air under pressure in a tank known as a *receiver*. Air is distributed from the receiver through a piping network to the actuators. Generally, the air is exhausted to atmosphere after passing through the actuator. Flow control valves, analogous to those studied for hydraulic systems, are available.

Pneumatic power is a key component of many manufacturing operations, particularly where high cycle speeds are required. As with hydraulic power, the advantages and disadvantages must be clearly understood to achieve optimal performance.

APPENDIX 10.1

Standard Conditions

Example: Conversion to Standard Conditions

Compressed air at 105 psi and 298 K is contained in a tank with an internal volume of 70 ft^3. What would be the volume of this air at standard conditions?

According to Boyle's law,

$$P_1 V_1 = P_2 V_2$$

$$P_1 = 15 \text{ psi (standard pressure)}$$

$$P_2 = 105 \text{ psi}$$

$$V_2 = 70 \text{ ft}^3$$

$$V_1 = \frac{P_2 V_2}{P} = \frac{105(70)}{15} = 490 \text{ ft}^3$$

We must now correct for temperature. Charles' law says that air expands by 1/273 of its volume for each 1 K it is heated. In general,

$$V_{T2} = V_{T1} + \frac{V_{T1}}{273}(T_2 - T_1)$$

where V_{T1} = volume at temperature T_1
V_{T2} = volume at temperature T_2

For our problem, T_1 = 298 K, T_2 = 289 K (standard temperature), and V_{T1} = 490 ft^3.

$$V_{T2} = 490 + \frac{490}{273}(289 - 298)$$

$$= 475 \text{ ft}^3$$

Air in the tank has a volume of 475 ft^3 at standard conditions.

Problems

10.1 The pneumatic system in a shop uses a 0.75-in. pipe to deliver compressed air to the various work stations on the shop floor. Maximum flow at any time is estimated to be 30 cfm. Supply pressure is 100 psi.

The following design data provide the pressure losses (psi) per 100 ft of pipe length. The data are for a 100-psi supply pressure.

	Pipe diameter (in.)			
Flow (cfm)	**0.5**	**0.75**	**1.0**	**1.25**
10	0.38	0.09	0.03	–
20	1.42	0.34	0.10	0.03
30	3.13	0.74	0.23	0.06
40	5.55	1.28	0.38	0.10
50	8.65	2.00	0.60	0.15

The size of the shop is 30 × 90 ft, and the delivery pipe is along one wall and across both ends. Total length is 155 ft.

1. Assume that the compressor efficiency is 85%. What size electric motor (hp) is required to ensure that 30 cfm can be delivered at 100 psi?
2. Calculate the maximum pneumatic power loss (hp) assuming that all 30 cfm travels 155 ft before being used at a pneumatic tool.

10.2 Receivers are pressure vessels, and their cost reflects this rating. They need to be large enough to meet the performance criteria, but cost considerations dictate that they be no larger than necessary.

A manufacturing operation uses an estimated 1000 cfm. On-off control of the compressor is used, and cycling less than once every four minutes is undesirable. Pressure can drop from 105 to 95 psi before the compressor cycles back on.

1. What size (ft^3) receiver is needed?
2. If the pressure can drop 15 psi before the compressor cycles on, what size receiver is needed?

10.3 A pneumatic cylinder is used for an "accept-reject" circuit. Parts pass the station on a conveyor at the rate of two parts per second. Approximately 10% have to be recycled, because the finish is not acceptable. Average number of strokes per minute is then 0.10 (2)(60) = 12.

The cylinder has a 3-in. bore and 1-in. rod. Stroke length is 8 in. Operating pressure is 100 psi. You may neglect any temperature rise.

1. Calculate the average free air intake (cfm) required for this cylinder.
2. Suppose the part size can be represented by a 5-in. cube, and they are spaced 2-in. apart on the conveyor. The cylinder must extend 8 in., push the part into the reject bin, and retract 8 in. to clear the conveyor before the next part reaches the accept-reject station. Calculate the maximum free air intake to meet this requirement when the conveyor speed is 840 in/min.

10.4 Calculate operating requirements of pneumatic motors.

1. A rotary vane motor with a displacement of 3 in^3/rev operates at 1500 rpm. Supply line pressure at the motor inlet is 90 psi. Compute the free air consumption. (Assume no temperature change.) If the motor efficiency is 85%, estimate the power output (hp).
2. Supply line pressure at a motor inlet is 100 psi. The motor must deliver a peak starting torque of 280 lb_f-in. Average torque requirement is estimated to be 210 lb_f-in. Compute the displacement required and estimate the free air consumption if the operating speed is 1000 rpm. Stall torque and operating torque efficiency are 89%. You may neglect temperature change.

10.5 A pneumatic motor has a displacement of 32.9 in^3/rev. Tests were run with an 100 psi supply pressure and the following data collected.

Motor speed (rpm)	Measured torque output (in-lb_f)	Free air consumption (cfm)
300	380	103
500	370	114
1000	345	162
1500	323	220
2000	295	275

Calculate the overall efficiency of this motor for each output speed. Report your results in a table, and also plot the overall efficiency vs. rpm.

11

Servo Valves

11.1 Introduction

Servo valves were developed to facilitate the adjustment of fluid flow based on changes in load motion. This chapter begins our study of closed-loop systems.

A typical open-loop system is shown in Fig. 11.1.

1. Input electric line voltage and current are converted to mechanical torque and rpm with an electric motor.
2. Input mechanical torque and rpm are converted to hydraulic pressure and flow with a pump.
3. Input pressure and flow are converted back to mechanical torque and rpm with a hydraulic motor.
4. The gearbox changes the ratio of torque and rpm to output the torque and rpm that will move the load in a way that meets the functional objective. Often, high speed and low torque are converted to low speed and high torque.

In many cases, it is important to maintain constant load speed as the magnitude of the load changes.

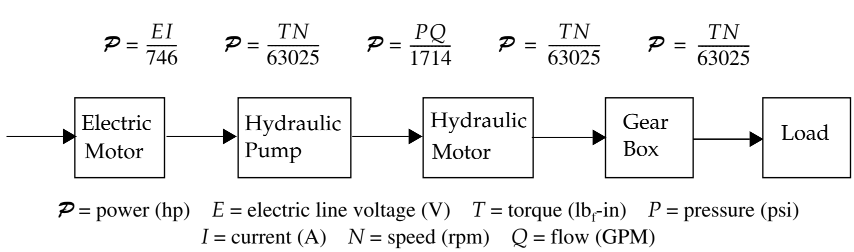


FIGURE 11.1
Typical open-loop system.

What happens when the load increases (torque requirement increases)? This increase reacts back through the system in the following way.

1. The torque requirement on the hydraulic motor increases. Pressure increases, and the leakage in the motor increases (volumetric efficiency decreases). The motor slows down.
2. The increase in pressure increase is immediately "seen" at the pump, and pump leakage increases. It delivers less flow for each revolution. This reduction in flow to the motor causes it to slow down even more.
3. The increase in pressure causes an increase in the input torque to drive the pump. This increase causes the electric motor to slow down. The subsequent reduction in pump speed reduces the pump output even more, which, in turn, reduces the motor speed even more.

With a closed-loop system, a feedback signal causes the pump displacement to increase until the motor output speed equals the desired output speed. Within a certain range, an increase in load does not produce a decrease in speed.

This chapter presents the fundamental information needed to design simple servo systems. The first half of the chapter deals primarily with concept, definition of terms, and the characteristics of servo valves. The second half presents an introduction to servo system analysis.

11.2 Concept

The meter-out circuit shown in Fig. 11.2 was discussed in detail earlier in the text. It is appropriate to use what we learned as a foundation for understanding how a servo valve functions in a circuit. In the circuit shown in Fig. 11.2, there is a provision to unload the pump when the closed center directional control valve (DCV) is centered. When extending, the flow from the cylinder passes through the flow control valve. The pressure drop, ΔP_{fc}, adds to the other pressure drops in the circuit to create a given pressure at the relief valve.

$$\Delta P_{tot} = \Delta P_{DCV} + \Delta P_{cyl} + \Delta P_{fc} \tag{11.1}$$

A force balance on the cylinder would reveal exactly how the pressure drop across the flow control affects the pressure at the relief valve. This detail is not needed here. Sufficient pressure is generated to cause the relief valve to crack

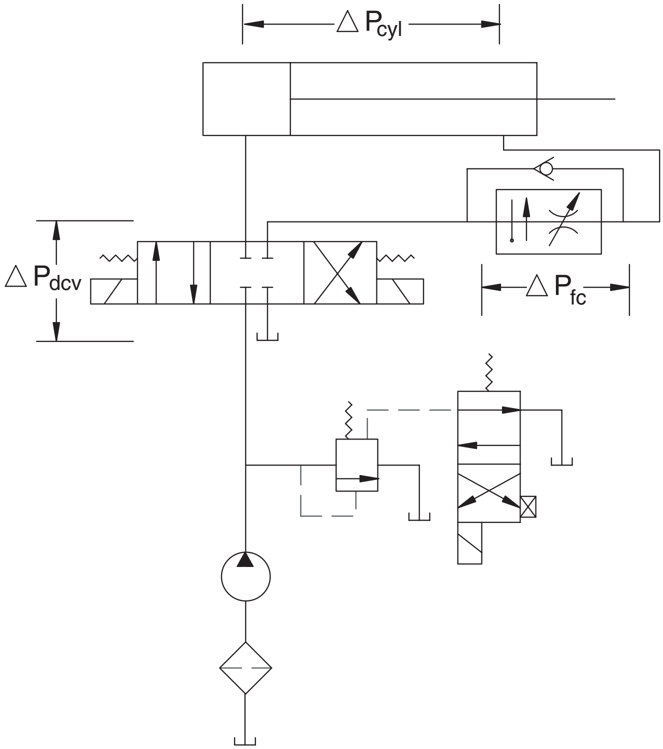


FIGURE 11.2
Meter-out circuit for controlling cylinder speed.

open and divert part of the flow back to tank. The subsequent reduction in flow to the cylinder causes it to slow to the desired speed.

Pressure compensation in the flow control valve causes it to increase, or decrease, the size of the orifice to maintain a constant ΔP_{fc} as flow changes. In addition, this particular flow control valve is also temperature compensated. It adjusts the orifice based on temperature change of the fluid.

One might think that the pressure-compensated, temperature-compensated flow control would maintain constant cylinder speed. Remember, however, that as pressure goes up due to a load increase, the output of the pump decreases due to increased leakage. In addition, there is increased leakage between the pressure line and the return line in the DCV. This leakage flow never makes it to the cylinder. These two effects both reduce cylinder speed. In many cases, the resultant reduction in cylinder speed is not critical. There are cases, however, where a more precise speed control is needed.

The circuit shown in Fig. 11.3 replaces the DCV and flow control valve with a servo valve. The rest of the circuit is the same except that a pressure line filter has been added. As explained in Chapter 8, servo valves are quickly damaged by contaminant in the fluid.

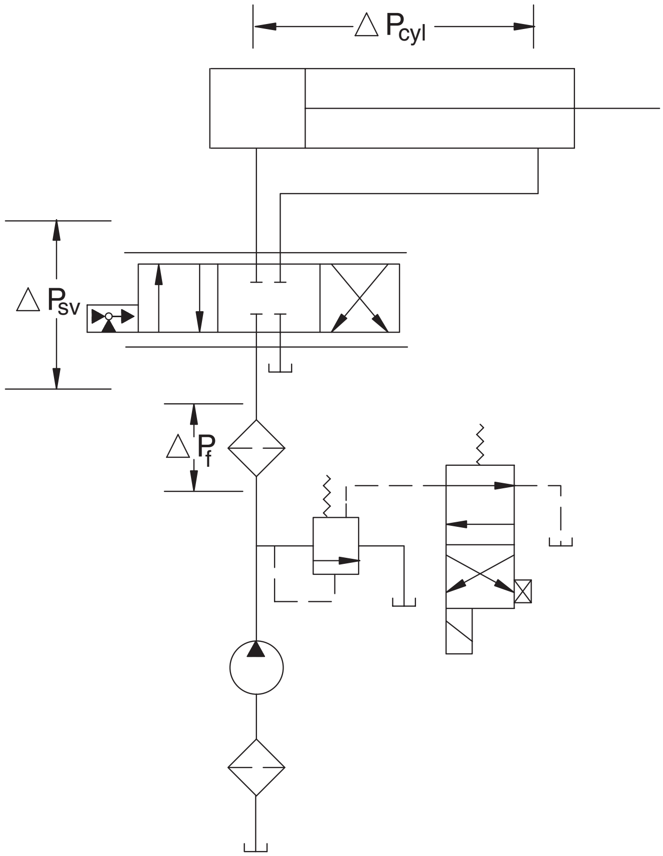


FIGURE 11.3
Servo valve used to control cylinder speed.

The characteristic flow across the relief valve in Fig. 11.3 is shown in Fig. 11.4. Total pressure at the relief valve for a given load is

$$\Delta P_{tot1} = \Delta P_{cyl1} + \Delta P_{sv1} + \Delta P_f \tag{11.2}$$

Corresponding flow across the relief valve is Q_{r1}. Suppose the load on the cylinder increases, and now the pressure drop across the cylinder is ΔP_{cyl2}. For the moment, we assume that the servo valve doesn't change. Total pressure at the relief valve is now

$$\Delta P_{tot2} = \Delta P_{cyl2} + \Delta P_{sv2} + \Delta P_f \tag{11.3}$$

and corresponding flow across the relief valve (Fig. 11.4) is Q_{r2}. Less flow is reaching the cylinder, and it slows down.

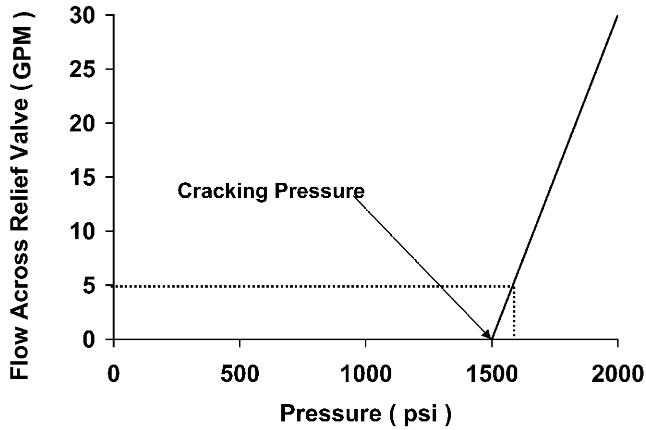


FIGURE 11.4
Characteristics of relief valve in circuit shown in Fig. 11.3.

Now we suppose that the servo valve is changed such that the pressure drop is

$$\Delta P_{tot2} = \Delta P_{cyl2} + \Delta P_{sv2} + \Delta P_f = \Delta P_{tot1} \tag{11.4}$$

The flow across the relief valve is Q_{r1}, and the speed of the cylinder is back to its original value.

The above example illustrates a key point. *In many applications, a servo valve (and related circuitry) is just a means of programming an orifice in the circuit.* The rest of the circuit, pump, relief valve, actuator, etc., have the characteristics we have discussed in previous chapters. These characteristics, along with the characteristics of the servo valve, define the performance of the circuit.

The servo valve is not a flow source. If all the fluid is being dumped across the relief valve, none will reach the actuator. Also, the servo valve can be completely open, and the actuator speed still be too slow if the pump output is too low. All we have previously learned about the various components in fluid power circuits will be used in this study of servo valve circuits.

11.2.1 Feedback

Feedback is an intuitive concept. Each time we reach for an object, we slow our hand as it approaches the object so that it is captured with minimal impact. There is visual feedback that our brain uses to control the arm and ensure that it reaches the target and stops.

An equipment operator, for example a backhoe operator, watches the bucket and controls the flow of oil to the cylinder. Experienced operators develop such a feel for the controls that they can control the bucket almost as they control their own arms. Let's review what happens when the operator is controlling cylinder speed. The pump is a fixed displacement pump, so it puts out a constant flow of oil. The operator cracks open the DCV to create an orifice, and resultant pressure drop causes the relief valve to open and bypass part of the pump flow back to tank. The orifice is continuously varied to obtain the desired cylinder speed. The operator provides the feedback, and the DCV functions like a servo valve. It is helpful to think of the servo valve as a *programmable* orifice. Feedback continuously programs the size of the orifice (required pressure drop) to achieve a given actuator speed or position.

11.2.2 Programmable Orifice

A servo valve with a line-to-line lap condition (the spool lines up precisely with the valve body) is shown in Fig. 11.5a. Just as with a four-way DCV, the valve has four ports: pressure, return, Port *A*, and Port *B*. When the spool is shifted, two orifices are created. There is an orifice between the pressure port and Port *A*, and an orifice between Port *B* and the return port.

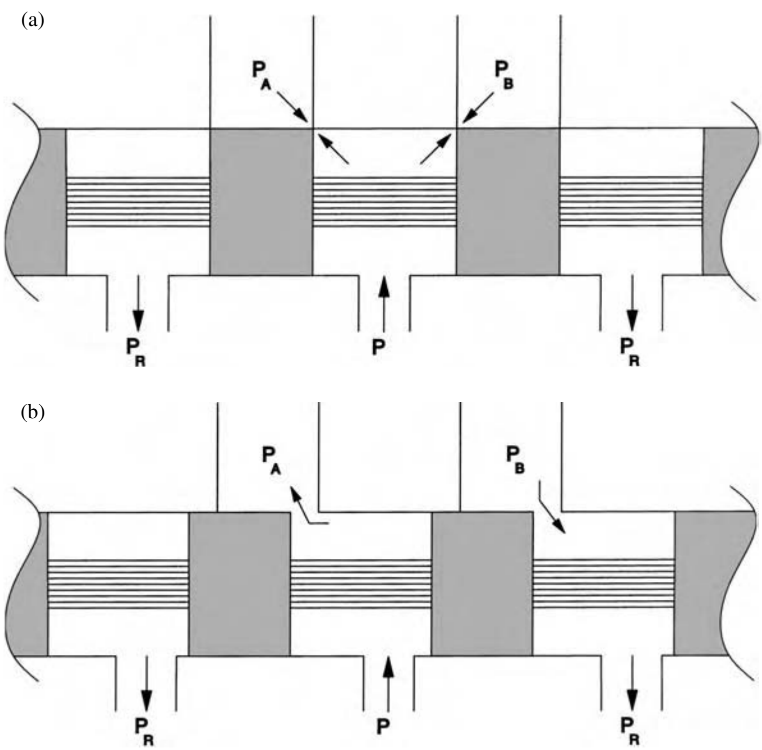


FIGURE 11.5
(a) Four-way servo valve with line-to-line lap condition and (b) shifted four-way servo valve showing pressure drops through the valve.

For purposes of discussion, we assume that Ports *A* and *B* are connected to a double-rod cylinder.

Flow through the valve to the cylinder is given by the orifice equation,

$$Q = C_d A_o \sqrt{2g/\gamma}\sqrt{P_s - P_A} \tag{11.5}$$

where Q = flow (in³/s)
C_d = orifice coefficient (decimal)
A_o = orifice area (in²)
γ = fluid density (lb_f/in^3)
P_s = supply pressure (psi)
P_A = Port *A* pressure (psi)
g = gravitation constant (in/s^2)

In like manner, flow from the cylinder back through the valve is given by

$$Q = C_d A_o \sqrt{2g/\gamma}\ \sqrt{P_B - P_R} \tag{11.6}$$

Load pressure is defined to be

$$P_L = P_A - P_B \tag{11.7}$$

It follows that

$$P_A = P_L + P_B \tag{11.8}$$

and substitution into Eq. (11.5) gives

$$Q = C_d A_o\ \sqrt{2g/\gamma}\ \sqrt{P_s - (P_L + P_B)} \tag{11.9}$$

We assume symmetry, meaning that the orifices on both sides have the same properties. Equating Eqs. (11.9) and (11.6),

$$\sqrt{P_s - (P_L + P_B)} = \sqrt{P_B - P_R} \tag{11.10}$$

Squaring and subtracting from both sides,

$$P_s - P_L - P_R = 2(P_B - P_R) \tag{11.11}$$

Substituting back into Eq. (11.6),

$$Q = C_d A_o\ \sqrt{g/\gamma}\ \sqrt{P_s - P_L - P_R} \tag{11.12}$$

Often, the equation is written

$$Q = k\sqrt{P_s - P_L - P_R} \tag{11.13}$$

where $k = C_d A_o \sqrt{g/\gamma}$

If the conductor from the servo valve back to the reservoir is properly sized, $P_R << P_s$, and the equation is written

$$Q = k\sqrt{P_s - P_L} \tag{11.14}$$

Servo valves are rated by the manufacturer at a given pressure drop, typically 1000 psi. Rated current is applied so that the valve is in its full open position. Pump speed is increased until a 1000 psi ΔP is measured across the servo valve. Once the 1000 psi ΔP is obtained, the flow is measured, and this flow is used to specify the valve *size*.

Typical performance curves reported in a servo valve technical specification sheet (tech sheet) are shown in Fig. 11.6. Four curves are shown—one for each of the 100%, 75%, 50%, and 25% open positions corresponding to input current of 100%, 75%, 50%, and 25% of rated current. Each of these curves is a plot of Eq. (11.14) with a different k corresponding to the given open position. Considering the 100% curve first, note that, when load pressure is 100% of supply pressure, $P_L = P_s$, flow through the valve is zero. When $P_L = 0$, the control flow is 100% of rated flow.

The curves on the left-hand side of Fig. 11.6 correspond to valve openings in the opposite direction. When pressure at Port *B* is greater than at Port *A*, the sign of the load pressure is reversed, and flow through the valve is in the reverse direction.

It is instructive to work a problem with some "typical" values for the various parameters. Suppose the supply pressure is $P_s = 1000$ psi, and the load pressure is $P_L = 600$ psi. The oil has a specific gravity of 0.83, and the orifice discharge coefficient is $C_d = 0.80$. At the fully open position (rated current applied to the valve), the area of the orifice is $A_o = 6.7 \times 10^{-3}$ in^2. What will be the flow of oil through this valve?

$$Q = k\sqrt{P_s - P_L} \tag{11.15}$$

where $k = A_o C_d \sqrt{g/\gamma}$

The density of the oil is

$$\gamma = 0.83(62.5) = 57.88 \text{ lb}_f/\text{ft}^3$$

$$= 0.030 \text{ lb/in}^3$$

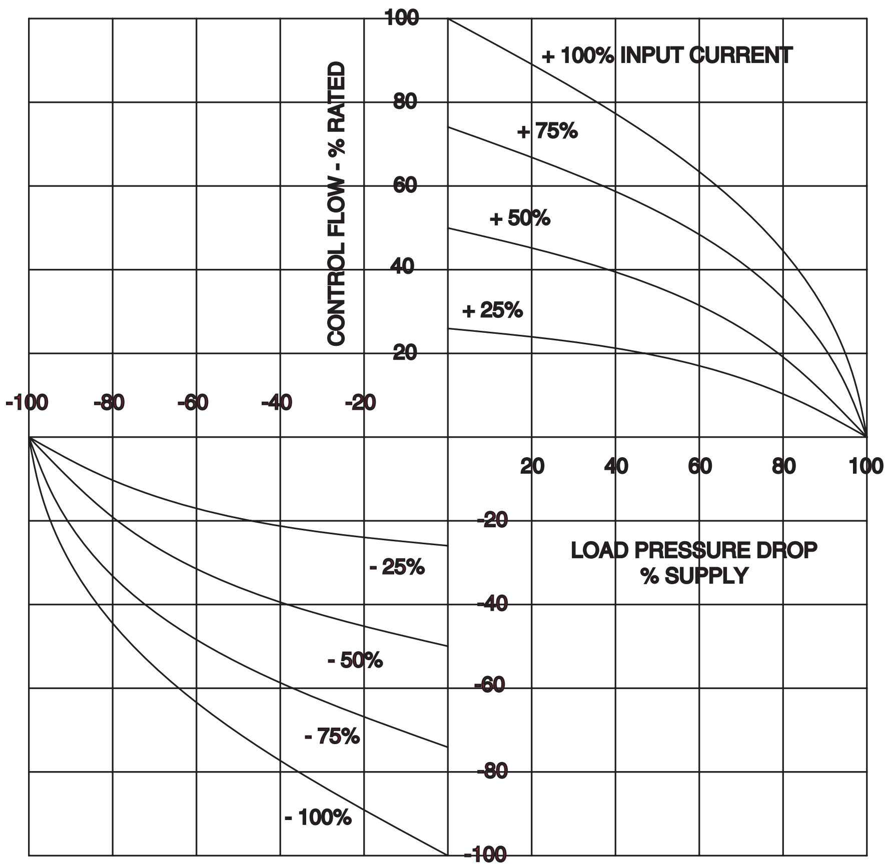


FIGURE 11.6
Typical flow vs. pressure curves for a servo valve.

Substituting,

$$k = 6.7 \times 10^{-3}(0.80)\sqrt{\frac{386\ \text{in/s}^2}{0.030\ \text{lb}_\text{f}/\text{in}^3}}$$

$$= 0.608\ \text{in}^4\text{lb}_\text{f}^{-0.5}s^{-1}$$

$$Q = 0.608\ \text{in}^4\text{lb}_\text{f}^{-0.5}s^{-1}(1000 - 600\ \text{lb}_\text{f}/\text{in}^2)^{0.5}$$

$$= 12.2\ \text{in}^3/s,\ \text{or}\ \ 3.17\ \text{GPM}$$

When pressure drop across the valve is $P_v = P_s - P_L = 1000 - 600 = 400$ psi, flow through the valve is Q = 12.2 in^3/s.

Suppose the relief valve is adjusted to increase supply pressure to P_s = 1500 psi, and the valve is opened to the same position, $A_o = 6.7 \times 10^{-3}$ in^2, what is the flow through the valve?

$$\begin{aligned} Q &= k\sqrt{P_v} \\ &= 0.608(900)^{0.5} \\ &= 18.2 \text{ in}^3/\text{s} \end{aligned} \tag{11.16}$$

Supply pressure increased from 1000 to 1500 psi, and flow increased from 12.2 to 18.2 in^3/s. At constant load pressure, flow through the valve increases as supply pressure increases.

This example problem reinforces the statement made earlier. The servo valve is one component (a programmable orifice) of the system. In this case, the relief valve setting was increased by 50%, and the flow through the valve was increased by 50%. Actuator speed is correspondingly increased.

11.2.3 Control of Pump Displacement

A variable displacement pump is more expensive, but it adds tremendous options for the design of a fluid power circuit. A servo valve can be used to change pump displacement and thus change actuator speed. The open-loop system shown in Fig. 11.1 can be made a closed-loop circuit by mounting a transducer on the output shaft of the gearbox. The resulting voltage is compared to a voltage corresponding to the set speed. The difference between the two voltages is used to increase (or decrease) pump displacement as shown in Fig. 6.27. The error signal causes the torque motor to rotate and move the servo valve spool. Fluid is directed to the control pistons, which extend (or retract) and change the angle of the swash plate, thus changing the displacement of the pump. If the various components (servo valve and associated circuitry) work together correctly, the pump output is continuously adjusted to maintain constant load speed as the load changes within some range.

When a servo valve is mounted on a variable displacement pump as shown in Fig. 6.27, the combination is known as a *servo pump.* The requirement to create a pressure drop across the servo valve (and dump fluid across the relief valve) to control actuator speed is eliminated. A servo pump is much more efficient for controlling actuator speed; however, it does not provide as accurate positioning as the servo-actuator configuration discussed in the previous section.

11.2.4 Basic Servo Systems

There are four basic servo systems as shown in Fig. 11.7. The two servo-actuator systems are (1) valve-motor and (2) valve-cylinder. These systems are

(1)

(2)

(3)

(4)

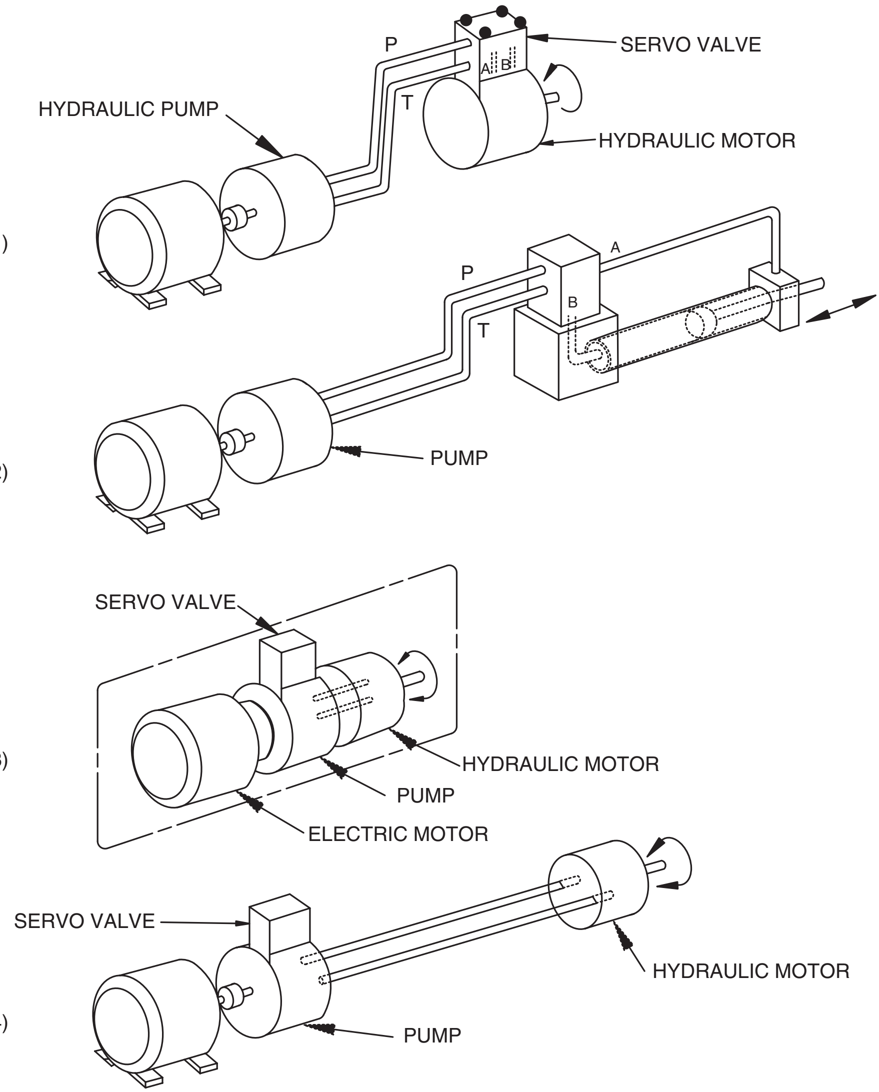


FIGURE 11.7
Four basic servo systems: (1) valve-motor, (2) valve-cylinder, (3) servo pump-motor, and (4) servo pump-motor (split). Reprinted with permission from Electrohydraulic Servo Systems, James E. Johnson, Penton Media, Inc., Cleveland, Ohio.

often referred to as *servo motors* and *servo cylinders*. The recommended procedure is to mount the valve directly on the actuator. This avoids a column of compressed fluid in the lines and increases the natural frequency of the system, which increases positioning accuracy, as later analysis will show.

The two servo pump systems are (3) servo pump-motor and (4) servo pump-motor (split). Most accurate speed control is given by the servo pump-motor, because this configuration avoids a column of compressed fluid in the lines and thus gives a higher natural frequency for the system. There are times, however, when the pump and motor cannot be packaged together. The split configuration has the largest position error of the four configurations.

11.3 Servo Valve Construction

This section gives some detail on the construction of servo valves. It is important to remember that a servo valve is really just a carefully machined spool-type directional control valve. The spool is shifted with a torque motor mounted on top of the valve rather than a solenoid mounted at the end of the spool.

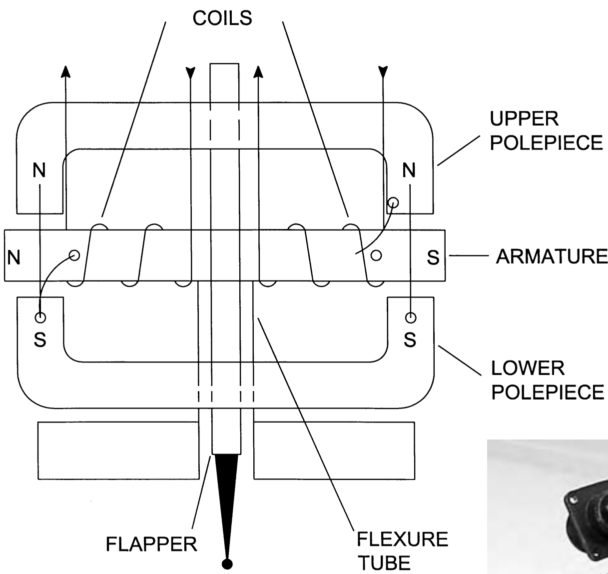
There are three types of servo valves.

1. *Single-stage.* This valve has one spool. The torque motor must supply enough torque to shift the spool against the pressures that act on the spool.
2. *Two-stage.* In this valve, the first stage is called the *pilot stage.* The torque motor shifts the pilot spool, which directs flow to shift the second stage. The second stage supplies flow to the actuator.
3. *Three-stage.* In this valve, the pilot stage shifts the second stage, which shifts the third stage. Three-stage valves are used for applications with high flow and high pressures. Large forces are required to shift the third stage, which directs the high-volume flow to the actuator.

11.3.1 Torque Motor

As shown in Fig. 11.8, a torque motor consists of an armature, two coils, and two pole pieces. The coils are wrapped on the armature, and the armature is mounted such that the ends are positioned in the center of the air gap between the upper and lower pole pieces. Permanent magnets are incorporated in the pole pieces. When current is supplied to the coils, the armature rotates clockwise or counterclockwise, depending on polarity produced in the armature. Current in the opposite direction produces the opposite polarity and the opposite rotation.

A key to the operation of the torque motor is the mounting of the armature to a *flexure tube.* This mount bends as the armature turns. The armature stops pivoting when the torque produced by magnetic attraction equals the restraining torque produced by deflection of the flexure tube. This design prevents the armature from touching the pole pieces.



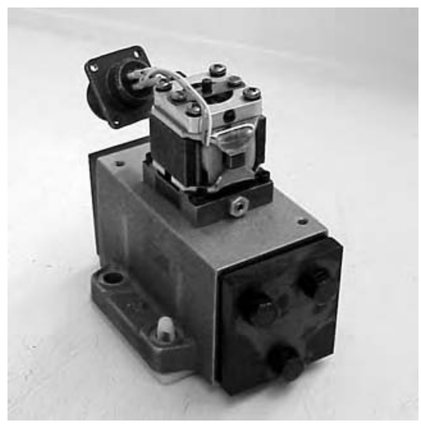


FIGURE 11.8
Construction of a torque motor. Photograph courtesy of Moog Inc.

The torque motor coil can be immersed in oil, classified as a *wet torque motor,* or operated dry. Even though a wet torque motor has the advantage of cooler operation, most servo valves use dry torque motors, because the magnets tend to attract metal particles circulating in the fluid, and this eventually causes failure.

A dry torque motor is a very reliable component. A current overload will burn out the coil; otherwise, the design life of a torque motor is indefinite.

The torque motor can move the pilot spool directly when it rotates. In Fig. 11.9, a rod connects the pilot stage spool to the armature. When the armature rotates, it shifts the pilot stage, which ports fluid to the second stage, causing it to shift. Note the feedback linkage between the pilot stage and the second stage. Suppose the second stage shifts to the right to connect *P* to *A* (the supply pressure to Port *A*). The feedback linkage rotates counterclockwise about the fulcrum, and the top slides the pilot stage sleeve to the left. When this sleeve centers on the pilot spool, flow is cut off to the second-stage spool, and it stops moving. The feedback linkage ensures that the second-stage spool always has the same relative position as the pilot-stage spool. If the pilot spool shifts 25% to the right, the second-stage spool shifts 25% to the left.

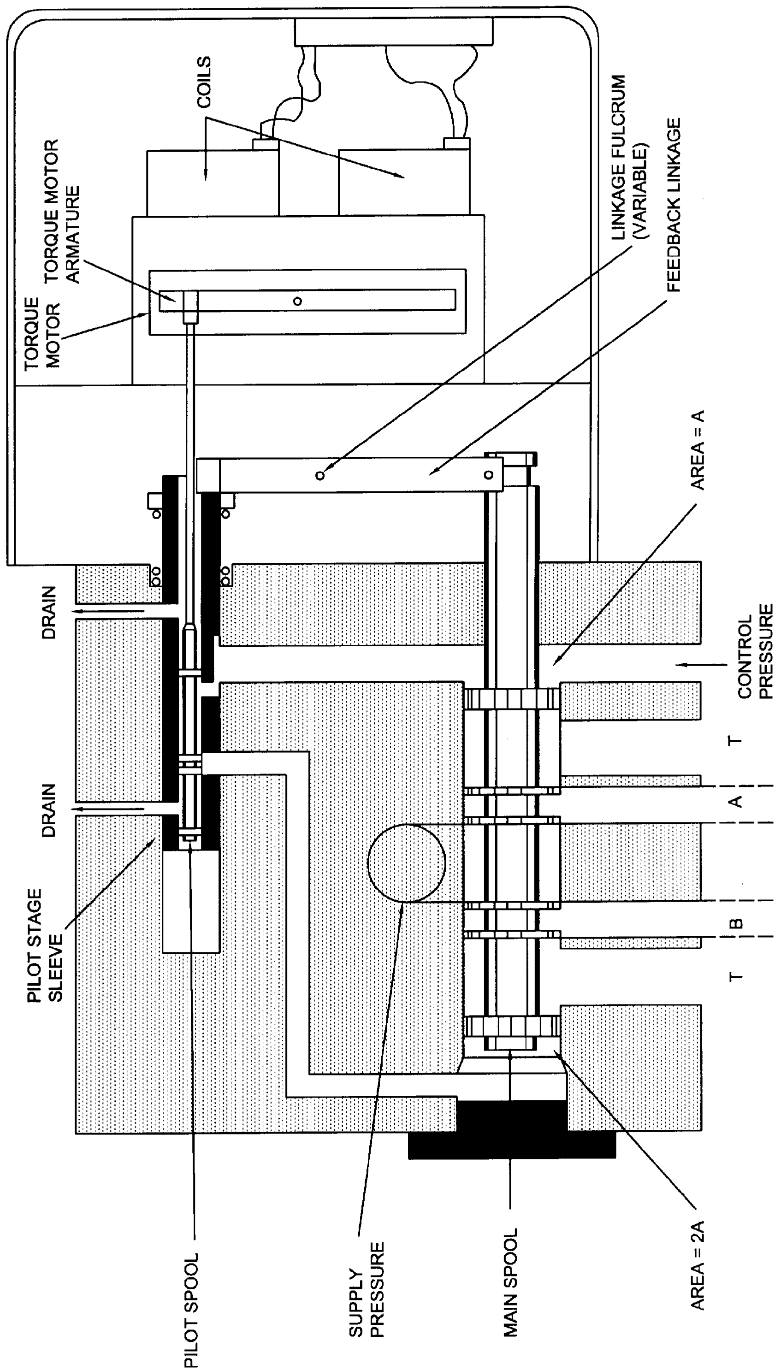


FIGURE 11.9
Two-stage servo valve with spool pilot stage directly linked to torque motor armature. Reprinted with permission from Electrohydraulic Servo Systems, James E. Johnson, Penton Media, Inc., Cleveland, Ohio.

Note that the linkage fulcrum is adjustable so that the valve can be "tuned" to exactly align the spools. More detail on this principle of providing feedback between components will be given later.

11.3.2 Methods for Shifting Servo Valve Spool

The torque motor can shift the spool directly, as discussed in the previous section (Fig. 11.9), or indirectly. There are three *indirect* methods for shifting the spool.

1. Single flapper nozzle
2. Double flapper nozzle
3. Jet pipe

These indirect methods were developed to provide a higher shifting force without the expense of a pilot-stage spool. The two most widely used methods are discussed here.

11.3.2.1 Double Flapper Nozzle

A diagram of the double flapper nozzle is shown in Fig. 11.10. Supply pressure (the setting of the relief valve or the setting of a pressure compensated pump) is supplied to the points identified with P_s (Fig. 11.10). Fluid flows across the fixed orifices and enters the center manifold. Orifices are formed on each side between the flapper and the opposing nozzles. As long as the flapper is centered, the orifice is the same on both sides and the pressure drop to the return is the same. Pressure at *A* equals the pressure at *B,* and the spool is in force balance. Suppose the torque motor rotates the flapper clockwise. Now, the orifice on the left is smaller than the orifice on the right, and the pressure at *A* will be greater than the pressure at *B*. This pressure differ-

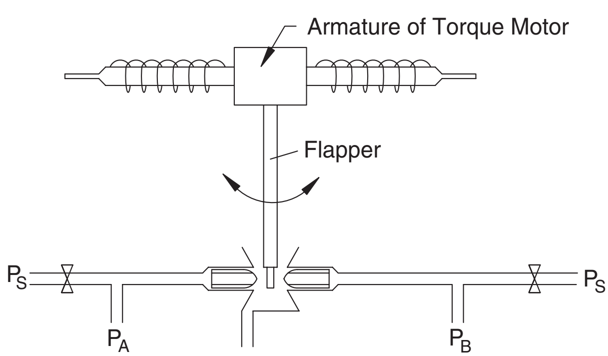


FIGURE 11.10
Diagram of double flapper nozzle as a first stage for a two-stage servo valve (second stage not shown). Reprinted with permission from Electrohydraulic Servo Systems, James E. Johnson, Penton Media, Inc., Cleveland, Ohio.

ence shifts the spool to the right. As the spool shifts, it deflects a feedback spring. The spool continues to move until the spring force produces a torque that equals the electromagnetic torque produced by the current flowing through the coil around the armature. At this point, the armature is moved back to the center position, the flapper is centered, the pressure becomes equal at *A* and *B*, and the spool stops. The spool stays in this position until the current through the coil changes. Because of the feedback spring, the spool has a unique position corresponding to each current through the coil ranging from 0 to rated current. At rated current, the spool is shifted to its full open position.

A cutaway of a two-stage valve with double flapper nozzle for the first stage is shown in Fig. 11.11. Note that the spool slides in a bushing. It is the relationship between this bushing and the spool that establishes the opening to Ports *A* and *B*. An adjustment, known as the null adjust, is provided to slide this bushing left or right and bring it into precise alignment with the spool when no current is supplied to the valve. This adjustment ensures that the valve is mechanically centered.

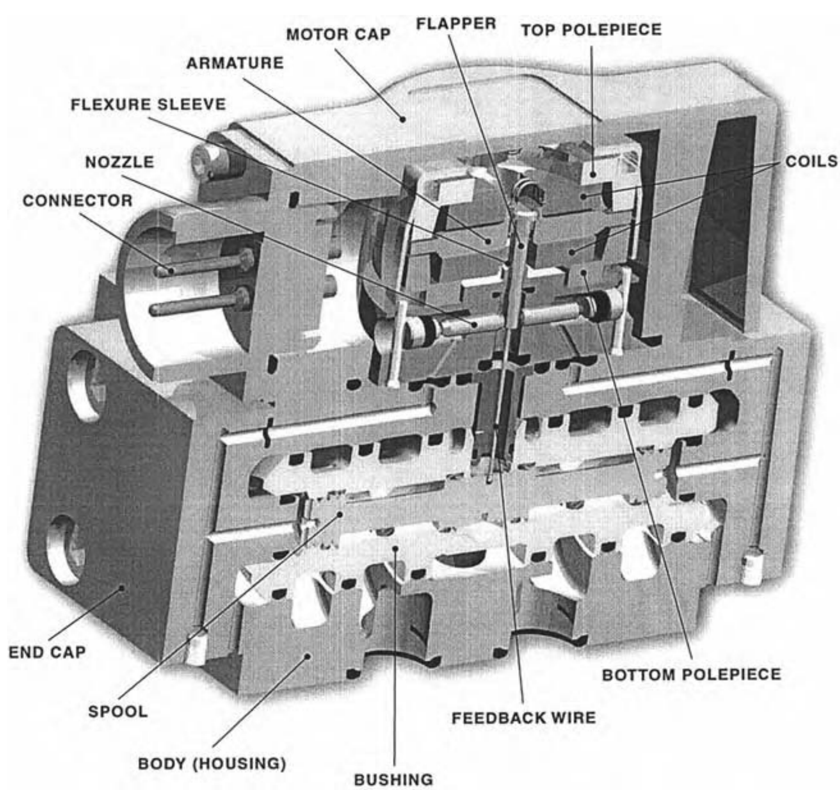


FIGURE 11.11
Cutaway of two-stage servo valve with double flapper nozzle for a first stage. Courtesy of Moog Inc.

11.3.2.2 Jet Pipe

The jet pipe design (Fig. 11.12) is less sensitive to contamination than the flapper nozzle. This design uses fluidics technology. A nozzle, attached to the armature, directs a stream of fluid (pressure P_s) at two receivers. When the nozzle is centered (no current through the armature coils to cause it to turn), the two receivers are hit by the stream of fluid in like manner, and the resultant pressure on the ends of the spool is equal. When the armature turns, the stream of fluid from the nozzle impinges more directly on one nozzle, and the resulting pressure imbalance shifts the second-stage spool. There is a feedback spring for this design that works the same as for the double flapper nozzle. The spool shifts until the spring force produces a torque

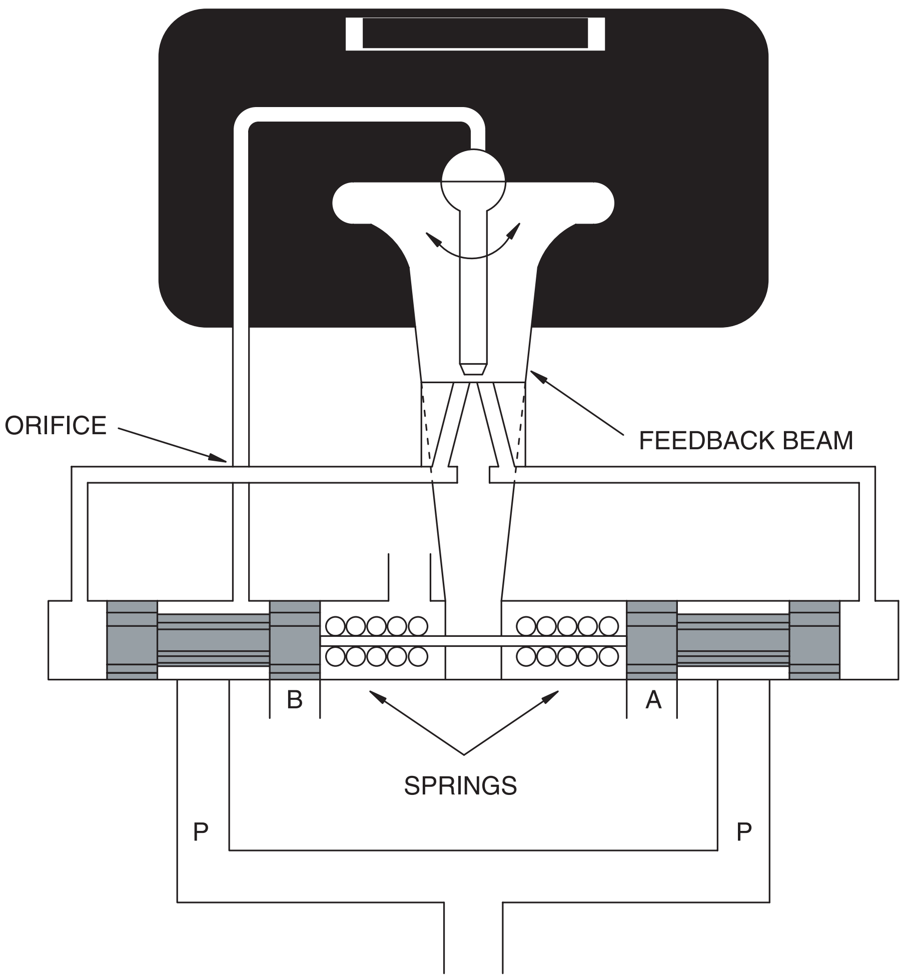


FIGURE 11.12
Diagram of jet pipe used as a first stage for a two-stage servo valve.

equal to the electromagnetic torque. At this point, the armature is moved back to the center position, the nozzle is centered, pressure at the two receivers is equal, the forces on the spool are equal, and it is held in position. A change in coil current changes the spool to a position that corresponds to the new level of current.

In summary, a two-stage servo valve can have four different first-stage designs. The torque motor can shift a pilot stage directly, which in turn shifts the second-stage spool. A single flapper nozzle, double flapper nozzle, or jet pipe can be used as a first stage to shift the second-stage spool indirectly.

11.3.3 Valve Construction

11.3.3.1 Port Shape

Servo valve bodies are machined with three common port shapes as shown in Fig. 11.13. The effect of port shape on flow is shown in Fig. 11.14. A full annulus port gives the highest flow per unit of spool displacement or, correspondingly, per unit of current input to the torque motor. The valve flow characteristics will be important when the valve transfer function is defined in a later section.

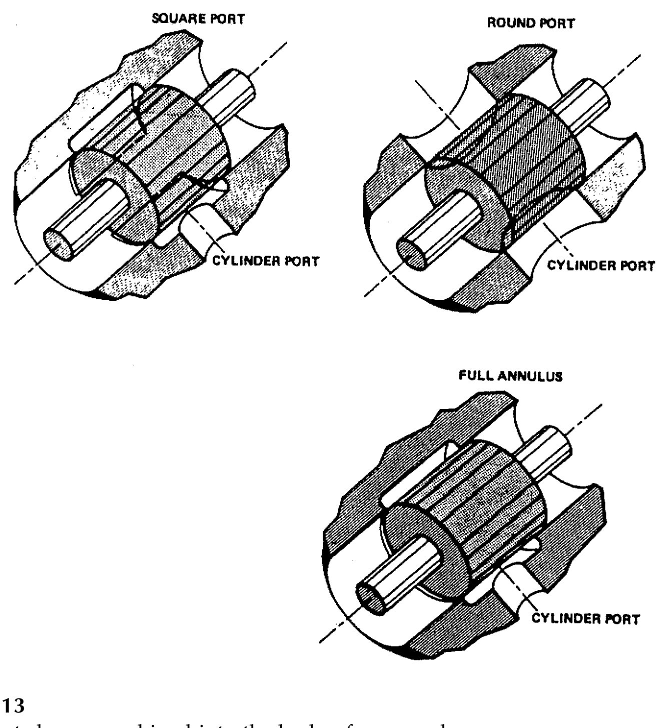


FIGURE 11.13
Common port shapes machined into the body of servo valves.

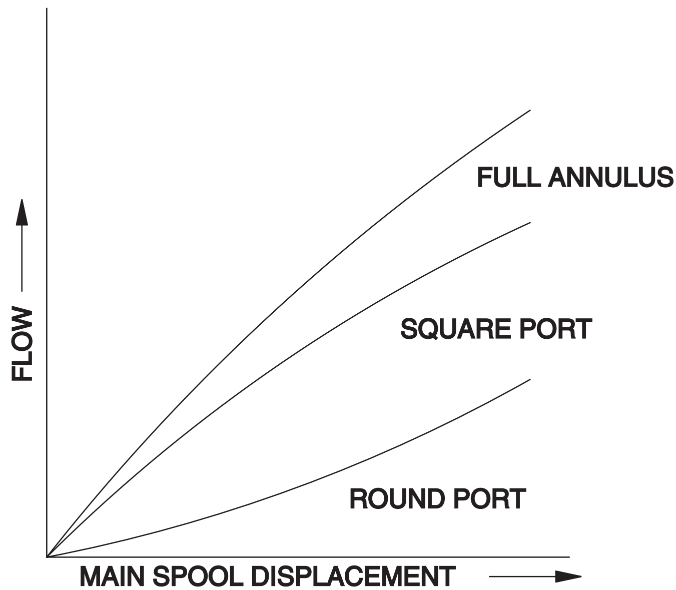


FIGURE 11.14
Flow as a function of spool displacement for common port shapes.

11.3.3.2 Lap Condition

Lap condition refers to position of the spool lands relative to the openings in the valve body. The most common condition is the line-to-line (Fig. 11.15) position. Here, the spool is machined very precisely to obtain a line-to-line fit with the opening in the valve body. The underlapped valve is sometimes referred to as an *open-center* valve, and the overlapped valve as a *closed center* valve. As shown in Fig. 11.16, the overlapped valve has a deadband. The spool must move a certain distance before any flow is delivered to the actuator ports.

11.3.3.3 Mount

It is typical for a servo valve to be mounted on a manifold block as shown in Fig. 11.17. This block has ports machined for the connection of hoses or hydraulic tubing to the valve. Internal passages are drilled to connect to four openings (pressure, return, Port *A*, and Port *B*) on the top of the manifold. The valve mounts in position over these four holes. When the mounting bolts are tightened, the matching holes in the valve and manifold are sealed with o-rings.

Typically, the manifold block will provide a passage between Ports *A* and *B*. When this passage is opened (by opening a manual valve), Ports *A* and *B* are connected, and the actuator is bypassed.

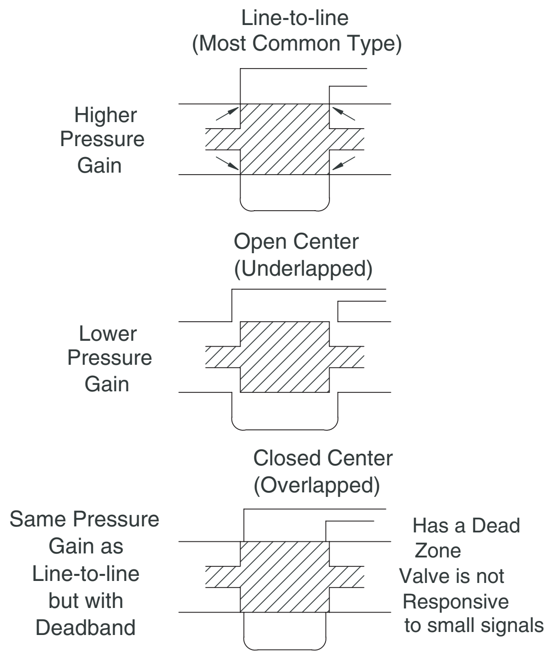


FIGURE 11.15
Three lap conditions for servo valves.

11.4 Valve Performance

Certain terms are used by the manufacturers to report the performance of their products. With an understanding of these terms, an engineer can select a valve with the needed characteristics.

11.4.1 Rated Flow

Rated flow is normally specified as the *no-load* flow. This is the flow measured with no actuator connected between Ports *A* and *B*. A flow meter is placed between these ports and the flow measured. Obviously, flow will increase as the pressure drop across the valve ($P_s - P_R$) increases. Typical flow vs. pressure drop curves for a "family" of different size valves is given in Fig. 11.18. Remember that the valves are in the full open position for the collection of this data.

A manufacturer usually labels their valves based on a 1000-psi pressure drop. You will note in Fig. 11.18 that, if you read up the 1000-psi vertical line to the intersection with the sloped line and read the ordinate, you will read the valve size designated by the manufacturer.

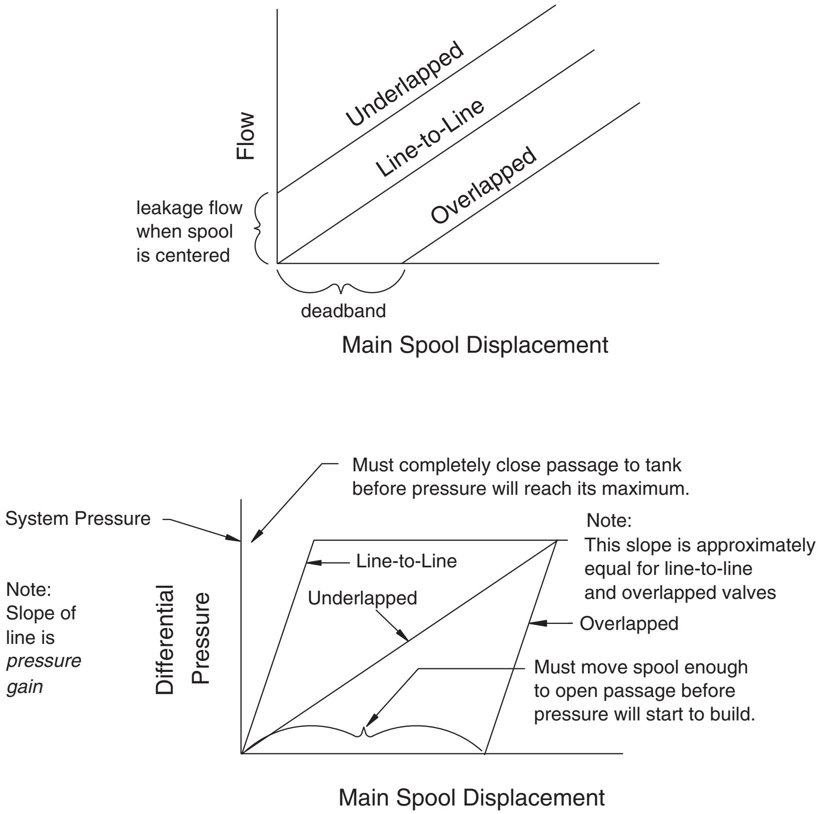


FIGURE 11.16
Flow and pressure as a function of lap condition for the three lap conditions.

11.4.2 Pressure Drop

Load pressure drop is the differential pressure between Ports *A* and *B*. For example,

$$P_L = P_A - P_B \tag{11.17}$$

Valve pressure drop is the sum of the differential pressures across the control orifices of the valve.

$$P_v = P_S - P_L - P_R \tag{11.18}$$

The reader is referred back to Sec.11.2.2 to understand the background for this definition.

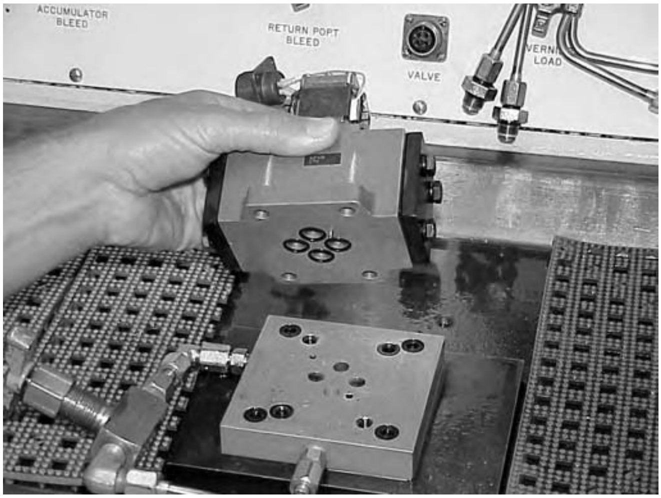


FIGURE 11.17
Servo valve being mounted on a manifold block. Courtesy of Moog, Inc.

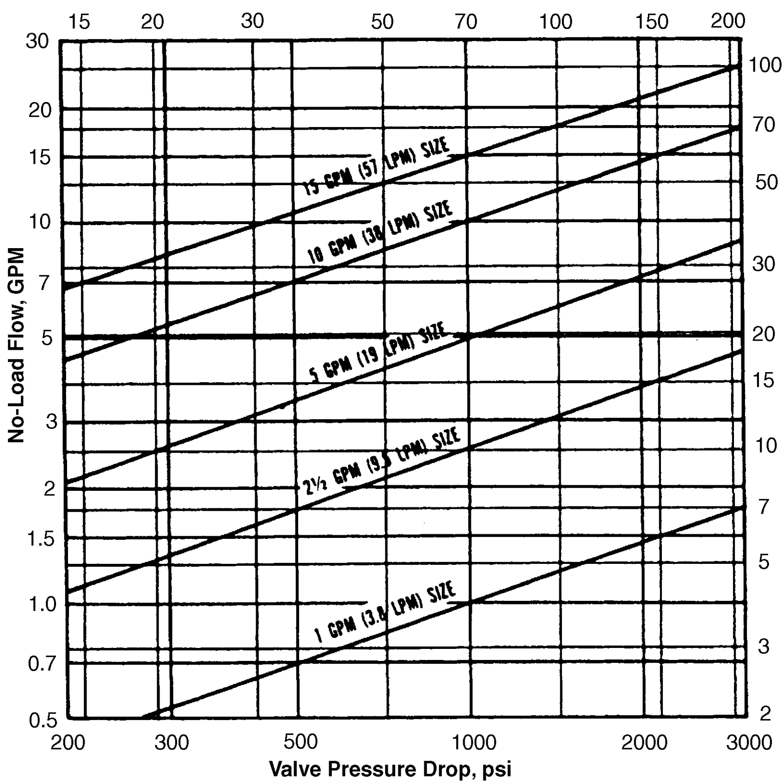


FIGURE 11.18
Change in rated flow with pressure drop across the valve for a family of different size valves.

11.4.3 Internal Leakage

The total flow from the pressure port to the return port with Ports *A* and *B* blocked is the internal leakage. This leakage is usually greatest with the valve in the null position. Referencing Fig. 11.5a, it is easy to see that a valve with line-to-line lap condition will leak more fluid between the pressure and return ports when it is in the null position.

11.4.4 Hysteresis

In an ideal world, a unit of current input would produce a unit of valve output. This result is closely approximated but never actually achieved. The curve shown in Fig. 11.19 was obtained by increasing current to the valve from 0 to + rated current and then returning through 0 to – rated current and back to 0. *Hysteresis* is defined as shown in the figure. The plot is not to scale; typically, hysteresis will be less than 3% of rated current.

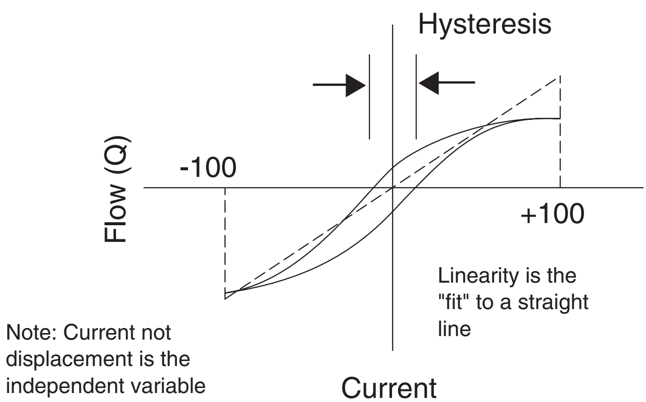


FIGURE 11.19
Flow vs. current input achieved with a constant supply pressure.

11.4.5 Threshold

Threshold is the current that must be applied before a response is detected. Good-quality two-stage valves have a threshold less than 0.5% of rated current.

11.4.6 Gain

Gain is defined as

$$G = \frac{\text{Output}}{\text{Input}}$$

Two gains are defined for servo valves: flow gain and pressure gain. Flow gain is always required to solve servo valve problems, thus it will be discussed first.

Flow gain is defined

$$G_{svf} = \frac{\text{Flow}}{\text{Input current}}$$

Typical units are in^3/s/mA. Flow gain is determined by measuring control flow vs. input current and plotting the curve shown in Fig. 11.20. Data is collected with no load connected between Ports *A* and *B*. The flow gain is the slope of the curve. Actual performance is expected to fall within ±10% of the linear curve shown in the figure. Most of the nonlinearity occurs in the null region, defined as ±5% of rated current. Flow gain in the null region may range from 50 to 200% of the expected gain based on the slope of the linear line. Special spool cuts can be ordered if performance in the null region is critical.

Pressure gain is defined as

$$G_{svp} = \frac{\text{Pressure}}{\text{Input current}}$$

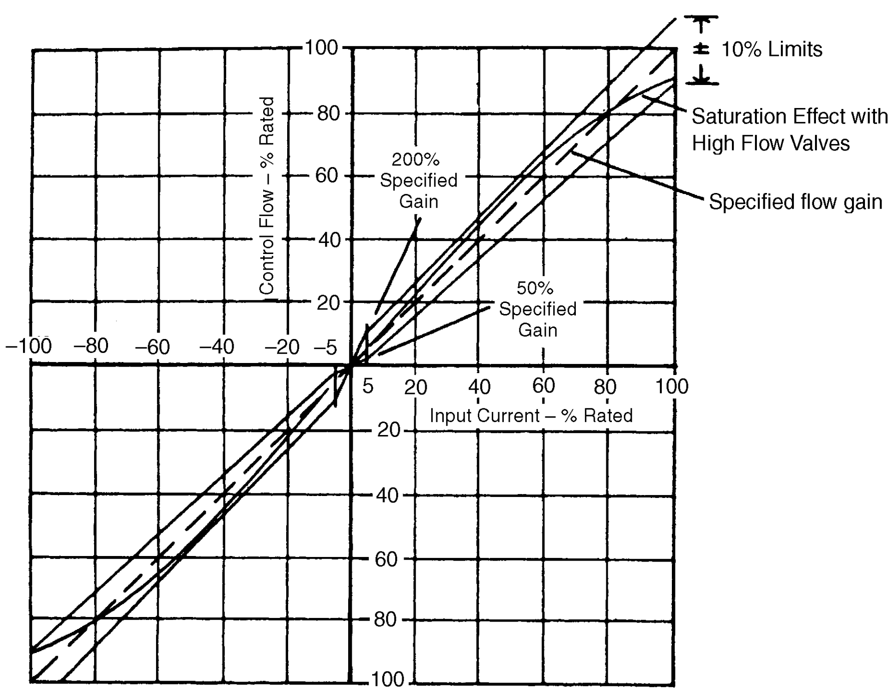


FIGURE 11.20
Data collected to define no-load flow gain.

with typical units being psi/mA. To measure pressure gain, Ports A and B are blocked. Pressure transducers are mounted in Ports *A* and *B*, and the pressure difference is measured as a function of input current. The range of input current to produce a load pressure change from –40% to +40% of supply pressure (P_s) is determined. Pressure gain is then

$$G_{svp} = \frac{0.8P_8}{\Delta I} \tag{11.19}$$

where ΔI = measured change in input current (mA)

Pressure gain is a difficult measurement to make. Once the valve opens enough to establish a continuous film of fluid between Ports *A* and *B*, the pressure difference equals P_s. A typical curve is shown in Fig. 11.21. A normal range of pressure gain measurements will be 30 to 100% of P_s for an input current equal to 1% of rated current.

11.4.7 Frequency Response

Frequency response relates to the ability of the servo system to respond to an input voltage. Before beginning this discussion, it is good to review our intuitive understanding of vibration.

A force *f(t)* is acting on a body that has mass and elasticity (Fig. 11.22). The model, known as a *lumped parameter* model, represents mass with *m* and elasticity with *k*. This model does not mean that a mass is held in position by a spring. It simply says that the system has two properties, mass and elasticity.

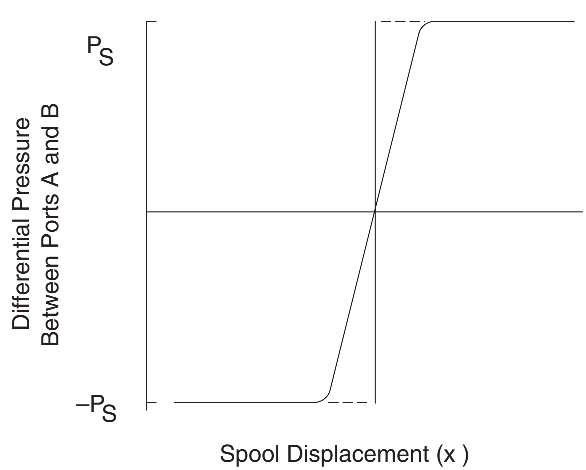


FIGURE 11.21
Typical pressure gain for servo valve.

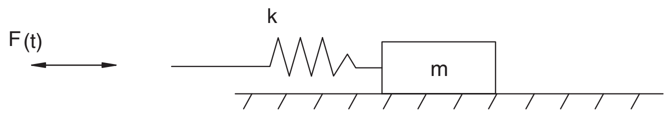


FIGURE 11.22
Time-varying force acting on a body that has elasticity and mass.

It may be helpful for the reader to literally think about a mass held in position with a spring as represented in Fig. 11.22. Suppose the force *f(t)* is sinusoidal. At a certain frequency, the spring deflects very little as the force is applied. The mass moves in phase with the applied force. As the frequency is increased, the movement of the mass is no longer in phase with the applied force. Eventually, a frequency is reached where the mass moves very little; all of the *f(t)* is absorbed by the spring deflection.

The spool of a servo valve has mass, and it is held in position by a spring. It also has oil in a cavity at both ends. Our intuitive understanding of the model in Fig. 11.22 can now be extended to the servo valve.

Suppose a step input current (abrupt application of rated current) is applied to the servo valve, and the spool position is plotted versus time (Fig. 11.23a). It takes 15 ms for the spool to move the completely open position. When rated current is abruptly removed, it takes 10 ms for the spool to return to the center position (Fig. 11.23b). Total time for one complete cycle is

$$15 \text{ ms} + 10 \text{ ms} = 25 \text{ ms}$$

The frequency is

$$\begin{aligned} \frac{1 \text{ cycle}}{25 \text{ ms}} \times \frac{1000 \text{ ms}}{s} &= 40 \text{ cycle/s} \\ &= 40 \text{ Hz} \end{aligned}$$

If the current is cycled faster than 40 Hz, the valve will not open fully before it begins to close.

The ratio of output to input is typically reported in decibels defined as follows:

$$\text{dB} = 20\log_{10}\left(\frac{\text{output}}{\text{input}}\right) \tag{11.20}$$

Suppose the output is 50% of the input.

$$\begin{aligned} \text{dB} &= 20\log_{10}\left(\frac{50}{100}\right) = 20(-0.3) \\ &= -6 \end{aligned}$$

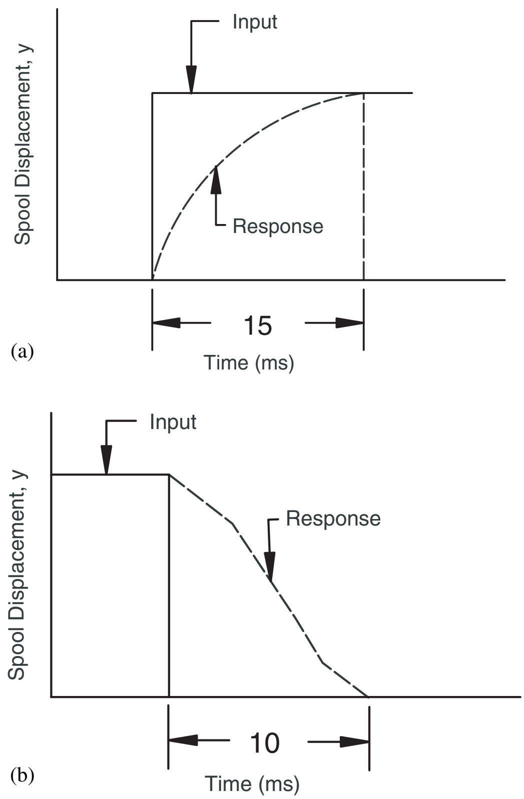


FIGURE 11.23
Time required for the spool of a servo valve to (a) fully open and (b) fully close when a step input current is applied.

There are several excitation signals used to determine the frequency response reported by a manufacturer. Several of these are shown in Fig. 11.24.

Measured frequency response for three different size valves excited with ±40% rated current is shown in Fig. 11.25. Results for ±100% rated current are shown on the same plot. Looking first at the response for the 5 GPM valve, output is less than input for frequencies above 40 Hz when the excitation is ±40%. With ±100% excitation, a resonant phenomenon develops, and the output is *greater than* input for frequencies between 40 and 70 Hz. Both excitations shown an attenuation of output at higher frequencies. The valve simply cannot move fast enough to track the input signal. It does not open completely before it is commanded to move in the opposite direction.

As might be expected, the larger valves (having greater mass) have a lower frequency response. The 10 GPM valve shows a resonant phenomenon for both the ±40% and ±100% excitation. Also, there is more difference in the frequency range with zero attenuation for the ±40% and ±100% excitations. The 15 GPM valve has the lowest frequency response and demon-

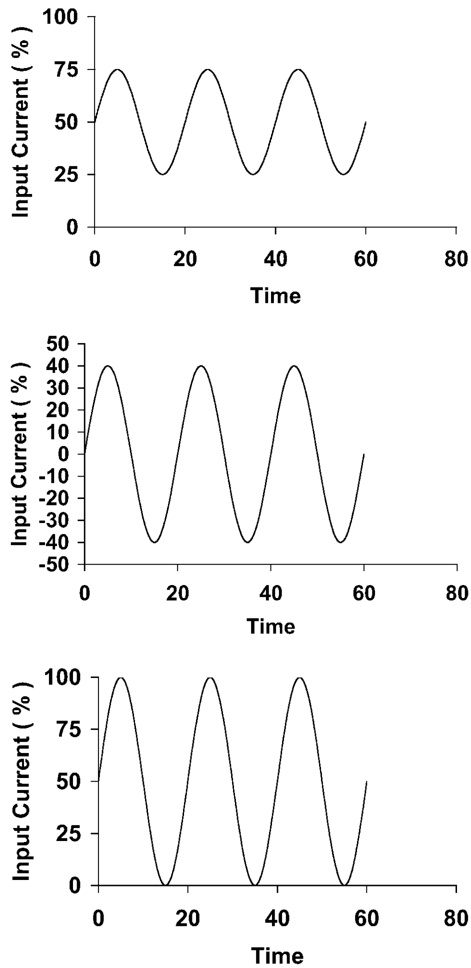


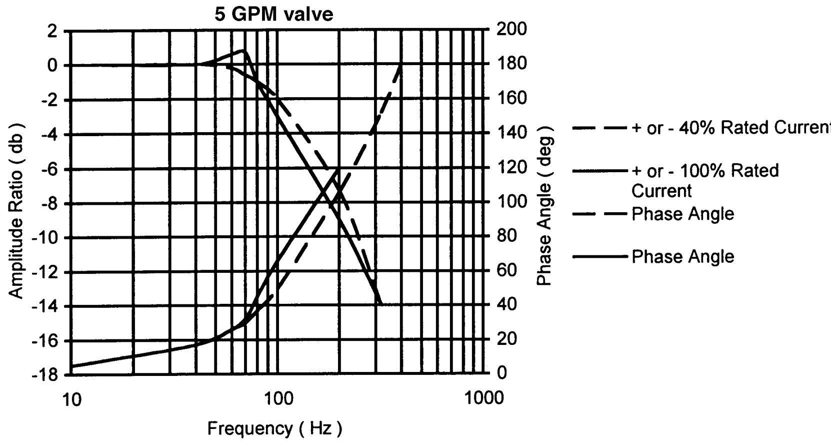
FIGURE 11.24
Excitation signals used to measure frequency response of servo valves.

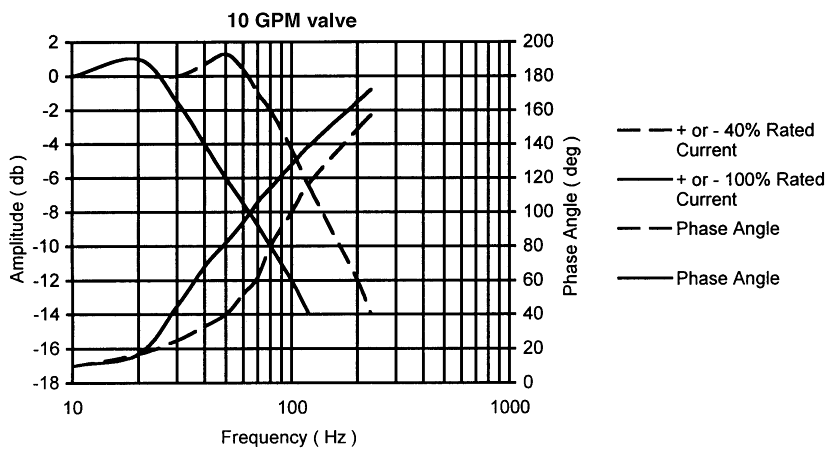
strates no resonant phenomena. Output is less than input for all frequencies above 10 Hz.

11.4.8 Phase Lag

Phase lag is the number of degrees that output lags behind input. Its meaning is best understood by comparing the two sinewaves shown in Fig. 11.26. The output is 30° behind the input, so the phase lag is defined to be 30°.

Phase lag is plotted on the frequency response curves in Fig. 11.25. It is interesting to note the frequency at which the phase lag is 90°, meaning that the valve is just beginning to open when the input signal is just beginning to cycle in the opposite direction.





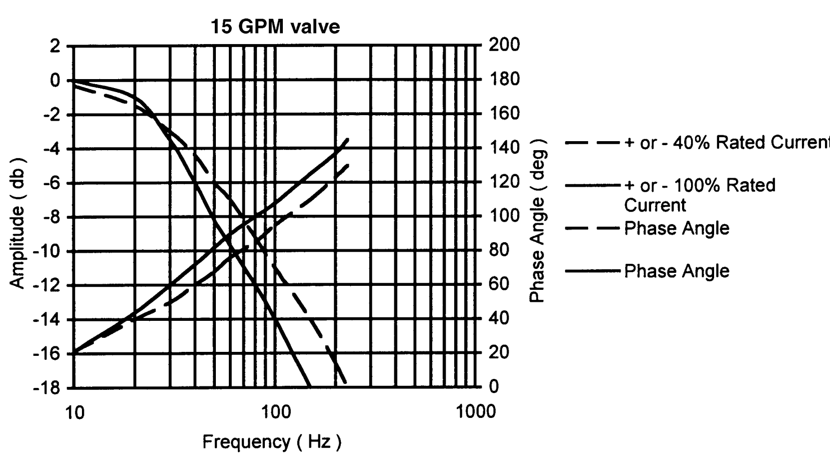


FIGURE 11.25
Frequency response for a "family" of servo valves. Reprinted by permission from Moog, Inc.

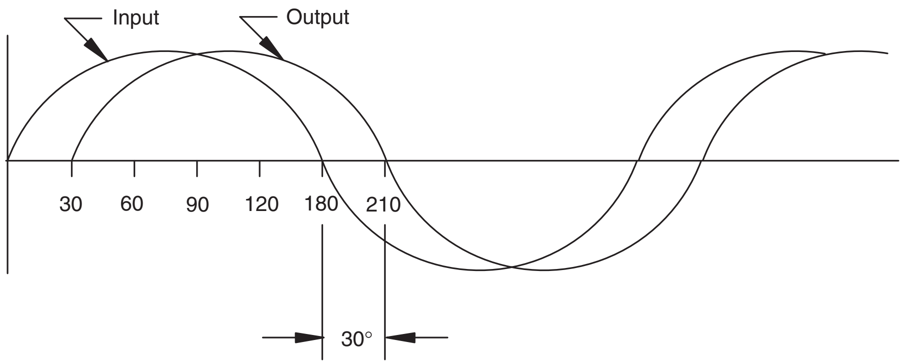


FIGURE 11.26
Output sinewave lags input sinewave by 30°.

11.4.9 Summary

The following terms were defined in this section:

Rated flow	Load pressure drop
Valve pressure drop	Internal leakage
Hysteresis	Threshold
Flow gain	Pressure gain
Frequency response	Phase lag

Other definitions and terminology used by manufacturers in their technical literature will be explained in Sec. 11.7.

11.5 Types of Servo Systems

Servo systems can be used to control position (linear or angular) and velocity (linear or angular). Positional systems will be discussed first.

11.5.1 Hydromechanical Servo System

On large machinery, the force required to shift transmissions and perform other control functions can be substantial. Ease of operation can give an increase in operator productivity.

The system shown in Fig. 11.27 is known as a *hydromechanical servo.* A mechanical input moves the servo cylinder to a new position. The input shifts

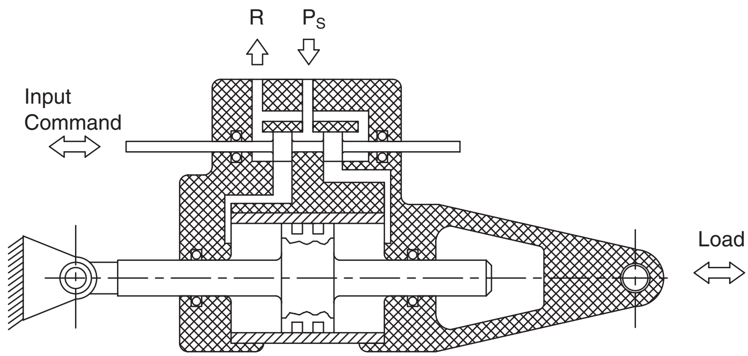


FIGURE 11.27
Hydromechanical servo.

the spool, and the cylinder moves until the valve body aligns with the new spool position.

The key to understanding the system in Fig. 11.27 is to realize that the valve body and cylinder body are one piece. Suppose the input command is to the right (spool is shifted to the right). Pressure is ported to the right end of the cylinder piston, and the left end is ported to the return. Since the piston is fixed in position, the cylinder housing moves to the right until the valve body is aligned with the new spool position.

A different type of hydromechanical servo is shown in Fig. 11.28. Here, a template moves the valve spool. Flow is directed to a cylinder, which moves the work table. (In this case, the valve and cylinder are mounted at separate locations.) The work table is mechanically connected to the valve body; thus, when the table moves, it moves the valve body. This arrangement produces a table movement equal to the template profile.

11.5.2 Electrohydraulic Servo Systems

Additional flexibility and versatility can be obtained by using an electrical input rather than a mechanical input. An example of an electrohydraulic servo is shown in Fig. 11.29. This servo is commonly referred to as a *pot-pot servo.*

11.5.2.1 Pot-Pot Servo

The pot-pot servo gets its name from the fact that the command and feedback signals are obtained from potentiometers. The command is a voltage obtained by rotating the command potentiometer. This command voltage is compared to a voltage obtained from the potentiometer mounted adjacent to the work table, known as the *feedback potentiometer.* As the work table moves, it slides the wiper along this potentiometer to change the feedback voltage. The difference between the command and feedback voltages is the "error" voltage, and

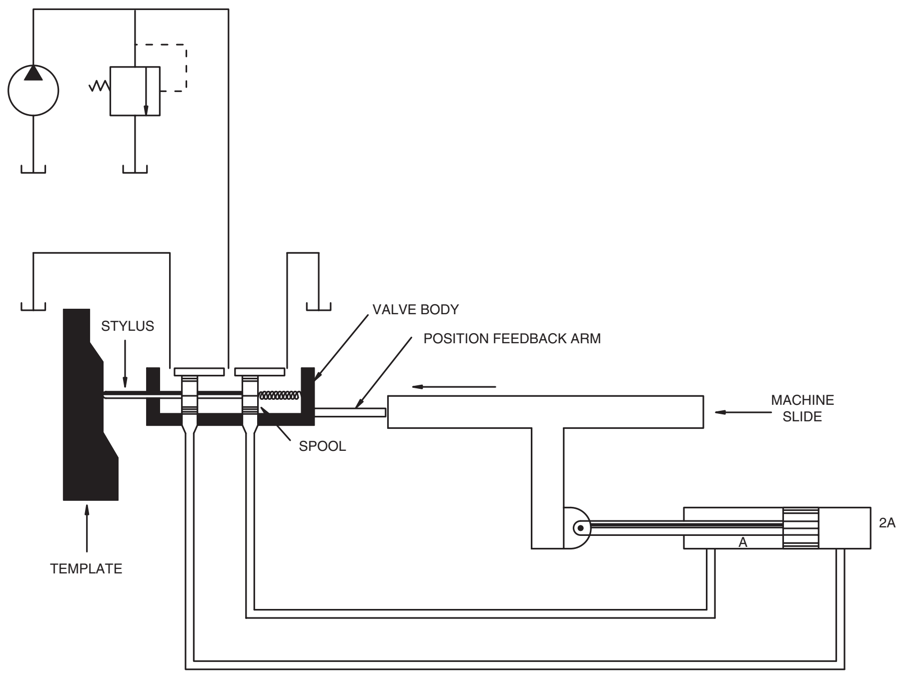


FIGURE 11.28
Electrohydraulic servo positioning system. Reprinted with permission from Electrohydraulic Servo Systems, James E. Johnson, Penton Media, Inc., Cleveland, Ohio.

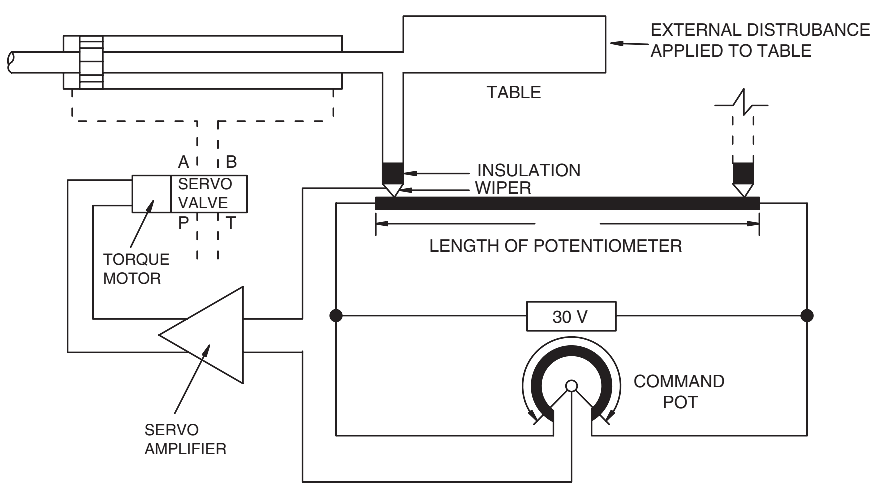


FIGURE 11.29
Electrohydraulic servo positioning system. Reprinted with permission from Electrohydraulic Servo Systems, James E. Johnson, Penton Media, Inc., Cleveland, Ohio.

this voltage is the input to an amplifier. The amplifier produces a milliamp current proportional to the error voltage, and this current is the input current to the servo valve torque motor. The servo valve opens to direct fluid to the cylinder, which moves the work table. The table moves, and thus moves the wiper on the feedback pot, until the feedback voltage equals the command voltage. Each table position corresponds to a unique command voltage.

The reader should be able to readily visualize how the system shown in Fig. 11.29 is used in modern manufacturing. The input voltage typically is a series of voltages produced by a digital-to-analog converter. This converter converts a series of computer instructions to the needed command voltages. A workpiece mounted on the table is positioned to be machined in accordance with the computer instructions. A series of operations, often with several actuators, are controlled in this manner.

Example Problem 11.1

The positioning system shown in Fig. 11.29 is subjected to an external disturbance. A 2000-lb_f force is applied to the table as shown. What displacement is produced by this force?

The pressure gain for the servo valve is quite high. An input current of only 5 mA produces a pressure differential between Ports *A* and *B* of 1000 psi.

The amplifier transfer function is 1000 mA/V. This means that a 1-V error signal will produce an output current of 1000 mA. The servo valve rated current is 30 mA. A higher current will burn out the coils of the torque motor. The amplifier is designed to saturate at 30 mA, and this protects the torque motor. Any input voltage equal to or greater than

$$\frac{30 \text{ ma}}{1000 \text{ ma/V}} = 0.03 \text{ V}$$

will produce an output current of 30 mA.

The cylinder stroke is 10 in., and the supply voltage is 30 V, thus the feedback gain is 30 V/10 in = 3 V/in. The cylinder bore is 3 in. and the rod diameter is 1.5 in., thus the annular area is 5.3 in^2.

If a 2000 lb_f external load is applied to the table, what position change will occur?

$$\frac{2000 \text{ lb}_f}{A_{cyl}} = \frac{2000}{5.3} = 377 \text{ psi}$$

The increase in pressure difference between Ports *A* and *B* due to the external load is 377 psi. How much does the valve have to open to create this pressure difference? The pressure gain for the servo valve is

$$\frac{1000 \text{ psi}}{5 \text{ ma}} = 200 \text{ psi/mA}$$

A pressure difference of 377 psi is created when the input current to the valve is

$$\frac{377 \text{ psi}}{200 \text{ psi/mA}} = 1.885 \text{ mA}$$
$$= 1.9 \text{ mA}$$

What error voltage is required at the amplifier to produce a current of 1.9 mA? Amplifier gain is 1000 mA/V.

$$\frac{1.9 \text{ mA}}{1000 \text{ mA/V}} = 0.0019\text{V}$$

What displacement of the work table will give an error voltage between the feedback and command voltages equal to 0.0019 V?

$$\frac{0.0019 \text{ V}}{3 \text{ V/in}} = 0.0006 \text{ in}$$

Table movement (and thus position error) due to a 2000 lb_f external disturbance is 6/10,000th in. Choosing system parameters correctly can give a very "stiff" position control system.

11.5.2.2 Force, Pressure, and Torque Servos

A force servo uses the signal from a force transducer as the feedback signal. In like manner, a pressure servo uses the signal from a pressure transducer and a torque servo the signal from a torque transducer. Only the force servo will be discussed here.

A force servo has a force transducer (strain gages mounted in a wheatstone bridge) mounted between the cylinder and the load (Fig. 11.30). In this case, the cylinder increases the force on the load until the transducer output voltage (feedback voltage) equals the command voltage. The system then holds this force until the command signal is changed. It is possible to cycle the force and program various force duration times by inputting the correct command voltage vs. time function.

11.5.2.3 Velocity Servos

Velocity servos are used to control both linear and rotational velocity. Only rotational velocity will be studied here. These servos are widely used in manufacturing to draw wire, buff steel sheets to a required finish, run printing presses, and for a variety of other applications where the rotational velocity of a drive is controlled to provide a certain linear velocity.

A typical velocity servo is shown in Fig. 11.31. There is a key difference between this servo and the positional servo; the amplifier output is not zero

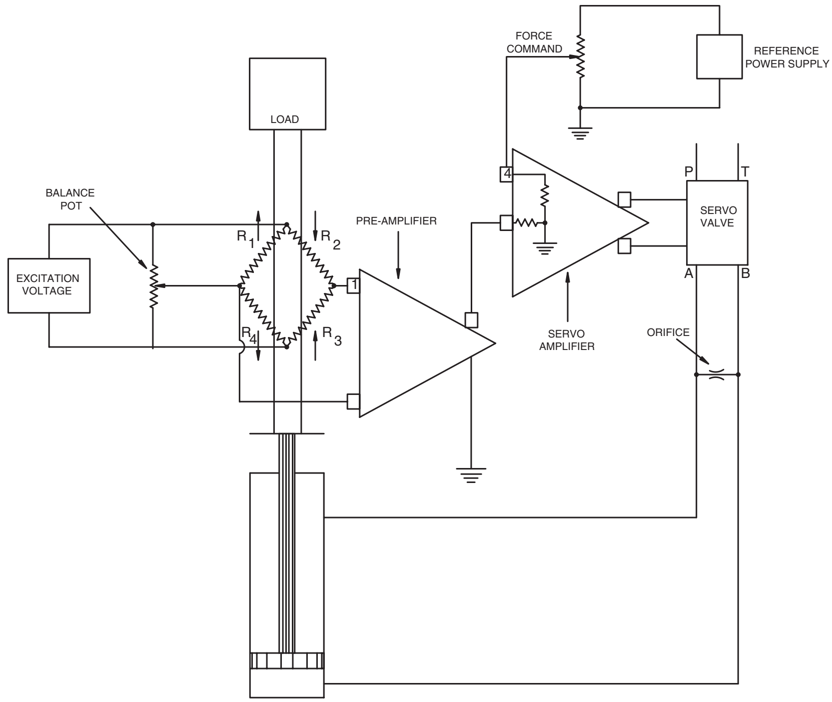


FIGURE 11.30
Force servo. Reprinted with permission from Electrohydraulic Servo Systems, James E. Johnson, Penton Media, Inc., Cleveland, Ohio.

for zero error voltage. If it were zero, the servo valve would be closed, and no fluid would flow to the motor to produce rotational velocity.

The servo amplifier for a velocity servo is an *integrating amplifier.* The output of an integrating amplifier ramps up when a positive voltage is applied to the input and ramps down when a negative voltage is applied. Its *non-zero output* changes based on the error voltage applied.

If the amplifier converts the error signal to produce an output current proportional to the first derivative of the error signal, then the system is said to be a *type one* servo. In the case of a velocity control, the first derivative of the error (velocity) is acceleration. As error goes to zero, acceleration goes to zero, and the system operates at the new velocity.

A dc tachometer is widely used as the feedback transducer for a rotary velocity servo, because its high-level dc output can be directly connected to the servo amplifier without any need for signal conditioning. This transducer can be expected to have a long service life if the tachometer shaft is connected to the load using a flexible coupling to avoid bearing wear due to misalignment. Because the output signal is dc, the output cable must be shielded to avoid signal errors from ambient electromagnetic noise. A shielded cable has a metal foil inside the insulation that encloses the signal wires. This metal foil

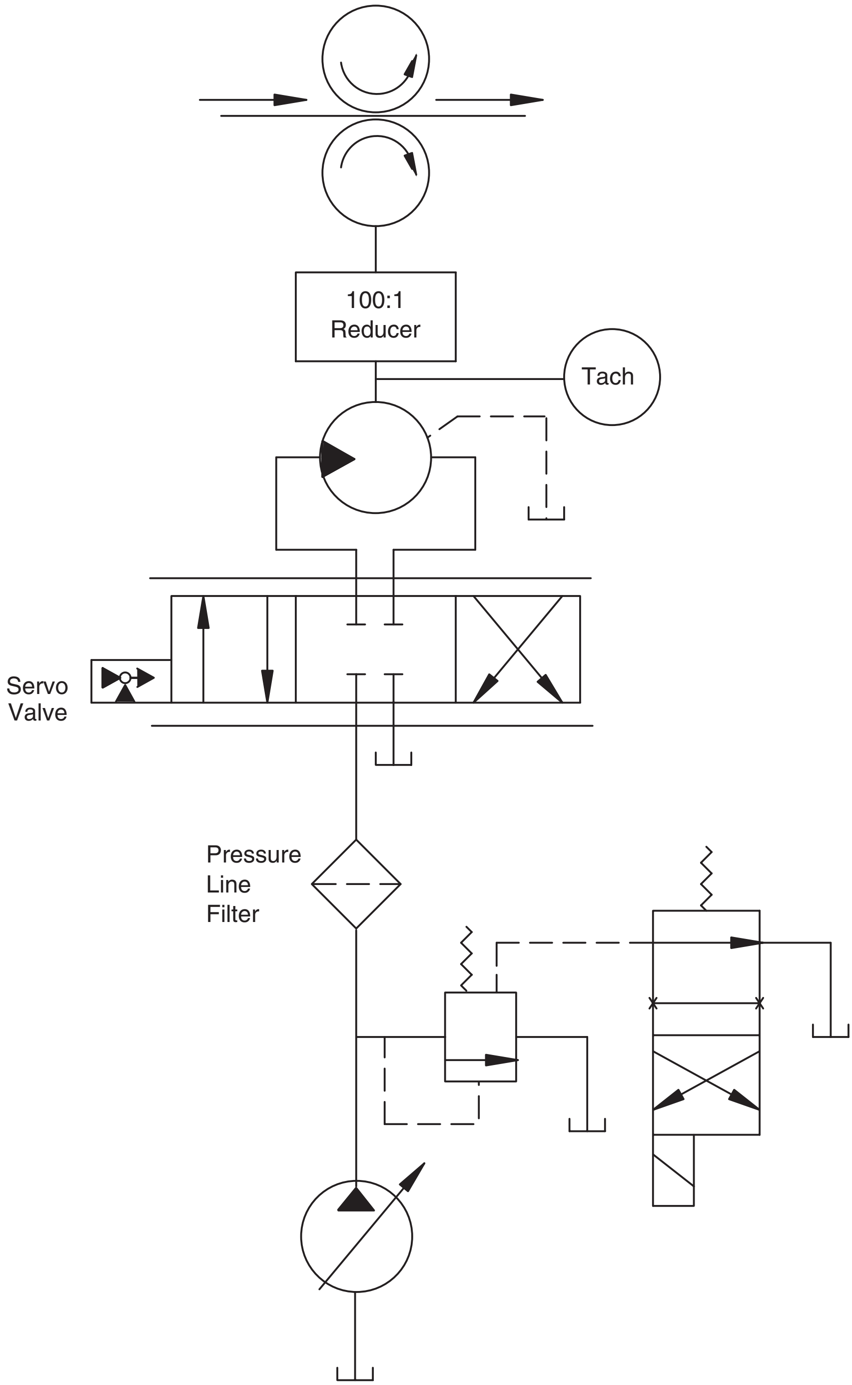


FIGURE 11.31
Servo motor used to control the speed of sheet steel for buffing to a required finish.

is grounded, and it tends to absorb interference that would otherwise be superimposed on the signal.

A typical example of a velocity servo is shown in Fig. 11.31. The entire hydraulic circuit is shown and will be discussed. The electrical circuit is not shown.

Much of this circuit is familiar from our study of circuits in previous chapters. The spring cavity of the pilot-operated relief valve is vented through a two-position, two-way DCV. When this line is blocked, the relief valve is set at the pilot setting. When it is open, the pump unloads at low pressure (the main spring setting). The pressure line filter protects the servo valve. The valve is mounted directly on the motor to create the valve-motor configuration shown in Fig. 11.7. Motor speed is controlled by opening the servo valve to provide a given pressure drop. Total pressure at the relief valve is

$$\Delta P_{tot} = \Delta P_f + \Delta P_{sv} + \Delta P_m \tag{11.21}$$

where ΔP_f = pressure drop across filter
ΔP_{sv} = pressure drop across servo valve
ΔP_m = pressure drop across the motor

The relief valve cracks open to divert part of flow back to the reservoir, and the remaining flow (less a small leakage in the servo valve) goes to the motor.

Pressure drop across the servo valve represents an energy loss. A more efficient design (lower energy loss) is the servo pump-motor (split). Here, the servo valve positions the swashplate of a variable displacement pump (Fig. 6.27) to deliver the correct flow to the motor. Unlike the velocity servo in Fig. 11.31, this servo system is not a *type one*; it is a positional servo. The amplifier output is zero when the error voltage (difference between command voltage and feedback voltage from dc tachometer) is zero. The servo valve is closed, and it holds the swashplate in the set position. A proportional amplifier rather than an integrating amplifier is used.

11.6 Servo Amplifiers

A servo amplifier has two main functions.

1. It provides a mA current proportional to an input voltage. Typical designs fall in the range 5 mA/V to 100 mA/V.
2. It saturates at the rated current of the servo valve. The amplifier is designed so that it cannot deliver enough current to the torque motor to burn out the coils.

The question can be asked, why is an amplifier needed for a pot-pot servo system? Why not connect the torque motor between the two potentiometers as shown in Fig. 11.32a? Suppose the command and feedback potentiometers both have a resistance of 40 Ω. Analysis of this circuit shows that the current through the coil would be 30 mA per inch of feedback pot movement. If the servo valve has a deadband of 5%, and the rated current is 30 mA, the deadband is $0.05 \times 30 = 1.5$ mA. This deadband corresponds to the following position error.

$$\text{Error} = \frac{1.5 \text{ ma}}{30 \text{ ma/in}} = 0.05 \text{ in}$$

A servo amplifier has very high input impedance, meaning that almost no current flows into the amplifier at the input. This feature allows the use of

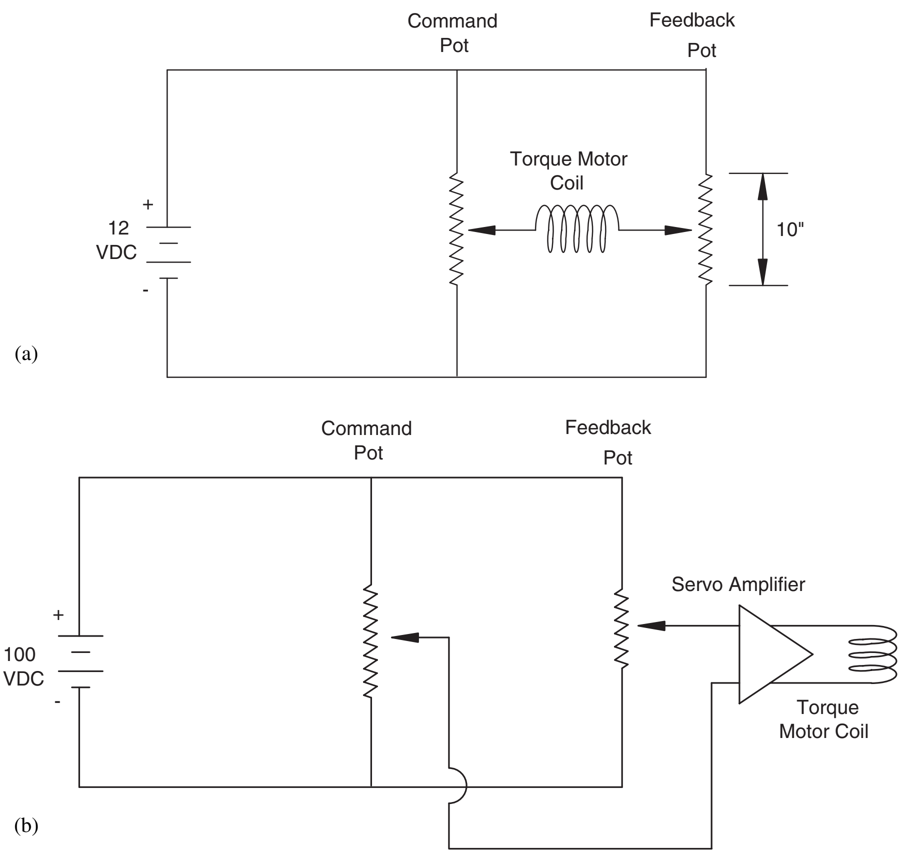


FIGURE 11.32
(a) Potential electrical circuit for a pot-pot servo and (b) electrical circuit for a pot-pot servo that includes a servo amplifier.

high-resistance pots and a high supply voltage as shown in Fig. 11.32b. If the feedback pot travel is 10 in., the feedback transfer function *(TF)* is 100 V/10 in. = 10 V/in.

Suppose the amplifier gain is the same as in Example Problem 11.1—1000 mA/V. Total current gain is defined by

$$G_I = G_a \times H \tag{11.22}$$

where G_I = total current gain (mA/in)
G_a = amplifier gain (mA/V)
H = feedback gain (V/in)

In this case,

$$\begin{aligned} G_I &= 1000 \times 10 \\ &= 10{,}000 \ \text{mA/in} \end{aligned}$$

Assuming the same servo valve deadband, the position error is

$$\text{Error} = \frac{1.5 \text{ ma}}{(10,000) \text{ mA/in}} = 0.00015 \text{ in.}$$

as compared to 0.05 in. without the servo amplifier. It is clear that a servo amplifier provides opportunity to significantly reduce position error.

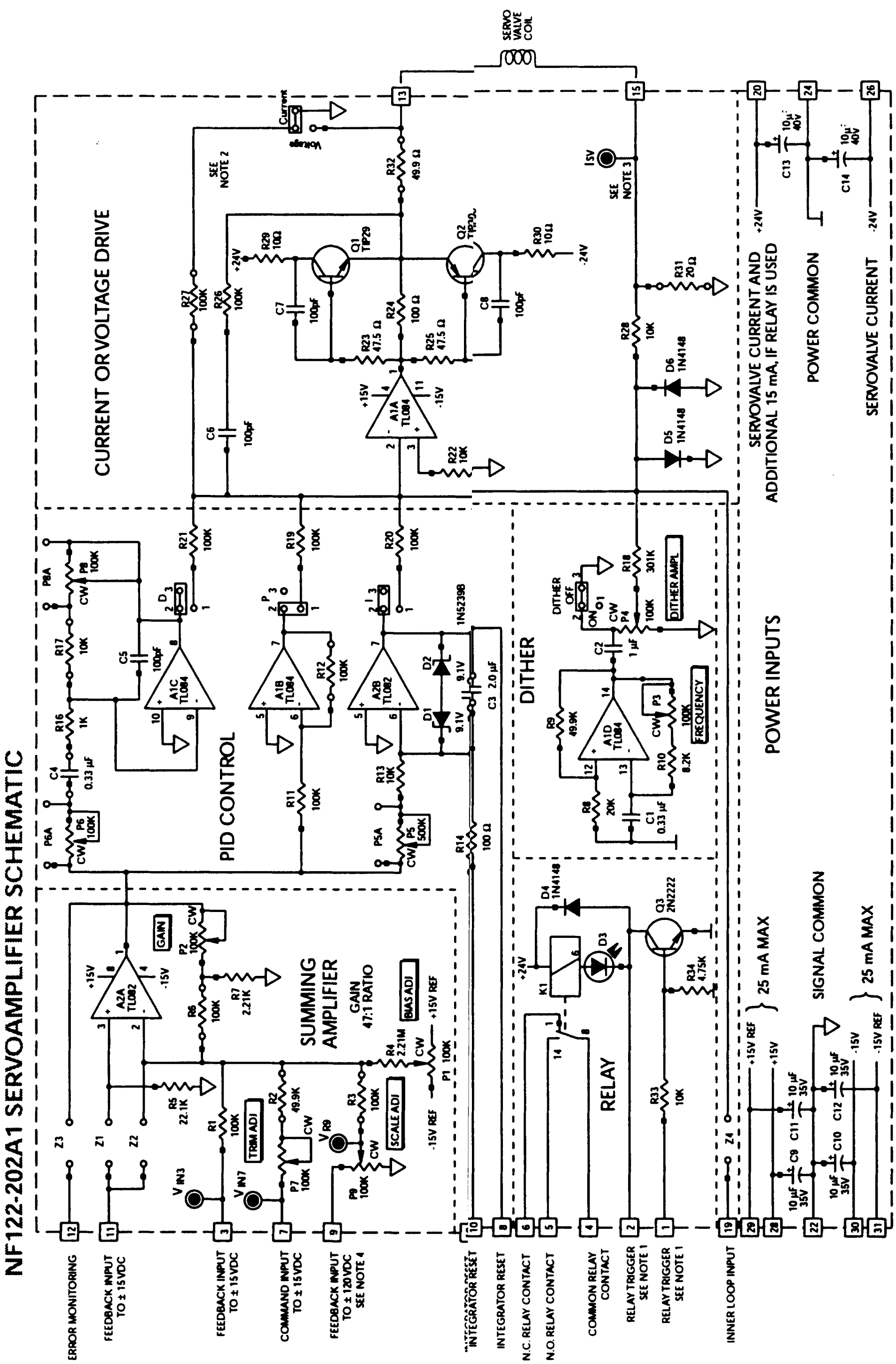
Detailed servo amplifier design is beyond the scope of this text. A circuit diagram for a servo amplifier is given in Fig. 11.33. The manufacturer provices the capability to switch between a proportional and an integrating amplifier. In some cases, a new circuit board is inserted to obtain the type of amplifier needed.

11.7 Servo Analysis

The concept of gain has previously been defined

$$G = \frac{\text{Output}}{\text{Input}}$$

Feedback is illustrated by the block diagram in Fig. 11.34. The principle is to sense the output and adjust the input to correct for changes in the output, thus feedback adds stability to any system. Each component changes as a function of the environment and as a function of normal wear over its design


NF122-202A1 SERVOAMPLIFIER SCHEMATIC
CURRENT OR VOLTAGE DRIVE
PID CONTROL
SUMMING AMPLIFIER
GAIN 47:1 RATIO
DITHER
RELAY
POWER INPUTS
SERVO VALVE COIL
ERROR MONITORING
FEEDBACK INPUT TO ± 15VDC
FEEDBACK INPUT TO ± 15VDC
COMMAND INPUT TO ± 15VDC
FEEDBACK INPUT TO ± 120VDC SEE NOTE 4
INTEGRATOR RESET
INTEGRATOR RESET
N.C. RELAY CONTACT
N.O. RELAY CONTACT
COMMON RELAY CONTACT
RELAY TRIGGER SEE NOTE 1
RELAY TRIGGER SEE NOTE 1
INNER LOOP INPUT
SERVOVALVE CURRENT AND ADDITIONAL 15 mA, IF RELAY IS USED
POWER COMMON
SERVOVALVE CURRENT
SIGNAL COMMON
25 mA MAX
25 mA MAX

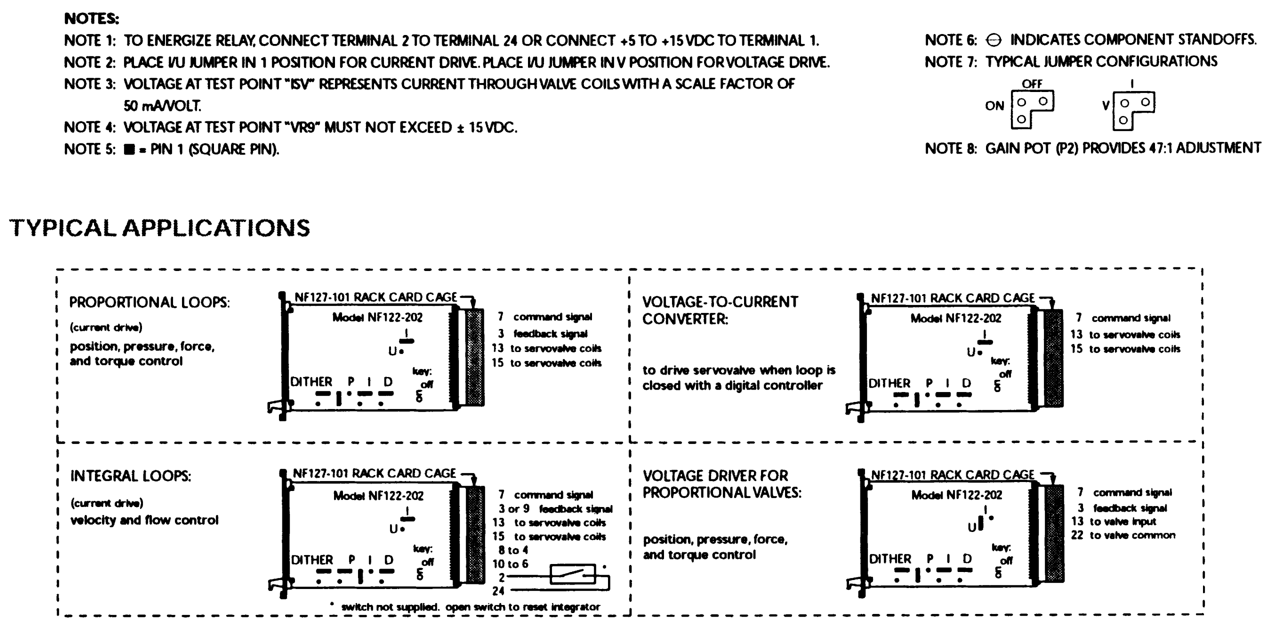


FIGURE 11.33
Typical proportional amplifier circuit. Reprinted by permission, Moog Inc.

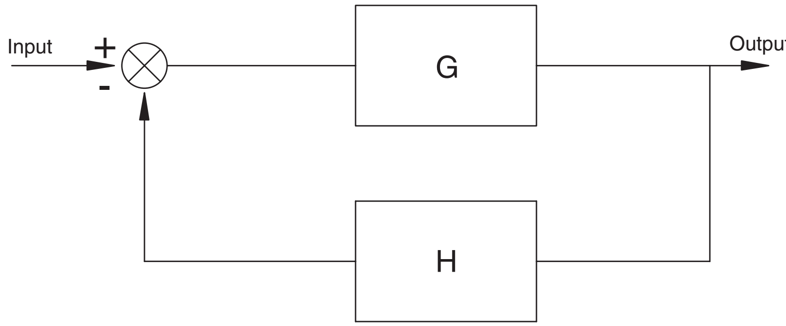


FIGURE 11.34
Block diagram of a system with feedback.

life. In addition, external disturbances always occur. These effects are overcome with feedback.

The feedback signal must be opposite in sign to the input (command) sign. We are correcting for a drift of the output, so we must move it in the direction opposite from the drift to get back to the set point. This correction is called *negative* feedback.

On many machines, the operator provides the feedback. When observing a drift in output, the operator changes a control to bring the output back to the set point; the operator "closes the loop." All of us have done this when driving our cars. We do it intuitively. It is important to realize that automatic control is just the replacement of the operator with hardware. A servo valve is the key piece of hardware, therefore, closed-loop systems are often called *servo* systems.

The feedback signal is most often a dc voltage. When it is a sinewave, the feedback signal must be shifted in phase 180° from the input sinewave. Then, when the amplitudes are equal, the two signals cancel each other, and the error is zero.

In previous sections, we discussed the individual components. It is now time to put these together in a closed-loop system.

A block diagram of a closed-loop system for a servo cylinder (servo valve mounted directly on a cylinder) is shown in Fig. 11.35. Note that the servo valve transfer function is the *flow* transfer function, not the *pressure* transfer function. Input to the cylinder is a flow, in^3/s, and output is a linear velocity, in/s. The transfer function is then

$$G_{cyl} = \frac{\text{output}}{\text{input}} = \frac{\text{in/s}}{\text{in}^3/\text{s}} = \frac{\text{in}}{\text{in}^3} = \frac{1}{\text{in}^2} = \frac{1}{A}$$

where A = cylinder area (in^2)

The feedback transducer is a potentiometer; thus, its transfer function is V/in. A linear velocity (in/s) is fed to the potentiometer to produce the feedback signal (V), not V/s.

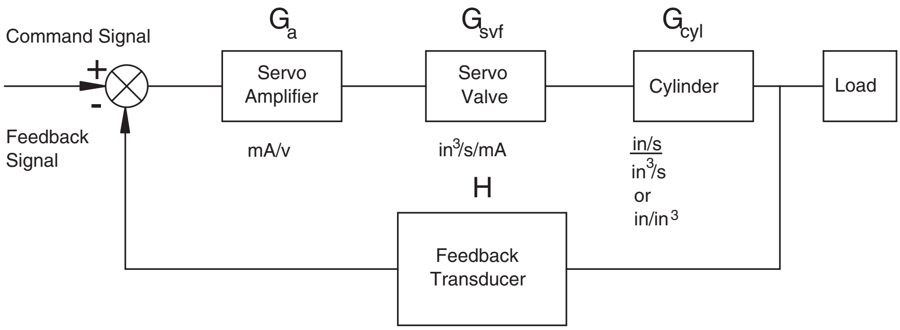


FIGURE 11.35
Block diagram of a servo cylinder closed-loop system.

11.7.1 Open-Loop Gain

By convention, open-loop gain is defined by

$$\begin{aligned} GH &= G_a \times G_{svf} \times G_{cyl} \times H \\ &= \frac{\text{mA}}{\text{V}} \times \frac{\text{in}^3/\text{s}}{\text{mA}} \times \frac{\text{in}}{\text{in}^3} \times \frac{V}{\text{in}} \\ &= \frac{1}{s} \end{aligned} \tag{11.23}$$

Correct units for open-loop gain are time^{-1}. By convention, the cylinder transfer function is defined as in^{-2}, or in/in^3. Another name for open-loop gain is the *velocity constant*. In this text, the velocity constant is denoted.

$$k_v = GH = G_a \times G_{svf} \times G_{cyl} \times H \tag{11.24}$$

11.7.2 Natural Frequency

The natural frequency of a servo cylinder is an important parameter in the design of the system. An analysis will now be done to obtain the natural frequency of a servo cylinder connected to a load. The model for this system is shown in Fig. 11.36. Note that the mass is held in position with a spring. As before, this representation denotes that the load has mass and elasticity.

Stiffness is a parameter that relates the force required to produce a unit deflection. Compliance is the reciprocal of stiffness, thus the units are deflection per unit force, or in/lb$_f$.

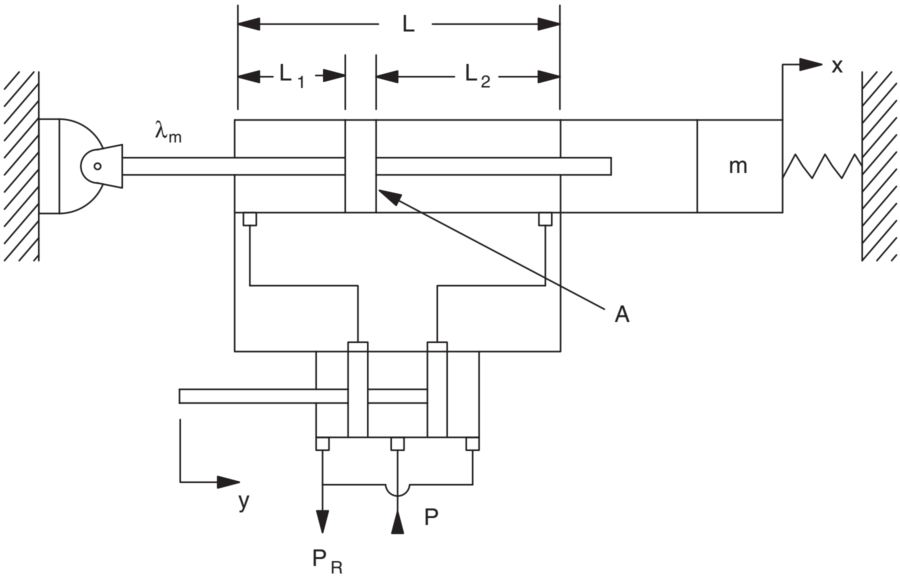


FIGURE 11.36
Diagram of servo cylinder moving a load [model of hydromechanical servo (Fig. 11.27)].

The system shown in Fig. 11.36 can be visualized as a mass held in position by a column of fluid. If compliance of this fluid is λ_o, and compliance of the rod is λ_m, then total cylinder compliance is

$$\lambda = \lambda_o + \lambda_m \tag{11.25}$$

Generally, $\lambda_m << \lambda_o$, so the compliance of the oil is used as the cylinder compliance with negligible error.

The concept of bulk modulus of a fluid was introduced in Chapter 2. Hydraulic oil does compress when pressure is applied.

The relationship is defined by

$$\beta = V\frac{\Delta P}{\Delta V} \tag{11.26}$$

where β = bulk modulus (psi)
V = original volume before pressure is applied (in^3)
ΔP = applied pressure (psi)
ΔV = change in volume (in^3)

Compliance is displacement divided by force. Suppose a force F is applied to the cylinder in Fig. 11.36. The effective piston area is A, and the column of fluid has a length L_1. Pressure resulting from application of the force is

$$\Delta P = \frac{F}{A} \tag{11.27}$$

or

$$F = \Delta P A$$

Displacement is ΔL. Using the definition of compliance,

$$\begin{aligned}\lambda &= \frac{\Delta L}{F} \\ &= \frac{\Delta L}{\Delta P A} \\ &= \frac{\Delta L A}{\Delta P A^2}\end{aligned} \tag{11.28}$$

Change in volume is

$$\Delta V = \Delta L A \tag{11.29}$$

Therefore,

$$\lambda = \frac{\Delta V}{\Delta P A^2} \tag{11.30}$$

From the definition of bulk modulus,

$$\frac{\Delta V}{\Delta P} = \frac{V}{\beta} \tag{11.31}$$

Substituting Eq. (11.31) into Eq. (11.30),

$$\lambda = \frac{V}{\beta A^2} \tag{11.32}$$

The original volume is $V = AL_1$, and substituting into Eq. (11.32),

$$\lambda = \frac{L_1}{\beta A} \tag{11.33}$$

Compliance of both columns of oil (both ends of cylinder) is given by

$$\lambda_{01} = \frac{L_1}{\beta A}, \quad \lambda_{02} = \frac{L_2}{\beta A}$$

The two columns of fluid are in series. The cylinder body with rigidly attached valve body and mass is represented by mass *m* in Fig. 11.37.

The equivalent stiffness is

$$k = k_1 + k_2$$

Remembering that compliance is the reciprocal of stiffness,

$$\frac{1}{\lambda_o} = \frac{1}{\lambda_{o1}} + \frac{1}{\lambda_{02}} \tag{11.34}$$

Solving for λ_o,

$$\lambda_o = \frac{\lambda_{o1}\lambda_{o2}}{\lambda_{o1} + \lambda_{o2}} \tag{11.35}$$

Substituting for λ_{01} and λ_{02},

$$\lambda_o = \frac{L_1 L_2}{\beta A[L_1 + L_2]} \tag{11.36}$$

When $L_1 = L_2 = L/2$, λ_o is a maximum.

$$\lambda_o(\text{max}) = \frac{L}{4\beta A} \tag{11.37}$$

Eq. (11.37) is normally used for the cylinder compliance.

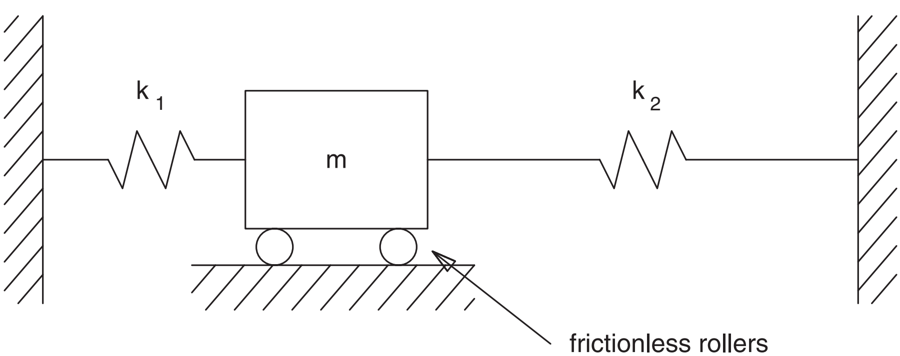


FIGURE 11.37
Model of servo cylinder showing actuator mass supported by columns of fluid on both sides.

There are times when the valve cannot be mounted directly on the cylinder, and oil in the connecting lines does affect performance. It is necessary to now rework the problem and account for oil in the lines between the valve and cylinder.

Suppose that the total oil volume in the lines is V_{line}. If volume change (swelling) of the lines is neglected, movement of the cylinder due to compressibility of this oil may be calculated as follows. Visualize the cylinder as being extended on both ends to provide an increase in volume equal to the volume of fluid in the lines on that end (Fig. 11.38). The length of the extension is

$$L_{line} = \frac{V_{line}/2}{A} \tag{11.38}$$

The effective length of the column of fluid on both ends is

$$L_{eff\,1} = L_1 + L_{line}$$

$$L_{eff\,2} = L_2 + L_{line}$$

and the compliance is

$$\lambda_{o1} = \frac{L_1 + L_{line}}{\beta A}$$

$$\lambda_{o2} = \frac{L_2 + L_{line}}{\beta A} \tag{11.39}$$

Substituting in Eq. (11.35),

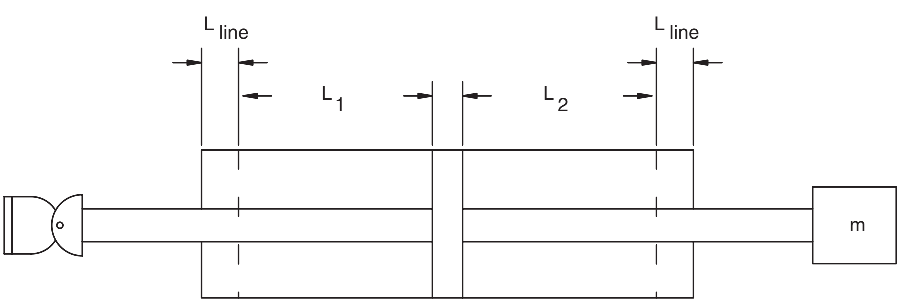


FIGURE 11.38
Diagram of servo valve cylinder showing increase in fluid column length to account for fluid in lines.

$$\lambda_o = \frac{\left(\frac{L_1 + L_{line}}{\beta A}\right)\left(\frac{L_2 + L_{line}}{\beta A}\right)}{\left(\frac{L_1 + L_{line}}{\beta A}\right) + \left(\frac{L_2 + L_{line}}{\beta A}\right)} \tag{11.40}$$

Substituting $L_1 = L_2 = L/2$ and simplifying,

$$\lambda_{o(\max)} = \frac{L/2 + L_{line}}{2\beta A}$$

$$= \left(\frac{L}{4\beta A} + \frac{L_{line}}{2\beta A}\right) \tag{11.41}$$

Substituting Eq. (11.38) into Eq. (11.41),

$$\lambda_o = \frac{L}{4\beta A} + \frac{(V_{line})}{4\beta A}$$

$$= \frac{(LA + V_{line})}{4\beta A^2} \tag{11.42}$$

The term $(LA + V_{line})$ is the *total* volume of fluid in the cylinder and lines.

Natural frequency is defined by

$$\omega = \sqrt{\frac{1}{\lambda_o m}} = \frac{1}{\sqrt{\lambda_o m}} \tag{11.43}$$

where ω = natural frequency (rad/s)
λ_o = compliance (in/lb_f)
m = mass ($lb_f \cdot s^2$/in)

Substituting from Eq. (11.42),

$$\omega = \sqrt{\frac{4\beta A^2}{Vm}} \tag{11.44}$$

where ω = natural frequency (rad/s)
A = cylinder area (in^2)
β = bulk modulus (lb_f/in^2)
V = total volume of fluid in cylinder and lines (in^3)
m = load mass ($lb_f \cdot s^2$/in)

Sometimes Eq. (11.44) is written in terms of oil compressibility, E (in^2/lb$_f$), and volume, V_c, equal to one-half the total volume of oil in the cylinder and the lines.

Substituting

$$V = 2V_c$$

$$\beta = 1/E$$

into Eq. (11.44) gives

$$\omega = \sqrt{\frac{2A^2}{V_c E m}} \tag{11.45}$$

Experience has shown that servo systems should be designed such that the velocity constant falls in the range between one-half and one-third the natural frequency. A velocity constant, $k_v > \omega/2$, will result in an unstable system.

Two types of instability can be observed. In one case, a step input will produce a damped vibration that settles out after a few oscillations. The system is said to *bounce.* It overshoots the desired position and then overshoots in the other direction before it eventually settles at the correct position. The second type of instability, corresponding to a higher k_v, is a continuous oscillation. The system never settles to the desired position.

In this text, we will design for a more conservative k_v and select an open-loop gain such that $k_v = \omega/3$. Generally, a set of components is selected, their transfer functions calculated, and then an amplifier gain is selected to ensure that k_v does not exceed $\omega/3$.

Example Problem 11.2

Design a pot-pot servo as described by the block diagram in Fig. 11.35.

Given:

Maximum mass to be moved: $16\ \text{lb}_f\cdot\text{s}^2/\text{in}$

Volume of fluid under compression:

$$\text{Cylinder} + \text{Line} = \text{Total}$$

$$24.6 + 9.4 = 34\ \text{in}^3$$

Oil compressibility: $5 \times 10^{-6}\ \text{in}^2/\text{lb}_f$

Maximum cylinder speed: 2 in/s

Cylinder area: $4.91\ \text{in}^2$

Servo valve: The valve opens completely with 6 mA input. The servo valve transfer function is calculated as follows. Find maximum flow required to cylinder. Assume that a valve is selected to give this flow at full open and calculate the flow gain.

Length of command pot and feedback pot: 8 in.
Power supply voltage: 15 V

Required:

Find the maximum amplifier gain for stable operation.

Solution:

Servo valve transfer function:

$$\text{Flow Gain} = \frac{\text{Full Flow}}{\text{Full Current}}$$

$$\text{Flow Flow} = \text{Max. Cylinder Velocity} \times A$$

$$= 2 \text{ in/s} \times 4.91 \text{ in}^2 = 9.82 \text{ in}^3\text{/s}$$

$$G_{sv} = \frac{9.82 \text{ in}^3\text{/s}}{6 \text{ mA}} = 1.637 \text{ in}^3\text{/s/mA}$$

Cylinder transfer function:

$$G_{cyl} = \frac{\text{Max. velocity}}{\text{Max. Flow}} = \frac{2 \text{ in/s}}{9.82 \text{ in}^3\text{/s}} = 0.204 \text{ in/in}^3$$

Pot transfer function:

$$H = \frac{15V}{8 \text{ in}} = 1.875 \ V/\text{in}$$

Natural frequency:

$$\omega = \sqrt{\frac{4A^2}{VEm}}$$

$$V = 34 \text{ in}^2$$

$$A = 4.91 \text{ in}^2$$

$$E = 5 \times 10^{-6} \text{in}^2/\text{lb}_\text{f}$$

$$m = 16 \text{ lb}_f \cdot s^2/\text{in}$$

$$\omega = \sqrt{\frac{4(4.91)^2}{34(5 \times 10^{-6})(16)}} = 188.3 \ r/s$$

$$k_v = (188.3)/3 = 62.76 \text{ 1/s}$$

$$k_v = G_a \times G_{svf} \times G_{cyl} \times H \text{or} G_a = \frac{k_v}{G_{svf} \times G_{cyl} \times H}$$

$$G_a = \frac{62.76(1/\text{s})}{\frac{1.637 \text{ in}^3/\text{s}}{\text{mA}} \times \frac{0.204 \text{ in}}{\text{in}^3} \times 1.875 \text{ V/in}}$$

$$= \frac{62.76(1/\text{s})}{0.626 \frac{V}{\text{mA} \cdot \text{s}}} = 100 \frac{\text{mA}}{V}$$

11.7.3 Error Terms

Servo systems can be designed to achieve extraordinary accuracy, but position (or velocity) control is never exact. There is always some error.

11.7.3.1 Position Error

Position error, also called repeatable error, is defined by

$$\text{Position error} = \frac{\sum \text{component deadbands}}{\text{Amplifier Gain} \times \text{Feedback Gain}} \tag{11.46}$$

Generally, the most significant component deadband is the servo valve threshold. The other component deadbands are smaller and are often ignored.

Suppose the servo valve threshold is 3 mA, the amplifier gain is 190 mA/V, and the feedback potentiometer transfer function is 6 V/in. The position error is

$$E_p = \frac{3 \text{ mA}}{190 \text{ mA/V} \times 6 \text{ V/in}} = 0.0026 \text{ in}$$

To achieve a smaller position error, one might think we can simply increase the amplifier gain. Remember, however, that k_v must be kept below $\omega/3$. Increasing G_a increases k_v, and we can drive the system unstable.

The following factors influence position error of a servo cylinder.

1. *Load mass.* An increase in mass gives an *increase* in position error. A larger mass reduces the natural frequency.

$$\omega = \sqrt{\frac{4A^2\beta}{Vm}}$$

Reducing ω reduces k_v. Solving for G_a

$$G_a = \frac{k_v}{G_{sv} \times G_{cyl} \times H}$$

A smaller k_v means a smaller G_a, and since

$$E_p = \frac{\text{deadband}}{G_a \times H}$$

a smaller G_a gives a larger E_p.

2. *Cylinder stroke*. An increase in cylinder stroke gives an *increase* in position error. A larger stroke gives a larger volume of fluid under compression and reduces the natural frequency. Reducing ω reduces k_v, which reduces G_a and thus increases E_p.
3. *Cylinder bore*. An increase in cylinder bore often gives a *decrease* in position error. Note that A is a squared term in the expression for natural frequency; thus, we might expect ω to increase linearly with A. Remember, however, that cylinder area is a term in the calculation of V. An increase in A does give an increase in ω and thus an increase in k_v. Amplifier gain can be higher, so position error will be lower.

11.7.3.2 Tracking Error

Tracking error is the error between the command and feedback *voltages while the command voltage is changing.* This type of error can be best understood by considering a system where one cylinder is "slaved" to another cylinder (Fig. 11.39). In this case, the slave cylinder is supposed to follow the movement of the master cylinder. The *ratio adjust* sets the ratio of the movement of the two cylinders. For example, if the slave cylinder is to move only 50% of the master cylinder movement, the ratio adjust is set at 50%.

When the two cylinders are moving, there has to be a difference in their position. Unless the master cylinder leads the slave cylinder by some amount, there is no error signal, and the servo valve does not open. Tracking error is defined as

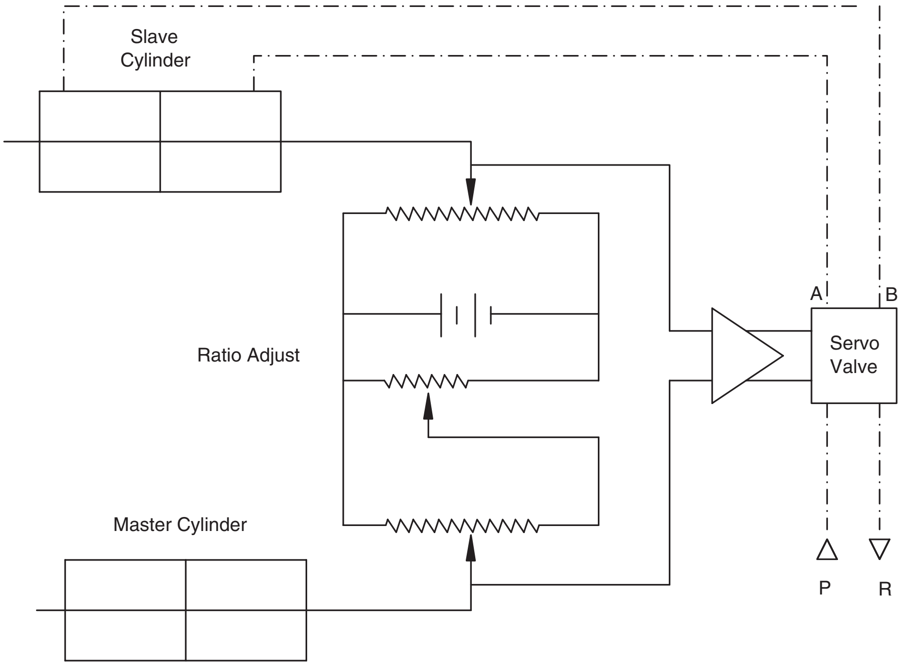


FIGURE 11.39
System for slaving one cylinder to another.

$$E_t = \frac{\text{cylinder velocity}}{G \times H}$$

$$= \frac{\text{cylinder velocity}}{G_a \times G_{svf} \times G_{cyl} \times H} \tag{11.47}$$

11.7.3.3 Example Problem 11.3

The components in a pot-pot tracking system have the following transfer functions:

$$G_a = 800 \text{ mA/V}$$

$$G_{svf} = \frac{0.1 \text{ in}^3\text{/s}}{\text{mA}}$$

$$H = 6 \text{ V/in}$$

The master cylinder has a 2-in. bore and maximum velocity is 5 in/s. Find the tracking error.

$$G_{cyl} = \frac{1}{A} = \frac{1}{\pi(2)^2/4} = 0.318\frac{\text{in}}{\text{in}^3}$$

$$E_t = \frac{5 \text{ in/s}}{800 \text{ mA/V} \times \dfrac{0.1 \text{ in}^3\text{/s}}{\text{mA}} \times 0.318 \text{ in/in}^3 \times 6 \text{ V/in}}$$

$$= 0.033 \text{ in}$$

The slave cylinder trails the master cylinder by 0.033 in. while the master cylinder is moving.

Suppose rated current for the servo valve is 40 mA, and the threshold is 3%. Find the position error when the master cylinder is not moving.

$$\text{Threshold} = 0.03 \times 40 = 1.2 \text{ mA}$$

$$E_p = \frac{\text{Deadband}}{G_a \times H}$$

$$= \frac{1.2}{800(6)} = 0.00025 \text{ in}$$

The position error is less than 1% of the tracking error.

11.7.4 Introduction to the Laplace Domain

This text does not presuppose a course in automatic control as a prerequisite. Many students will have had an introduction to classical control theory and thus will have an appreciation for the techniques used when the transfer function is written in the Laplace domain.

The Laplace transform of *F(t)* is given by

$$L[F(t)] = \int_o^\infty e^{-st}F(t)dt = f(s) \tag{11.48}$$

Suppose $F(t) = 1$.

$$L[F(t)] = \int_o^\infty e^{-st}dt = -\frac{1}{s}e^{-st}\Big]_o^\infty$$

$$= -\frac{1}{s}e^{-s(\infty)} - \left(-\frac{1}{s}e^{-s(o)}\right) = \frac{1}{s}$$

There is a theorem that gives the Laplace transform of an nth derivative as

$$L[F^{(n)}(t)] = s^n f(s) - s^{n-1}F(o) - s^{n-2}F^{(1)}(o) - s^{n-3}F^{(2)}(o)\ldots - F^{(n-1)}(o) \tag{11.49}$$

The notation $F^{(n)}(t)$ means the nth derivative of $F(t)$. The notation $F^{(n-1)}(o)$ means the $n - 1$st derivative of $F(t)$ evaluated at $t = 0$.

The equation of motion for a system with mass (m), elasticity (k), and energy dissipation (c) is

$$F(t) = m\ddot{x} + c\dot{x} + kx \tag{11.50}$$

In this equation, a single dot is used to denote the first derivative with respect to time, and a double dot is used to denote the second derivative with respect to time. The initial velocity is

$$\dot{x}(o) = v_o$$

and the initial displacement is

$$x(o) = x_o$$

Using Eq. (11.49),

$$L[\ddot{x}(t)] = s^2 x(s) - sx_o - v_o$$

$$L[\dot{x}(t)] = sx(s) - x_o$$

$$L[x(t)] = x(s)$$

$$L[F(t)] = f(s)$$

Substituting into Eq. (11.50),

$$f(s) = m[s^2 x(s) - sx_o - v_o] + \mathrm{c}[sx(s) - x_o] + kx(s) \tag{11.51}$$

It is expedient to start the system at rest when $t = 0$; therefore, $v_o = 0$, and if the system is also at zero displacement, $x_o = 0$.

$$f(s) = [ms^2 + cs + k]x(s) \tag{11.52}$$

The definition for a transfer function is output/input. In this case, the input is the force function, $F(t)$, and the output is the displacement, $x(t)$. The transfer function in the Laplace domain is

$$\begin{aligned} G &= \frac{x(s)}{f(s)} \\ &= \frac{x(s)}{[ms^2 + cs + k]x(s)} \\ &= \frac{1}{ms^2 + cs + k} \end{aligned} \tag{11.53}$$

With this brief introduction to transfer functions in the Laplace domain, we are now ready to derive the transfer function for the servo cylinder.

In the case of the servo cylinder shown in Fig. 11.36, the input is the spool displacement (y), and the output is the load displacement (x). The transfer function for the servo cylinder combination is therefore

$$G = \frac{\text{output}}{\text{input}} = \frac{x}{y} \tag{11.54}$$

Load displacement is given by

$$x = \frac{1}{A}\int Q dt \tag{11.55}$$

where x = displacement (in)
A = cylinder area (in^2)
Q = flow rate (in^3/s)
t = time(s)

This displacement is not fully achieved, because the fluid compresses. Remembering the definition for compliance, *actual* displacement is

$$x = \frac{1}{A}\int Q dt - F\lambda \tag{11.56}$$

where F = force (lb_f)
λ = compliance (in/lb_f)

To transform into the Laplace domain, an integral is replaced with $1/s$; therefore, Eq. (11.56) becomes

$$x = \frac{1}{As}Q - F\lambda \tag{11.57}$$

If supply pressure is a constant, which is a good assumption if a good quality relief valve is used in the supply circuit, flow is a function of two variables, spool displacement and load pressure.

$$Q = f(y, P_L)$$

Suppose the servo cylinder is operating at a steady-state operating point. If the changes in y and P_L about this point are small, the flow can be approximated by

$$Q = \frac{\Delta Q}{\Delta y} y + \frac{\Delta Q}{\Delta P_L} P_L \tag{11.58}$$

Substituting Eq. (11.58) into Eq. (11.57),

$$x = \frac{1}{As}\left[\frac{\Delta Q}{\Delta y} y + \frac{\Delta Q}{\Delta P_L} P_L\right] - \lambda F \tag{11.59}$$

Before proceeding further, several definitions are required.

11.7.4.1 Pressure Factor

The pressure factor is the change in flow through the valve per unit change in load pressure. It is defined by

$$P_f = \frac{\Delta Q}{\Delta P_L} \tag{11.60}$$

Thus, it is the slope of the curve shown in Fig. 11.6. (Each valve opening, 25%, 50%, 75%, or 100%, will have a different value for the pressure factor at a given load pressure.) When load pressure is close to supply pressure, the pressure factor is a maximum (slope is maximum), because a small ΔP_L will produce a large change in flow, ΔQ. Conversely, the pressure factor is at a minimum when load pressure is small, because it takes a large ΔP_L to produce a small increase in flow.

11.7.4.2 Flow Gain

Flow gain was previously defined (Sec. 11.4.6) as

$$G_{svf} = \frac{\text{Flow}}{\text{Input current}}$$

In this case, the input is not a current but a physical displacement of the spool.

$$G_{svf} = \frac{\Delta Q}{\Delta y} \tag{11.61}$$

11.7.4.3 Valve Stiffness

Valve stiffness is obtained by blocking Ports A and B with pressure transducers and measuring load pressure, $P_L = P_A - P_B = \Delta P_L$, as the valve is opened. Spool travel is the displacement, y; therefore, valve stiffness is defined by

$$S_v = \frac{\Delta P_L}{\Delta y} \tag{11.62}$$

We can now continue with the derivation of the servo cylinder transfer function. Using the definition of pressure factor and flow gain, Eqs. (11.60) and (11.61), respectively, Eq. (11.59) may be rewritten as

$$\begin{aligned} x &= \frac{1}{As}[G_{svf}y + P_f P_L] - \lambda F \\ &= \frac{G_{svf}}{As}\left[y + \frac{P_f}{G_{svf}}P_L\right] - \lambda F \end{aligned} \tag{11.63}$$

Load pressure is

$$P_L = F/A$$

And, substituting into Eq. (11.63),

$$x = \frac{G_{svf}}{As}\left[y + \frac{P_f}{G_{svf}}\frac{F}{A}\right] - \lambda F \tag{11.64}$$

To simplify notation, define

$$K_a = G_{svf}/A$$

and use the expression for valve stiffness [Eq. (11.62)],

$$S_v = \frac{\Delta P_L}{\Delta y} = \frac{\Delta Q/\Delta y}{\Delta Q/\Delta P_L} = \frac{G_{svf}}{P_f} \tag{11.65}$$

Eq. (11.64) now becomes

$$x = \frac{K_a}{s}\left(y + \frac{F}{S_v A}\right) - \lambda F \tag{11.66}$$

Defining

$$K_b = -S_v A$$

and substituting gives

$$x = \frac{K_a}{s}\left(y - \frac{F}{K_b}\right) - \lambda F \tag{11.67}$$

In general, the load will have mass, elasticity, and energy dissipation characteristics, and the equation of motion is

$$F = m\ddot{x} + c\ddot{x} + kx \tag{11.68}$$

In the Laplace domain,

$$F = (ms^2 + cs + k)x \tag{11.69}$$

The most adverse stability condition arises when the load is primarily an inertia load. Neglecting elasticity ($k = 0$) and energy dissipation ($c = 0$),

$$F = ms^2 x \tag{11.70}$$

Substituting Eq. (11.70) into Eq. (11.67),

$$x = \frac{K_a}{s}\left(y - \frac{ms^2 x}{K_b}\right) - \lambda m s^2 x$$

$$x\left(\lambda m s^2 + \frac{K_a\, ms}{K_b} + 1\right) = \frac{K_a}{s} y \tag{11.71}$$

$$\frac{x}{y} = \frac{K_a}{s[\lambda m s^2 + (K_a m / K_b)s + 1]} \tag{11.72}$$

This transfer function is applicable for simple control systems using a servo cylinder. More complex systems will have a more complex transfer function. Derivation of these transfer functions requires a first course in automatic controls as a prerequisite.

11.8 Summary

Functionally, a servo valve is a *programmable* orifice. It is opened and closed to decrease or increase the total pressure drop in the circuit. This pressure drop at the relief valve causes some portion of the total pump flow to be bypassed back to the reservoir, thus controlling the speed of the actuator.

Servo valve circuits are used to precisely control linear position and velocity and angular position and velocity. This control is accomplished by measuring the load motion and feeding this signal (voltage) to a comparator, where it is compared to a command signal (voltage). When these two voltages (opposite in sign) directly cancel, the load response equals the desired response. Feedback adds stability and stiffness to a motion control system. In this case, stiffness is a term that refers to the fact that a relatively large external disturbance produces a relatively small error in desired motion.

There are four basic servo systems: (1) valve-motor; (2) valve-cylinder; (3) servo pump-motor; and (4) servo pump-motor (split). The servo pump has a servo valve that controls the small cylinders that position the swashplate in the pump. This arrangement is used to control pump displacement and thus control flow to the actuator. It does not dump part of the pump flow across the relief valve as is done with the valve-motor and valve-cylinder systems; thus, it is more efficient. Accuracy of position or speed control is less with the servo pump systems than the valve systems.

Servo valve spools are precisely machined; thus, the valves are more expensive than spool-type directional control valves. Extra filtration requirements also add to the cost of a servo circuit.

Servo valve spools are machined with three lap conditions: line-to-line (most common), underlapped (open center) and overlapped (closed center). The spool slides in a bore machined in the body of the valve. The servo valve transfer function, defined as output divided by input, is a function of the lap condition and the bore fit.

Two transfer functions are used in the analysis of servo valve circuits. Flow gain is defined as

$$G_{svf} = \frac{\text{Flow}}{\text{Input current}}$$

and typical units are in^3/s/mA. Pressure gain is defined as

$$G_{svp} = \frac{\text{Pressure}}{\text{Input current}}$$

and typical units are psi/mA. Here, pressure is the pressure difference between Port *A* and *B*.

The concept of feedback was previously mentioned; thus, it is understood that the output (load response) is measured, and this signal is fed back for comparison with the input (command signal). A system with feedback is termed a *closed-loop* system.

Servo analysis of a servo-cylinder system makes use of the following definition of open-loop gain:

$$GH = G_a \times G_{svf} \times G_{cyl} \times H$$

where GH = open-loop gain
G_a = servo amplifier gain
G_{svf} = servo valve flow gain
G_{cyl} = cylinder gain
H = feedback transducer transfer function

(Note that the terms *gain* and *transfer function* are used interchangeably.) *Velocity constant* is another name for open-loop gain. Experience has shown that a system should be designed such that the velocity constant is less than one-half the natural frequency, and sometimes it must be less than one-third to prevent instability. Typically, amplifier gain is adjusted to reduce the velocity constant to the point where stable operation is achieved.

Servo valves are widely used in industry for precise manufacturing of parts used in every segment of the economy. Servo valves and servo circuits are the key components in the use of fluid power actuators to implement the decisions made by automatic control systems.

References

Johnson, J. E. 1977. *Electrohydraulic servo systems.* Penton/IPC, 1111 Chester Ave., Cleveland, OH 44114.

Moog, Inc. 1999. Technical product literature. Moog Inc., Industrial Division, East Aurora, NY 14052.

Problems

11.1 You are required to design a pot-pot servo positioning system.

Given:

Electric supply: 15 VDC

Command pot and feedback pot length: 6 in.

Servo valve requires 100 mA to open to the full open position

Servo valve deadband 5 mA

Maximum flow 10 GPM

Cylinder: double rod, 3.25-in. bore, 1.75-in. rod, 6.5-in. stroke

Maximum load 3500 lb_f

Total volume of fluid in lines between the cylinder and servo valve 9.42 in^3

Fluid is oil with compressibility $E = 5 \times 10^{-6}\ in^2/lb_f$

Required: Find the amplifier gain that gives a velocity constant equal to $\omega/3$ where ω is the natural frequency of the system.

11.2 Design a pot-pot positioning system using the following information.

Load:

a. Cylinder area: $A = 5.83\ in^2$
b. Maximum mass to be moved: $M = 9\ lb_f \cdot s^2/in$
c. Volume of fluid under compression: $V = 60\ in^3$

Servo-Cylinder:

a. Flow gain for the servo valve: 0.25 $in^3/s/mA$
b. Maximum cylinder speed: 500 in/min
c. Servo valve is fully open at 300 mA

Potentiometers:

a. Length of command pot and feedback pot: 8 in
b. Power supply voltage: 15 VDC

Required:

a. Find the maximum amplifier gain for stable operation of this system. *Hint: Use the "rule of thumb" that the velocity constant should not be greater than one-third the natural frequency.*
b. If the valve deadband is 3 mA, find the position error for this system.

11.3 Design a pot-pot positioning system using the following information:

Load:

a. Maximum mass to be moved: 16 $lb_f \cdot s^2/in.$
b. Volume of fluid under compression:

Cylinder	+	Lines	=	Total
24.6	+	9.4	=	34 in^3

Cylinder:

a. Maximum cylinder speed: 2 in/s

b. Cylinder area: 4.91/in^2

Servo valve:

The valve opens completely with 6-mA input. Assume that a valve is selected to give the maximum flow to the cylinder at full open.

Potentiometers:

a. Length of command pot and feedback pot: 8 in.

b. Power supply voltage: 15 VDC

Required:

a. Find the maximum amplifier gain for stable operation.
Hint: Use the "rule of thumb" that the velocity constant should not be greater than one-third the natural frequency.

b. If the valve deadband is 1% of rated current, find the position error for this system.

11.4 Analyze the electrohydraulic servo system shown in Fig. 11.40. The fluid is oil and the actuator is a double rod cylinder with 10-in. stroke. The servo valve is mounted directly on the cylinder so fluid in the lines can be neglected.

Given:

Amplifier transfer function: 80 mA/V

Servo valve characteristics: full open flow of 5 GPM at rated current of 30 mA.

Cylinder transfer function: 0.318 in^{-2}.

Feedback transducer: 60-V power supply, 10 in. potentiometer displacement

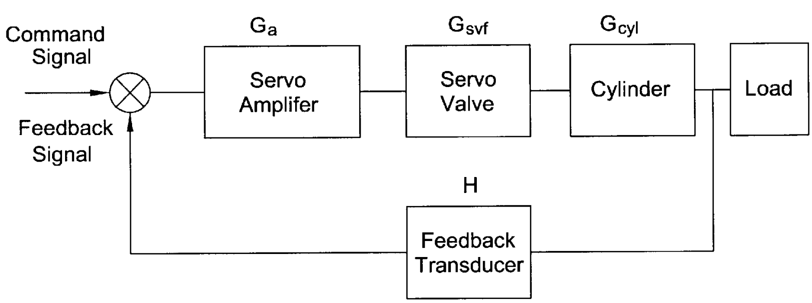


FIGURE 11.40
Block diagram of servo cylinder (Problem 11.4).

Required:

a. Find the velocity constant, k_v.

b. The design rule of thumb requires that the natural frequency be below $\omega = 3\,k_v$. What is the maximum mass that can be moved by this servo cylinder?

Hint: You will have to make certain assumptions. Your instructor may choose to supply the missing parameters. If not, make sure your assumptions are consistent with what you learned in solving Problems 11.1 through 11.3.

11.5 Design a pot-pot positioning system to move a vehicle frame into position for welding. The following design parameters are given.

Maximum supply pressure:

3000 psi

Load:

The part has a mass, $M = 87.6\ \text{lb}_f\cdot\text{s}^2/\text{in}$. It is pushed into position on a surface with a coefficient of friction, $\mu = 0.4$. Estimated load force is

$$\begin{aligned} F_L &= mg\mu \\ &= 87.6(386)(0.4) \\ &= 13{,}525\ \text{lb}_f \end{aligned}$$

Load pressure:

The decision is made to design for a maximum load pressure equal to $2P_s/3$.

$$P_L = \frac{2(3000)}{3} = 2000\ \text{psi}$$

Cylinder:

A double-rod cylinder of the approximate size needed has been found. The estimated friction for this cylinder is

$$F_f = 85\ \text{lb}_f$$

The force balance for the cylinder is

$$P_A A = F_L + F_f + P_B A$$

or

$$(P_A - P_B)A = F_L + F_f$$

Since $P_A - P_B = P_L = 2000\text{ psi}$,

$$A = \frac{13{,}525 + 85}{2000} = 6.8\text{ in}^2$$

The cylinder chosen has a 3.25-in. bore and 1.375-in. rod diameter.

$$A = \frac{\pi}{4}[(3.25)^2 - (1.375)^2]$$

$$= 6.8\text{ in}^2$$

Maximum cylinder velocity when moving the part into position is

$$v = 275\text{ in/min}$$
$$= 4.58\text{ in/s}$$

Required load flow:

$$Q_L = \frac{60Av}{231}$$
$$= \frac{60(6.8)(4.58)}{231} = 8.09\text{ GPM}$$

The servo valve is mounted directly on the cylinder, so the volume of fluid under compression in the lines is negligible. Cylinder stroke is 8 in., so the total fluid under compression in the cylinder is

$$V = As$$
$$= 6.8(8) = 54.4\text{ in}^3$$

Servo valve:

A "family" of servo valves are available with rated flow at 1000 psi of 1, 2.5, 5, 7.5, and 10 GPM. Which of these valves should be selected?

The decision is made to choose the servo valve rated for 10 GPM rather than the one rated for 7.5 GPM. It is important not to oversize servo valve flow capacity as this will needlessly reduce system accuracy. In this case, we desire to operate below the P_s = 3000 psi maximum supply pressure, so we choose a larger valve to achieve a smaller pressure drop across the valve.

The servo valve chosen has a rated current of 14 mA. When 15 mA is applied, the valve moves to the full open position. The deadband for the valve is 1% of rated current.

Potentiometers:

a. Length of command pot and feedback pot: 8 in.

b. Power supply voltage: 15 VDC.

Required:

You are asked to select an amplifier for this installation and then do an analysis to determine the position error.

a. Find the maximum amplifier gain for stable operation. *Hint: Use the "rule of thumb" that the velocity constant should not be greater than one-third the natural frequency.*

b. Find the position error for this system.

12

Proportional Valves

12.1 Introduction

There are many applications in which the precision of servo valves is not needed, but more accuracy is needed than can be obtained with conventional valves. Proportional valves were developed to fill this gap. Their use has increased significantly since the mid 1970s.

A comparison of proportional valves and servo valves is given in Table 12.1. Proportional valves are less accurate, but they cost less. They are not used for closed-loop circuits; there is no feedback of the output to change the setting of the valve.

TABLE 12.1

Comparison of Characteristics of Proportional Valves and Servo Valves

	Proportional valves	**Servo valves**
Type of Loop	Open	Closed
Feedback	No	Yes
Accuracy	Moderate error factor ≥ 3%	Extremely high error factor < 1%
Frequency response	Low: <10 Hz	Very high: 60–400 Hz
Cost	Moderate	High
Need for auxiliary electronic equipment	Moderate	Substantial
Sensitivity to contamination	Tolerant	High

12.2 Types of Proportional Valves

There are two types of proportional valves.

1. Force-controlled
2. Stroke-controlled

When a conventional solenoid is energized, the plunger travels its full stroke. A given force is developed at the moment actuation occurs. With a force-controlled proportional valve, a provision is provided to increase the force output by the solenoid proportional to the input signal. With a stroke-controlled valve, the stroke *distance* is proportional to the input signal. Both types provide an opening of the valve (movement of the valve spool) proportional to the magnitude of the milliamp current applied to the valve. Spool movement is proportional to input current.

12.2.1 Force-Controlled Proportional Valves

A functional diagram of a force-controlled proportional valve is given in Fig. 12.1. The spool is held in position by a spring. When current is applied to the solenoid coil, the plunger presses against the spool. An increase in current produces an increase in force, which produces a greater deflection of the spring, and the spool moves to open the valve. Force produced by the solenoid is almost a linear function of input current (Fig. 12.2).

Force-controlled solenoids can replace the mechanical spring in several types of pressure control valves to create a valve that can be adjusted with an electrical signal rather than a manual input. It is readily apparent that this

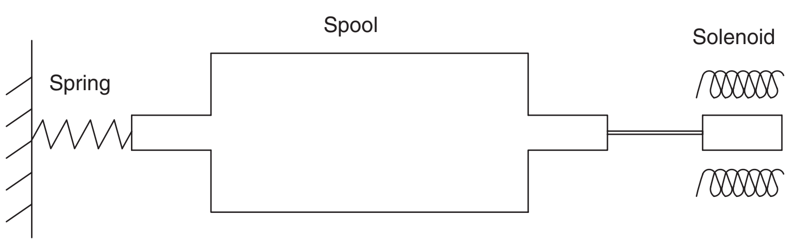


FIGURE 12.1
Functional diagram of force-controlled proportional valve.

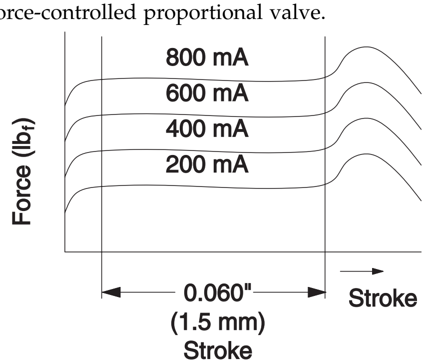


FIGURE 12.2
Force vs. input current developed by Rexroth proportional valve. Reprinted with permission from Mannesmann Rexroth.

replacement provides many control options. Now, the setting of a circuit relief valve can be changed based on a signal from a transducer that is measuring some variable, for example, the torque to turn a screw. In this case, torque is limited by changing the setting of the relief valve.

12.2.1.1 Proportional Pilot-Operated Relief Valve

A functional diagram of a pilot-operated relief valve with a force solenoid replacing the manually adjusted pilot spring is shown in Fig. 12.3a, and a cross section of an actual valve is shown in Fig. 12.3b. [Note that the pilot poppet (Fig. 12.3b) is equivalent to the pilot dart (Fig. 12.3a). The term *pilot poppet* will be used in this chapter.] The functional diagram shows an internal drain. In the actual proportional relief valve (Fig. 12.3b), the pilot section is drained through Port Y to a drain line connected directly to the reservoir. The sensitivity of the proportional solenoid requires that the pilot be drained directly back to the reservoir. Any back pressure on an internal drain could cause erratic valve operation.

A power amplifier (like the servo amplifier discussed in Chapter 11) is used to produce the milliamp signal to the force solenoid. Response time is good. Time to change from one pressure setting to another in response to the milliamp signal from the amplifier ranges from 50 to 150 ms, depending on valve size. In the case of a power failure, the milliamp input signal drops to zero, and the pilot poppet opens at very low pressure. If there is concern that an electronic failure will send a high current to the solenoid and thus set the pressure setting too high, a mechanically adjusted pilot can be installed and

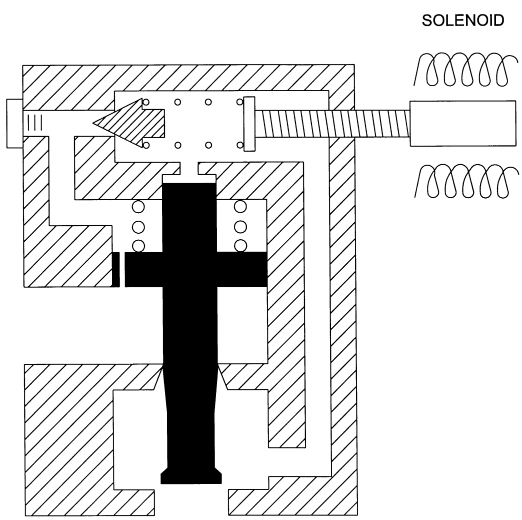


FIGURE 12.3a
Functional diagram of pilot-operated relief valve with a force solenoid in place of a screw adjustment for the pilot spring. Reprinted with permission from Parker Hannifin Corp.

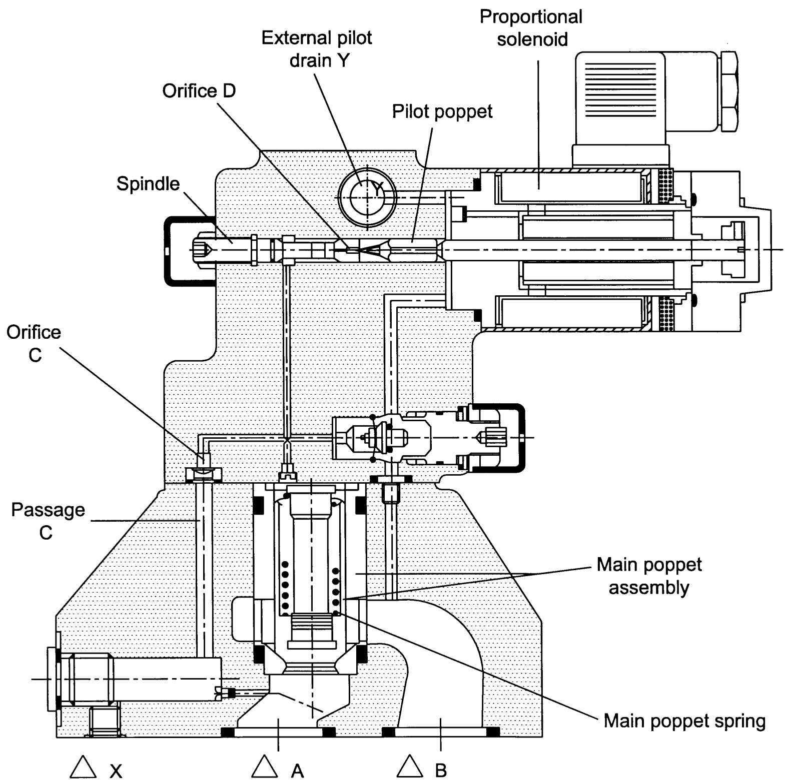


FIGURE 12.3b
Cross-sectional view of proportional pressure relief valve. With permission from Mannesmann Rexroth.

set to relieve at a pressure just above the force solenoid setting. This backup ensures hydraulic circuit protection.

12.2.1.2 Proportional Pressure Reducing Valve

A cross-sectional view of a proportional pressure reducing valve is shown in Fig. 12.4. The cross section looks very similar to the cross section of the pilot-operated relief valve (Fig. 12.3b). The same pilot poppet is shown and the same main spool. In this cross section, however, flow goes from Port *B* to Port *A*.

The proportional pressure reducing valve works as follows. When pressure rises above the set point at Port *A*, this pressure acts on the seat of the pilot poppet and creates a force large enough to offset the solenoid force. The pilot poppet unseats, and fluid drains out of the pilot chamber into Port Y. Fluid flows through the pilot passage C to replace the lost fluid. There is a pressure drop across orifices 6 and 7 because of this flow. (With zero flow, the pressure is the same on both sides of the orifice.) Now, the pressure on top of the main

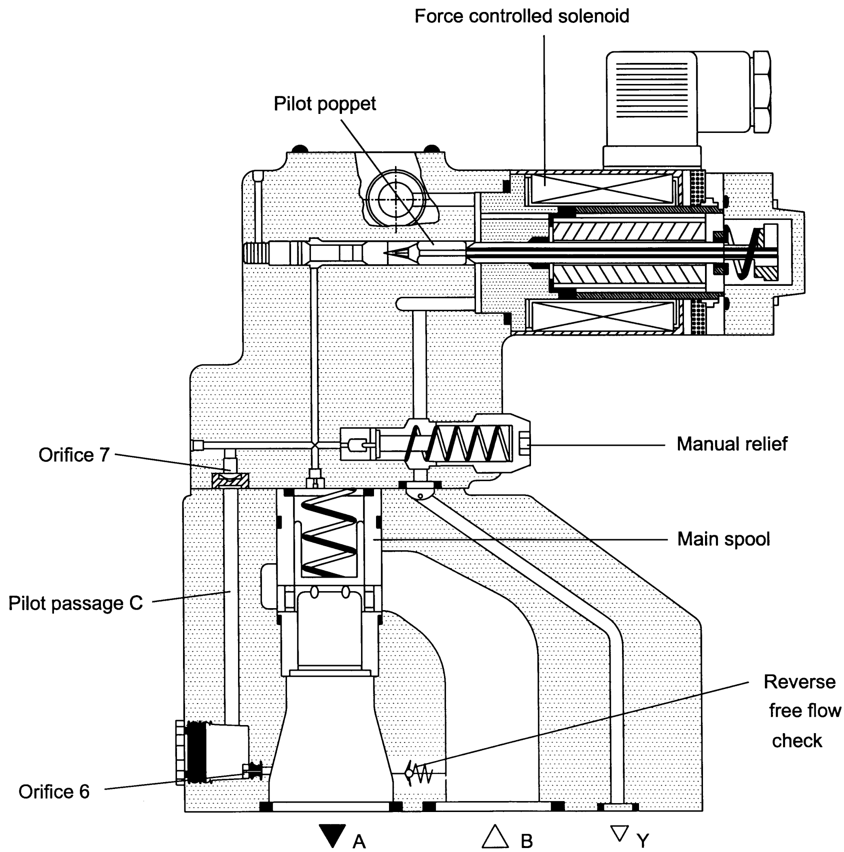


FIGURE 12.4
Cross-sectional view of proportional pressure reducing valve. Reprinted with permission from Mannesmann Rexroth.

spool is lower than the pressure on the bottom. A little more pressure at Port *A*, and the hydraulic force on the bottom of the main spool is enough to overcome the hydraulic force on the top plus the spring force. The main spool moves up and reduces the area of the opening between Port *B* and Port *A*. The pressure drop across this reduced opening reduces the pressure at Port *A*. The valve modulates the position of the main spool in this manner to maintain the set pressure at Port *A*. This set pressure can be quickly and accurately changed by changing the milliamp signal to the force solenoid, and thus changing the force it exerts on the pilot poppet. Response time ranges from 100 to 300 ms, depending on valve size, to move from a minimum to a maximum setting. Solenoid force can also be increased or decreased gradually, resulting in a gradual increase or decrease in pressure, as needed.

At high flows, the pressure at Port *A* may vary because of turbulence produced by the partly closed main spool. This pressure fluctuation will affect operation of the reducing valve. The solution is to take pilot oil from Port *B* (the primary port). This pilot oil flows across a pressure-compensated flow control valve (Fig. 12.5) to the top of the main spool. Remember that no flow occurs until the pressure increases to the point where the pilot poppet is unseated. Once flow is established, the pressure-compensated flow control maintains a *constant* flow of pilot oil. Resulting regulated pressure on the top of the main spool gives a smoother regulation of the set pressure at Port *A*.

12.2.1.3 Proportional Directional Control Valves

A proportional directional control valve (DCV) has many of the same features of a conventional four-way, three-position directional control valve. The graphic symbols have orifices shown on the arrows to indicate that the spool has been machined to allow metering of the flow (Fig. 12.6).

The spool in a proportional DCV can be machined with different shaped notches in the spool lands. For the following discussion, we will assume that these are triangular notches. When the spool is shifted, openings are pro-

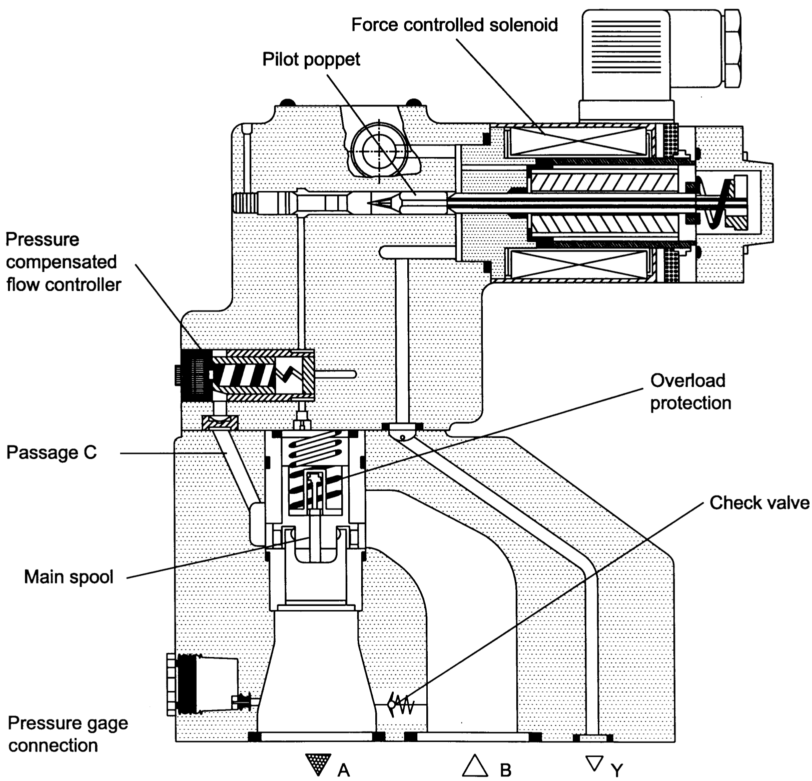


FIGURE 12.5
Cross-sectional view of proportional pressure reducing valve showing pressure-compensated flow valve used to smooth pilot flow and thus smooth the pressure exerted on the top of the main spool. Reprinted with permission from Mannesmann Rexroth.

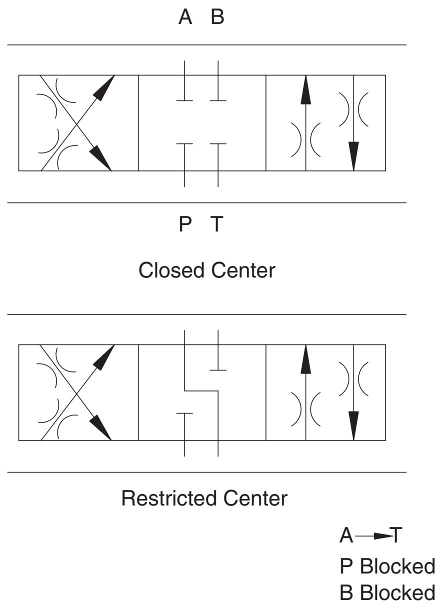


FIGURE 12.6
Graphic symbols used for proportional directional valves.

duced as shown in Fig. 12.7. Note the matched notches; flow is metered when the spool is shifted in both directions. Overlap, which gives the deadband shown, is generally about 10% of the total spool travel. In most valves, spool movement stops just short of the full open position so that some metering function is maintained. Because the metering notches are the same on both sides of the spool lands (Fig. 12.8), pressure drop from P to A is approximately equal to the pressure drop from B to T.

A special spool is needed to control a cylinder with a 2:1 area ratio. Flow from the rod end during extension is half the flow to the cap end. A spool with equal triangular notches will have unequal pressure drops across the valve because of the unequal flow through equal notch areas.

To control a cylinder with 2:1 area ration, a spool is machined with twice the number of notches on one side of the land as the other side. The best way to visualize this is to think about a piece of tape wrapped around the spool. The outlines of the v-notches are printed on this tape. When it is unrolled, the image shown in Fig. 12.9 is produced. The example has eight notches on one side and four on the other. Total area opened per unit of spool displacement is $2A$ on the right and $1A$ on the left. This spool configuration keeps the total pressure drop on both sides of the proportional DCV fairly equal, and good controllability of cylinders with 2:1 area ratios is maintained.

A cross-sectional view of a proportional DCV is given in Fig. 12.10. Note that there is a force solenoid on both ends to produce proportional control in both directions. The valve has a main spool and a pilot spool like the two-

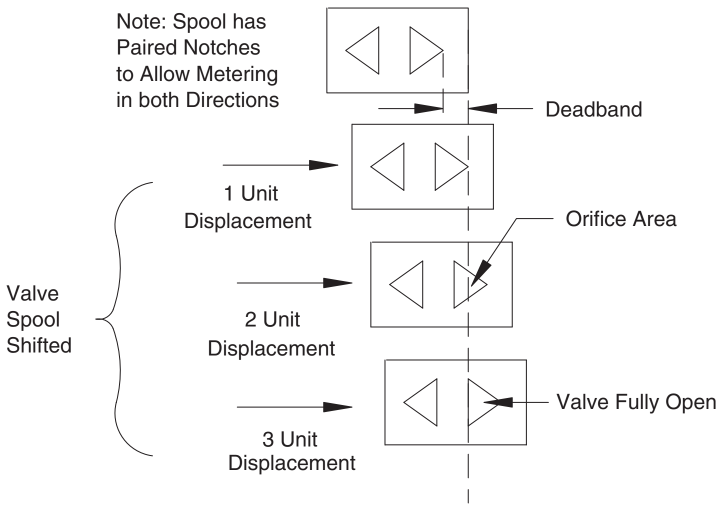


FIGURE 12.7
Openings produced when the spool of a proportional directional control valve is shifted.

FIGURE 12.8
Actual proportional directional control valve spool with triangular notches. Reprinted with permission from Mannesmann Rexroth.

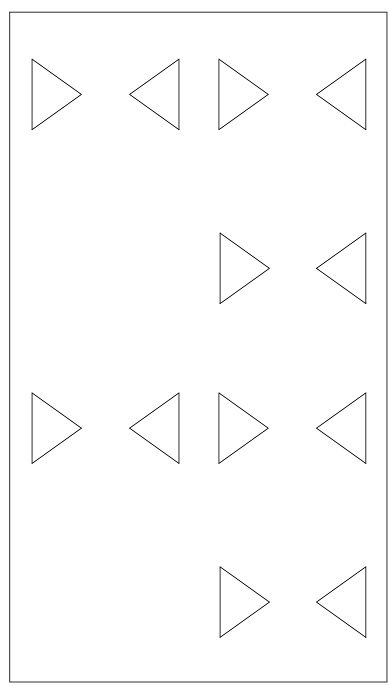


FIGURE 12.9
Diagram showing how notches are machined in the spool of a proportional directional control valve used to control a cylinder with a 2:1 area ratio.

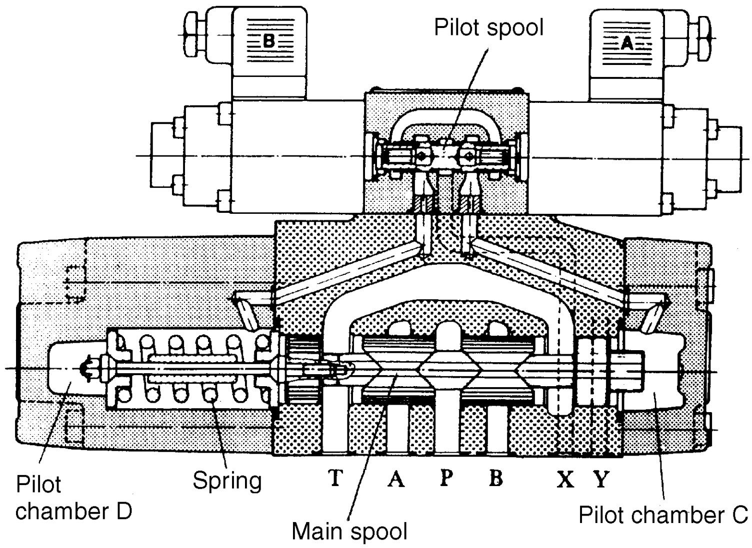


FIGURE 12.10
Cross-sectional view of a proportional directional control valve. Reprinted with permission from Mannesmann Rexroth.

stage servo valve discussed in Chapter 11. The force solenoids shift the pilot spool to supply pilot pressure to the pilot chambers. Pressure in Chamber *C* acts against the right end of the spool causing it to move to the left against the main spool spring. The main spool spring is a push-pull spring. Pressure in Chamber *D* acts on the end of the piston that seals the left end of the spring cavity. (The right end is sealed by the end of the main spool.) This piston compresses the spring, which presses against the left end of the main spool causing it to shift toward the right. The higher the pilot pressure, the greater the spool displacement. The spring holds the main spool in the center position until it is acted on by pilot pressure in one of the pilot chambers.

It is appropriate to use specific pressure values for the following discussion. These values were supplied by Manesmann Rexroth Corp. and are appropriate for their valves.

Maximum pilot pressure needed in either pilot chamber to produce full main spool displacement is 365 psi. This pilot pressure is produced when the solenoid develops 14 lb_f force. As the solenoid force increases from 0 to 14 lb_f, the pilot pressure increases from 0 to 365 psi. Thus, pilot pressure, and subsequently spool displacement, is proportional to the input signal to the solenoid. The triangular notches in the spool open to provide a progressively larger orifice as the spool shifts. Orifice size is programmed by the signal to the solenoid.

Like the servo valve, a proportional DCV is really just a programmable orifice. As discussed in Chapter 11, it is important to remember that a servo valve (and a proportional DCV) work in conjunction with the relief valve.

The programmable orifice is set to create a ΔP, which, when added to other ΔPs in the circuit, creates a total pressure that opens the relief valve. Flow to the actuator is controlled by dumping part of the pump flow across the relief valve; thus, a proportional DCV controls flow by converting hydraulic energy to heat energy. As with servo valves, provision for adequate cooling is an important issue when designing a circuit with a proportional DCV.

It is possible to program a gradually increasing signal to the proportional DCV, hold this signal constant at some value for an interval of time, and then gradually decrease the signal back to zero. This procedure will gradually increase flow to accelerate a load, hold a constant velocity, and then gradually decelerate the load to a stop. Many manufacturing operations require this type of load control, and a proportional DCV works quite well for these applications. Adequate control is achieved without the expense and strict contamination control requirements of a servo valve.

Typical frequency response for a proportional DCV is shown in Fig. 12.11a. The signal used to generate curve (a) was ±25% of rated input current about 50%, and the signal to generate curve (b) was ±50% about 50% (Fig. 12.11b). As with servo valves, the industry standard method for rating proportional valves is the frequency at which attenuation is – 3 dB. In this case, the rated frequency response using a ±25% signal [curve (a)] is 12 Hz, and using a ±50% signal [curve (b)], it is 9 Hz. The results are consistent with the results shown in Table 12.1, where the frequency response for proportional valves is listed as < 10 Hz.

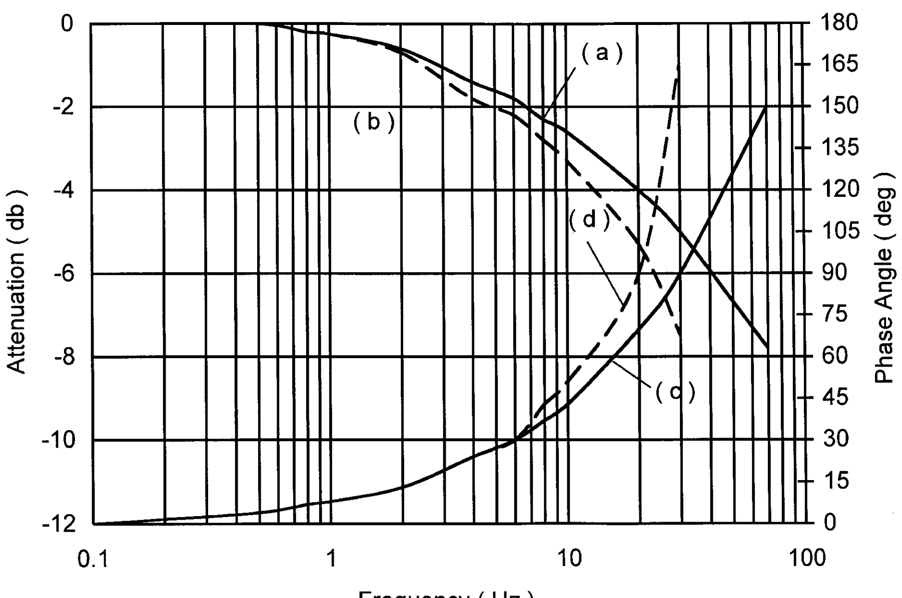


FIGURE 12.11a
Frequency response curves for proportional directional control valve.

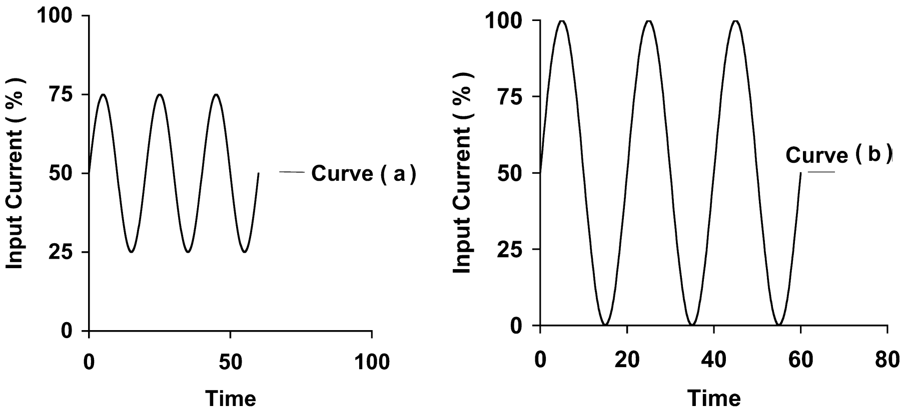


FIGURE 12.11b
Input signals used to obtain frequency response data in Fig. 12.11a.

Phase lag curves are plotted for the ±25% signal [curve (c)] and the ±50% signal [curve (d)]. At 12 Hz, the phase lag for the ±25% signal is approximately 45°, meaning that the output lags the input by 45°. The phase lag at 9 Hz is also approximately 45° for the ±50% signal. As signal frequency increases, the ability of the valve to "keep up" decreases (phase angle increases) and control is lost. If the frequency increases to the point where the phase angle is 180°, meaning that the output is moving in the opposite direction from the input, the system is unstable.

12.2.2 Summary

Force solenoids are used to adjust the compression of the spring in a relief valve and a pressure reducing valve. These valves can then be adjusted with an electrical signal instead of a manual adjustment.

A proportional DCV uses a force solenoid to adjust the pilot pressure used to shift the main spool. Spool displacement is proportional to pilot pressure, which is proportional to the electrical signal applied to the force solenoid. Like a servo valve, a proportional DCV is a programmable orifice. It is important to remember, however, that the force solenoid proportional DCV is an open-loop system. No feedback is provided; thus, the high precision achieved with a servo valve is not possible.

12.2.3 Stroke-Controlled Proportional Valves

The stroke-controlled solenoid was developed to provide feedback of the actual movement (the stroke) of the solenoid. The solenoid acts directly on the spool of a directional control valve. A linear variable differential transformer (LVDT) is mechanically linked to the spool. As the spool moves, it moves the core of the LVDT and produces a feedback signal proportional to spool movement. This feedback signal is fed back to the amplifier (Fig. 12.12). As with the closed-loop servo circuit discussed in Chapter 11, the amplifier output increases until the error signal (error = command – feedback) goes to zero. The solenoid is held in this position until a new command signal is introduced.

12.2.3.1 Direct-Operated Proportional Directional Control Valve

A proportional DCV with stroke-controlled solenoid is known as a *direct-operated* valve, because the solenoid moves the spool directly; there is no pilot stage. These valves have a solenoid on both ends to move the spool in either direction. There is a spring on both sides to keep the spool centered when no signal is applied.

Stroke-controlled proportional DCVs are the most accurate of all proportional directional valves. Their main drawback is a limitation in size. The largest valve available has a nominal flow rating of 16 GPM with a 150 psi pressure drop across the valve. For comparison, force-controlled solenoid valves with pilot stage are available with nominal flow ratings up to 140 GPM.

High flow forces are generated by the high flows. In the stroke-controlled solenoid valves, the solenoid acts directly on the spool, and it must develop the force needed to overcome the flow forces. A very large solenoid would be needed for large valves, and this is not practical.

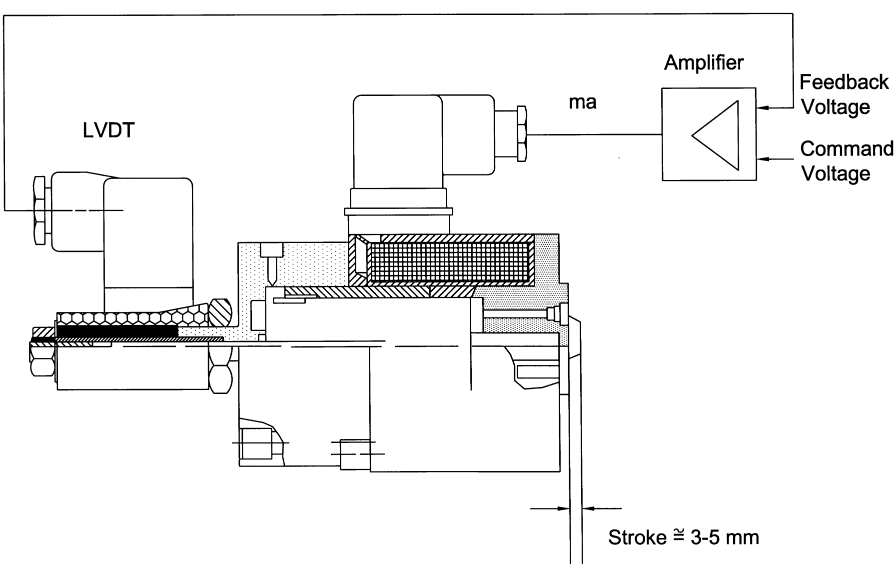


FIGURE 12.12
Feedback of spool position in a direct-operated proportional directional control valve. Reprinted with permission from Mannesmann Rexroth.

The spool is larger in a force-solenoid proportional DCVs; thus, the area acted on by the pilot pressure is larger. Enough force is generated to overcome the flow forces without increasing pilot pressure. Rexroth uses a maximum pilot pressure of 435 psi for its complete size range of force-solenoid proportional DCVs.

12.2.3.2 Direct-Operated Proportional Relief Valves

As might be expected, the stroke-controlled solenoid can be used to adjust spring compression in a direct-acting relief valve. The resulting valve is called a *direct-operated proportional relief valve* (Fig. 12.13). The solenoid acts on a *pressure pad*. This pad is mounted in a cavity such that it can move back and forth a distance equal to the stroke length of the solenoid. A very precise spring is used in this valve. Each increment of pressure pad movement gives a new spring setting and thus a new pressure setting. The key advantage, of course, is that an electrical signal is used to change the pressure setting. Feedback from the LVDT ensures that a specific spring position is always achieved with a given command signal.

12.3 Analysis of Proportional Directional Control Valve

A functional diagram of a proportional valve in a simple cylinder circuit is shown in Fig. 12.14. The pressure drop between Port P and Port A is shown

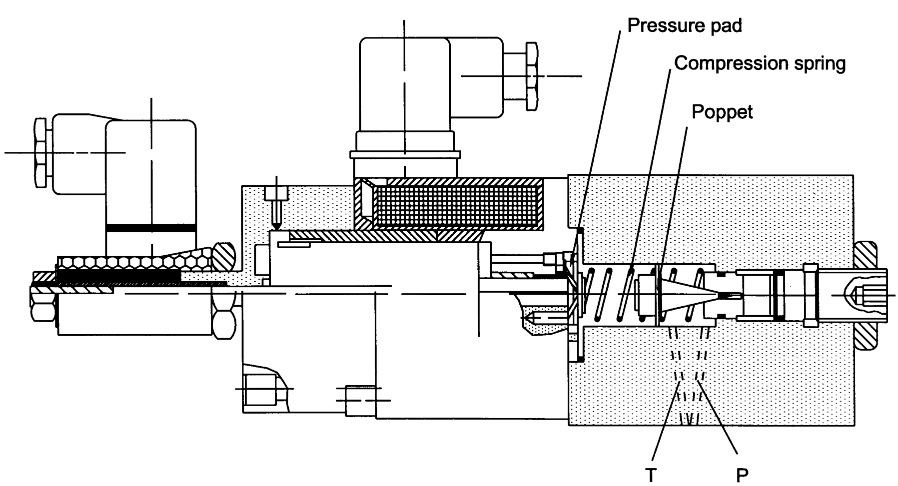


FIGURE 12.13
Direct-operated proportional relief valve. Reprinted with permission from Mannesmann Rexroth.

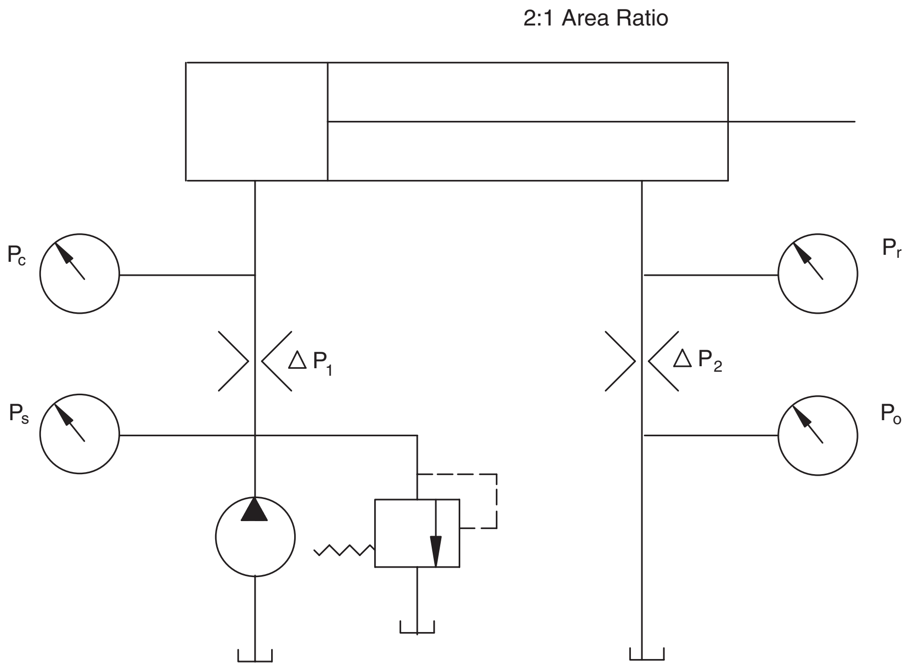


FIGURE 12.14
Functional diagram of a cylinder circuit with proportional valve.

as ΔP_1, and the pressure drop between Port *B* and Port *T* is shown as ΔP_2. This notation will be used throughout this chapter. Flow corresponding to ΔP_1 will be designated Q_1, and flow corresponding to ΔP_2 will be designated Q_2. The cylinder has a 2:1 area ratio, and a proportional valve with a 2:1 area ratio is used.

As shown in Fig. 12.15, the orifice equation applies for both sides of the valve.

$$Q_1 = CA_1\sqrt{\Delta P_1} \tag{12.1}$$

$$Q_2 = CA_2\sqrt{\Delta P_2} \tag{12.2}$$

where Q_1 = flow into cap end of cylinder (in^3/s)
Q_2 = flow out of rod end of cylinder (in^3/s)
A_1 = area of orifice between Port *P* and Port *A* (in^2)
A_2 = area of orifice between Port *B* and Port *T* (in^2)
C = orifice coefficient ($\text{in}^2 \cdot \text{s}^{-1} \cdot \text{lb}_f^{-0.5}$)
ΔP_1 = pressure drop between Ports *P* and *A* (psi)
ΔP_2 = pressure drop between Ports *B* and *T* (psi)

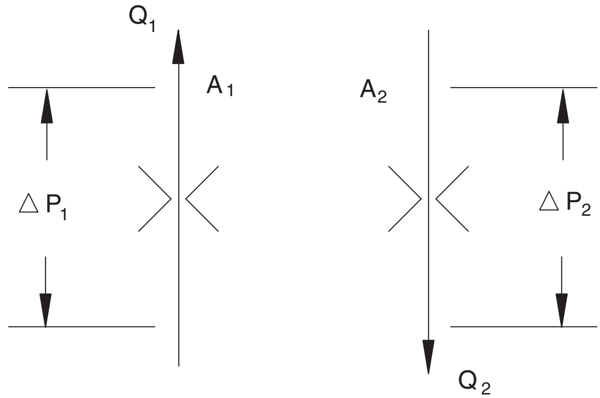


FIGURE 12.15
Orifices created when a proportional valve is opened.

The orifices have the same shape on both sides of the spool land. There are just twice as many grooves on one side for a valve with a 2:1 ratio. Note that the same orifice coefficient (C) is used for both sides.

The area ratio of the cylinder is 2:1; therefore, during extension, $Q_2 = Q_1/2$. Flow out the rod end for each inch of movement is half the flow into the cap end. The valve has a 2:1 area ratio; therefore, $A_2 = A_1/2$. Substituting into Eqs. (12.1) and (12.2),

$$Q_1 = CA_1\sqrt{\Delta P_1} \tag{12.3}$$

$$Q_2 = Q_1/2 = C(A_1/2)\sqrt{\Delta P_2} \tag{12.4}$$

or

$$Q_1 = CA_1\sqrt{\Delta P_2}$$

These two equations are satisfied only if $\Delta P_1 = \Delta P_2$. Having an equal, or approximately equal, pressure drop on both sides of the valve gives good controllability of a cylinder with 2:1 area ratio. It is left as an exercise to show that $\Delta P_2 = \Delta P_1/4$ if $A_1 = A_2$.

12.3.1 Overrunning Load

Problems can result if a valve with 1:1 area ratio is used with a cylinder with 2:1 area ratio. Suppose the circuit shown in Fig. 12.14 has an overrunning load during extension (Fig. 12.16). The force balance on the cylinder is

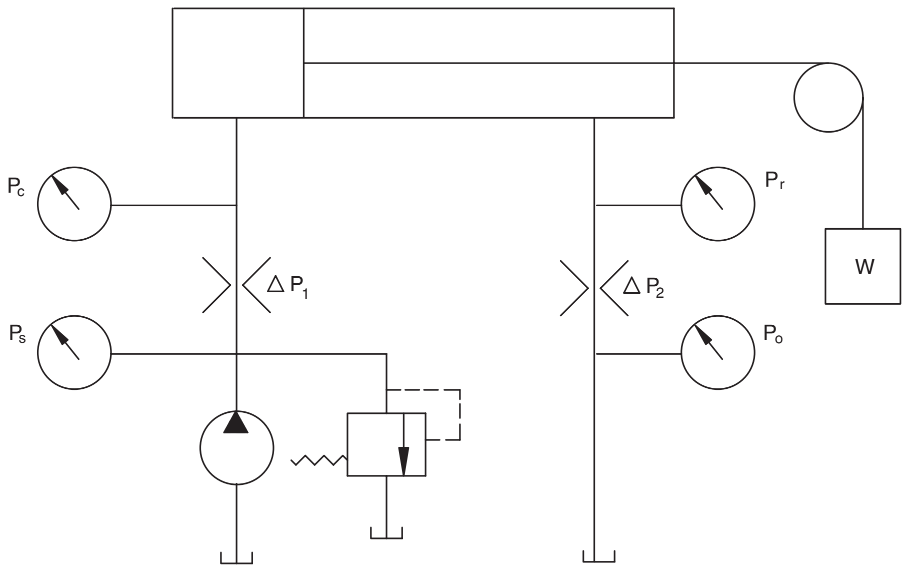


FIGURE 12.16
Cylinder with 2:1 area ratio and valve with 1:1 area ratio controlling an overrunning load.

$$P_c A_c = P_r A_r + F_f + F_L \tag{12.5}$$

where $F_L = W$ = load on the cylinder (lb_f)
F_f = friction force (lb_f)

In this case, F_L is negative, since the load is overrunning, i.e., it is acting in the direction of the cylinder movement. Solving for P_r,

$$P_r = (P_c A_c + F_L - F_f)/A_r \tag{12.6}$$

The pressure drop across the Port P to A orifice in the proportional valve is

$$\Delta P_1 = P_s - P_c$$

Neglecting any pressure drop between the proportional valve outlet and the reservoir, $P_o = 0$, the pressure drop from Port A to T is

$$\Delta P_2 = P_r - P_o = P_r$$

With a 1:1 area ratio valve, $A_1 = A_2 = A$, and the orifice equations become

$$Q_1 = CA\sqrt{\Delta P_1} \tag{12.7}$$

$$Q_2 = CA\sqrt{\Delta P_2} \tag{12.8}$$

Solving for *CA* and equating the two expressions,

$$Q_1/\sqrt{\Delta P_1} = Q_2/\sqrt{\Delta P_2} \tag{12.9}$$

or

$$Q_1/Q_2 = \sqrt{\Delta P_1}/\sqrt{\Delta P_2}$$

Squaring both sides,

$$Q_1^2/Q_2^2 = \Delta P_1/\Delta P_2 \tag{12.10}$$

or

$$\Delta P_2 = \Delta P_1 Q_2^2/Q_1^2$$

Substituting for ΔP_1 and ΔP_2,

$$P_r = (P_s - P_c)Q_2^2/Q_1^2 \tag{12.11}$$

Equating Eq. (12.6) and Eq. (12.11), we can now solve for the pressure at the cap end of the cylinder.

$$\frac{P_c A_c + F_L - F_f}{A_r} = (P_s - P_c)Q_2^2/Q_1^2$$

$$P_c = \frac{P_s(Q_2^2/Q_1^2) - (F_L - F_f)/A_r}{(A_c/A_r) + (Q_2^2/Q_1^2)} \tag{12.12}$$

Under certain conditions, P_c can be negative. This means a vacuum will exist in the cap end of the cylinder; the cylinder will not be completely filled with oil. When this condition develops, positive control of the load is lost during extension.

It is instructive to determine what load will cause P_c to go negative. Setting $P_c = 0$ and solving for F_L,

$$F_L = P_s A_r \frac{Q_2^2}{Q_1^2} + F_f \tag{12.13}$$

The following parameter values are assigned:

$$P_s = 1500 \text{ psi}$$

$$A_c = 3.14 \text{ in}^2$$

$$A_r = 1.66 \text{ in}^2$$

$$F_f = 65 \text{ lb}_\text{f}$$

The flow is not critical to the solution of this problem. Since the cylinder area ratio is 2:1, $Q_2 = Q_1/2$, and $Q_2^2/Q_1^2 = 0.25$.

$$F_L = 1500(1.66)(0.25) + 65 = 687.5 \text{ lb}_\text{f}$$

Any load greater than 687.5 lb_f will cause a negative pressure at the cap end of the cylinder.

If the overrunning load is 687.5 lb_f ($P_c = 0$), the pressure at the rod end given by Eq. (12.6) is

$$P_r = (F_L - F_f)/A_r = (687.5 - 65)/1.66 = 375 \text{ psi}$$

The pressure drop across the Port P to Port A orifice is

$$\Delta P_1 = P_s - P_c = 1500 - 0 = 1500 \text{ psi}$$

The pressure drop across the Port B to T orifice is

$$\Delta P_2 = P_r - 0 = 375 \text{ psi}$$

For any overrunning load greater than 687.5 lb_f, the valve will not create enough pressure drop across the Port B to Port T orifice to maintain control of the load. Suppose the load is $F_L = 1000$ lb_f. Using Eq. (12.12), $P_c = -88$ psi. To create $P_c = -88$ psi, the pressure drop across the Port P to Port A orifice must be

$$\Delta P_1 = P_s - P_c = 1500 - (-88) = 1588 \text{ psi}$$

This is impossible.

Using Eq. (12.6), the needed pressure at the rod end is

$$P_r = (P_c A_c + F_L - F_f)/A_r = [-88(3.14) + 1000 - 65]/1.66 = 397 \text{ psi}$$

This pressure cannot be developed by the size orifice in a 1:1 area ratio valve. A smaller orifice is needed.

We will now rework the problem using a valve with a 2:1 area ratio.

$$A_1 = 2A_2$$

Using the orifice equation

$$Q_1 = CA_1\sqrt{\Delta P_1}$$

$$Q_2 = CA_2\sqrt{\Delta P_2}$$

$$= C(A_1/2)\sqrt{\Delta P_2}$$

Solving for CA_1 and equating the two expressions,

$$Q_1/\sqrt{\Delta P_1} = 2Q_2/\sqrt{\Delta P_2}$$

or

$$\Delta P_2/\Delta P_1 = (2Q_2)^2/Q_1^2 \tag{12.14}$$

As before,

$$\Delta P_1 = P_s - P_c$$

$$\Delta P_2 = P_r - 0 = P_r$$

Substituting in Eq. (12.14),

$$P_r = (P_s - P_c)(2Q_2)^2/Q_1^2 \tag{12.15}$$

Equating Eq. (12.6) and Eq. (12.15), and solving for P_c,

$$P_c = \frac{P_s[(2Q_2)^2/Q_1^2] - (F_L - F_f)/A_r}{(A_c/A_r) + [(2Q_2)^2/Q_1^2]} \tag{12.16}$$

Setting $P_c = 0$, we find the maximum overrunning load that can be controlled is

$$F_L = P_s A_r \frac{(2Q_2)^2}{Q_1^2} + F_f \tag{12.17}$$

$Q_2 = Q_1/2$ for a 2:1 area ratio cylinder, therefore

$$(2Q_2)^2/Q_1^2 = 1.0$$

$$F_L = 1500(1.66)(1.0) + 65 = 2555 \text{ lb}_6$$

A 2:1 area ratio valve can control an overrunning load 3.7 times the size load controlled with a 1:1 area ratio valve.

Using Eq. (12.17), $F_L = 1000$ lb$_f$ gives a cap end pressure of $P_c = 324$ psi. Rod end pressure is given by Eq. (12.6).

$$P_r = (P_c A_c + F_L - F_f)/A_r = [324(3.14) + 1000 - 65]/1.66 = 1176 \text{ psi} \quad (12.18)$$

The pressure drops across the valve are

$$\Delta P_1 = P_s - P_c = 1500 - 324 = 1176 \text{ psi}$$

$$\Delta P_2 = P_r - 0 = 1176 \text{ psi}$$

Total pressure drop across the valve is

$$\Delta P_{totv} = 1176 + 1176 = 2352 \text{ psi}$$

What is the consequence of a total ΔP this large across a proportional valve?

Data for a valve rated at 22.5 GPM nominal flow at 150 psi pressure drop across the valve is given in Fig. 12.17. Curve no. 1 was obtained by opening the valve with some input current in the range 25 to 100% rated current. Flow was increased until the measured pressure drop ($\Delta P_{tot} = \Delta P_1 + \Delta P_2$) equaled 150 psi. The flow to obtain this pressure drop was recorded and plotted vs. current. Curve no. 2 was obtained for a $\Delta P_{tot} = 300$ psi, curve no. 3 for 450 psi, curve no. 4 for 725 psi, and curve no. 5 for 1450 psi.

Suppose there is a requirement to control flow at 30 GPM with this valve. Following the 30 GPM line rightward until it intersects with curve no. 5 and reading down, we determine that 70% current will open the valve for 30 GPM flow. The resulting pressure drop is $\Delta P_{tot} = 1450$ psi. Continuing over to curve no. 4, a 76% current will limit flow to 30 GPM and result in a $\Delta P_{tot} = 725$ psi. Curve no. 3 shows that flow can be limited to 30 GPM with 87% current and the resulting $\Delta P_{tot} = 450$ psi. At 100% current (valve fully open) the flow is 30 GPM with $\Delta P_{tot} = 300$ psi (curve no. 2).

No curve gives a 30 GPM flow with $\Delta P_{tot} = 2352$ psi, the requirement to control the overrunning load. A higher ΔP curve is needed. Because the opening is so small, only a small part of the full spool stroke is available to control flow. It is recommended that an application use as much of the total stroke as possible. In this case, counterbalancing of the overrunning load should be part of the overall system design.

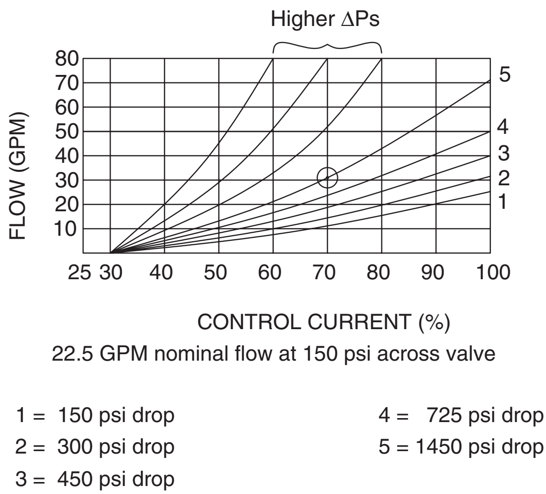


FIGURE 12.17
Family of curves for proportional DCV with 22.5 GPM nominal rating at 150 psi pressure drop across valve.

12.3.2 Resistive Load

A simple circuit with a resistive load is shown in Fig. 12.18. A force balance on the cylinder gives

$$P_c A_c = P_r A_r + F_f + F_L$$

Solving for P_c,

$$P_c = (P_r A_r + F_L + F_f)/A_c \tag{12.19}$$

Note that this equation is the same as Eq. (12.6) except for the change in sign for F_L.

Since the valve has a 1:1 area ratio, Eq. (12.10) applies.

$$\Delta P_2 = \Delta P_1 Q_2^2/Q_1^2$$

or

$$\Delta P_1 = \Delta P_2 Q_1^2/Q_2^2 \tag{12.20}$$

As previously defined,

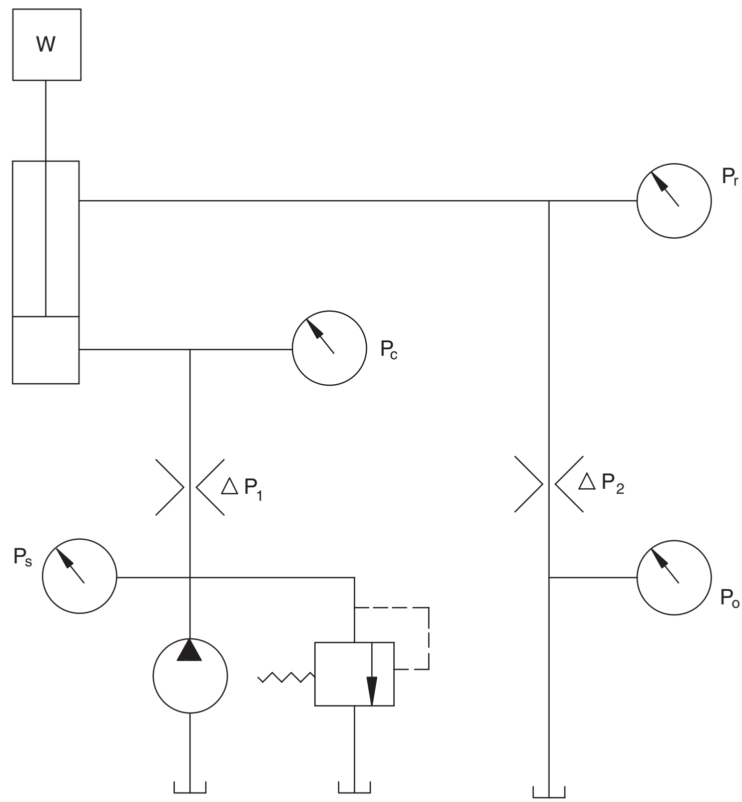


FIGURE 12.18
Cylinder with 2:1 area ratio and valve with 1:1 area ratio controlling a resistive load.

$$\Delta P_1 = P_s - P_c$$

$$\Delta P_2 = P_r$$

Substituting into Eq. (12.20), and solving for P_c,

$$P_c = P_s - P_r Q_1^2 / Q_2^2 \tag{12.21}$$

Equating Eqs. (12.19) and (12.21) and solving for P_r,

$$P_r = \frac{P_s - (F_L + F_f)/A_c}{\dfrac{A_r}{A_c} + \dfrac{Q_1^2}{Q_2^2}} \tag{12.22}$$

If $F_L = 1000$ lb_f, and the other parameters are the same as for the previous example,

$$P_r = \frac{1500 - (1000 + 65)/3.14}{\dfrac{1.66}{3.14} + 4.0} = 256 \text{ psi}$$

Substituting back into Eq. (12.19),

$$P_c = [256(1.66) + 1000 + 65]/3.14 = 475 \text{ psi}$$

The pressure drops across the valve are

$$\Delta P_1 = P_s - P_c = 1500 - 475 = 1025$$

$$\Delta P_2 = P_r = 256$$

$$\Delta P_{totv} = \Delta P_1 + \Delta P_2 = 1025 + 256 = 1281 \text{ psi}$$

If the valve has a 2:1 area ratio,

$$\Delta P_1 = \Delta P_2 Q_1^2/(2Q_2)^2$$

and the expression for becomes

$$P_r = \frac{P_s - (F_L + F_f)/A_c}{\dfrac{A_r}{A_c} + \dfrac{Q_1^2}{(2Q_2)^2}} \tag{12.23}$$

Reworking the problem for the same parameters,

$$P_r = \frac{1500 - (1000 + 65)/3.14}{\dfrac{1.66}{3.14} + 1.0} = 759 \text{ psi}$$

Substituting back into Eq. (12.19),

$$P_c = [759(1.66) + 1000 + 65]/3.14 = 740 \text{ psi}$$

The pressure drops across the valve are

$$\Delta P_1 = P_s - P_c = 1500 - 740 = 760 \text{ psi}$$

$$\Delta P_2 = P_r = 759 \text{ psi}$$

$$\Delta P_{totv} = \Delta P_1 + \Delta P_2 = 760 + 759 = 1519 \text{ psi}$$

To select a valve for this application, the designer will look in the manufacturer's literature and choose a valve with 2:1 area ratio spool having an operating curve for 1500 psi pressure drop. The best control is achieved if full spool stroke, or almost full spool stroke, is used to obtain the desired flow at the desired pressure drop.

For large pressure drops, like the 1500 psi here, you may have to use less than the full spool stroke. Data for a Rexroth valve with 2:1 spool area ratio rated for 26.4 GPM flow with 145 psi pressure drop is given in Fig. 12.19. A curve corresponding to 1520 psi total pressure drop was added to the published "family" of curves. As shown, a 60% control current will give 30 GPM flow at 1520 psi total pressure drop.

12.3.3 How a Proportional Directional Control Valve Functions in a Circuit

The reader may be wondering how a valve that develops a 1200 or 1500 psi total pressure drop can be used in a circuit. Pressure drops this large represent a sizeable loss of efficiency. As discussed in Chapter 11, use of a servo valve, or proportional valve, means that efficiency is traded for control accuracy. We give up efficiency to gain accuracy in position, velocity, or acceleration control.

To explain how a valve works in a circuit, let's assume that the valve used to move the resistive load in the previous example is in a circuit where the pump is developing 30 GPM, and the system relief valve is set to crack open

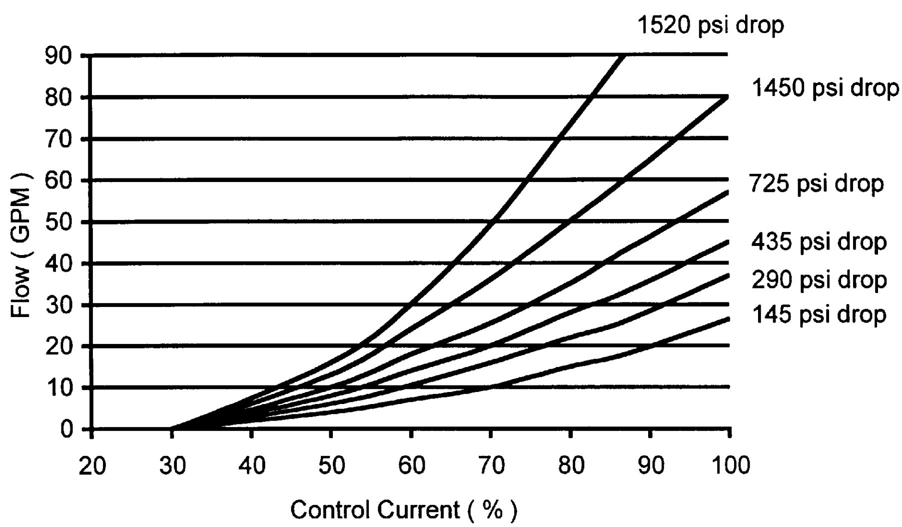


FIGURE 12.19
Flow at different control current settings for different total pressure drops across a proportional directional control valve.

at 1500 psi. The relief valve is direct acting and is fully open at 2000 psi. An idealized curve is shown in Fig. 12.20.

Previously, pressure drops were calculated for a circuit in which a cylinder extends against a resistive load. A valve with 1:1 area ratio was used. These pressure drops are repeated below.

$$\Delta P_1 = P_s - P_c = 1500 - 475 = 1025$$

$$\Delta P_{cyl} = P_c - P_r = 475 - 256 = 219$$

$$\Delta P_2 = P_r = 256$$

Total pressure "seen" at the relief valve is

$$\Delta P_{tot} = \Delta P_1 + \Delta P_{cyl} + \Delta P_2 = 1025 + 219 + 256 = 1500 \text{ psi}$$

It is necessary to change the flow to the cylinder from 30 GPM to 25 GPM. To achieve this change, 5 GPM must be dumped across the relief valve. A pressure of 1550 psi at the relief valve is needed (Fig. 12.20) to cause it to open enough to pass 5 GPM back to the reservoir. The cylinder and load are the same; therefore, ΔP_{cyl} is the same. The additional pressure must be generated with the proportional valve. Substituting $P_s = 1550$ psi into Eq. (12.22),

$$P_r = \frac{1550 - (1000 + 65)/3.14}{\dfrac{1.66}{3.14} + 4.0} = 267 \text{ psi}$$

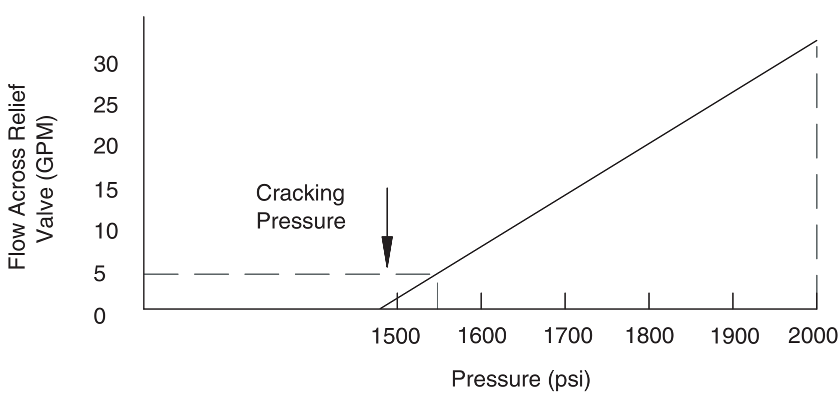


FIGURE 12.20
Idealized curve for a direct-acting relief valve used in a circuit with a proportional directional valve.

Substituting back into Eq. (12.19),

$$P_c = [267(1.66) + 1000 + 65]/3.14 = 480 \text{ psi}$$

The pressure drops around the circuit are now

$$\Delta P_1 = P_s - P_c = 1550 - 480 = 1070$$

$$\Delta P_{cyl} = P_c - P_r = 480 - 267 = 213$$

$$\Delta P_2 = P_r = 267$$

$$\Delta P_{tot} = \Delta P_1 + \Delta P_2 = 1070 + 213 + 267 = 1550 \text{ psi}$$

Total pressure drop across the valve is

$$\Delta P_{totv} = \Delta P_1 + \Delta P_2 = 1070 + 267 = 1337 \text{ psi}$$

A smaller control current must be chosen to allow the valve to close a small increment. The resulting increase in ΔP_{totv} from 1281 to 1337 psi will dump 5 GPM across the relief valve and achieve the desired 25 GPM to the cylinder.

We will now consider the same circuit with a 2:1 area ratio valve. Total pressure drops around the circuit, calculated for P_s = 1500 psi and a flow of 30 GPM, are

$$\Delta P_1 = P_s - P_c = 1500 - 740 = 760$$

$$\Delta P_{cyl} = P_c - P_r = 740 - 759 = -19$$

$$\Delta P_2 = P_r = 759$$

$$\Delta P_{tot} = \Delta P_1 + \Delta P_{cyl} + \Delta P_2 = 760 - 19 + 759 = 1500 \text{ psi}$$

How can a 1000 lb_f resistive load be lifted with a –19 psi pressure drop across the cylinder? The force balance equation is

$$P_c A_c = F_L + F_f + P_r A_r$$

$$740(3.14) = 1000 + 65 + 759(1.66)$$

$$2324 = 2325$$

Within round-off error, the force balance equation is satisfied. Because the cap end area is twice the rod end area, a pressure of only 740 psi on the cap end can lift the 1000 lb_f load, overcome the friction force, and overcome a 759 psi back pressure.

Now consider the case in which flow must be reduced to 25 GPM. The required pressure at the relief valve is 1550 psi, and, substituting into Eq. (12.23),

$$P_r = 1550 - \frac{(1000 + 65)/3.14}{\frac{1.66}{3.14} + 1.0} = 792 \text{ psi}$$

Substituting back into Eq. (12.19),

$$P_c = [792(1.66) + 1000 + 65]/3.14 = 758 \text{ psi}$$

Total pressure drops around the circuit are

$$\Delta P_1 = P_s - P_c = 1550 - 758 = 792$$

$$\Delta P_{cyl} = P_c - P_r = 758 - 792 = -34$$

$$\Delta P_2 = P_r = 792$$

$$\Delta P_{tot} = \Delta P_1 + \Delta P_{cyl} + \Delta P_2 = 792 - 34 + 792 = 1500 \text{ psi}$$

Total pressure drop across the valve is

$$\Delta P_{totv} = \Delta P_1 + \Delta P_2 = 792 + 792 = 1584 \text{ psi}$$

A smaller control current must be chosen to close the valve a small increment and increase ΔP_{totv} from 1519 psi to 1584 psi.

The proportional valve works with the relief valve in the same way for both the 1:1 spool area ratio and the 2:1 ratio. Calculated increase in ΔP_{totv} for the 1:1 ratio was 56 psi, and the increase was 65 psi for the 2:1 ratio.

12.4 Comparison of Servo and Proportional Valves

The control current to a proportional valve can be programmed to position the spool and create the orifice area, and thus the pressure drops, needed to accelerate a load, hold a constant velocity, and then decelerate the load to a stop. The required settings are determined by operating the circuit to cycle the cylinder under load. Remember that circuit performance is affected by the pump, cylinder, and relief valve characteristics as well as the proportional direction control valve characteristics. The interaction of these characteristics determines the final control settings that are programmed into the proportional valve amplifier. Typically, there is a certain amount of "tweaking" of proportional valve and relief valve settings before the final control settings are defined.

Once a proportional valve circuit has been programmed, it will cycle the cylinder in the same way each time the program is initiated. For many manufacturing applications, the same part is manipulated the same way each time; thus, the load function is approximately the same for each cycle, and a proportional valve circuit is very satisfactory. Its initial cost is less than that of a servo valve circuit, and it costs less to operate, because the cleanliness requirements are not as stringent.

The proportional valve circuit has no feedback. When load increases beyond the level at which the programmed settings were defined, pressure goes up. Pump leakage increases with pressure, flow across the relief valve increases with pressure, and flow to the cylinder changes. The controller does not compensate for these changes. It positions the spool of the proportional valve in accordance with the programmed settings. A servo valve circuit must be used to maintain a precise output when the load changes outside a very narrow plus or minus band about the design load.

12.5 Summary

Proportional valves are programmable orifices like servo valves. A proportional valve circuit does not have feedback; therefore, it cannot compensate for external disturbances and still maintain the desired output like a servo valve circuit. Proportional valves are less expensive and are more tolerant of contaminant.

There are two types of proportional valves: force-controlled and stroke-controlled. The solenoid in the force-controlled valve moves in proportion to the magnitude of a milliamp input current. The spool is held in position by a spring, and this spring is compressed by the solenoid in order to shift the spool. A stroke-controlled valve has a built-in LVDT, which feeds back the position of the solenoid. The solenoid moves to a position that gives a feedback signal equal to the command signal. The feedback provides for very precise positioning of the valve spool and thus offsets the influence of changes in flow forces acting on the spool.

Force-controlled solenoids can replace the mechanical spring in several types of pressure control valves to create a valve that can be adjusted with an electrical signal rather than manual input. Two examples are relief valves and pressure reducing valves.

The spool in a proportional direction control valve can be machined with different-shaped notches in the spool lands. A spool with a 1:1 spool area ratio has equal area notches on both sides of the spool land; thus, the pressure drop between Ports P and A and Ports B and T are approximately equal when flows on both sides of the valve are equal.

During extension, flow from the rod end of a cylinder with 2:1 area ratio is one-half the flow into the cap end. Better controllability is achieved if a proportional valve with 2:1 area ratio spool is used with a 2:1 area ratio cylinder. A 2:1 area ratio spool is one having notches with twice the area on one side of the land as the notches on the other side.

Equations of the pressure drops on both sides of a proportional directional control valve (Ports P to A and B to T) were derived for an overrunning load on a cylinder and a resistive load (Appendix 12.1). Some pressure drop is required for the metering that is needed for good control. These pressure drops represent an unavoidable loss in efficiency. Efficiency is given up in exchange for the increased control provided by proportional directional control valves.

Unlike servo valve circuits, a proportional valve circuit does not have feedback of the required output. (A stroke-controlled solenoid has feedback of the spool position, not feedback of the load position.) Proportional valve circuits do not compensate for external disturbances; consequently, they are recommended for applications in which the load is approximately constant for each cycle. Many manufacturing applications, where the same part is manipulated in the same way during each cycle, have an approximately constant load cycle. A series of settings can be programmed into the proportional valve controller, and it will cycle the cylinder with good repeatability.

References

Tonyan, M.J, 1985. *Electronically Controlled Proportional Valves: Selection and Application.* Marcel Dekker, Inc. New York.

APPENDIX 12.1

Summary of Equations Derived for Analysis of Proportional Directional Control Valves

TABLE A12.1

Summary of Equations Derived for Analysis of Proportional DCV Circuits with 2:1 Area Ratio Cylinder subjected to An Overrunning Load

Application	Extension	Retraction
	2:1 Area Ratio Valve	
Pressure at P_2	$P_2 = \dfrac{P_s Q_2^2/Q_1^2 - (F_L - F_f)/A_r}{(A_c/A_r) + Q_2^2/Q_1^2}$	$P_2 = \dfrac{P_s Q_2^2/(2Q_1)^2 - (F_L - F_f)/A_c}{(A_r/A_c) + Q_2^2/(2Q_1)^2}$
Pressure at P_3	$P_3 = \dfrac{F_L - F_f + P_2 A_c}{A_r}$	$P_3 = \dfrac{F_L - F_f + P_2 A_r}{A_c}$
	1:1 Area Ratio Valve	
Pressure at P_2	$P_2 = \dfrac{P_s Q_2^2/Q_1^2 - (F_L - F_f)/A_r}{(A_c/A_r) + Q_2^2/Q_1^2}$	$P_2 = \dfrac{P_s Q_2^2/{Q_1}^2 - (F_L - F_f)/A_c}{(A_r/A_c) + Q_2^2/{Q_1}^2}$
Pressure at P_3	$P_3 = \dfrac{F_L - F_f + P_2 A_c}{A_r}$	$P_3 = \dfrac{F_L - F_f + P_2 A_r}{A_c}$

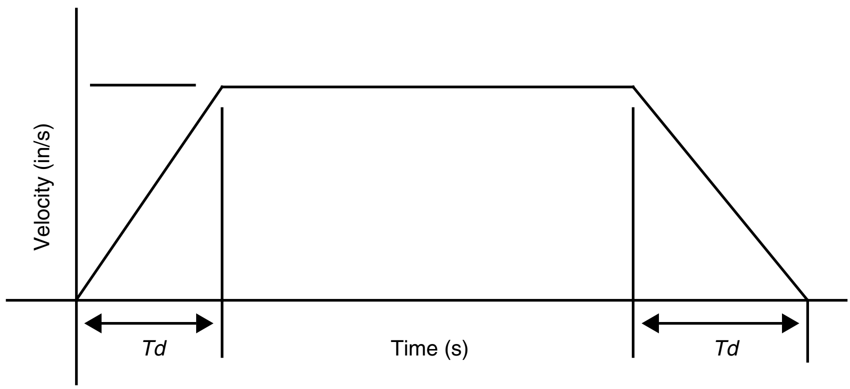
TABLE A12.2

Summary of Equations Derived for Analysis of Proportional DCV Circuits with 2:1 Area Ratio Cylinder Subjected to a Resistive Load

Application	Extension	Retraction
	2:1 Area Ratio Valve	
Pressure at P_2	$P_2 = \dfrac{F_L + F_f + P_3 A_r}{A_c}$	$P_2 = \dfrac{F_L + F_f + P_3 A_c}{A_r}$
Pressure at P_3	$P_3 = \dfrac{P_s - (F_L + F_f)/A_c}{Q_1^2/(2Q_2)^2 + A_r/A_c}$	$P_3 = \dfrac{P_s - (F_L + F_f)/A_r}{(2Q_1)^2/Q_2^2 + A_c/A_r}$
	1:1 Area Ratio Valve	
Pressure at P_2	$P_2 = \dfrac{F_L + F_f + P_3 A_r}{A_c}$	$P_2 = \dfrac{F_L + F_f + P_3 A_c}{A_r}$
Pressure at P_3	$P_3 = \dfrac{P_s - (F_L + F_f)/A_c}{Q_1^2/{Q_2}^2 + A_r/A_c}$	$P_3 = \dfrac{P_s - (F_L + F_f)/A_r}{{Q_1}^2/Q_2^2 + A_c/A_r}$

Problems

12.1 A proportional valve is used to extend and retract a cylinder. It is necessary to accelerate and decelerate as rapidly as possible to achieve a minimum cycle time. The desired velocity vs. time plot for the mass being cycled is given below:



Given:

Cylinder: 2-in. bore, 1-in. rod

Cap end area A_c = 3.14 in^2

Rod end area A_r = 2.36 in^2

Stroke distance: x = 26 in.

I.D. of line between valve and cylinder: d_1 = 0.50

Length of line to cap end: 26 in.

Length of line to rod end: 18.5 in.

Volume of fluid in cap end line:

$$V_3 = \frac{\pi}{4}d_1^2(26) = 5.1 \text{ in}^3$$

Volume of fluid in rod end line:

$$V_4 = \frac{\pi}{4}d_1^2(18.5) = 3.63 \text{ in}^3$$

Bulk modulus of oil:

$$\beta = 2 \times 10^5 \text{ lb}_f/\text{in}^2$$

Mass to be moved:

$$\text{Weight} = 1875 \text{ lb}_f$$

$$m = 1875 \text{ lb}_f/386 \text{ in/s}^2 = 4.85 \text{ lb}_\text{f} \cdot \text{s}^2/\text{in}$$

Maximum pressure available at rod end: 1000 psi

Coefficient of friction: $\mu = 0.58$

Required:

Find the design acceleration time, T_d. (Remember that design natural frequency is one-third calculated natural frequency, and that design acceleration time is $6T$, where T is the period of the design natural frequency).

Find the total time to extend and retract the cylinder. (Assume the retraction velocity plot is equal to the extension velocity plot.)

Index

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