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Naval Education and Training Command

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Training Manual (TRAMAN)

Fluid Power

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Naval Education And Training Program Management Support Activity

FLUID POWER

NAVEDTRA 12964

1990 Edition Prepared by
MWC Albert Beasley, Jr.

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PREFACE

This training manual (TRAMAN) and the associated nonresident training course (NRTC) are written for Navy and Naval Reserve personnel whose duties require them to have a knowledge of the fundamentals of hydraulics and pneumatics. The TRAMAN is based on occupational standards for advancement as stated in the *Navy Enlisted Manpower and Personnel Classifications and Occupational Standards*, NAVPERS 18068E; and numerous personnel qualifications standards (PQS) published by the Naval Education and Training Support Center, Pacific.

The completion of this TRAMAN and the associated NRTC, combined with the necessary practical experience, will assist personnel preparing for advancement in ratings requiring a knowledge of fluid power. Additionally, completion of this TRAMAN and the associated NRTC will assist personnel preparing for watch station qualification for PQSs requiring a knowledge of fluid power.

This self-study package was prepared by the Naval Education and Training Program Management Support Activity, Pensacola, Florida, for the Chief of Naval Education and Training. Technical assistance was provided by Naval Safety Center, Norfolk, Virginia; Naval Sea Systems Command, Washington, D.C.; Chief of Naval Operations (OP-03), Washington, D.C.; Commander, Naval Air Systems Command, Patuxent River, MD; Naval Supply Center, Pensacola, FL; and the subject matter experts for the AMH, AS, EN, and GSM ratings at NETPMSA, Pensacola, FL.

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THE UNITED STATES NAVY

GUARDIAN OF OUR COUNTRY

The United States Navy is responsible for maintaining control of the sea and is a ready force on watch at home and overseas, capable of strong action to preserve the peace or of instant offensive action to win in war.

It is upon the maintenance of this control that our country's glorious future depends; the United States Navy exists to make it so.

WE SERVE WITH HONOR

Tradition, valor, and victory are the Navy's heritage from the past. To these may be added dedication, discipline, and vigilance as the watchwords of the present and the future.

At home or on distant stations we serve with pride, confident in the respect of our country, our shipmates, and our families.

Our responsibilities sober us; our adversities strengthen us.

Service to God and Country is our special privilege. We serve with honor.

THE FUTURE OF THE NAVY

The Navy will always employ new weapons, new techniques, and greater power to protect and defend the United States on the sea, under the sea, and in the air.

Now and in the future, control of the sea gives the United States her greatest advantage for the maintenance of peace and for victory in war.

Mobility, surprise, dispersal, and offensive power are the keynotes of the new Navy. The roots of the Navy lie in a strong belief in the future, in continued dedication to our tasks, and in reflection on our heritage from the past.

Never have our opportunities and our responsibilities been greater.

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CHAPTER 1

INTRODUCTION TO FLUID POWER

Fluid power is a term which was created to include the generation, control, and application of smooth, effective power of pumped or compressed fluids (either liquids or gases) when this power is used to provide force and motion to mechanisms. This force and motion maybe in the form of pushing, pulling, rotating, regulating, or driving. Fluid power includes hydraulics, which involves liquids, and pneumatics, which involves gases. Liquids and gases are similar in many respects. The differences are pointed out in the appropriate areas of this manual.

This manual presents many of the fundamental concepts in the fields of hydraulics and pneumatics. It is intended as a basic reference for all personnel of the Navy whose duties and responsibilities require them to have a knowledge of the fundamentals of fluid power. Consequently, emphasis is placed primarily on the theory of operation of typical fluid power systems and components that have applications in naval equipment. Many applications of fluid power are presented in this manual to illustrate the functions and operation of different systems and components. However, these are only representative of the many applications of fluid power in naval equipment. Individual training manuals for each rate provide information concerning the application of fluid power to specific equipment for which the rating is responsible.

A brief summary of the contents of this training manual is given in the following paragraphs:

Chapter 2 covers the characteristics of liquids and the factors affecting them. It also explains the behavior of liquids at rest, identifies the characteristics of liquids in motion, and explains the operation of basic hydraulic components.

Chapter 3 discusses the qualities of fluids acceptable for hydraulic systems and the types of fluids used. Included are sections on safety precautions to follow when handling potentially hazardous fluids, liquid contamination, and control of contaminants.

Chapter 4 covers the hydraulic pump, the component in the hydraulic system which generates the force required for the system to perform its design function. The information provided covers classifications, types, operation, and construction of pumps.

Chapter 5 deals with the piping, tubing and flexible hoses, and connectors used to carry fluids under pressure.

Chapter 6 discusses the classification, types, and operation of valves used in the control of flow, pressure, and direction of fluids.

Chapter 7 covers the types and purposes of sealing devices used in fluid power systems, including the different materials used in their construction. Additionally, the guidelines for selecting, installing, and removing O-rings are included.

Chapter 8 discusses the operation of devices used to measure and regulate the pressure of fluids and to measure the temperature of fluids.

Chapter 9 describes the functions and types of reservoirs, strainers, filters, and accumulators, and their uses in fluid power systems.

Chapter 10 discusses the types and operation of actuators used to transform the energy generated by hydraulic systems into mechanical force and motion.

Chapter 11 deals with pneumatics. It discusses the origin of pneumatics, the characteristics and compressibility of gases, and the most commonly used gases in pneumatic systems. Also, sections are included to cover safety precautions and the potential hazards of compressed gases.

Chapter 12 identifies the types of diagrams encountered in fluid power systems. This chapter also discusses how components of chapters 4, 5, 6, 8, 9, and 10 are combined to form and operate together as a system.

A glossary of terms commonly used in fluid power is provided in appendix I. Appendix II provides symbols used in aeronautical mechanical

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systems, and appendix III provides symbols used in nonaeronautical mechanical systems.

The remainder of chapter 1 is devoted to the advantages and problems of fluid power applications. Included are brief sections on the history, development, and applications of hydraulics, and the states of matter.

ADVANTAGES OF FLUID POWER

The extensive use of hydraulics and pneumatics to transmit power is due to the fact that properly constructed fluid power systems possess a number of favorable characteristics. They eliminate the need for complicated systems of gears, cams, and levers. Motion can be transmitted without the slack inherent in the use of solid machine parts. The fluids used are not subject to breakage as are mechanical parts, and the mechanisms are not subjected to great wear.

The different parts of a fluid power system can be conveniently located at widely separated points, since the forces generated are rapidly transmitted over considerable distances with small loss. These forces can be conveyed up and down or around corners with small loss in efficiency and without complicated mechanisms. Very large forces can be controlled by much smaller ones and can be transmitted through comparatively small lines and orifices.

If the system is well adapted to the work it is required to perform, and if it is not misused, it can provide smooth, flexible, uniform action without vibration, and is unaffected by variation of load. In case of an overload, an automatic release of pressure can be guaranteed, so that the system is protected against breakdown or strain. Fluid power systems can provide widely variable motions in both rotary and straight-line transmission of power. The need for control by hand can be minimized. In addition, fluid power systems are economical to operate.

The question may arise as to why hydraulics is used in some applications and pneumatics in others. Many factors are considered by the user and/or the manufacturer when determining which type of system to use in a specific application. There are no hard and fast rules to follow; however, past experience has provided some sound ideas that are usually considered when such decisions are made. If the application requires speed, a medium amount of pressure, and only fairly accurate control, a pneumatic system may be used. If the application requires only a medium amount of pressure and a more accurate control, a combination of hydraulics and pneumatics may be used. If the application requires a great amount of pressure and/or extremely accurate control, a hydraulic system should be used.

SPECIAL PROBLEMS

The extreme flexibility of fluid power elements presents a number of problems. Since fluids have no shape of their own, they must be positively confined throughout the entire system. Special consideration must be given to the structural integrity of the parts of a fluid power system. Strong pipes and containers must be provided. Leaks must be prevented. This is a serious problem with the high pressure obtained in many fluid power installations.

The operation of the system involves constant movement of the fluid within the lines and components. This movement causes friction within the fluid itself and against the containing surfaces which, if excessive, can lead to serious losses in efficiency. Foreign matter must not be allowed to accumulate in the system, where it will clog small passages or score closely fitted parts. Chemical action may cause corrosion. Anyone working with fluid power systems must know how a fluid power system and its components operate, both in terms of the general principles common to all physical mechanisms and of the peculiarities of the particular arrangement at hand.

HYDRAULICS

The word *hydraulics* is based on the Greek word for water, and originally covered the study of the physical behavior of water at rest and in motion. Use has broadened its meaning to include the behavior of all liquids, although it is primarily concerned with the motion of liquids.

Hydraulics includes the manner in which liquids act in tanks and pipes, deals with their properties, and explores ways to take advantage of these properties.

DEVELOPMENT OF HYDRAULICS

Although the modern development of hydraulics is comparatively recent, the ancients were familiar with many hydraulic principles and their applications. The Egyptians and the ancient people of Persia, India, and China conveyed water

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along channels for irrigation and domestic purposes, using dams and sluice gates to control the flow. The ancient Cretans had an elaborate plumbing system. Archimedes studied the laws of floating and submerged bodies. The Romans constructed aqueducts to carry water to their cities.

After the breakup of the ancient world, there were few new developments for many centuries. Then, over a comparatively short period, beginning near the end of the seventeenth century, Italian physicist, Evangelista Torricelle, French physicist, Edme Mariotte, and later, Daniel Bernoulli conducted experiments to study the elements of force in the discharge of water through small openings in the sides of tanks and through short pipes. During the same period, Blaise Pascal, a French scientist, discovered the fundamental law for the science of hydraulics.

Pascal's law states that increase in pressure on the surface of a confined fluid is transmitted undiminished throughout the confining vessel or system (fig. 1-1). (This is the basic principle of hydraulics and is covered in detail in chapter 2 of this manual.)

For Pascal's law to be made effective for practical applications, it was necessary to have a piston that "fit exactly." It was not until the latter part of the eighteenth century that methods were found to make these snugly fitted parts required in hydraulic systems. This was accomplished by the invention of machines that were used to cut and shape the necessary closely fitted parts and, particularly, by the development of gaskets and packings. Since that time, components such as valves, pumps, actuating cylinders, and motors have been developed and refined to make hydraulics one of the leading methods of transmitting power.

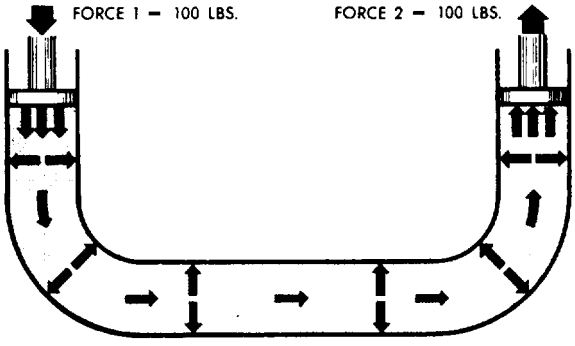

Figure 1-1.—Force transmitted through fluid.

Use of Hydraulics

The hydraulic press, invented by Englishman John Brahmah, was one of the first workable pieces of machinery developed that used hydraulics in its operation. It consisted of a plunger pump piped to a large cylinder and a ram. This press found wide use in England because it provided a more effective and economical means of applying large forces in industrial uses.

Today, hydraulic power is used to operate many different tools and mechanisms. In a garage, a mechanic raises the end of an automobile with a hydraulic jack. Dentists and barbers use hydraulic power, through a few strokes of a control lever, to lift and position their chairs to a convenient working height. Hydraulic doorstops keep heavy doors from slamming. Hydraulic brakes have been standard equipment on automobiles since the 1930s. Most automobiles are equipped with automatic transmissions that are hydraulically operated. Power steering is another application of hydraulic power. Construction workers depend upon hydraulic power for the operation of various components of their equipment. For example, the blade of a bulldozer is normally operated by hydraulic power.

During the period preceding World War II, the Navy began to apply hydraulics to naval mechanisms extensively. Since then, naval applications have increased to the point where many ingenious hydraulic devices are used in the solution of problems of gunnery, aeronautics, and navigation. Aboard ship, hydraulic power is used to operate such equipment as anchor windlasses, cranes, steering gear, remote control devices, and power drives for elevating and training guns and rocket launchers. Elevators on aircraft carriers use hydraulic power to transfer aircraft from the hangar deck to the flight deck and vice versa.

Hydraulics and pneumatics (chapter 11) are combined for some applications. This combination is referred to as *hydropneumatics*. An example of this combination is the lift used in garages and service stations. Air pressure is applied to the surface of hydraulic fluid in a reservoir. The air pressure forces the hydraulic fluid to raise the lift.

STATES OF MATTER

The material that makes up the universe is known as *matter*. Matter is defined as any substance that occupies space and has weight.

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Matter exists in three states: solid, liquid, and gas; each has distinguishing characteristics. Solids have a definite volume and a definite shape; liquids have a definite volume, but take the shape of their containing vessels; gases have neither a definite shape nor a definite volume. Gases not only take the shape of the containing vessel, but also expand and fill the vessel, regardless of its volume. Examples of the states of matter are iron, water, and air.

Matter can change from one state to another. Water is a good example. At high temperatures it is in the gaseous state known as steam. At moderate temperatures it is a liquid, and at low temperatures it becomes ice, which is definitely a solid state. In this example, the temperature is the dominant factor in determining the state the substance assumes.

Pressure is another important factor that will affect changes in the state of matter. At pressures lower than atmospheric pressure, water will boil and thus change into steam at temperatures lower than 212° Fahrenheit (F). Pressure is also a critical factor in changing some gases to liquids or solids. Normally, when pressure and chilling are both applied to a gas, the gas assumes a liquid state. Liquid air, which is a mixture of oxygen and nitrogen, is produced in this manner.

In the study of fluid power, we are concerned primarily with the properties and characteristics of liquids and gases. However, you should keep in mind that the properties of solids also affect the characteristics of liquids and gases. The lines and components, which are solids, enclose and control the liquid or gas in their respective systems.

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CHAPTER 2

FORCES IN LIQUIDS

The study of liquids is divided into two main parts: liquids at rest (hydrostatics) and liquids in motion (hydraulics).

The effects of liquids at rest can often be expressed by simple formulas. The effects of liquids in motion are more difficult to express due to frictional and other factors whose actions cannot be expressed by simple mathematics.

In chapter 1 we learned that liquids have a definite volume but take the shape of their containing vessel. There are two additional characteristics we must explore prior to proceeding.

Liquids are almost incompressible. For example, if a pressure of 100 pounds per square inch (psi) is applied to a given volume of water that is at atmospheric pressure, the volume will decrease by only 0.03 percent. It would take a force of approximately 32 tons to reduce its volume by 10 percent; however, when this force is removed, the water immediately returns to its original volume. Other liquids behave in about the same manner as water.

Another characteristic of a liquid is the tendency to keep its free surface level. If the surface is not level, liquids will flow in the direction which will tend to *make* the surface level.

LIQUIDS AT REST

In studying fluids at rest, we are concerned with the transmission of force and the factors which affect the forces in liquids. Additionally, pressure in and on liquids and factors affecting pressure are of great importance.

PRESSURE AND FORCE

The terms *force* and *pressure* are used extensively in the study of fluid power. It is essential that we distinguish between the terms. Force means a total push or pull. It is the push or pull exerted against the total area of a particular surface and is expressed in pounds or grams. Pressure means the amount of push or pull (force) applied to each unit area of the surface and is expressed in pounds per square inch (lb/in^2) or grams per square centimeter (gm/cm^2). Pressure maybe exerted in one direction, in several directions, or in all directions.

Computing Force, Pressure, and Area

A formula is used in computing force, pressure, and area in fluid power systems. In this formula, P refers to pressure, F indicates force, and A represents area.

Force equals pressure times area. Thus, the formula is written

$$F = P \times A \qquad \text{Equation 2-1.}$$

Pressure equals force divided by area. By rearranging the formula, this statement may be condensed into

$$P = \frac{F}{A} \qquad \text{Equation 2-2.}$$

Since area equals force divided by pressure, the formula is written

$$A = \frac{F}{P} \qquad \text{Equation 2-3.}$$

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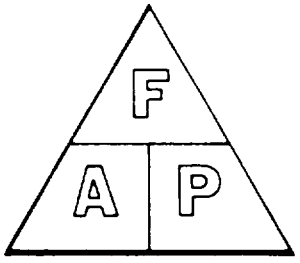

Figure 2-1.—Device for determining the arrangement of the force, pressure, and area formula.

Figure 2-1 illustrates a memory device for recalling the different variations of this formula. Any letter in the triangle may be expressed as the product or quotient of the other two, depending on its position within the triangle.

For example, to find area, consider the letter A as being set off to itself, followed by an equal sign. Now look at the other two letters. The letter F is above the letter P; therefore,

$$A = \frac{F}{P}$$

NOTE: Sometimes the area may not be expressed in square units. If the surface is rectangular, you can determine its area by multiplying its length (say, in inches) by its width (also in inches). The majority of areas you will consider in these calculations are circular in shape. Either the radius or the diameter may be given, but you must know the radius in inches to find the area. The radius is one-half the diameter. To determine the area, use the formula for finding the area of a circle. This is written $A = \pi r^2$, where A is the area, π is is 3.1416 (3.14 or 3 1/7 for most calculations), and r^2 indicates the radius squared.

Atmospheric Pressure

The atmosphere is the entire mass of air that surrounds the earth. While it extends upward for about 500 miles, the section of primary interest is the portion that rests on the earth's surface and extends upward for about 7 1/2 miles. This layer is called the troposphere.

If a column of air 1-inch square extending all the way to the "top" of the atmosphere could be weighed, this column of air would weigh approximately 14.7 pounds at sea level. Thus, atmospheric pressure at sea level is approximately 14.7 psi.

As one ascends, the atmospheric pressure decreases by approximately 1.0 psi for every 2,343 feet. However, below sea level, in excavations and depressions, atmospheric pressure increases. Pressures under water differ from those under air only because the weight of the water must be added to the pressure of the air.

Atmospheric pressure can be measured by any of several methods. The common laboratory method uses the mercury column barometer. The height of the mercury column serves as an indicator of atmospheric pressure. At sea level and at a temperature of 0° Celsius (C), the height of the mercury column is approximately 30 inches, or 76 centimeters. This represents a pressure of approximately 14.7 psi. The 30-inch column is used as a reference standard.

Another device used to measure atmospheric pressure is the aneroid barometer. The aneroid barometer uses the change in shape of an evacuated metal cell to measure variations in atmospheric pressure (fig. 2-2). The thin metal of the aneroid cell moves in or out with the variation of pressure on its external surface. This movement is transmitted through a system of levers to a pointer, which indicates the pressure.

The atmospheric pressure does not vary uniformly with altitude. It changes more rapidly at lower altitudes because of the compressibility of the air, which causes the air layers close to the earth's surface to be compressed by the air masses above them. This effect, however, is partially counteracted by the contraction of the upper

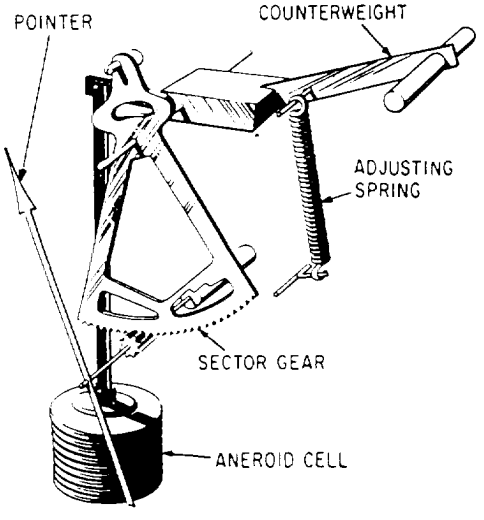

Figure 2-2.—Simple diagram of the aneroid barometer.

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layers due to cooling. The cooling tends to increase the density of the air.

Atmospheric pressures are quite large, but in most instances practically the same pressure is present on all sides of objects so that no single surface is subjected to a great load.

Atmospheric pressure acting on the surface of a liquid (fig. 2-3, view A) is transmitted equally throughout the liquid to the walls of the container, but is balanced by the same atmospheric pressure acting on the outer walls of the container. In view B of figure 2-3, atmospheric pressure acting on the surface of one piston is balanced by the same pressure acting on the surface of the other piston. The different areas of the two surfaces make no difference, since for a unit of area, pressures are balanced.

TRANSMISSION OF FORCES THROUGH LIQUIDS

When the end of a solid bar is struck, the main force of the blow is carried straight through the bar to the other end (fig. 2-4, view A). This happens because the bar is rigid. The direction of the blow almost entirely determines the direction of the transmitted force. The more rigid the bar, the less force is lost inside the bar or transmitted outward at right angles to the direction of the blow.

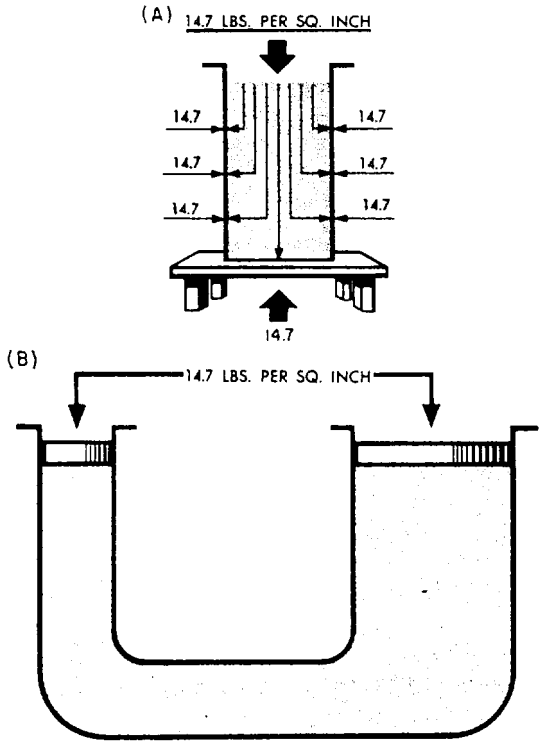

Figure 2-3.—Effects of atmospheric pressure.

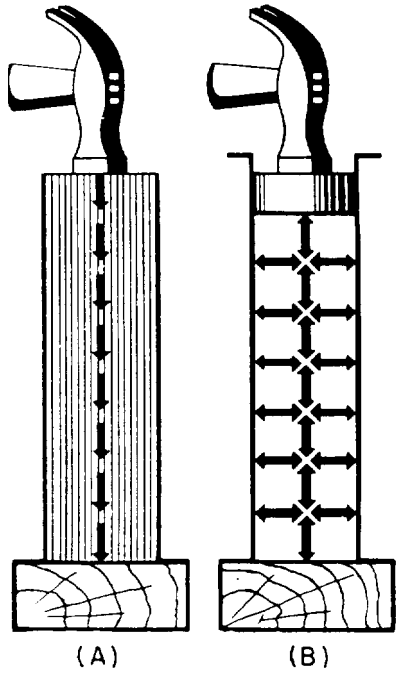

Figure 2-4.—Transmission of force: (A) solid; (B) fluid.

When a force is applied to the end of a column of confined liquid (fig. 2-4, view B), it is transmitted straight through to the other end and also equally and undiminished in every direction throughout the column—forward, backward, and sideways—so that the containing vessel is literally filled with pressure.

An example of this distribution of force is illustrated in figure 2-5. The flat hose takes on

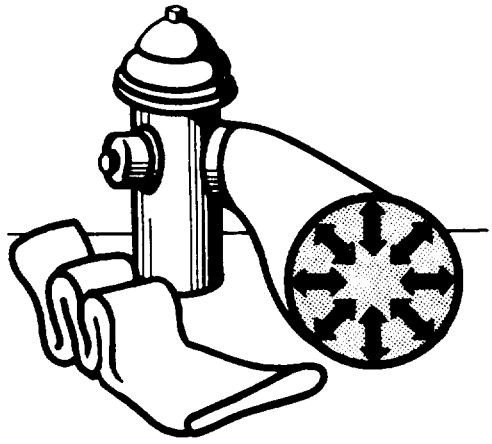

Figure 2-5.—Distribution of force.

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a circular cross section when it is filled with water under pressure. The outward push of the water is equal in every direction.

So far we have explained the effects of atmospheric pressure on liquids and how external forces are distributed through liquids. Let us now focus our attention on forces generated by the weight of liquids themselves. To do this, we must first discuss density, specific gravity, and Pascal's law.

Density and Specific Gravity

The density of a substance is its weight per unit volume. The unit volume in the English system of measurement is 1 cubic foot. In the metric system it is the cubic centimeter; therefore, density is expressed in pounds per cubic foot or in grams per cubic centimeter.

To find the density of a substance, you must know its weight and volume. You then divide its weight by its volume to find the weight per unit volume. In equation form, this is written as

$$D = \frac{W}{V} \qquad \text{Equation 2-4.}$$

EXAMPLE: The liquid that fills a certain container weighs 1,497.6 pounds. The container is 4 feet long, 3 feet wide, and 2 feet deep. Its volume is 24 cubic feet (4 ft x 3 ft x 2 ft). If 24 cubic feet of this liquid weighs 1,497.6 pounds, then 1 cubic foot weighs

$$\frac{1{,}497.6}{24}$$

or 62.4 pounds. Therefore, the density of the liquid is 62.4 pounds per cubic foot.

This is the density of water at 4°C and is usually used as the standard for comparing densities of other substances. The temperature of 4°C was selected because water has its maximum density at this temperature. In the metric system, the density of water is 1 gram per cubic centimeter. The standard temperature of 4°C is used whenever the density of liquids and solids is measured. Changes in temperature will not change the weight of a substance but will change the volume of the substance by expansion or contraction, thus changing the weight per unit volume.

In physics, the word *specific* implies a ratio. Weight is the measure of the earth's attraction for a body. The earth's attraction for a body is called gravity. Thus, the ratio of the weight of a unit volume of some substance to the weight of an equal volume of a standard substance, measured under standard pressure and temperature conditions, is called specific gravity. The terms *specific weight* and *specific density* are sometimes used to express this ratio.

The following formulas are used to find the specific gravity (sp gr) of solids and liquids, with water used as the standard substance.

$$\text{sp gr} = \frac{\text{Weight of the substance}}{\text{Weight of an equal volume of water}}$$

or,

$$\text{sp gr} = \frac{\text{Density of the substance}}{\text{Density of water}}$$

The same formulas are used to find the specific gravity of gases by substituting air, oxygen, or hydrogen for water.

$$\text{The specific gravity of water is 1, } \frac{62.4}{62.4}.$$

If a cubic foot of a certain liquid weighs 68.64 pounds, then its specific gravity is 1.1,

$$\frac{68.64}{62.4}.$$

Thus, the specific gravity of the liquid is the ratio of its density to the density of water. If the specific gravity of a liquid or solid is known, the density of the liquid or solid maybe obtained by multiplying its specific gravity by the density of water. For example, if a certain hydraulic liquid has a specific gravity of 0.8, 1 cubic foot of the liquid weighs 0.8 times as much as a cubic foot of water—0.8 times 62.4, or 49.92 pounds. In the metric system, 1 cubic centimeter of a substance with a specific gravity of 0.8 weighs 1 times 0.8, or 0.8 grams. (Note that in the metric system the specific gravity of a liquid or solid has the same numerical value as its density, because water weighs 1 gram per cubic centimeter.)

Specific gravity and density are independent of the size of the sample under consideration and depend only on the substance of which it is made.

A device called a hydrometer is used for measuring the specific gravity of liquids.

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Pascal's Law

Recall from chapter 1 that the foundation of modern hydraulics was established when Pascal discovered that pressure in a fluid acts equally in all directions. This pressure acts at right angles to the containing surfaces. If some type of pressure gauge, with an exposed face, is placed beneath the surface of a liquid (fig. 2-6) at a specific depth and pointed in different directions, the pressure will read the same. Thus, we can say that pressure in a liquid is independent of direction.

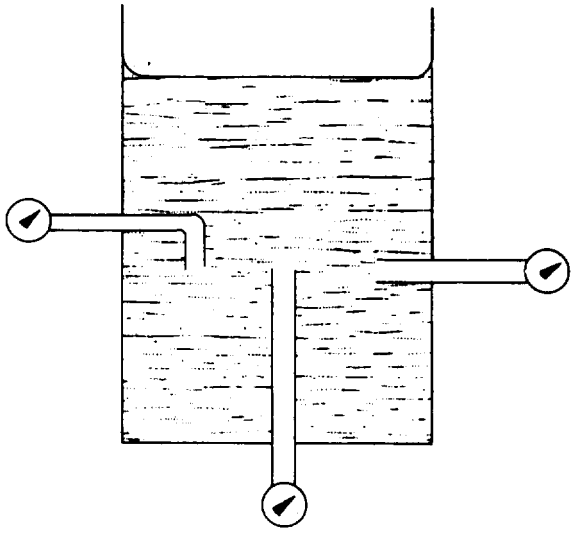

Figure 2-6.—Pressure of a liquid is independent of direction.

Pressure due to the weight of a liquid, at any level, depends on the depth of the fluid from the surface. If the exposed face of the pressure gauges, figure 2-6, are moved closer to the surface of the liquid, the indicated pressure will be less. When the depth is doubled, the indicated pressure is doubled. Thus the pressure in a liquid is directly proportional to the depth.

Consider a container with vertical sides (fig. 2-7) that is 1 foot long and 1 foot wide. Let it be filled with water 1 foot deep, providing 1 cubic foot of water. We learned earlier in this chapter that 1 cubic foot of water weighs 62.4 pounds. Using this information and equation 2-2, P = F/A, we can calculate the pressure on the bottom of the container.

$$P = \frac{F}{A}$$

$$= \frac{62.4 \text{ lb}}{1 \text{ ft}^2}$$

$$= 62.4 \text{ lb/ft}^2.$$

Since there are 144 square inches in 1 square foot,

$$P = \frac{62.4}{144} = 0.433 \text{ lb/in}^2.$$

This can be stated as follows: the weight of a column of water 1 foot high, having a cross-sectional area of 1 square inch, is 0.433 pound.

If the depth of the column is tripled, the weight of the column will be 3 x 0.433, or 1.299 pounds, and the pressure at the bottom will be 1.299 lb/in^2(psi), since pressure equals the force divided by the area. Thus, the pressure at any depth in a liquid is equal to the weight of the column of liquid at that depth divided by the cross-sectional area of the column at that depth. The volume of a liquid that produces the pressure is referred to as the fluid head of the liquid. The pressure of a liquid due to its fluid head is also dependent on the density of the liquid.

If we let A equal any cross-sectional area of a liquid column and h equal the depth of the column, the volume becomes Ah. Using equation 2-4, D = W/V, the weight of the liquid above area A is equal to AhD.

$$D = W/V,\ D = \frac{W}{Ah},\ W = AhD$$

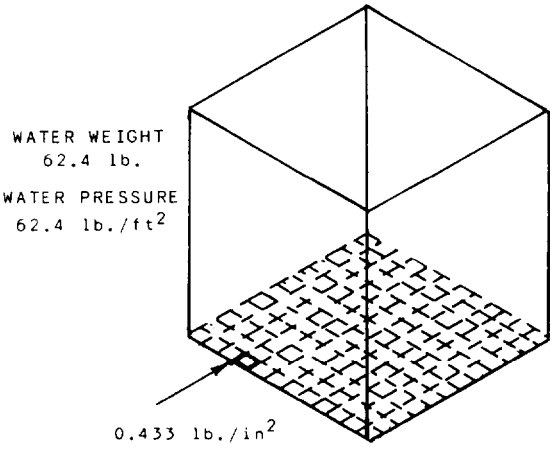

Figure 2-7.—Water pressure in a 1-cubic-foot container.

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Since pressure is equal to the force per unit area, set A equal to 1. Then the formula pressure becomes

$$P = h\,D \qquad \text{Equation 2-5.}$$

It is essential that h and D be expressed in similar units. That is, if D is expressed in pounds per cubic foot, the value of h must be expressed in feet. If the desired pressure is to be expressed in pounds per square inch, the pressure formula, equation 2-5, becomes

$$P = \frac{hD}{144} \qquad \text{Equation 2-6.}$$

Pascal was also the first to prove by experiment that the shape and volume of a container in no way alters pressure. Thus in figure 2-8, if the pressure due to the weight of the liquid at a point on horizontal line H is 8 psi, the pressure is 8 psi everywhere at level H in the system. Equation 2-5 also shows that the pressure is independent of the shape and volume of a container.

Pressure and Force in Fluid Power Systems

Recall that, according to Pascal's law, any force applied to a confined fluid is transmitted in all directions throughout the fluid regardless of the shape of the container. Consider the effect of this in the system shown in figure 2-9. If there is a resistance on the output piston and the input piston is pushed downward, a pressure is created through the fluid, which acts equally at right angles to surfaces in all parts of the container.

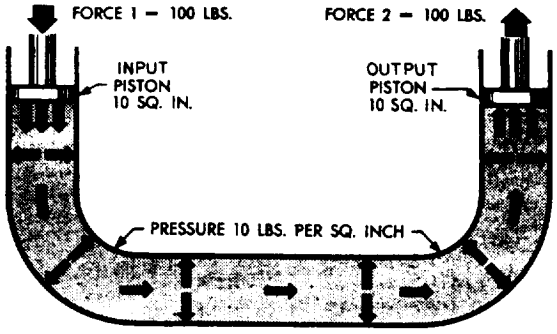

Figure 2-9.—Force transmitted through fluid.

If force 1 is 100 pounds and the area of the input piston is 10 square inches, then the pressure in the fluid is 10 psi

$$\frac{100 \text{ lb}}{10 \text{ sq. in}}$$

NOTE: Fluid pressure cannot be created without resistance to flow. In this case, resistance

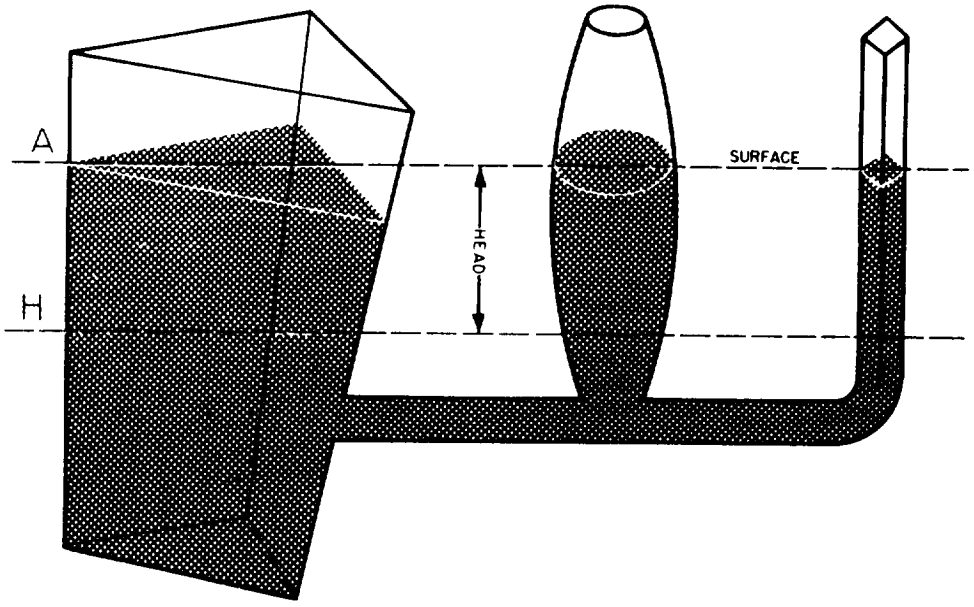

Figure 2-8.—Pressure relationship with shape.

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is provided by the equipment to which the output piston is attached. The force of resistance acts against the top of the output piston. The pressure created in the system by the input piston pushes on the underside of the output piston with a force of 10 pounds on each square inch.

In this case, the fluid column has a uniform cross section, so the area of the output piston is the same as the area of the input piston, or 10 square inches. Therefore, the upward force on the output piston is 100 pounds (10 psi x 10 sq. in.), the same as the force applied to the input piston. All that was accomplished in this system was to transmit the 100-pound force around the bend. However, this principle underlies practically all mechanical applications of fluid power.

At this point you should note that since Pascal's law is independent of the shape of the container, it is not necessary that the tube connecting the two pistons have the same cross-sectional area of the pistons. A connection of any size, shape, or length will do, as long as an unobstructed passage is provided. Therefore, the system shown in figure 2-10, with a relatively small, bent pipe connecting two cylinders, will act exactly the same as the system shown in figure 2-9.

MULTIPLICATION OF FORCES.— Consider the situation in figure 2-11, where the input piston is much smaller than the output piston. Assume that the area of the input piston is 2 square inches. With a resistant force on the output piston a downward force of 20 pounds acting on the input piston creates a pressure of $\frac{20}{2}$ or 10 psi

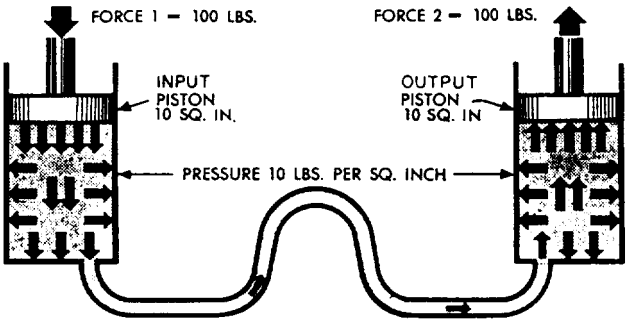

Figure 2-10.—Transmitting force through a small pipe.

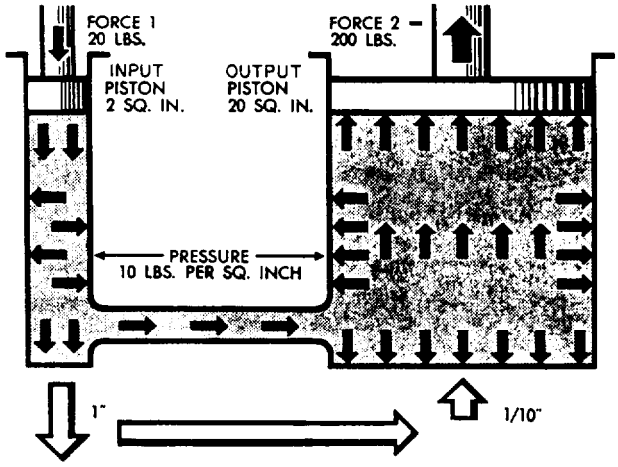

Figure 2-11.—Multiplication of forces.

in the fluid. Although this force is much smaller than the force applied in figures 2-9 and 2-10, the pressure is the same. This is because the force is applied to a smaller area.

This pressure of 10 psi acts on all parts of the fluid container, including the bottom of the output piston. The upward force on the output piston is 200 pounds (10 pounds of pressure on each square inch). In this case, the original force has been multiplied tenfold while using the same pressure in the fluid as before. In any system with these dimensions, the ratio of output force to input force is always ten to one, regardless of the applied force. For example, if the applied force of the input piston is 50 pounds, the pressure in the system will be 25 psi. This will support a resistant force of 500 pounds on the output piston.

The system works the same in reverse. If we change the applied force and place a 200-pound force on the output piston (fig. 2-11), making it the input piston, the output force on the input piston will be one-tenth the input force, or 20 pounds. (Sometimes such results are desired.) Therefore, if two pistons are used in a fluid power system, the force acting on each piston is directly proportional to its area, and the magnitude of each force is the product of the pressure and the area of each piston.

Note the white arrows at the bottom of figure 2-11 that indicate up and down movement. The movement they represent will be explained later in the discussion of volume and distance factors.

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DIFFERENTIAL AREAS.— Consider the special situation shown in figure 2-12. Here, a single piston (1) in a cylinder (2) has a piston rod (3) attached to one of its sides. The piston rod extends out of one end of the cylinder. Fluid under pressure is admitted equally to both ends of the cylinder. The opposed faces of the piston (1) behave like two pistons acting against each other. The area of one face is the full cross-sectional area of the cylinder, say 6 square inches, while the area of the other face is the area of the cylinder minus the area of the piston rod, which is 2 square inches. This leaves an effective area of 4 square inches on the right face of the piston. The pressure on both faces is the same, in this case, 20 psi. Applying the rule just stated, the force pushing the piston to the right is its area times the pressure, or 120 pounds (20 x 6). Likewise, the force pushing the piston to the left is its area times the pressure, or 80 pounds (20 x 4). Therefore, there is a net unbalanced force of 40 pounds acting to the right, and the piston will move in that direction. The net effect is the same as if the piston and the cylinder had the same cross-sectional area as the piston rod.

VOLUME AND DISTANCE FACTORS.— You have learned that if a force is applied to a system and the cross-sectional areas of the input and output pistons are equal, as in figures 2-9 and 2-10, the force on the input piston will support an equal resistant force on the output piston. The pressure of the liquid at this point is equal to the force applied to the input piston divided by the piston's area. Let us now look at what happens when a force greater than the resistance is applied to the input piston.

In the system illustrated in figure 2-9, assume that the resistance force on the output piston is 100 psi. If a force slightly greater than 100 pounds is applied to the input piston, the pressure in the system will be slightly greater than 10 psi. This increase in pressure will overcome the resistance force on the output piston. Assume that the input piston is forced downward 1 inch. The movement displaces 10 cubic inches of fluid. The fluid must go somewhere. Since the system is closed and the fluid is practically incompressible, the fluid will move to the right side of the system. Because the output piston also has a cross-sectional area of 10 square inches, it will move 1 inch upward to accommodate the 10 cubic inches of fluid. You may generalize this by saying that if two pistons in a closed system have equal cross-sectional areas and one piston is pushed and moved, the other piston will move the same distance, though in the opposite direction. This is because a decrease in volume in one part of the system is balanced by one equal increase in volume in another part of the system.

Apply this reasoning to the system in figure 2-11. If the input piston is pushed down a distance

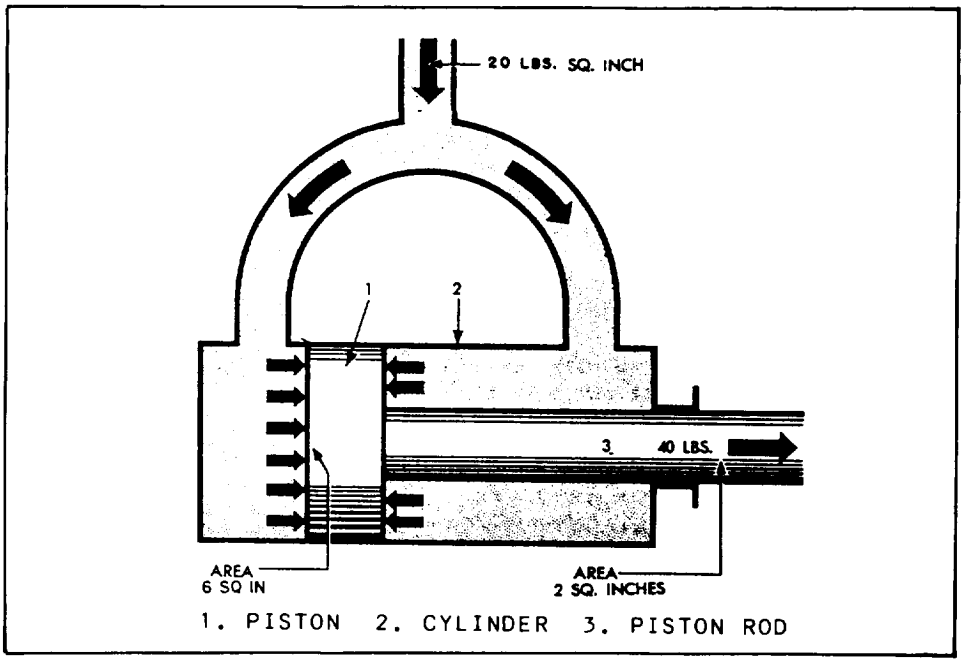

Figure 2-12.—Differential areas on a piston.

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of 1 inch, the volume of fluid in the left cylinder will decrease by 2 cubic inches. At the same time, the volume in the right cylinder will increase by 2 cubic inches. Since the diameter of the right cylinder cannot change, the piston must move upward to allow the volume to increase. The piston will move a distance equal to the volume increase divided by the surface area of the piston (equal to the surface area of the cylinder). In this example, the piston will move one-tenth of an inch (2 cu. in. ÷ 20 sq. in.). This leads to the second basic rule for a fluid power system that contains two pistons: The distances the pistons move are inversely proportional to the areas of the pistons. Or more simply, if one piston is smaller than the other, the smaller piston must move a greater distance than the larger piston any time the pistons move.

LIQUIDS IN MOTION

In the operation of fluid power systems, there must be a flow of fluid. The amount of flow will vary from system to system. To understand fluid power systems in action, it is necessary to understand some of the characteristics of liquids in motion.

Liquids in motion have characteristics different from liquids at rest. Frictional resistances within a fluid (viscosity) and inertia contribute to these differences. (Viscosity is discussed in chapter 3.) *Inertia,* which means the resistance a mass offers to being set in motion, will be discussed later in this section. There are other relationships of liquids in motion with which you must become familiar. Among these are volume and velocity of flow, flow rate and speed, laminar and turbulent flow, and more importantly, the force and energy changes which occur in flow.

VOLUME AND VELOCITY OF FLOW

The volume of a liquid passing a point in a given time is known as its *volume of flow* or flow rate. The volume of flow is usually expressed in gallons per minute (gpm) and is associated with relative pressures of the liquid, such as 5 gpm at 40 psi.

The *velocity of flow* or velocity of the fluid is defined as the average speed at which the fluid moves past a given point. It is usually expressed in feet per second (fps) or feet per minute (fpm). Velocity of flow is an important consideration in sizing the hydraulic lines. (Hydraulic lines are discussed in chapter 5.)

Volume and velocity of flow are often considered together. With other conditions unaltered—that is, with volume of input unchanged—the velocity of flow increases as the cross section or size of the pipe decreases, and the velocity of flow decreases as the cross section increases. For example, the velocity of flow is slow at wide parts of a stream and rapid at narrow parts, yet the volume of water passing each part of the stream is the same.

In figure 2-13, if the cross-sectional area of the pipe is 16 square inches at point A and 4 square inches at point B, we can calculate the relative velocity of flow using the flow equation

$$Q = v\,A \qquad \text{Equation 2-7.}$$

where Q is the volume of flow, v is the velocity of flow and A is the cross-sectional area of the liquid. Since the volume of flow at point A, Q_1, is equal to the volume of flow at point B, Q_2, we can use equation 2-7 to determine the ratio of the

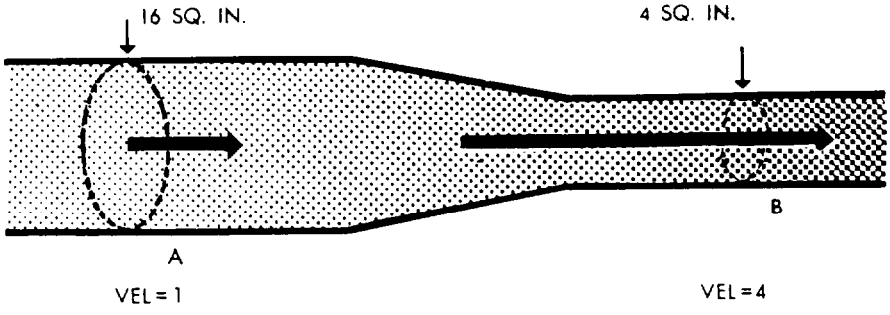

Figure 2-13.—Volume and velocity of flow.

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velocity of flow at point A, v_1, to the velocity of flow at point B, v_2.

Since $Q_1 = Q_2$, $A_1v_1 = A_2v_2$

From figure 2-13; $A_1 = 16$sq. in., $A_2 = 4$sq. in.

Substituting: $16v_1 = 4v_2$ or $v_2 = 4v_1$

Therefore, the velocity of flow at point B is four times the velocity of flow at point A.

VOLUME OF FLOW AND SPEED

If you consider the cylinder volume you must fill and the distance the piston must travel, you can relate the volume of flow to the speed of the piston. The volume of the cylinder is found by multiplying the piston area by the length the piston must travel (stroke).

Suppose you have determined that two cylinders have the same volume and that one cylinder is twice as long as the other. In this case, the cross-sectional area of the longer tube will be half of the cross-sectional area of the other tube. If fluid is pumped into each cylinder at the same rate, both pistons will reach their full travel at the same time. However, the piston in the smaller cylinder must travel twice as fast because it has twice as far to go.

There are two ways of controlling the speed of the piston, (1) by varying the size of the cylinder and (2) by varying the volume of flow (gpm) to the cylinders. (Hydraulic cylinders are discussed in detail in chapter 10.)

STREAMLINE AND TURBULENT FLOW

At low velocities or in tubes of small diameter, flow is streamlined. This means that a given particle of fluid moves straight forward without bumping into other particles and without crossing their paths. Streamline flow is often referred to as laminar flow, which is defined as a flow situation in which fluid moves in parallel lamina or layers. As an example of streamline flow, consider figure 2-14, which illustrates an open stream flowing at a slow, uniform rate with logs floating on its surface. The logs represent particles of fluid. As long as the stream flows at a slow, uniform rate, each log floats downstream in its own path, without crossing or bumping into the other.

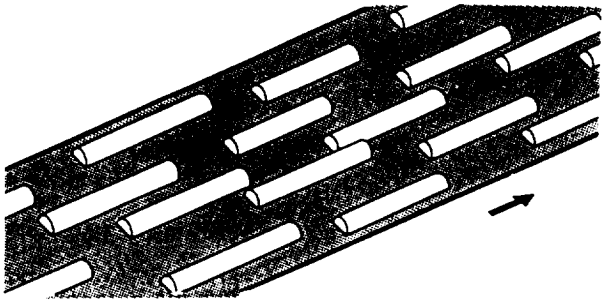

Figure 2-14.—Streamline flow.

If the stream narrows, however, and the volume of flow remains the same, the velocity of flow increases. If the velocity increases sufficiently, the water becomes turbulent. (See fig. 2-15.) Swirls, eddies, and cross-motions are set up in the water. As this happens, the logs are thrown against each other and against the banks of the stream, and the paths followed by different logs will cross and recross.

Particles of fluid flowing in pipes act in the same manner. The flow is streamlined if the fluid flows slowly enough, and remains streamlined at greater velocities if the diameter of the pipe is small. If the velocity of flow or size of pipe is increased sufficiently, the flow becomes turbulent.

While a high velocity of flow will produce turbulence in any pipe, other factors contribute to turbulence. Among these are the roughness of the inside of the pipe, obstructions, the degree of curvature of bends, and the number of bends in the pipe. In setting up or maintaining fluid power systems, care should be taken to eliminate or

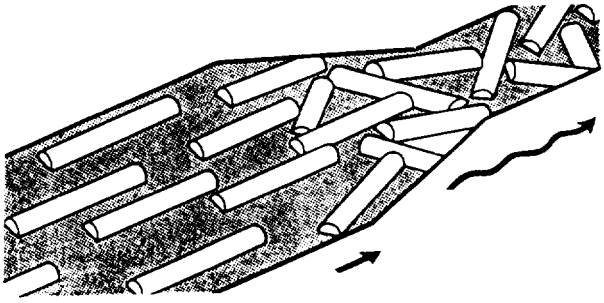

Figure 2-15.—Turbulent flow.

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minimize as many causes of turbulence as possible, since the energy consumed by turbulence is wasted. Limitations related to the degree and number of bends of pipe are discussed in chapter 5.

While designers of fluid power equipment do what they can to minimize turbulence, it cannot be avoided. For example, in a 4-inch pipe at 68°F, flow becomes turbulent at velocities over approximately 6 inches per second or about 3 inches per second in a 6-inch pipe. These velocities are far below those commonly encountered in fluid power systems, where velocities of 5 feet per second and above are common. In streamlined flow, losses due to friction increase directly with velocity. With turbulent flow these losses increase much more rapidly.

FACTORS INVOLVED IN FLOW

An understanding of the behavior of fluids in motion, or solids for that matter, requires an understanding of the term *inertia*. Inertia is the term used by scientists to describe the property possessed by all forms of matter that makes the matter resist being moved if it is at rest, and likewise, resist any change in its rate of motion if it is moving.

The basic statement covering inertia is Newton's first law of motion—inertia. Sir Isaac Newton was a British philosopher and mathematician. His first law states: *A body* at *rest tends to remain at rest, and a body in motion tends to remain in motion at the same speed and direction, unless acted on by some unbalanced force.* This simply says what you have learned by experience—that you must push an object to start it moving and push it in the opposite direction to stop it again.

A familiar illustration is the effort a pitcher must exert to make a fast pitch and the opposition the catcher must put forth to stop the ball. Similarly, considerable work must be performed by the engine to make an automobile begin to roll; although, after it has attained a certain velocity, it will roll along the road at uniform speed if just enough effort is expended to overcome friction, while brakes are necessary to stop its motion. Inertia also explains the kick or recoil of guns and the tremendous striking force of projectiles.

Inertia and Force

To overcome the tendency of an object to resist any change in its state of rest or motion, some force that is not otherwise canceled or unbalanced must act on the object. Some unbalanced force must be applied whenever fluids are set in motion or increased in velocity; while conversely, forces are made to do work elsewhere whenever fluids in motion are retarded or stopped.

There is a direct relationship between the magnitude of the force exerted and the inertia against which it acts. This force is dependent on two factors: (1) the mass of the object (which is proportional to its weight), and (2) the rate at which the velocity of the object is changed. The rule is that the force in pounds required to overcome inertia is equal to the weight of the object multiplied by the change in velocity, measured in feet per second, and divided by 32 times the time in seconds required to accomplish the change. Thus, the rate of change in velocity of an object is proportional to the force applied. The number 32 appears because it is the conversion factor between weight and mass.

There are five physical factors that can act on a fluid to affect its behavior. All of the physical actions of fluids in all systems are determined by the relationships of these five factors to each other. Summarizing, these five factors are as follows:

1. Gravity, which acts at all times on all bodies, regardless of other forces

2. Atmospheric pressure, which acts on any part of a system exposed to the open air

3. Specific applied forces, which mayor may not be present, but which, in any event, are entirely independent of the presence or absence of motion

4. Inertia, which comes into play whenever there is a change from rest to motion or the opposite, or whenever there is a change in direction or in rate of motion

5. Friction, which is always present whenever there is motion

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Figure 2-16 illustrates a possible relationship of these factors with respect to a particle of fluid (P) in a system. The different forces are shown in terms of head, or in other words, in terms of vertical columns of fluid required to provide the forces. At the particular moment under consideration, a particle of water (P) is being acted on by applied force (A), by atmospheric pressure (B), and by gravity (C) produced by the weight of the fluid standing over it. The particle possesses sufficient inertia or velocity head to rise to level P1, since head equivalent to F was lost in friction as P passed through the system. Since atmospheric pressure (B) acts downward on both sides of the system, what is gained on one side is lost on the other.

If all the pressure acting on P to force it through the nozzle could be recovered in the form of elevation head, it would rise to level Y. If account is taken of the balance in atmospheric pressure, in a frictionless system, P would rise to level X, or precisely as high as the sum of the gravity head and the head equivalent to the applied force.

Kinetic Energy

It was previously pointed out that a force must be applied to an object in order to give it a velocity or to increase the velocity it already has. Whether the force begins or changes velocity, it acts over a certain distance. A force acting over a certain distance is work. Work and all forms into which it can be changed are classified as energy. Obviously then, energy is required to give an object velocity. The greater the energy used, the greater the velocity will be.

Disregarding friction, for an object to be brought to rest or for its motion to be slowed down, a force opposed to its motion must be applied to it. This force also acts over some distance. In this way energy is given up by the object and delivered in some form to whatever opposes its continuous motion. The moving object is therefore a means of receiving energy at one place (where its motion is increased) and delivering it to another point (where it is stopped or retarded). While it is in motion, it is said to contain this energy as energy of motion or *kinetic* energy.

Since energy can never be destroyed, it follows that if friction is disregarded the energy delivered to stop the object will exactly equal the energy that was required to increase its speed. At all times the amount of kinetic energy possessed by an object depends on its weight and the velocity at which it is moving.

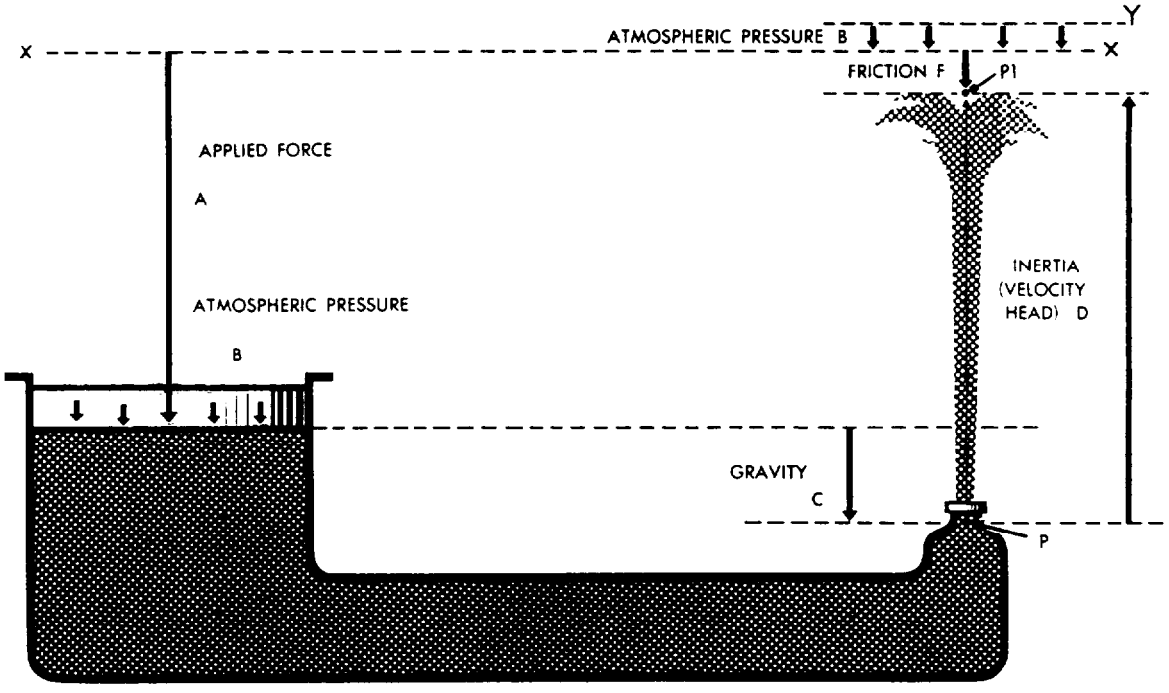

Figure 2-16.—Physical factors governing fluid flow.

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The mathematical relationship for kinetic energy is stated in the rule: "Kinetic energy in foot-pounds is equal to the force in pounds which created it, multiplied by the distance through which it was applied, or to the weight of the moving object in pounds, multiplied by the square of its velocity in feet per second, and divided by 64.s"

The relationship between inertia forces, velocity, and kinetic energy can be illustrated by analyzing what happens when a gun fires a projectile against the armor of an enemy ship. (See fig. 2-17.) The explosive force of the powder in the breach pushes the projectile out of the gun, giving it a high velocity. Because of its inertia, the projectile offers opposition to this sudden velocity and a reaction is set up that pushes the gun backward (kick or recoil). The force of the explosion acts on the projectile throughout its movement in the gun. This is force acting through a distance producing work. This work appears as kinetic energy in the speeding projectile. The resistance of the air produces friction, which uses some of the energy and slows down the projectile. Eventually, however, the projectile hits its target and, because of the inertia, tries to continue moving. The target, being relatively stationary, tends to remain stationary because of its inertia. The result is that a tremendous force is set up that either leads to the penetration of the armor or the shattering of the projectile. The projectile is simply a means of transferring energy, in this instance for destructive purpose, from the gun to the enemy ship. This energy is transmitted in the form of energy of motion or kinetic energy.

A similar action takes place in a fluid power system in which the fluid takes the place of the projectile. For example, the pump in a hydraulic system imparts energy to the fluid, which overcomes the inertia of the fluid at rest and causes it to flow through the lines. The fluid flows against some type of actuator that is at rest. The fluid tends to continue flowing, overcomes the inertia of the actuator, and moves the actuator to do work. Friction uses up a portion of the energy as the fluid flows through the lines and components.

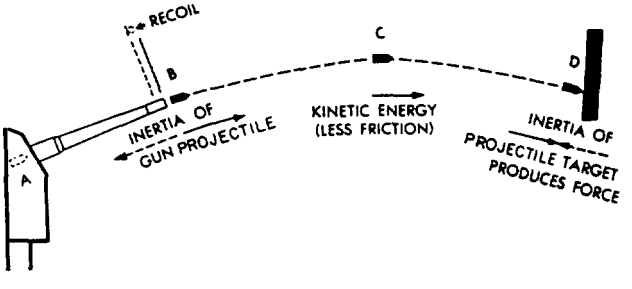

Figure 2-17.—Relationship of inertia, velocity, and kinetic energy.

RELATIONSHIP OF FORCE, PRESSURE, AND HEAD

In dealing with fluids, forces are usually considered in relation to the areas over which they are applied. As previously discussed, a force acting over a unit area is a pressure, and pressure can alternately be stated in pounds per square inch or in terms of head, which is the vertical height of the column of fluid whose weight would produce that pressure.

In most of the applications of fluid power in the Navy, applied forces greatly outweigh all other forces, and the fluid is entirely confined. Under these circumstances it is customary to think of the forces involved in terms of pressures. Since the term *head* is encountered frequently in the study of fluid power, it is necessary to understand what it means and how it is related to pressure and force.

All five of the factors that control the actions of fluids can, of course, be expressed either as force, or in terms of equivalent pressures or head. In each situation, the different factors are referred to in the same terms, since they can be added and subtracted to study their relationship to each other.

At this point you need to review some terms in general use. Gravity head, when it is important enough to be considered, is sometimes referred to as head. The effect of atmospheric pressure is referred to as atmospheric pressure. (Atmospheric pressure is frequently and improperly referred to as suction.) Inertia effect, because it is always directly related to velocity, is usually called velocity head; and friction, because it represents a loss of pressure or head, is usually referred to as friction head.

STATIC AND DYNAMIC FACTORS

Gravity, applied forces, and atmospheric pressure are static factors that apply equally to

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fluids at rest or in motion, while inertia and friction are dynamic factors that apply only to fluids in motion. The mathematical sum of gravity, applied force, and atmospheric pressure is the static pressure obtained at any one point in a fluid at any given time. Static pressure exists in addition to any dynamic factors that may also be present at the same time.

Remember, Pascal's law states that a pressure set up in a fluid acts equally in all directions and at right angles to the containing surfaces. This covers the situation only for fluids at rest or practically at rest. It is true only for the factors making up static head. Obviously, when velocity becomes a factor it must have a direction, and as previously explained, the force related to the velocity must also have a direction, so that Pascal's law alone does not apply to the dynamic factors of fluid power.

The dynamic factors of inertia and friction are related to the static factors. Velocity head and friction head are obtained at the expense of static head. However, a portion of the velocity head can always be reconverted to static head. Force, which can be produced by pressure or head when dealing with fluids, is necessary to start a body moving if it is at rest, and is present in some form when the motion of the body is arrested; therefore, whenever a fluid is given velocity, some part of its original static head is used to impart this velocity, which then exists as velocity head.

BERNOULLI'S PRINCIPLE

Consider the system illustrated in figure 2-18. Chamber A is under pressure and is connected by a tube to chamber B, which is also under pressure. The pressure in chamber A is static pressure of 100 psi. The pressure at some point (X) along the connecting tube consists of a velocity pressure of 10 psi exerted in a direction parallel to the line of flow, plus the unused static pressure of 90 psi, which still obeys Pascal's law and operates equally in all directions. As the fluid enters chamber B it is slowed down, and its velocity is changed back to pressure. The force required to absorb its inertia equals the force required to start the fluid moving originally, so that the static pressure in chamber B is equal to that in chamber A.

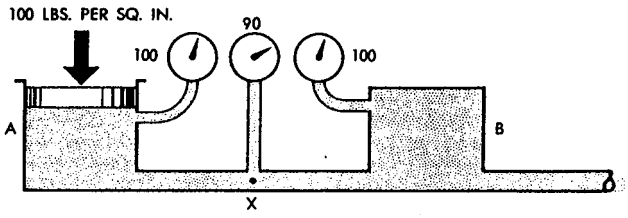

Figure 2-18.—Relation of static and dynamic factors—Bernoulli's principle.

This situation (fig. 2-18) disregards friction; therefore, it would not be encountered in actual practice. Force or head is also required to overcome friction but, unlike inertia effect, this force cannot be recovered again, although the energy represented still exists somewhere as heat. Therefore, in an actual system the pressure in chamber B would be less than in chamber A by the amount of pressure used in overcoming friction along the way.

At all points in a system the static pressure is always the original static pressure, less any velocity head at the point in question and less the friction head consumed in reaching that point. Since both the velocity head and the friction head represent energy that came from the original static head, and since energy cannot be destroyed, the sum of the static head, the velocity head, and the friction head at any point in the system must add up to the original static head. This is known as Bernoulli's principle, which states: *For the horizontal flow of fluid through a tube, the sum of the pressure and the kinetic energy per unit volume of the fluid is constant.* This principle governs the relations of the static and dynamic factors concerning fluids, while Pascal's law states the manner in which the static factors behave when taken by themselves.

MINIMIZING FRICTION

Fluid power equipment is designed to reduce friction to the lowest possible level. Volume and velocity of flow are made the subject of careful study. The proper fluid for the system is chosen. Clean, smooth pipe of the best dimensions for the particular conditions is used, and it is installed along as direct a route as possible. Sharp bends and sudden changes in cross-sectional areas are avoided. Valves, gauges, and other components are designed to interrupt flow as little as possible. Careful thought is given to the size and shape of the openings. The systems are designed so they

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can be kept clean inside and variations from normal operation can easily be detected and remedied.

OPERATION OF HYDRAULIC COMPONENTS

To transmit and control power through pressurized fluids, an arrangement of interconnected components is required. Such an arrangement is commonly referred to as a system. The number and arrangement of the components vary from system to system, depending on the particular application. In many applications, one main system supplies power to several subsystems, which are sometimes referred to as circuits. The complete system may be a small compact unit; more often, however, the components are located at widely separated points for convenient control and operation of the system.

The basic components of a fluid power system are essentially the same, regardless of whether the system uses a hydraulic or a pneumatic medium. There are five basic components used in a system. These basic components are as follows:

1. Reservoir or receiver
2. Pump or compressor
3. Lines (pipe, tubing, or flexible hose)
4. Directional control valve
5. Actuating device

Several applications of fluid power require only a simple system; that is, a system which uses only a few components in addition to the five basic components. A few of these applications are presented in the following paragraphs. We will explain the operation of these systems briefly at this time so you will know the purpose of each component and can better understand how hydraulics is used in the operation of these systems. More complex fluid power systems are described in chapter 12.

HYDRAULIC JACK

The hydraulic jack is perhaps one of the simplest forms of a fluid power system. By moving the handle of a small device, an individual can lift a load weighing several tons. A small initial force exerted on the handle is transmitted by a fluid to a much larger area. To understand this better, study figure 2-19. The small input piston has an area of 5 square inches and is directly connected to a large cylinder with an output piston having an area of 250 square inches. The top of this piston forms a lift platform.

If a force of 25 pounds is applied to the input piston, it produces a pressure of 5 psi in the fluid, that is, of course, if a sufficient amount of resistant force is acting against the top of the output piston. Disregarding friction loss, this pressure acting on the 250 square inch area of the output piston will support a resistance force of 1,250 pounds. In other words, this pressure could overcome a force of slightly under 1,250 pounds. An input force of 25 pounds has been transformed into a working force of more than half a ton; however, for this to be true, the distance traveled by the input piston must be 50 times greater than the distance traveled by the output piston. Thus, for every inch that the input piston moves, the output piston will move only one-fiftieth of an inch.

This would be ideal if the output piston needed to move only a short distance. However, in most instances, the output piston would have to be capable of moving a greater distance to serve a practical application. The device shown in figure 2-19 is not capable of moving the output piston farther than that shown; therefore, some other means must be used to raise the output piston to a greater height.

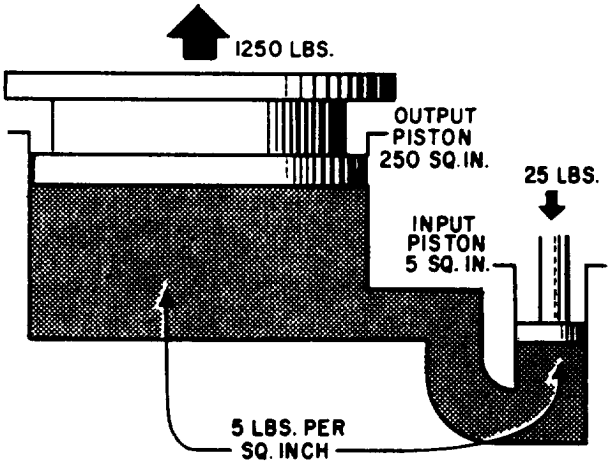

Figure 2-19.—Hydraulic jack.

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The output piston can be raised higher and maintained at this height if additional components are installed as shown in figure 2-20. In this illustration the jack is designed so that it can be raised, lowered, or held at a constant height. These results are attained by introducing a number of valves and also a reserve supply of fluid to be used in the system.

Notice that this system contains the five basic components—the reservoir; cylinder 1, which serves as a pump; valve 3, which serves as a directional control valve; cylinder 2, which serves as the actuating device; and lines to transmit the fluid to and from the different components. In addition, this system contains two valves, 1 and 2, whose functions are explained in the following discussion.

As the input piston is raised (fig. 2-20, view A), valve 1 is closed by the back pressure from the weight of the output piston. At the same time, valve 2 is opened by the head of the fluid in the reservoir. This forces fluid into cylinder 1. When the input piston is lowered (fig. 2-20, view B), a pressure is developed in cylinder 1. When this pressure exceeds the head in the reservoir, it closes valve 2. When it exceeds the back pressure from the output piston, it opens valve 1, forcing fluid into the pipeline. The pressure from cylinder 1 is thus transmitted into cylinder 2, where it acts to raise the output piston with its attached lift platform. When the input piston is again raised, the pressure in cylinder 1 drops below that in cylinder 2, causing valve 1 to close. This prevents the return of fluid and holds the output piston with its attached lift platform at its new level. During this stroke, valve 2 opens again allowing a new supply of fluid into cylinder 1 for the next power (downward) stroke of the input piston. Thus, by repeated strokes of the input piston, the lift platform can be progressively raised. To lower the lift platform, valve 3 is opened, and the fluid from cylinder 2 is returned to the reservoir.

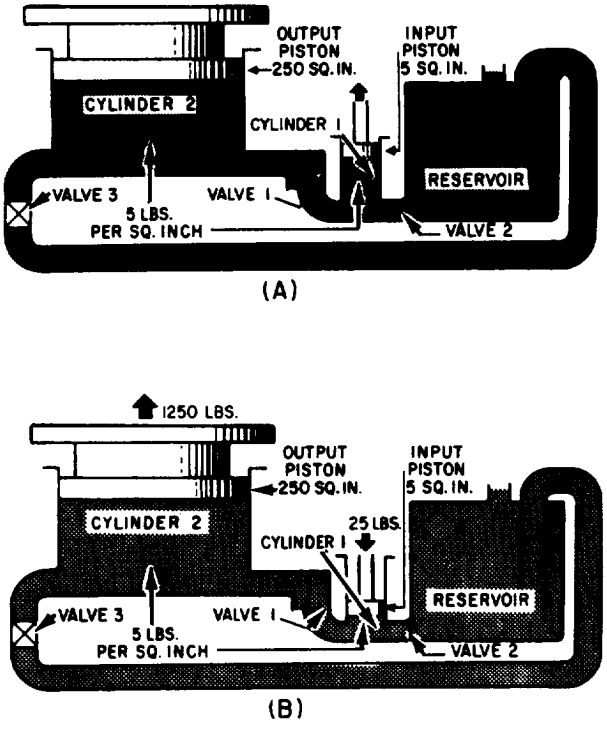

Figure 2-20.—Hydraulic jack; (A) up stroke; (B) downstroke.

HYDRAULIC BRAKES

The hydraulic brake system used in the automobile is a multiple piston system. A multiple piston system allows forces to be transmitted to two or more pistons in the manner indicated in figure 2-21. Note that the pressure set up by the force applied to the input piston (1) is transmitted undiminished to both output pistons (2 and 3), and that the resultant force on each piston is proportional to its area. The multiplication of forces from the input piston to each output piston is the same as that explained earlier.

The hydraulic brake system from the master cylinders to the wheel cylinders on most

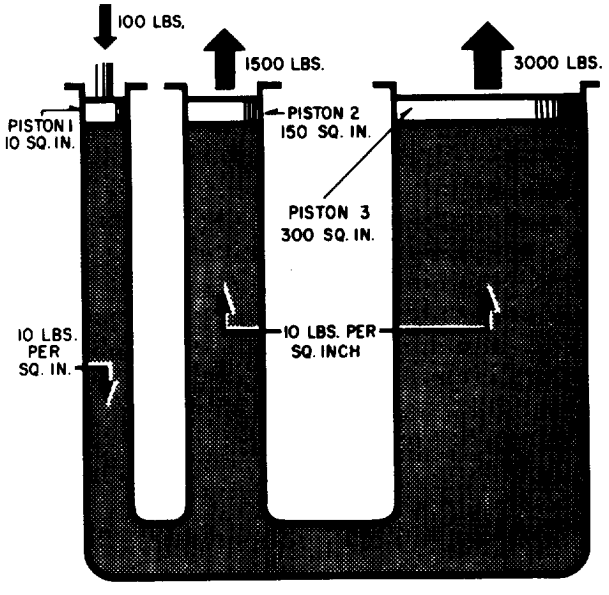

Figure 2-21.—Multiple piston system.

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automobiles operates in a way similar to the system illustrated in figure 2-22.

When the brake pedal is depressed, the pressure on the brake pedal moves the piston within the master cylinder, forcing the brake fluid from the master cylinder through the tubing and flexible hose to the wheel cylinders. The wheel cylinders contain two opposed output pistons, each of which is attached to a brake shoe fitted inside the brake drum. Each output piston pushes the attached brake shoe against the wall of the brake drum, thus retarding the rotation of the wheel. When pressure on the pedal is released, the springs on the brake shoes return the wheel cylinder pistons to their released positions. This action forces the displaced brake fluid back through the flexible hose and tubing to the master cylinder.

The force applied to the brake pedal produces a proportional force on each of the output pistons, which in turn apply the brake shoes frictionally to the turning wheels to retard rotation.

As previously mentioned, the hydraulic brake system on most automobiles operates in a similar way, as shown in figure 2-22. It is beyond the scope of this manual to discuss the various brake systems.

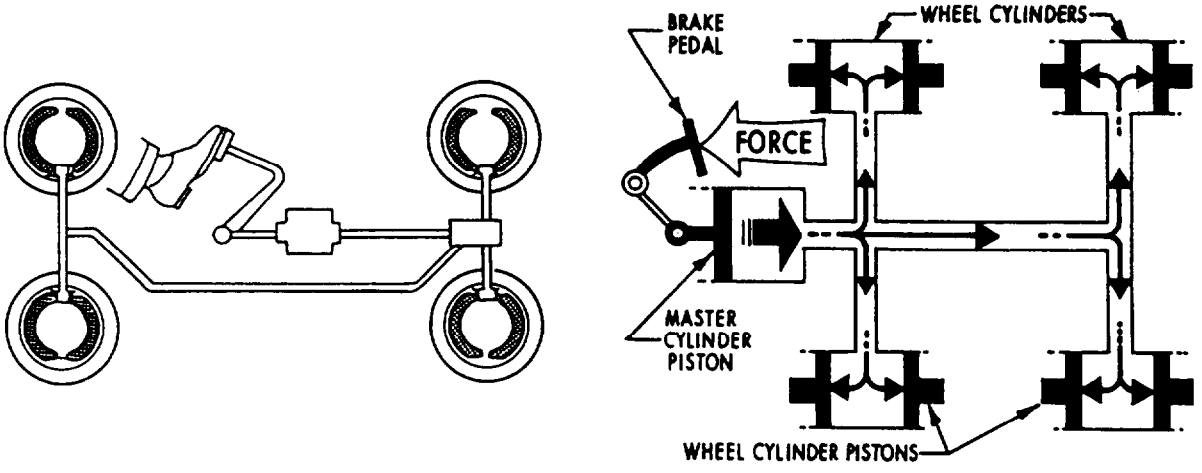

Figure 2-22.—An automobile brake system.

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CHAPTER 3

HYDRAULIC FLUIDS

During the design of equipment that requires fluid power, many factors are considered in selecting the type of system to be used—hydraulic, pneumatic, or a combination of the two. Some of the factors are required speed and accuracy of operation, surrounding atmospheric conditions, economic conditions, availability of replacement fluid, required pressure level, operating temperature range, contamination possibilities, cost of transmission lines, limitations of the equipment, lubricity, safety to the operators, and expected service life of the equipment.

After the type of system has been selected, many of these same factors must be considered in selecting the fluid for the system. This chapter is devoted to hydraulic fluids. Included in it are sections on the properties and characteristics desired of hydraulic fluids; types of hydraulic fluids; hazards and safety precautions for working with, handling, and disposing of hydraulic liquids; types and control of contamination; and sampling.

PROPERTIES

If fluidity (the physical property of a substance that enables it to flow) and incompressibility were the only properties required, any liquid not too thick might be used in a hydraulic system. However, a satisfactory liquid for a particular system must possess a number of other properties. The most important properties and some characteristics are discussed in the following paragraphs.

VISCOSITY

Viscosity is one of the most important properties of hydraulic fluids. It is a measure of a fluid's resistance to flow. A liquid, such as gasoline, which flows easily has a low viscosity; and a liquid, such as tar, which flows slowly has a high viscosity. The viscosity of a liquid is affected by changes in temperature and pressure. As the temperature of a liquid increases, its viscosity decreases. That is, a liquid flows more easily when it is hot than when it is cold. The viscosity of a liquid increases as the pressure on the liquid increases.

A satisfactory liquid for a hydraulic system must be thick enough to give a good seal at pumps, motors, valves, and so on. These components depend on close fits for creating and maintaining pressure. Any internal leakage through these clearances results in loss of pressure, instantaneous control, and pump efficiency. Leakage losses are greater with thinner liquids (low viscosity). A liquid that is too thin will also allow rapid wearing of moving parts, or of parts that operate under heavy loads. On the other hand, if the liquid is too thick (viscosity too high), the internal friction of the liquid will cause an increase in the liquid's flow resistance through clearances of closely fitted parts, lines, and internal passages. This results in pressure drops throughout the system, sluggish operation of the equipment, and an increase in power consumption.

Measurement of Viscosity

Viscosity is normally determined by measuring the time required for a fixed volume of a fluid (at a given temperature) to flow through a calibrated orifice or capillary tube. The instruments used to measure the viscosity of a liquid are known as viscometers or viscosimeters.

Several types of viscosimeters are in use today. The Saybolt viscometer, shown in figure 3-1, measures the time required, in seconds, for 60 milliliters of the tested fluid at 100°F to pass through a standard orifice. The time measured is

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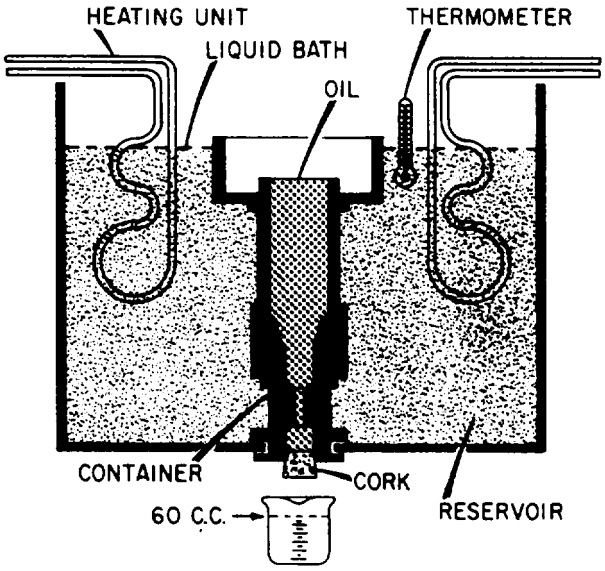

Figure 3-1.—Saybolt viscometer.

used to express the fluid's viscosity, in Saybolt universal seconds or Saybolt furol seconds.

The glass capillary viscometers, shown in figure 3-2, are examples of the second type of viscometer used. These viscometers are used to measure kinematic viscosity. Like the Saybolt viscometer, the glass capillary measures the time in seconds required for the tested fluid to flow through the capillary. This time is multiplied by the temperature constant of the viscometer in use to provide the viscosity, expressed in centistrokes.

The following formulas may be used to convert centistrokes (cSt units) to approximate Saybolt universal seconds (SUS units).

For SUS values between 32 and 100:

$$cST = 0.226 \times SUS - \frac{195}{SUS}$$

For SUS values greater than 100:

$$cST = 0.220 \times SUS - \frac{135}{SUS}$$

Although the viscometers discussed above are used in laboratories, there are other viscometers in the supply system that are available for local use. These viscometers can be used to test the viscosity of hydraulic fluids either prior to their being added to a system or periodically after they have been in an operating system for a while.

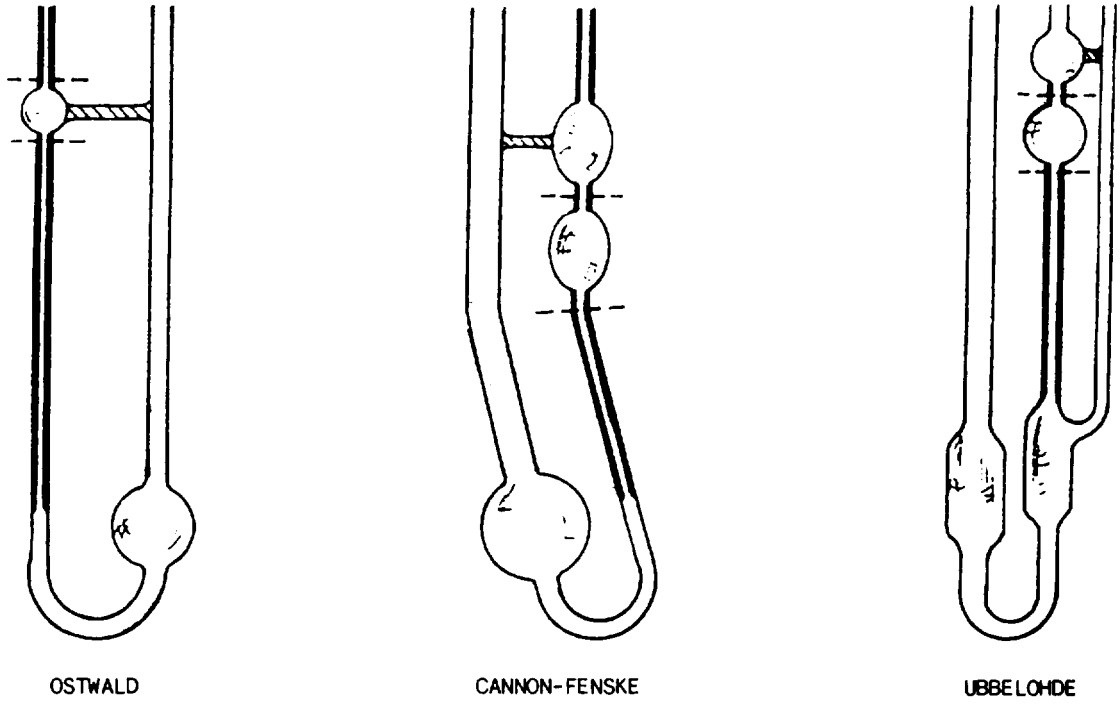

Figure 3-2.–Various styles of glass capillary viscometers.

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Additional information on the various types of viscometers and their operation can be found in the *Physical Measurements Training Manual*, NAVAIR 17-35QAL-2.

Viscosity Index

The viscosity index (V.I.) of an oil is a number that indicates the effect of temperature changes on the viscosity of the oil. A low V.I. signifies a relatively large change of viscosity with changes of temperature. In other words, the oil becomes extremely thin at high temperatures and extremely thick at low temperatures. On the other hand, a high V.I. signifies relatively little change in viscosity over a wide temperature range.

An ideal oil for most purposes is one that maintains a constant viscosity throughout temperature changes. The importance of the V.I. can be shown easily by considering automotive lubricants. An oil having a high V.I. resists excessive thickening when the engine is cold and, consequently, promotes rapid starting and prompt circulation; it resists excessive thinning when the motor is hot and thus provides full lubrication and prevents excessive oil consumption.

Another example of the importance of the V.I. is the need for a high V.I. hydraulic oil for military aircraft, since hydraulic control systems may be exposed to temperatures ranging from below –65°F at high altitudes to over 100°F on the ground. For the proper operation of the hydraulic control system, the hydraulic fluid must have a sufficiently high V.I. to perform its functions at the extremes of the expected temperature range.

Liquids with a high viscosity have a greater resistance to heat than low viscosity liquids which have been derived from the same source. The average hydraulic liquid has a relatively low viscosity. Fortunately, there is a wide choice of liquids available for use in the viscosity range required of hydraulic liquids.

The V.I. of an oil may be determined if its viscosity at any two temperatures is known. Tables, based on a large number of tests, are issued by the American Society for Testing and Materials (ASTM). These tables permit calculation of the V.I. from known viscosities.

LUBRICATING POWER

If motion takes place between surfaces in contact, friction tends to oppose the motion. When pressure forces the liquid of a hydraulic system between the surfaces of moving parts, the liquid spreads out into a thin film which enables the parts to move more freely. Different liquids, including oils, vary greatly not only in their lubricating ability but also in film strength. Film strength is the capability of a liquid to resist being wiped or squeezed out from between the surfaces when spread out in an extremely thin layer. A liquid will no longer lubricate if the film breaks down, since the motion of part against part wipes the metal clean of liquid.

Lubricating power varies with temperature changes; therefore, the climatic and working conditions must enter into the determination of the lubricating qualities of a liquid. Unlike viscosity, which is a physical property, the lubricating power and film strength of a liquid is directly related to its chemical nature. Lubricating qualities and film strength can be improved by the addition of certain chemical agents.

CHEMICAL STABILITY

Chemical stability is another property which is exceedingly important in the selection of a hydraulic liquid. It is defined as the liquid's ability to resist oxidation and deterioration for long periods. All liquids tend to undergo unfavorable changes under severe operating conditions. This is the case, for example, when a system operates for a considerable period of time at high temperatures.

Excessive temperatures, especially extremely high temperatures, have a great effect on the life of a liquid. The temperature of the liquid in the reservoir of an operating hydraulic system does not always indicate the operating conditions throughout the system. Localized hot spots occur on bearings, gear teeth, or at other points where the liquid under pressure is forced through small orifices. Continuous passage of the liquid through these points may produce local temperatures high enough to carbonize the liquid or turn it into sludge, yet the liquid in the reservoir may not indicate an excessively high temperature.

Liquids may break down if exposed to air, water, salt, or other impurities, especially if they are in constant motion or subjected to heat. Some metals, such as zinc, lead, brass, and copper, have undesirable chemical reactions with certain liquids.

These chemical reactions result in the formation of sludge, gums, carbon, or other deposits which clog openings, cause valves and pistons to stick or leak, and give poor lubrication to moving

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parts. Once a small amount of sludge or other deposits is formed, the rate of formation generally increases more rapidly. As these deposits are formed, certain changes in the physical and chemical properties of the liquid take place. The liquid usually becomes darker, the viscosity increases and damaging acids are formed.

The extent to which changes occur in different liquids depends on the type of liquid, type of refining, and whether it has been treated to provide further resistance to oxidation. The stability of liquids can be improved by the addition of oxidation inhibitors. Inhibitors selected to improve stability must be compatible with the other required properties of the liquid.

FREEDOM FROM ACIDITY

An ideal hydraulic liquid should be free from acids which cause corrosion of the metals in the system. Most liquids cannot be expected to remain completely noncorrosive under severe operating conditions. The degree of acidity of a liquid, when new, may be satisfactory; but after use, the liquid may tend to become corrosive as it begins to deteriorate.

Many systems are idle for long periods after operating at high temperatures. This permits moisture to condense in the system, resulting in rust formation.

Certain corrosion- and rust-preventive additives are added to hydraulic liquids. Some of these additives are effective only for a limited period. Therefore, the best procedure is to use the liquid specified for the system for the time specified by the system manufacturer and to protect the liquid and the system as much as possible from contamination by foreign matter, from abnormal temperatures, and from misuse.

FLASHPOINT

Flashpoint is the temperature at which a liquid gives off vapor in sufficient quantity to ignite momentarily or flash when a flame is applied. A high flashpoint is desirable for hydraulic liquids because it provides good resistance to combustion and a low degree of evaporation at normal temperatures. Required flashpoint minimums vary from 300°F for the lightest oils to 510°F for the heaviest oils.

FIRE POINT

Fire point is the temperature at which a substance gives off vapor in sufficient quantity to ignite and continue to burn when exposed to a spark or flame. Like flashpoint, a high fire point is required of desirable hydraulic liquids.

MINIMUM TOXICITY

Toxicity is defined as the quality, state, or degree of being toxic or poisonous. Some liquids contain chemicals that are a serious toxic hazard. These toxic or poisonous chemicals may enter the body through inhalation, by absorption through the skin, or through the eyes or the mouth. The result is sickness and, in some cases, death. Manufacturers of hydraulic liquids strive to produce suitable liquids that contain no toxic chemicals and, as a result, most hydraulic liquids are free of harmful chemicals. Some fire-resistant liquids are toxic, and suitable protection and care in handling must be provided.

DENSITY AND COMPRESSIBILITY

A fluid with a specific gravity of less than 1.0 is desired when weight is critical, although with proper system design, a fluid with a specific gravity greater than one can be tolerated. Where avoidance of detection by military units is desired, a fluid which sinks rather than rises to the surface of the water is desirable. Fluids having a specific gravity greater than 1.0 are desired, as leaking fluid will sink, allowing the vessel with the leak to remain undetected.

Recall from chapter 2 that under extreme pressure a fluid may be compressed up to 7 percent of its original volume. Highly compressible fluids produce sluggish system operation. This does not present a serious problem in small, low-speed operations, but it must be considered in the operating instructions.

FOAMING TENDENCIES

Foam is an emulsion of gas bubbles in the fluid. Foam in a hydraulic system results from compressed gases in the hydraulic fluid. A fluid under high pressure can contain a large volume of air bubbles. When this fluid is depressurized, as when it reaches the reservoir, the gas bubbles in the fluid expand and produce foam. Any amount of foaming may cause pump cavitation and produce poor system response and spongy

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control. Therefore, defoaming agents are often added to fluids to prevent foaming. Minimizing air in fluid systems is discussed later in this chapter.

CLEANLINESS

Cleanliness in hydraulic systems has received considerable attention recently. Some hydraulic systems, such as aerospace hydraulic systems, are extremely sensitive to contamination. Fluid cleanliness is of primary importance because contaminants can cause component malfunction, prevent proper valve seating, cause wear in components, and may increase the response time of servo valves. Fluid contaminants are discussed later in this chapter.

The inside of a hydraulic system can only be kept as clean as the fluid added to it. Initial fluid cleanliness can be achieved by observing stringent cleanliness requirements (discussed later in this chapter) or by filtering all fluid added to the system.

TYPES OF HYDRAULIC FLUIDS

There have been many liquids tested for use in hydraulic systems. Currently, liquids being used include mineral oil, water, phosphate ester, water-based ethylene glycol compounds, and silicone fluids. The three most common types of hydraulic liquids are petroleum-based, synthetic fire-resistant, and water-based fire-resistant.

PETROLEUM-BASED FLUIDS

The most common hydraulic fluids used in shipboard systems are the petroleum-based oils. These fluids contain additives to protect the fluid from oxidation (antioxidant), to protect system metals from corrosion (anticorrosion), to reduce tendency of the fluid to foam (foam suppressant), and to improve viscosity.

Petroleum-based fluids are used in surface ships' electrohydraulic steering and deck machinery systems, submarines' hydraulic systems, and aircraft automatic pilots, shock absorbers, brakes, control mechanisms, and other hydraulic systems using seal materials compatible with petroleum-based fluids.

SYNTHETIC FIRE-RESISTANT FLUIDS

Petroleum-based oils contain most of the desired properties of a hydraulic liquid. However, they are flammable under normal conditions and can become explosive when subjected to high pressures and a source of flame or high temperatures. Nonflammable synthetic liquids have been developed for use in hydraulic systems where fire hazards exist.

Phosphate Ester Fire-Resistant Fluid

Phosphate ester fire-resistant fluid for shipboard use is covered by specification MIL-H-19457. There are certain trade names closely associated with these fluids. However, the only acceptable fluids conforming to MIL-H-19457 are the ones listed on the current Qualified Products List (QPL) 19457. These fluids will be delivered in containers marked MIL-H-19457C or a later specification revision. Phosphate ester in containers marked by a brand name without a specification identification must not be used in shipboard systems, as they may contain toxic chemicals.

These fluids will burn if sufficient heat and flame are applied, but they do not support combustion. Drawbacks of phosphate ester fluids are that they will attack and loosen commonly used paints and adhesives, deteriorate many types of insulations used in electrical cables, and deteriorate many gasket and seal materials. Therefore, gaskets and seals for systems in which phosphate ester fluids are used are manufactured of specific materials. *Naval Ships' Technical Manual,* chapter 262, specifies paints to be used on exterior surfaces of hydraulic systems and components in which phosphate ester fluid is used and on ship structure and decks in the immediate vicinity of this equipment. *Naval Ships' Technical Manual,* chapter 078, specifies gasket and seal materials used. NAVAIR 01-1A-17 also contains a list of materials resistant to phosphate ester fluids.

Trade names for phosphate ester fluids, which do not conform to MIL-H-19457 include Pydraul, Skydrol, and Fyre Safe.

PHOSPHATE ESTER FLUID SAFETY.— As a maintenance person, operator, supervisor, or crew member of a ship, squadron, or naval shore installation, you must understand the hazards associated with hydraulic fluids to which you may be exposed.

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Phosphate ester fluid conforming to specification MIL-H-19457 is used in aircraft elevators, ballast valve operating systems, and replenishment-at-sea systems. This type of fluid contains a controlled amount of neurotoxic material. Because of the neurotoxic effects that can result from ingestion, skin absorption, or inhalation of these fluids, be sure to use the following precautions:

1. Avoid contact with the fluids by wearing protective clothing.
2. Use chemical goggles or face shields to protect your eyes.
3. If you are expected to work in an atmosphere containing a fine mist or spray, wear a continuous-flow airline respirator.
4. Thoroughly clean skin areas contaminated by this fluid with soap and water.
5. If you get any fluid in your eyes, flush them with running water for at least 15 minutes and seek medical attention.

If you come in contact with MIL-H-19457 fluid, report the contact when you seek medical aid and whenever you have a routine medical examination.

Naval Ships' Technical Manual, chapter 262, contains a list of protective clothing, along with national stock numbers (NSN), for use with fluids conforming to MIL-H-19457. It also contains procedures for repair work and for low-level leakage and massive spills cleanup.

PHOSPHATE ESTER FLUID DISPOSAL.— Waste MIL-H-19457 fluids and refuse (rags and other materials) must not be dumped at sea. Fluid should be placed in bung-type drums. Rags and other materials should be placed in open top drums for shore disposal. These drums should be marked with a warning label stating their content, safety precautions, and disposal instructions. Detailed instructions for phosphate ester fluids disposal can be found in *Naval Ships' Technical Manual,* chapter 262, and OPNAVINST 5090.1.

Silicone Synthetic Fire-Resistant Fluids

Silicone synthetic fire-resistant fluids are frequently used for hydraulic systems which require fire resistance, but which have only marginal requirements for other chemical or physical properties common to hydraulic fluids. Silicone fluids do not have the detrimental characteristics of phosphate ester fluids, nor do they provide the corrosion protection and lubrication of phosphate ester fluids, but they are excellent for fire protection. Silicone fluid conforming to MIL-S-81087 is used in the missile holddown and lockout system aboard submarines.

Lightweight Synthetic Fire-Resistant Fluids

In applications where weight is critical, lightweight synthetic fluid is used in hydraulic systems. MIL-H-83282 is a synthetic, fire-resistant hydraulic fluid used in military aircraft and hydrofoils where the requirement to minimize weight dictates the use of a low-viscosity fluid. It is also the most commonly used fluid in aviation support equipment. NAVAIR 01-1A-17 contains additional information on fluids conforming to specification MIL-H-83282.

WATER-BASED FIRE-RESISTANT FLUIDS

The most widely used water-based hydraulic fluids may be classified as water-glycol mixtures and water-synthetic base mixtures. The water-glycol mixture contains additives to protect it from oxidation, corrosion, and biological growth and to enhance its load-carrying capacity.

Fire resistance of the water mixture fluids depends on the vaporization and smothering effect of steam generated from the water. The water in water-based fluids is constantly being driven off while the system is operating. Therefore, frequent checks to maintain the correct ratio of water are important.

The water-based fluid used in catapult retracting engines, jet blast deflectors, and weapons elevators and handling systems conforms to MIL-H-22072.

The safety precautions outlined for phosphate ester fluid and the disposal of phosphate ester fluid also apply to water-based fluid conforming to MIL-H-22072.

CONTAMINATION

Hydraulic fluid contamination may be described as any foreign material or substance whose presence in the fluid is capable of adversely affecting system performance or reliability. It may assume many different forms, including liquids, gases, and solid matter of various composition, sizes, and shapes. Solid matter is the type most often found in hydraulic systems and is generally

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referred to as particulate contamination. Contamination is always present to some degree, even in new, unused fluid, but must be kept below a level that will adversely affect system operation. Hydraulic contamination control consists of requirements, techniques, and practices necessary to minimize and control fluid contamination.

CLASSIFICATION

There are many types of contaminants which are harmful to hydraulic systems and liquids. These contaminants may be divided into two different classes—particulate and fluid.

Particulate Contamination

This class of contaminants includes organic, metallic solid, and inorganic solid contaminants. These contaminants are discussed in the following paragraphs.

ORGANIC CONTAMINATION.— Organic solids or semisolids found in hydraulic systems are produced by wear, oxidation, or polymerization. Minute particles of O-rings, seals, gaskets, and hoses are present, due to wear or chemical reactions. Synthetic products, such as neoprene, silicones, and hypalon, though resistant to chemical reaction with hydraulic fluids, produce small wear particles. Oxidation of hydraulic fluids increases with pressure and temperature, although antioxidants are blended into hydraulic fluids to minimize such oxidation. The ability of a hydraulic fluid to resist oxidation or polymerization in service is defined as its oxidation stability. Oxidation products appear as organic acids, asphaltics, gums, and varnishes. These products combine with particles in the hydraulic fluid to form sludge. Some oxidation products are oil soluble and cause the hydraulic fluid to increase in viscosity; other oxidation products are not oil soluble and form sediment.

METALLIC SOLID CONTAMINATION.— Metallic contaminants are almost always present in a hydraulic system and will range in size from microscopic particles to particles readily visible to the naked eye. These particles are the result of wearing and scoring of bare metal parts and plating materials, such as silver and chromium. Although practically all metals commonly used for parts fabrication and plating may be found in hydraulic fluids, the major metallic materials found are ferrous, aluminum, and chromium particles. Because of their continuous high-speed internal movement, hydraulic pumps usually contribute most of the metallic particulate contamination present in hydraulic systems. Metal particles are also produced by other hydraulic system components, such as valves and actuators, due to body wear and the chipping and wearing away of small pieces of metal plating materials.

INORGANIC SOLID CONTAMINATION.— This contaminant group includes dust, paint particles, dirt, and silicates. Glass particles from glass bead peening and blasting may also be found as contaminants. Glass particles are very undesirable contaminants due to their abrasive effect on synthetic rubber seals and the very fine surfaces of critical moving parts. Atmospheric dust, dirt, paint particles, and other materials are often drawn into hydraulic systems from external sources. For example, the wet piston shaft of a hydraulic actuator may draw some of these foreign materials into the cylinder past the wiper and dynamic seals, and the contaminant materials are then dispersed in the hydraulic fluid. Contaminants may also enter the hydraulic fluid during maintenance when tubing, hoses, fittings, and components are disconnected or replaced. It is therefore important that all exposed fluid ports be sealed with approved protective closures to minimize such contamination.

Fluid Contamination

Air, water, solvent, and other foreign fluids are in the class of fluid contaminants.

AIR CONTAMINATION.— Hydraulic fluids are adversely affected by dissolved, entrained, or free air. Air may be introduced through improper maintenance or as a result of system design. Any maintenance operation that involves breaking into the hydraulic system, such as disconnecting or removing a line or component will invariably result in some air being introduced into the system. This source of air can and must be minimized by prebilling replacement components with new filtered fluid prior to their installation. Failing to prefill a filter element bowl with fluid is a good example of how air can be introduced into the system. Although prebilling will minimize introduction of air, it is still important to vent the system where venting is possible.

Most hydraulic systems have built-in sources of air. Leaky seals in gas-pressurized accumulators and reservoirs can feed gas into a system faster

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than it can be removed, even with the best of maintenance. Another lesser known but major source of air is air that is sucked into the system past actuator piston rod seals. This usually occurs when the piston rod is stroked by some external means while the actuator itself is not pressurized.

WATER CONTAMINATION.— Water is a serious contaminant of hydraulic systems. Hydraulic fluids are adversely affected by dissolved, emulsified, or free water. Water contamination may result in the formation of ice, which impedes the operation of valves, actuators, and other moving parts. Water can also cause the formation of oxidation products and corrosion of metallic surfaces.

SOLVENT CONTAMINATION.— Solvent contamination is a special form of foreign fluid contamination in which the original contaminating substance is a chlorinated solvent. Chlorinated solvents or their residues may, when introduced into a hydraulic system, react with any water present to form highly corrosive acids.

Chlorinated solvents, when allowed to combine with minute amounts of water often found in operating hydraulic systems, change chemically into hydrochloric acids. These acids then attack internal metallic surfaces in the system, particularly those that are ferrous, and produce a severe rust-like corrosion. NAVAIR 01-1A-17 and *NSTM*, chapter 556, contain tables of solvents for use in hydraulic maintenance.

FOREIGN-FLUIDS CONTAMINATION.— Hydraulic systems can be seriously contaminated by foreign fluids other than water and chlorinated solvents. This type of contamination is generally a result of lube oil, engine fuel, or incorrect hydraulic fluid being introduced inadvertently into the system during servicing. The effects of such contamination depend on the contaminant, the amount in the system, and how long it has been present.

NOTE: It is extremely important that the different types of hydraulic fluids are not mixed in one system. If different type hydraulic fluids are mixed, the characteristics of the fluid required for a specific purpose are lost. Mixing the different types of fluids usually will result in a heavy, gummy deposit that will clog passages and require a major cleaning. In addition, seals and packing installed for use with one fluid usually are not compatible with other fluids and damage to the seals will result.

ORIGIN OF CONTAMINATION

Recall that contaminants are produced from wear and chemical reactions, introduced by improper maintenance, and inadvertently introduced during servicing. These methods of contaminant introduction fall into one of the four major areas of contaminant origin.

1. Particles originally contained in the system. These particles originate during the fabrication and storage of system components. Weld spatter and slag may remain in welded system components, especially in reservoirs and pipe assemblies. The presence is minimized by proper design. For example, seam-welded overlapping joints are preferred, and arc welding of open sections is usually avoided. Hidden passages in valve bodies, inaccessible to sand blasting or other methods of cleaning, are the main source of introduction of core sand. Even the most carefully designed and cleaned casting will almost invariably free some sand particles under the action of hydraulic pressure. Rubber hose assemblies always contain some loose particles. Most of these particles can be removed by flushing the hose before installation; however, some particles withstand cleaning and are freed later by the action of hydraulic pressure.

Particles of lint from cleaning rags can cause abrasive damage in hydraulic systems, especially to closely fitted moving parts. In addition, lint in a hydraulic system packs easily into clearances between packing and contacting surfaces, leading to component leakage and decreased efficiency. Lint also helps clog filters prematurely. The use of the proper wiping materials will reduce or eliminate lint contamination. The wiping materials to be used for a given application will be determined by

a. substances being wiped or absorbed,
b. the amount of absorbency required, and/or
c. the required degree of cleanliness.

These wiping materials are categorized for contamination control by the degree of lint or debris that they may deposit during use. For internal hydraulic repairs, this factor itself will determine the choice of wiping material.

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NAVAIR 01-1A-17 and *NSTM,* chapter 556, provides information on low-lint wiping cloths.

Rust or corrosion initially present in a hydraulic system can usually be traced to improper storage of materials and component parts. Particles can range in size from large flakes to abrasives of microscopic dimensions. Proper preservation of stored parts is helpful in eliminating corrosion.

2. Particles introduced from outside sources. Particles can be introduced into hydraulic systems at points where either the liquid or certain working parts of the system (for example, piston rods) are at least in temporary contact with the atmosphere. The most common contaminant introduction areas are at the refill and breather openings, cylinder rod packings, and open lines where components are removed for repair or replacement. Contamination arising from carelessness during servicing operations is minimized by the use of filters in the system fill lines and finger strainers in the filler adapter of hydraulic reservoirs. Hydraulic cylinder piston rods incorporate wiper rings and dust seals to prevent the dust that settles on the piston rod during its outward stroke from entering the system when the piston rod retracts. Caps and plugs are available and should be used to seal off the open lines when a component is removed for repair or replacement.

3. Particles created within the system during operation. Contaminants created during system operation are of two general types—mechanical and chemical. Particles of a mechanical nature are formed by wearing of parts in frictional contact, such as pumps, cylinders, and packing gland components. These wear particles can vary from large chunks of packings down to steel shavings that are too small to be trapped by filters.

The major source of chemical contaminants in hydraulic liquid is oxidation. These contaminants are formed under high pressure and temperatures and are promoted by the chemical action of water and air and of metals like copper and iron oxides. Liquid-oxidation products appear initially as organic acids, asphaltines, gums, and varnishes—sometimes combined with dust particles as sludge. Liquid-soluble oxidation products tend to increase liquid viscosity, while insoluble types separate and form sediments, especially on colder elements such as heat exchanger coils.

Liquids containing antioxidants have little tendency to form gums and sludge under normal operating conditions. However, as the temperature increases, resistance to oxidation diminishes. Hydraulic liquids that have been subjected to excessively high temperatures (above 250°F for most liquids) will break down, leaving minute particles of asphaltines suspended in the liquids. The liquid changes to brown in color and is referred to as decomposed liquid. This explains the importance of keeping the hydraulic liquid temperature below specific levels.

The second contaminant-producing chemical action in hydraulic liquids is one that permits these liquids to react with certain types of rubber. This reaction causes structural changes in the rubber, turning it brittle, and finally causing its complete disintegration. For this reason, the compatibility of system liquid with seals and hose material is a very important factor.

4. Particles introduced by foreign liquids. One of the most common foreign-fluid contaminants is water, especially in hydraulic systems that require petroleum-based liquids. Water, which enters even the most carefully designed system by condensation of atmospheric moisture, normally settles to the bottom of the reservoir. Oil movement in the reservoir disperses the water into fine droplets, and agitation of the liquid in the pump and in high-speed passages forms an oil-water-air emulsion. This emulsion normally separates during the rest period in the system reservoir; but when fine dust and corrosion particles are present, the emulsion is chemically changed by high pressures into sludge. The damaging action of sludge explains the need for effective filtration, as well as the need for water separation qualities in hydraulic liquids.

CONTAMINATION CONTROL

Maintaining hydraulic fluid within allowable contamination limits for both water and particulate matter is crucial to the care and protection of hydraulic equipment.

Filters (discussed in chapter 9) will provide adequate control of the particular contamination problem during all normal hydraulic system operations if the filtration system is installed properly and filter maintenance is performed properly. Filter maintenance includes changing elements at proper intervals. Control of the size and amount of contamination entering the system from any other source is the responsibility

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of the personnel who service and maintain the equipment. During installation, maintenance, and repair of hydraulic equipment, the retention of cleanliness of the system is of paramount importance for subsequent satisfactory performance.

The following maintenance and servicing procedures should be adhered to at all times to provide proper contamination control:

1. All tools and the work area (workbenches and test equipment) should be kept in a clean, dirt-free condition.
2. A suitable container should always be provided to receive the hydraulic liquid that is spilled during component removal or disassembly.

NOTE: The reuse of drained hydraulic liquid is prohibited in most hydraulic systems. In some large-capacity systems the reuse of fluid is permitted. When liquid is drained from these systems for reuse, it must be stored in a clean and suitable container. The liquid must be strained and/or filtered when it is returned to the system reservoir.

3. Before hydraulic lines or fittings are disconnected, the affected area should be cleaned with an approved dry-cleaning solvent.
4. All hydraulic lines and fittings should be capped or plugged immediately after disconnection.
5. Before any hydraulic components are assembled, their parts should be washed with an approved dry-cleaning solvent.
6. After the parts have been cleaned in dry-cleaning solvent, they should be dried thoroughly with clean, low-lint cloths and lubricated with the recommended preservative or hydraulic liquid before assembly.

NOTE: Only clean, low lint type I or II cloths as appropriate should be used to wipe or dry component parts.

7. All packings and gaskets should be replaced during the assembly procedures.
8. All parts should be connected with care to avoid stripping metal slivers from threaded areas. All fittings and lines should be installed and torqued according to applicable technical instructions.
9. All hydraulic servicing equipment should be kept clean and in good operating condition.

Some hydraulic fluid specifications, such as MIL-H-6083, MIL-H-46170, and MIL-H-83282, contain particle contamination limits that are so low that the products are packaged under clean room conditions. Very slight amounts of dirt, rust, and metal particles will cause them to fail the specification limit for contamination. Since these fluids are usually all packaged in hermetically sealed containers, the act of opening a container may allow more contaminants into the fluid than the specification allows. Therefore, extreme care should be taken in the handling of these fluids. In opening the container for use, observation, or tests, it is extremely important that the can be opened and handled in a clean environment. The area of the container to be opened should be flushed with filtered solvent (petroleum ether or isopropyl alcohol), and the device used for opening the container should be thoroughly rinsed with filtered solvent. After the container is opened, a small amount of the material should be poured from the container and disposed of prior to pouring the sample for analysis. Once a container is opened, if the contents are not totally used, the unused portion should be discarded. Since the level of contamination of a system containing these fluids must be kept low, maintenance on the system's components must be performed in a clean environment commonly known as a controlled environment work center. Specific information about the controlled environment work center can be found in the *Aviation Hydraulics Manual,* NAVAIR 01-1A-17.

HYDRAULIC FLUID SAMPLING

The condition of a hydraulic system, as well as its probable future performance, can best be determined by analyzing the operating fluid. Of particular interest are any changes in the physical and chemical properties of the fluid and excessive particulate or water contamination, either of which indicates impending trouble.

Excessive particulate contamination of the fluid indicates that the filters are not keeping the system clean. This can result from improper filter maintenance, inadequate filters, or excessive ongoing corrosion and wear.

Operating equipment should be sampled according to instructions given in the operating

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and maintenance manual for the particular equipment or as directed by the MRCs.

1. All samples should be taken from circulating systems, or immediately upon shutdown, while the hydraulic fluid is within 5°C (9°F) of normal system operating temperature. Systems not up to temperature may provide nonrepresentative samples of system dirt and water content, and such samples should either be avoided or so indicated on the analysis report. The first oil coming from the sampling point should be discarded, since it can be very dirty and does not represent the system. As a general rule, a volume of oil equivalent to one to two times the volume of oil contained in the sampling line and valve should be drained before the sample is taken.

2. Ideally, the sample should be taken from a valve installed specifically for sampling. When sampling valves are not installed, the taking of samples from locations where sediment or water can collect, such as dead ends of piping, tank drains, and low points of large pipes and filter bowls, should be avoided if possible. If samples are taken from pipe drains, sufficient fluid should be drained before the sample is taken to ensure that the sample actually represents the system. Samples are not to be taken from the tops of reservoirs or other locations where the contamination levels are normally low.

3. Unless otherwise specified, a minimum of one sample should be taken for each system located wholly within one compartment. For ship's systems extending into two or more compartments, a second sample is required. An exception to this requirement is submarine external hydraulic systems, which require only one sample. Original sample points should be labeled and the same sample points used for successive sampling. If possible, the following sampling locations should be selected:

a. A location that provides a sample representative of fluid being supplied to system components
b. A return line as close to the supply tank as practical but upstream of any return line filter
c. For systems requiring a second sample, a location as far from the pump as practical

Operation of the sampling point should not introduce any significant amount of external contaminants into the collected fluid. Additional information on hydraulic fluid sampling can be found in NAVAIR 01-1A-17.

Most fluid samples are submitted to shore laboratories for analysis. NAVAIR 17-15-50-1 and *NSTM*, chapter 556, contain details on collecting, labeling, and shipping samples.

NAVAIR 01-1A-17 contains procedures for unit level, both aboard ship and ashore, testing of aviation hydraulic fluids for water, particulate, and chlorinated solvent contamination.

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CHAPTER 4

PUMPS

Pumps are used for some essential services in the Navy. Pumps supply water to the boilers, draw condensation from the condensers, supply sea water to the firemain, circulate cooling water for coolers and condensers, pump out bilges, transfer fuel, supply water to the distilling plants, and serve many other purposes. Although the pumps discussed in this chapter are used primarily in hydraulic systems, the principles of operation apply as well to the pumps used in other systems.

PURPOSE

The purpose of a hydraulic pump is to supply a flow of fluid to a hydraulic system. The pump does not create system pressure, since pressure can be created only by a resistance to the flow. As the pump provides flow, it transmits a force to the fluid. As the fluid flow encounters resistance, this force is changed into a pressure. Resistance to flow is the result of a restriction or obstruction in the path of the flow. This restriction is normally the work accomplished by the hydraulic system, but can also be restrictions of lines, fittings, and valves within the system. Thus, the pressure is controlled by the load imposed on the system or the action of a pressure-regulating device.

OPERATION

A pump must have a continuous supply of fluid available to the inlet port to supply fluid to the system. As the pump forces fluid through the outlet port, a partial vacuum or low-pressure area is created at the inlet port. When the pressure at the inlet port of the pump is lower than the local atmospheric pressure, atmospheric pressure acting on the fluid in the reservoir forces the fluid into the pump's inlet. If the pump is located at a level lower than the reservoir, the force of gravity supplements atmospheric pressure on the reservoir. Aircraft and missiles that operate at high altitudes are equipped with pressurized hydraulic reservoirs to compensate for low atmospheric pressure encountered at high altitudes.

PERFORMANCE

Pumps are normally rated by their volumetric output and pressure. Volumetric output is the amount of fluid a pump can deliver to its outlet port in a certain period of time at a given speed. Volumetric output is usually expressed in gallons per minute (gpm). Since changes in pump speed affect volumetric output, some pumps are rated by their displacement. Pump displacement is the amount of fluid the pump can deliver per cycle. Since most pumps use a rotary drive, displacement is usually expressed in terms of cubic inches per revolution.

As we stated previously, a pump does not create pressure. However, the pressure developed by the restrictions in the system is a factor that affects the volumetric output of the pump. As the system pressure increases, the volumetric output decreases. This drop in volumetric output is the result of an increase in the amount of internal leakage from the outlet side to the inlet side of the pump. This leakage is referred to as pump slippage and is a factor that must be considered in all pumps. This explains why most pumps are rated in terms of volumetric output at a given pressure.

CLASSIFICATION OF PUMPS

Many different methods are used to classify pumps. Terms such as *nonpositive displacement, positive displacement, fixed displacement, variable displacement, fixed delivery, variable delivery, constant volume,* and others are used to describe pumps. The first two of these terms describe the fundamental division of pumps; that

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is, all pumps are either nonpositive displacement or positive displacement.

Basically, pumps that discharge liquid in a continuous flow are referred to as nonpositive displacement, and those that discharge volumes separated by a period of no discharge are referred to as positive displacement.

Although the nonpositive-displacement pump normally produces a continuous flow, it does not provide a positive seal against slippage; therefore, the output of the pump varies as system pressure varies. In other words, the volume of fluid delivered for each cycle depends on the resistance to the flow. This type of pump produces a force on the fluid that is constant for each particular speed of the pump. Resistance in the discharge line produces a force in a direction opposite the direction of the force produced by the pump. When these forces are equal, the fluid is in a state of equilibrium and does not flow.

If the outlet of a nonpositive-displacement pump is completely closed, the discharge pressure will increase to the maximum for that particular pump at a specific speed. Nothing more will happen except that the pump will churn the fluid and produce heat.

In contrast to the nonpositive-displacement pump, the positive-displacement pump provides a positive internal seal against slippage. Therefore, this type of pump delivers a definite volume of fluid for each cycle of pump operation, regardless of the resistance offered, provided the capacity of the power unit driving the pump is not exceeded. If the outlet of a positive-displacement pump were completely closed, the pressure would instantaneously increase to the point at which the unit driving the pump would stall or something would break.

Positive-displacement pumps are further classified as fixed displacement or variable displacement. The fixed-displacement pump delivers the same amount of fluid on each cycle. The output volume can be changed only by changing the speed of the pump. When a pump of this type is used in a hydraulic system, a pressure regulator (unloading valve) must be incorporated in the system. A pressure regulator or unloading valve is used in a hydraulic system to control the amount of pressure in the system and to unload or relieve the pump when the desired pressure is reached. This action of a pressure regulator keeps the pump from working against a load when the hydraulic system is at maximum pressure and not functioning. During this time the pressure regulator bypasses the fluid from the pump back to the reservoir. (See chapter 6 for more detailed information concerning pressure regulators.) The pump continues to deliver a fixed volume of fluid during each cycle. Such terms as *fixed delivery*, *constant delivery*, and *constant volume* are all used to identify the fixed-displacement pump.

The variable-displacement pump is constructed so that the displacement per cycle can be varied. The displacement is varied through the use of an internal controlling device. Some of these controlling devices are described later in this chapter.

Pumps may also be classified according to the specific design used to create the flow of fluid. Practically all hydraulic pumps fall within three design classifications-centrifugal, rotary, and reciprocating. The use of centrifugal pumps in hydraulics is limited and will not be discussed in this text.

ROTARY PUMPS

All rotary pumps have rotating parts which trap the fluid at the inlet (suction) port and force it through the discharge port into the system. Gears, screws, lobes, and vanes are commonly used to move the fluid. Rotary pumps are positive displacement of the fixed displacement type.

Rotary pumps are designed with very small clearances between rotating parts and stationary parts to minimize slippage from the discharge side back to the suction side. They are designed to operate at relatively moderate speeds. Operating at high speeds causes erosion and excessive wear which results in increased clearances.

There are numerous types of rotary pumps and various methods of classification. They may be classified by the shaft position—either vertically or horizontally mounted; the type of drive—electric motor, gasoline engine, and so forth; their manufacturer's name; or their service application. However, classification of rotary pumps is generally made according to the type of rotating element. A few of the most common types of rotary pumps are discussed in the following paragraphs.

GEAR PUMPS

Gear pumps are classified as either external or internal gear pumps. In external gear pumps the teeth of both gears project outward from their

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centers (fig, 4-1). External pumps may use spur gears, herringbone gears, or helical gears to move the fluid. In an internal gear pump, the teeth of one gear project outward, but the teeth of the other gear project inward toward the center of the pump (fig. 4-2, view A). Internal gear pumps may be either centered or off-centered.

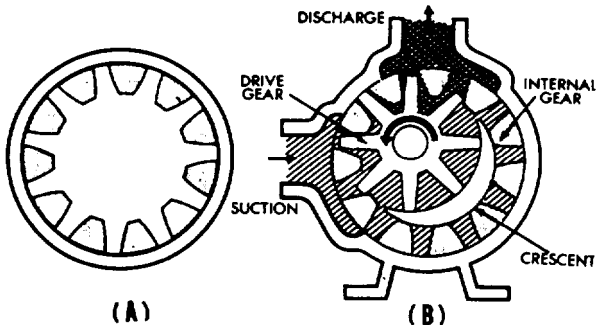

Figure 4-2.—Off-centered internal gear pump.

Spur Gear Pump

The spur gear pump (fig. 4-1) consists of two meshed gears which revolve in a housing. The drive gear in the illustration is turned by a drive shaft which is attached to the power source. The clearances between the gear teeth as they mesh and between the teeth and the pump housing are very small.

The inlet port is connected to the fluid supply line, and the outlet port is connected to the pressure line. In figure 4-1 the drive gear is turning in a counterclockwise direction, and the driven (idle) gear is turning in a clockwise direction. As the teeth pass the inlet port, liquid is trapped between the teeth and the housing. This liquid is carried around the housing to the outlet port. As the teeth mesh again, the liquid between the teeth is pushed into the outlet port. This action produces a positive flow of liquid into the system. A shearpin or shear section is incorporated in the drive shaft. This is to protect the power source

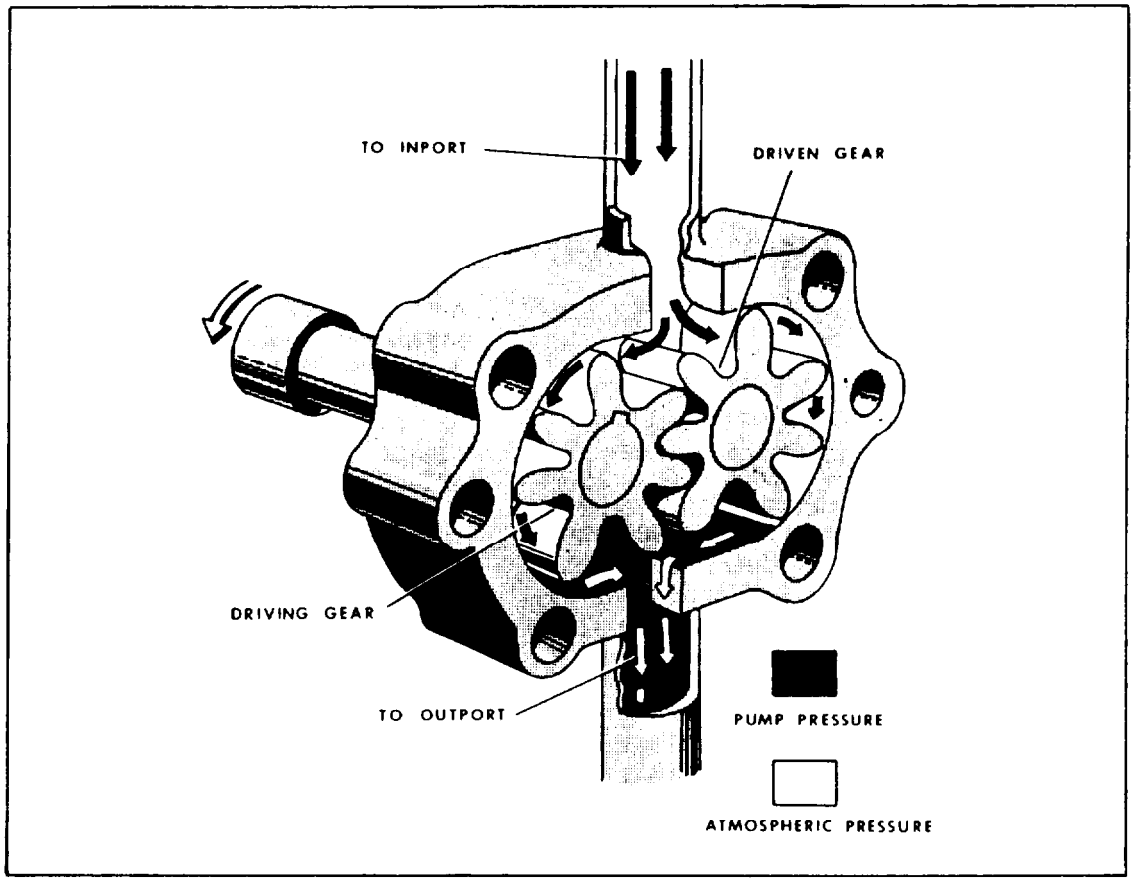

Figure 4-1.—Gear-type rotary pump.

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or reduction gears if the pump fails because of excessive load or jamming of parts.

Herringbone Gear Pump

The herringbone gear pump (fig. 4-3) is a modification of the spur gear pump. The liquid is pumped in the same manner as in the spur gear pump. However, in the herringbone pump, each set of teeth begins its fluid discharge phase before the previous set of teeth has completed its discharge phase. This overlapping and the relatively larger space at the center of the gears tend to minimize pulsations and give a steadier flow than the spur gear pump.

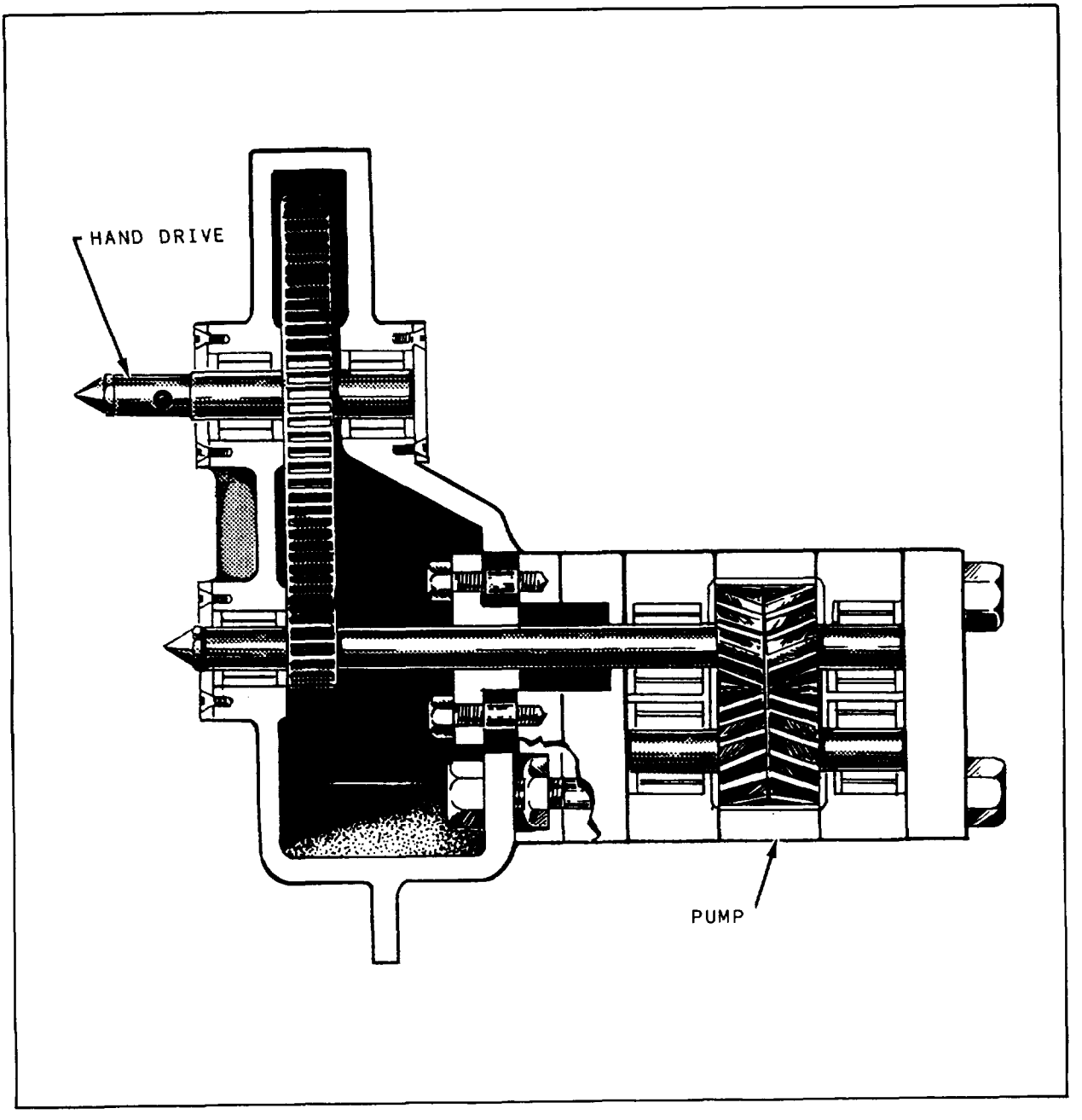

Figure 4-3.—Herringbone gear pump.

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Helical Gear Pump

The helical gear pump (fig. 4-4) is still another modification of the spur gear pump. Because of the helical gear design, the overlapping of successive discharges from spaces between the teeth is even greater than it is in the herringbone gear pump; therefore, the discharge flow is smoother. Since the discharge flow is smooth in the helical pump, the gears can be designed with a small number of large teeth—thus allowing increased capacity without sacrificing smoothness of flow.

The pumping gears of this type of pump are driven by a set of timing and driving gears that help maintain the required close clearances without actual metallic contact of the pumping gears. (Metallic contact between the teeth of the pumping gears would provide a tighter seal against slippage; however, it would cause rapid wear of the teeth, because foreign matter in the liquid would be present on the contact surfaces.)

Roller bearings at both ends of the gear shafts maintain proper alignment and minimize the friction loss in the transmission of power. Suitable packings are used to prevent leakage around the shaft.

Off-centered Internal Gear Pump

This pump is illustrated in figure 4-2, view B. The drive gear is attached directly to the drive shaft of the pump and is placed off-center in relation to the internal gear. The two gears mesh on one side of the pump, between the suction (inlet) and discharge ports. On the opposite side of the chamber, a crescent-shaped form fitted to a close tolerance fills the space between the two gears.

The rotation of the center gear by the drive shaft causes the outside gear to rotate, since the two are meshed. Everything in the chamber rotates except the crescent. This causes liquid to be trapped in the gear spaces as they pass the crescent. The liquid is carried from the suction port to the discharge port where it is forced out of the pump by the meshing of the gears. The size of the crescent that separates the internal and external gears determines the volume delivery of the pump. A small crescent allows more volume of liquid per revolution than a larger crescent.

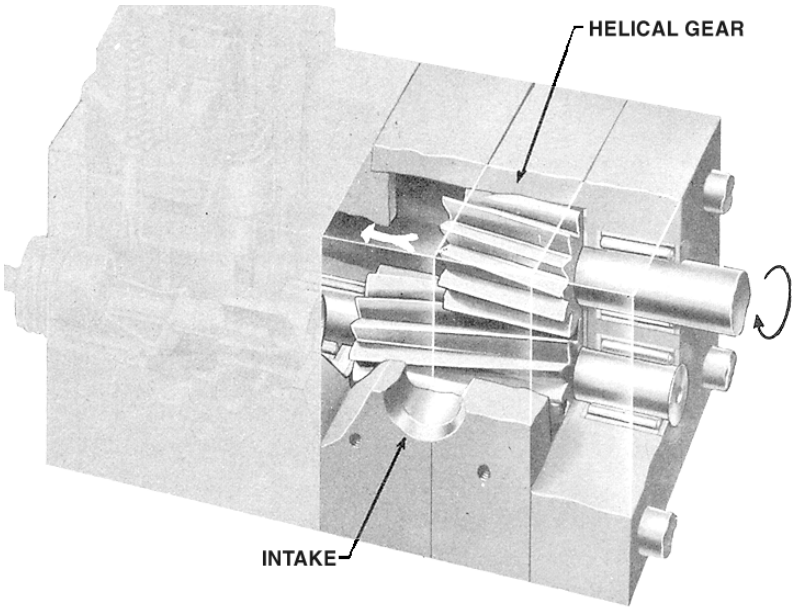

Figure 4-4.—Helical gear pump.

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Centered Internal Gear Pump

Another design of internal gear pump is illustrated in figures 4-5 and 4-6. This pump consists of a pair of gear-shaped elements, one within the other, located in the pump chamber. The inner gear is connected to the drive shaft of the power source.

The operation of this type of internal gear pump is illustrated in figure 4-6. To simplify the explanation, the teeth of the inner gear and the spaces between the teeth of the outer gear are numbered. Note that the inner gear has one less tooth than the outer gear. The tooth form of each gear is related to that of the other in such a way that each tooth of the inner gear is always in sliding contact with the surface of the outer gear. Each tooth of the inner gear meshes with the outer gear at just one point during each revolution. In the illustration, this point is at the X. In view A, tooth 1 of the inner gear is meshed with space 1 of the outer gear. As the gears continue to rotate in a clockwise direction and the teeth approach point X, tooth 6 of the inner gear will mesh with space 7 of the outer gear, tooth 5 with space 6, and so on. During this revolution, tooth 1 will mesh with space 2; and during the following revolution, tooth 1 will mesh with space 3. As a result, the outer gear will rotate at just six-sevenths the speed of the inner gear.

At one side of the point of mesh, pockets of increasing size are formed as the gears rotate, while on the other side the pockets decrease in size. In figure 4-6, the pockets on the right-hand side of the drawings are increasing in size toward the bottom of the illustration, while those on the left-hand side are decreasing in size toward the top of the illustration. The intake side of the pump would therefore be on the right and the discharge side on the left. In figure 4-5, since the right-hand side of the drawing was turned over to show the ports, the intake and discharge appear reversed. Actually, A in one drawing covers A in the other.

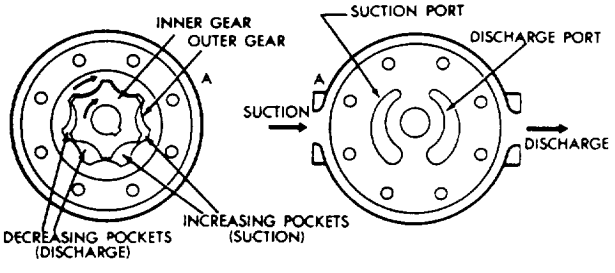

Figure 4-5.—Centered internal gear pump.

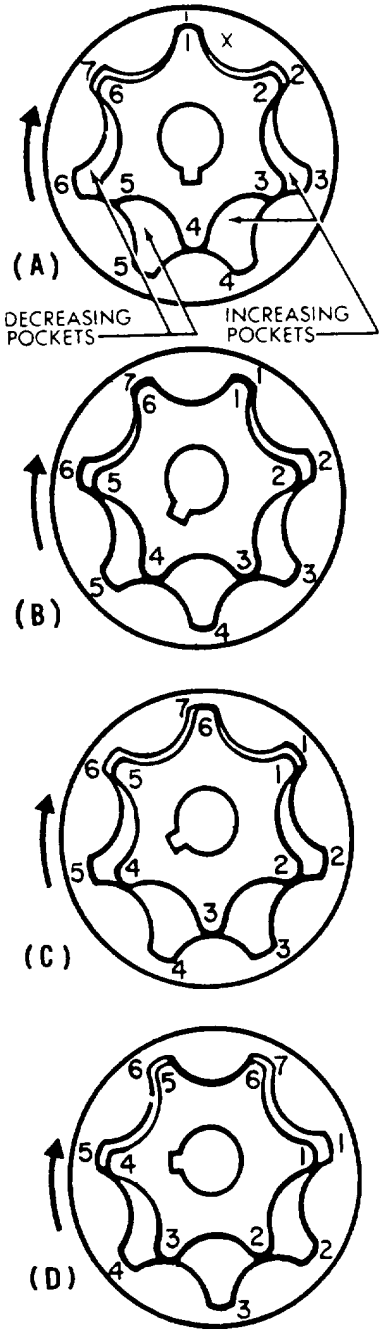

Figure 4-6.—Principles of operation of the internal gear pump.

LOBE PUMP

The lobe pump uses the same principle of operation as the external gear pump described

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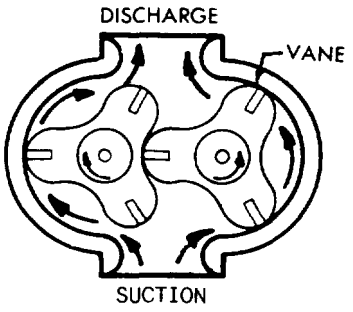

Figure 4-7.—Lobe pump.

previously. The lobes are considerably larger than gear teeth, but there are only two or three lobes on each rotor. A three-lobe pump is illustrated in figure 4-7. The two elements are rotated, one directly driven by the source of power, and the other through timing gears. As the elements rotate, liquid is trapped between two lobes of each rotor and the walls of the pump chamber and carried around from the suction side to the discharge side of the pump. As liquid leaves the suction chamber, the pressure in the suction chamber is lowered, and additional liquid is forced into the chamber from the reservoir.

The lobes are constructed so there is a continuous seal at the points where they meet at the center of the pump. The lobes of the pump illustrated in figure 4-7 are fitted with small vanes at the outer edge to improve the seal of the pump. Although these vanes are mechanically held in their slots, they are, to some extent, free to move outward. Centrifugal force keeps the vanes snug against the chamber and the other rotating members.

SCREW PUMP

Screw pumps for power transmission systems are generally used only on submarines. Although low in efficiency and expensive, the screw pump is suitable for high pressures (3000 psi), and delivers fluid with little noise or pressure pulsation.

Screw pumps are available in several different designs; however, they all operate in a similar manner. In a fixed-displacement rotary-type screw pump (fig. 4-8, view A), fluid is propelled axially

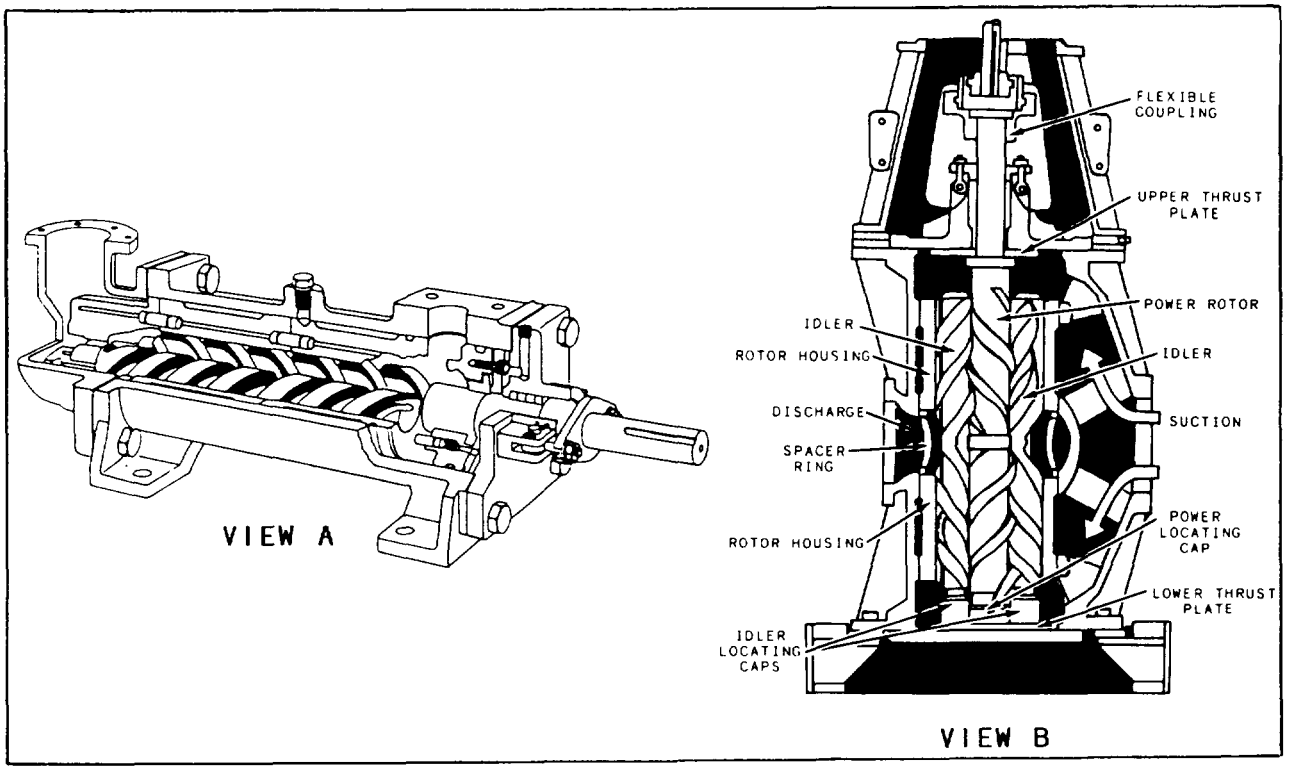

Figure 4-8.—Screw pumps.

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in a constant, uniform flow through the action of just three moving parts-a power rotor and two idler rotors. The power rotor is the only driven element, extending outside the pump casing for power connections to an electrical motor. The idler rotors are turned by the power rotor through the action of the meshing threads. The fluid pumped between the meshing helical threads of the idler and power rotors provides a protective film to prevent metal-to-metal contact. The idler rotors perform no work; therefore, they do not need to be connected by gears to transmit power. The enclosures formed by the meshing of the rotors inside the close clearance housing contain the fluid being pumped. As the rotors turn, these enclosures move axially, providing a continuous flow. Effective performance is based on the following factors:

1. The rolling action obtained with the thread design of the rotors is responsible for the very quiet pump operation. The symmetrical pressure loading around the power rotor eliminates the need for radial bearings because there are no radial loads. The cartridge-type ball bearing in the pump positions the power rotor for proper seal operation. The axial loads on the rotors created by discharge pressure are hydraulically balanced.

2. The key to screw pump performance is the operation of the idler rotors in their housing bores. The idler rotors generate a hydrodynamic film to support themselves in their bores like journal bearings. Since this film is self-generated, it depends on three operating characteristics of the pump—speed, discharge pressure, and fluid viscosity. The strength of the film is increased by increasing the operating speed, by decreasing pressure, or by increasing the fluid viscosity. This is why screw pump performance capabilities are based on pump speed, discharge pressure, and fluid viscosity.

The supply line is connected at the center of the pump housing in some pumps (fig. 4-8, view B). Fluid enters into the pump's suction port, which opens into chambers at the ends of the screw assembly. As the screws turn, the fluid flows between the threads at each end of the assembly. The threads carry the fluid along within the housing toward the center of the pump to the discharge port.

VANE PUMP

Vane-type hydraulic pumps generally have circularly or elliptically shaped interior and flat end plates. (Figure 4-9 illustrates a vane pump with a circular interior.) A slotted rotor is fixed to a shaft that enters the housing cavity through one of the end plates. A number of small rectangular plates or vanes are set into the slots of the rotor. As the rotor turns, centrifugal force causes the outer edge of each vane to slide along the surface of the housing cavity as the vanes slide in and out of the rotor slots. The numerous cavities, formed by the vanes, the end plates, the housing, and the rotor, enlarge and shrink as the rotor and vane assembly rotates. An inlet port is installed in the housing so fluid may flow into the cavities as they enlarge. An outlet port is provided to allow the fluid to flow out of the cavities as they become small.

The pump shown in figure 4-9 is referred to as an unbalanced pump because all of the pumping action takes place on one side of the rotor. This causes a side load on the rotor. Some vane pumps are constructed with an elliptically shaped housing that forms two separate pumping areas on opposite sides of the rotor. This cancels out the side loads; such pumps are referred to as balanced vane.

Usually vane pumps are fixed displacement and pump only in one direction. There are, however, some designs of vane pumps that provide variable flow. Vane pumps are generally restricted to service where pressure demand does not exceed 2000 psi. Wear rates, vibration, and noise levels increase rapidly in vane pumps as pressure demands exceed 2000 psi.

RECIPROCATING PUMPS

The term *reciprocating* is defined as back-and-forth motion. In the reciprocating pump it is this

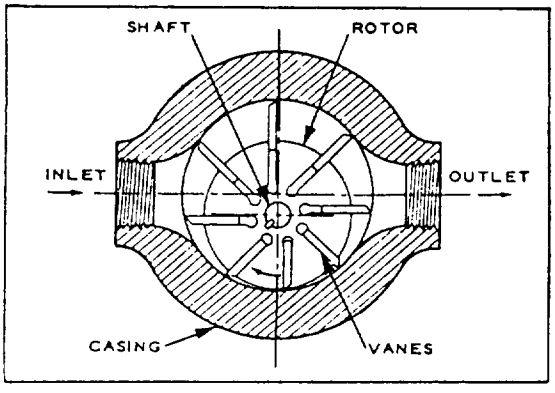

Figure 4-9.—Vane pump.

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back-and-forth motion of pistons inside of cylinders that provides the flow of fluid. Reciprocating pumps, like rotary pumps, operate on the positive principle—that is, each stroke delivers a definite volume of liquid to the system.

The master cylinder of the automobile brake system, which is described and illustrated in chapter 2, is an example of a simple reciprocating pump. Several types of power-operated hydraulic pumps, such as the radial piston and axial piston, are also classified as reciprocating pumps. These pumps are sometimes classified as rotary pumps, because a rotary motion is imparted to the pumps by the source of power. However, the actual pumping is performed by sets of pistons reciprocating inside sets of cylinders.

HAND PUMPS

There are two types of manually operated reciprocating pumps—the single-action and the double-action. The single-action pump provides flow during every other stroke, while the double-action provides flow during each stroke. Single-action pumps are frequently used in hydraulic jacks.

A double-action hand pump is illustrated in figure 4-10. This type of pump is used in some aircraft hydraulic systems as a source of hydraulic power for emergencies, for testing certain subsystems during preventive maintenance inspections, and for determining the causes of malfunctions in these subsystems.

This pump (fig. 4-10) consists of a cylinder, a piston containing a built-in check valve (A), a piston rod, an operating handle, and a check valve (B) at the inlet port. When the piston is moved to the left, the force of the liquid in the outlet chamber and spring tension cause valve A to close. This movement causes the piston to force the liquid in the outlet chamber through the outlet port and into the system. This same piston movement causes a low-pressure area in the inlet chamber. The difference in pressure between the inlet chamber and the liquid (at atmospheric pressure) in the reservior acting on check valve B causes its spring to compress; thus, opening the check valve. This allows liquid to enter the inlet chamber.

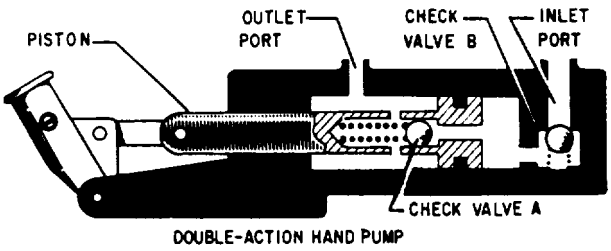

Figure 4-10.—Hydraulic hand pump.

When the piston completes this stroke to the left, the inlet chamber is full of liquid. This eliminates the pressure difference between the inlet chamber and the reservior, thereby allowing spring tension to close check valve B.

When the piston is moved to the right, the force of the confined liquid in the inlet chamber acts on check valve A. This action compresses the spring and opens check valve A which allows the liquid to flow from the intake chamber to the outlet chamber. Because of the area occupied by the piston rod, the outlet chamber cannot contain all the liquid discharged from the inlet chamber. Since liquids do not compress, the extra liquid is forced out of the outlet port into the system.

PISTON PUMPS

Piston pumps are made in a variety of types and configurations. A basic distinction is made between axial and radial pumps. The axial piston pump has the cylinders parallel to each other and the drive shaft. The radial piston design has the cylinders extending radially outward from the drive shaft like the spokes of a wheel. A further distinction is made between pumps that provide a fixed delivery and those able to vary the flow of the fluid. Variable delivery pumps can be further divided into those able to pump fluid from zero to full delivery in one direction of flow and those able to pump from zero the full delivery in either direction.

All piston pumps used in Navy shipboard systems have the cylinders bored in a cylinder block that is mounted on bearings within a housing. This cylinder block assembly rotates with the pump drive shaft.

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Radial Piston Pumps

Figure 4-11 illustrates the operation of the radial piston pump. The pump consists of a pintle, which remains stationary and acts as a valve; a cylinder block, which revolves around the pintle and contains the cylinders in which the pistons operate; a rotor, which houses the reaction ring of hardened steel against which the piston heads press; and a slide block, which is used to control the length of the piston strokes. The slide block does not revolve but houses and supports the rotor, which does revolve due to the friction set up by the sliding action between the piston heads and the reaction ring. The cylinder block is attached to the drive shaft.

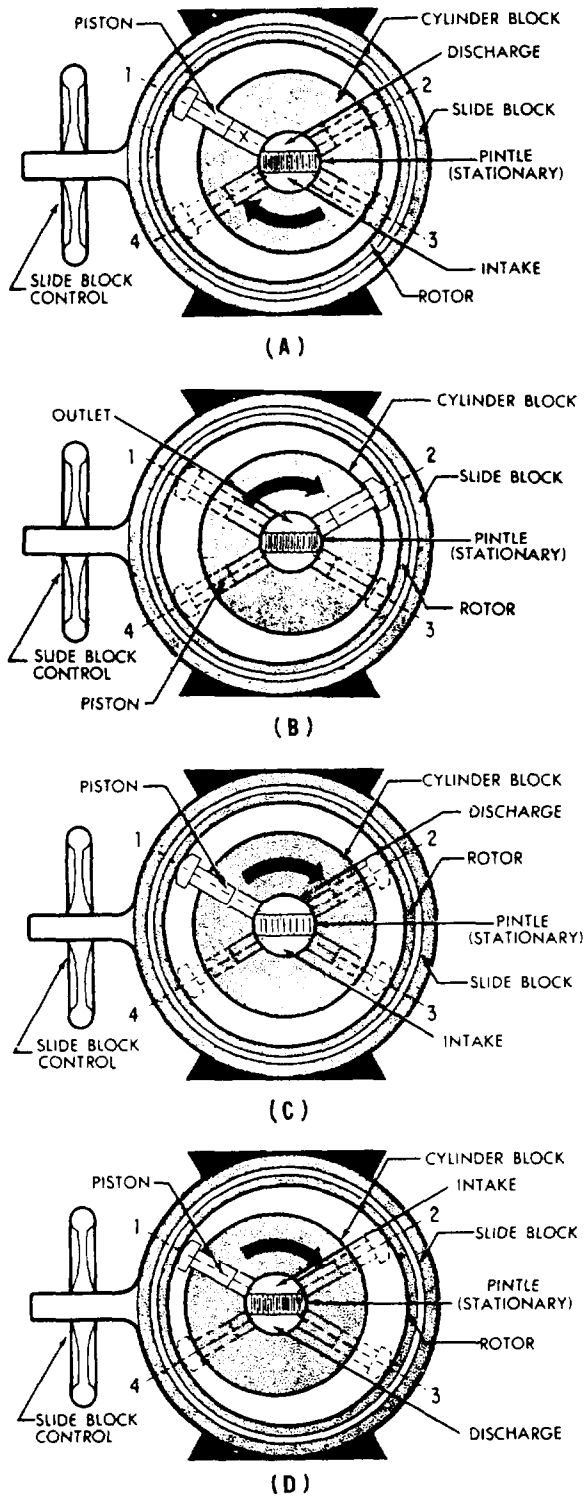

Figure 4-11.—Principles of operation of the radial piston pump.

Referring to view A of figure 4-11, assume that space X in one of the cylinders of the cylinder block contains liquid and that the respective piston of this cylinder is at position 1. When the cylinder block and piston are rotated in a clockwise direction, the piston is forced into its cylinder as it approaches position 2. This action reduces the volumetric size of the cylinder and forces a quantity of liquid out of the cylinder and into the outlet port above the pintle. This pumping action is due to the rotor being off-center in relation to the center of the cylinder block.

In figure 4-11 view B, the piston has reached position 2 and has forced the liquid out of the open end of the cylinder through the outlet above the pintle and into the system. While the piston moves from position 2 to position 3, the open end of the cylinder passes over the solid part of the pintle; therefore, there is no intake or discharge of liquid during this time. As the piston and cylinder move from position 3 to position 4, centrifugal force causes the piston to move outward against the reaction ring of the rotor. During this time the open end of the cylinder is open to the intake side of the pintle and, therefore, fills with liquid. As the piston moves from position 4 to position 1, the open end of the cylinder is against the solid side of the pintle and no intake or discharge of liquid takes place. After the piston has passed the pintle and starts toward position 2, another discharge of liquid takes place. Alternate intake and discharge continues as the rotor revolves about its axis-intake on one side of the pintle and discharge on the other, as the piston slides in and out.

Notice in views A and B of figure 4-11 that the center point of the rotor is different from the center point of the cylinder block. The difference of these centers produces the pumping action. If the rotor is moved so that its center point is the same as that of the cylinder block, as shown in figure 4-11, view C, there is no pumping action, since the piston does not move back and forth in the cylinder as it rotates with the cylinder block.

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The flow in this pump can be reversed by moving the slide block, and therefore the rotor, to the right so the relation of the centers of the rotor and the cylinder block is reversed from the position shown in views A and B of figure 4-11. View D shows this arrangement. Liquid enters the cylinder as the piston travels from position 1 to position 2 and is discharged from the cylinder as the piston travels from position 3 to 4.

In the illustrations the rotor is shown in the center, the extreme right, or the extreme left in relation to the cylinder block. The amount of adjustment in distance between the two centers determines the length of the piston stroke, which controls the amount of liquid flow in and out of the cylinder. Thus, this adjustment determines the displacement of the pump; that is, the volume of liquid the pump delivers per revolution. This adjustment may be controlled in different ways. Manual control by a handwheel is the simplest. The pump illustrated in figure 4-11 is controlled in this way. For automatic control of delivery to accommodate varying volume requirements during the operating cycle, a hydraulically controlled cylinder may be used to position the slide block. A gear-motor controlled by a push button or a limit switch is sometimes used for this purpose.

Figure 4-11 is shown with four pistons for the sake of simplicity. Radial pumps are actually designed with an odd number of pistons (fig. 4-12). This is to ensure that no more than one cylinder is completely blocked by the pintle at any one time. If there were an even number of pistons spaced evenly around the cylinder block (for example, eight), there would be occasions when two of the cylinders would be blocked by the pintle, while at other times none would be blocked. This would cause three cylinders to discharge at one time and four at one time, causing pulsations in flow. With an odd number of pistons spaced evenly around the cylinder block, only one cylinder is completely blocked by the pintle at any one time. This reduces pulsations of flow.

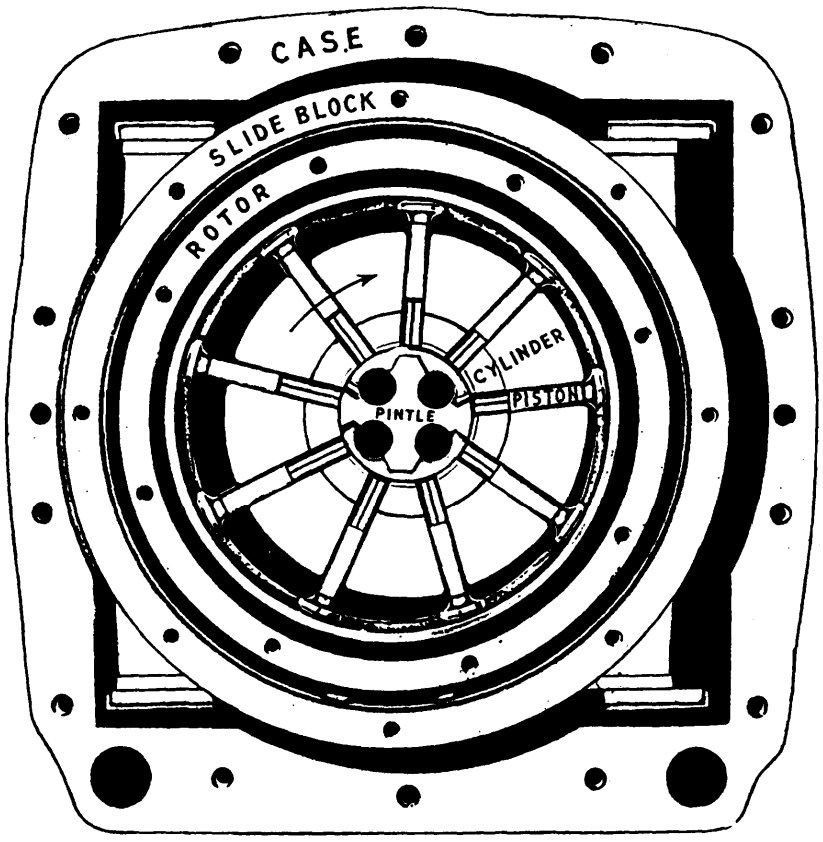

Figure 4-12.—Nine-piston radial piston pump.

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Axial Piston Pumps

In axial piston pumps of the in-line type, where the cylinders and the drive shaft are parallel (fig. 4-13), the reciprocating motion is created by a cam plate, also known as a wobble plate, tilting plate, or swash plate. This plate lies in a plane that cuts across the center line of the drive shaft and cylinder barrel and does not rotate. In a fixed-displacement pump, the cam plate will be rigidly mounted in a position so that it intersects the center line of the cylinder barrel at an angle approximately 25 degrees from perpendicular. Variable-delivery axial piston pumps are designed so that the angle that the cam plate makes with a perpendicular to the center line of the cylinder barrel may be varied from zero to 20 or 25 degrees to one or both sides. One end of each piston rod is held in contact with the cam plate as the cylinder block and piston assembly rotates with the drive shaft. This causes the pistons to reciprocate within the cylinders. The length of the piston stroke is proportional to the angle that the cam plate is set from perpendicular to the center line of the cylinder barrel.

A variation of axial piston pump is the bent-axis type shown in figure 4-14. This type does not have a tilting cam plate as the in-line pump does. Instead, the cylinder block axis is varied from the drive shaft axis. The ends of the connecting rods are retained in sockets on a disc that turns with the drive shaft. The cylinder block is turned with the drive shaft by a universal joint assembly at the intersection of the drive shaft and the cylinder block shaft. In order to vary the pump displacement, the cylinder block and valve plate are mounted in a yoke and the entire assembly is swung in an are around a pair of mounting pintles attached to the pump housing.

The pumping action of the axial piston pump is made possible by a universal joint or link.

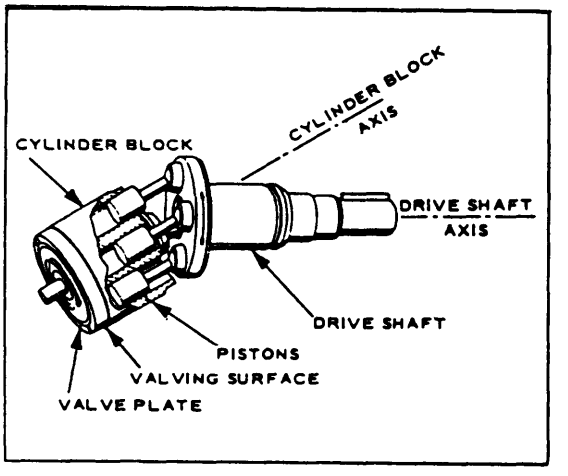

Figure 4-14.—Bent-axis axial piston pump.

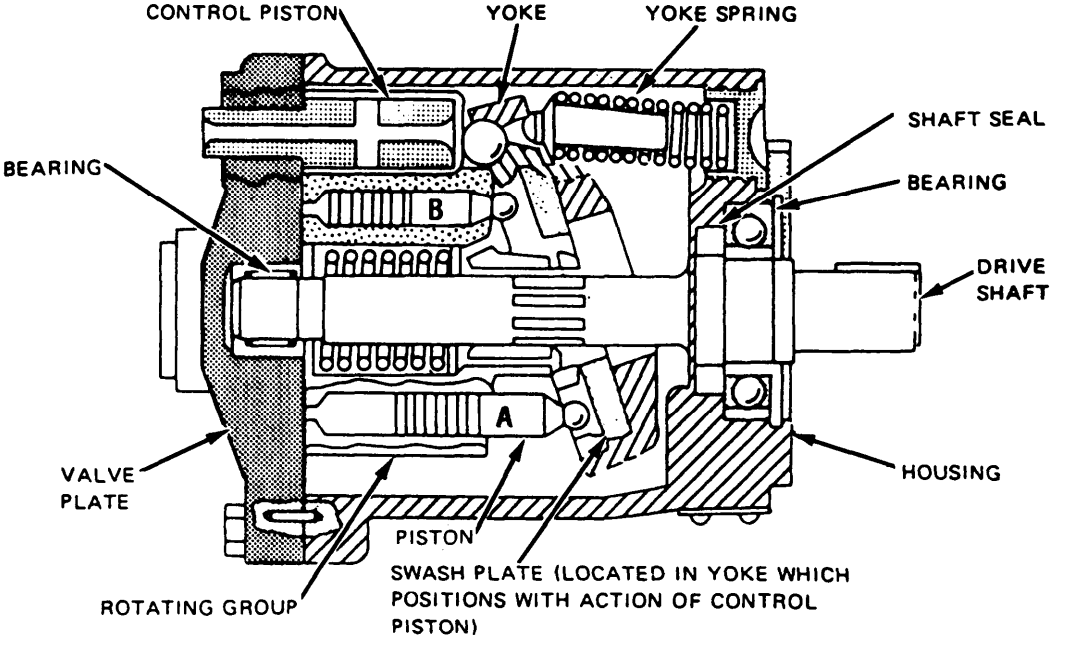

Figure 4-13.—In-line axial piston pump.

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Figure 4-15 is a series of drawings that illustrates how the universal joint is used in the operation of this pump.

First, a rocker arm is installed on a horizontal shaft. (See fig. 4-15, view A.) The arm is joined to the shaft by a pin so that it can be swung back and forth, as indicated in view B. Next, a ring is placed around the shaft and secured to the rocker arm so the ring can turn from left to right as shown in view C. This provides two rotary motions in different planes at the same time and in varying proportions as may be desired. The rocker arm can swing back and forth in one arc, and the ring can simultaneously move from left to right in another arc, in a plane at right angles to the plane in which the rocker arm turns.

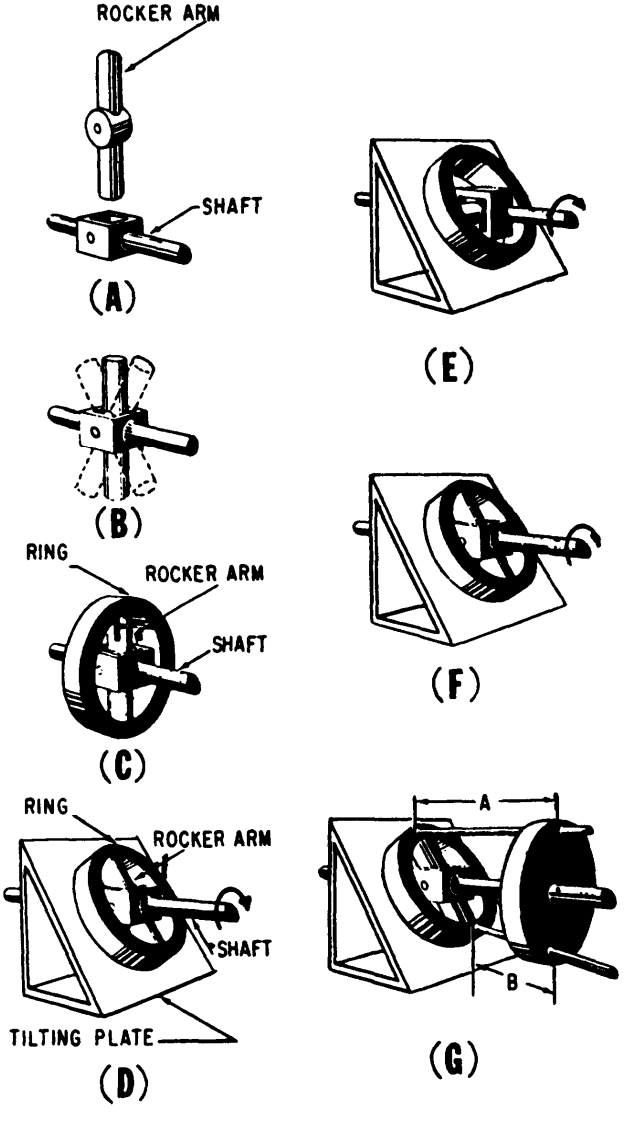

Figure 4-15.–Relationship of the universal joint in operation of the axial piston pump.

Next, a tilting plate is added to the assembly. The tilting plate is placed at a slant to the axis of the shaft, as depicted in figure 4-15, view D. The rocker arm is then slanted at the same angle as the tilting plate, so that it lies parallel to the tilting plate. The ring is also parallel to, and in contact with, the tilting plate. The position of the ring in relation to the rocker arm is unchanged from that shown in figure 4-15, view C.

Figure 4-15, view E, shows the assembly after the shaft, still in a horizontal position, has been rotated a quarter turn. The rocker arm is still in the same position as the tilting plate and is now perpendicular to the axis of the shaft. The ring has turned on the rocker pins, so that it has changed its position in relation to the rocker arm, but it remains parallel to, and in contact with, the tilting plate.

View F of figure 4-15 shows the assembly after the shaft has been rotated another quarter turn. The parts are now in the same position as shown in view D, but with the ends of the rocker arm reversed. The ring still bears against the tilting plate.

As the shaft continues to rotate, the rocker arm and the ring turn about their pivots, with each changing its relation to the other and with the ring always bearing on the plate.

Figure 4-15, view G, shows a wheel added to the assembly. The wheel is placed upright and fixed to the shaft, so that it rotates with the shaft. In addition, two rods, A and B, are loosely connected to the tilting ring and extend through two holes standing opposite each other in the fixed wheel. As the shaft is rotated, the fixed wheel turns perpendicular to the shaft at all times. The tilting ring rotates with the shaft and always remains tilted, since it remains in contact with the tilting plate. Referring to view G, the distance along rod A, from the tilting ring to the fixed wheel, is greater than the distance along rod B. As the assembly is rotated, however, the distance along rod A decreases as its point of attachment to the tilting ring moves closer to the fixed wheel, while the distance along rod B increases. These changes continue until after a half revolution, at which time the initial positions of the rods have been reversed. After another half revolution, the two rods will again be in their original positions.

As the assembly rotates, the rods move in and out through the holes in the fixed wheel. This is the way the axial piston pump works. To get a pumping action, place pistons at the ends of the

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rods, beyond the fixed wheel, and insert them into cylinders. The rods must be connected to the pistons and to the wheel by ball and socket joints. As the assembly rotates, each piston moves back and forth in its cylinder. Suction and discharge lines can be arranged so that liquid enters the cylinders while the spaces between the piston heads and the bases of the cylinders are increasing, and leaves the cylinders during the other half of each revolution when the pistons are moving in the opposite direction.

The main parts of the pump are the drive shaft, pistons, cylinder block, and valve and swash plates. There are two ports in the valve plate. These ports connect directly to openings in the face of the cylinder block. Fluid is drawn into one port and forced out the other port by the reciprocating action of the pistons.

IN-LINE VARIABLE-DISPLACEMENT AXIAL PISTON PUMP.— When the drive shaft is rotated, it rotates the pistons and the cylinder block with it. The swash plate placed at an angle causes the pistons to move back and forth in the cylinder block while the shaft, piston, cylinder block, and swash plate rotate together. (The shaft, piston, cylinder block, and swash plate together is sometimes referred to as the rotating group or assembly.) As the pistons reciprocate in the cylinder block, fluid enters one port and is forced out the other.

Figure 4-13 shows piston A at the bottom of its stroke. When piston A has rotated to the position held by piston B, it will have moved upward in its cylinder, forcing fluid through the outlet port during the entire distance. During the remainder of the rotation back to it original position, the piston travels downward in the cylinder. This action creates a low-pressure area in the cylinder. The difference in pressure between the cylinder inlet and the reservoir causes fluid to flow into the inlet port to the cylinder. Since each one of the pistons performs the same operation in succession, fluid is constantly being taken into the cylinder bores through the inlet port and discharged from the cylinder bores into the system. This action provides a steady, nonpulsating flow of fluid.

The tilt or angle of the swash plate determines the distance the pistons move back and forth in their cylinders; thereby, controlling the pump output.

When the swash plate is at a right angle to the shaft, and the pump is rotating, the pistons do not reciprocate; therefore, no pumping action takes place. When the swash plate is tilted away from a right angle, the pistons reciprocate and fluid is pumped.

Since the displacement of this type of pump is varied by changing the angle of the tilting box, some means must be used to control the changes of this angle. Various methods are used to control this movement—manual, electric, pneumatic, or hydraulic.

STRATOPOWER PUMP.— Another type of axial piston pump, sometimes referred to as an in-line pump, is commonly referred to as a Stratopower pump. This pump is available in either the fixed-displacement type or the variable-displacement type.

Two major functions are performed by the internal parts of the fixed-displacement Stratopower pump. These functions are mechanical drive and fluid displacement.

The mechanical drive mechanism is shown in figure 4-16. In this type of pump, the pistons and block do not rotate. Piston motion is caused by rotating the drive cam displacing each piston the full height of the drive cam during each revolution of the shaft. The ends of the pistons are attached to a wobble plate supported by a freed center pivot and are held inconstant contact with the cam face. As the high side of the rotating drive cam depresses one side of the wobble plate, the other side of the wobble plate is withdrawn an equal amount, moving the pistons with it. The two creep plates are provided to decrease wear on the revolving cam.

A schematic diagram of the displacement of fluid is shown in figure 4-17. Fluid is displaced by axial motion of the pistons. As each piston advances in its respective cylinder block bore, pressure opens the check valve and a quantity of fluid is forced past it. Combined back pressure and check valve spring tension close the check

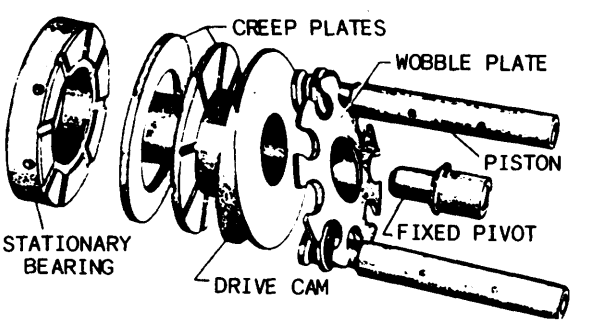

Figure 4-16.—Mechanical drive—Stratopower pump.

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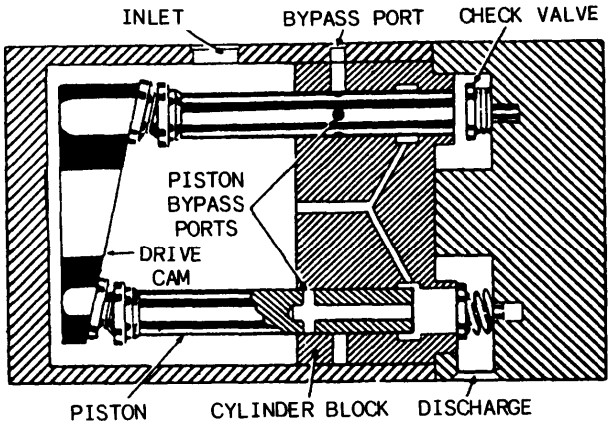

Figure 4-17.—Fluid displacement—Stratopower pump.

valve when the piston advances to its foremost position. The low-pressure area occurring in the cylinder during the piston return causes fluid to flow from the reservoir into the cylinder.

The internal features of the variable-displacement Stratopower pump are illustrated in figure 4-18. This pump operates similarly to the fixed-displacement Stratopower pump; however, this pump provides the additional function of automatically varying the volume output.

This function is controlled by the pressure in the hydraulic system. For example, let us take a pump rated at 3000 psi, and providing flow to a 3000 psi system. As system pressure approaches, say 2850 psi, the pump begins to unload (deliver less flow to the system) and is fully unloaded (zero flow) at 3000 psi.

The pressure regulation and flow are controlled by internal bypasses that automatically adjust fluid delivery to system demands.

The bypass system is provided to supply self-lubrication, particularly when the pump is in nonflow operation. The ring of bypass holes in the pistons are aligned with the bypass passage each time a piston reaches the very end of its forward travel. This pumps a small quantity of fluid out of the bypass passage back to the supply reservoir and provides a constant changing of fluid in the pump. The bypass is designed to pump against a considerable back pressure for use with pressurized reservoirs.

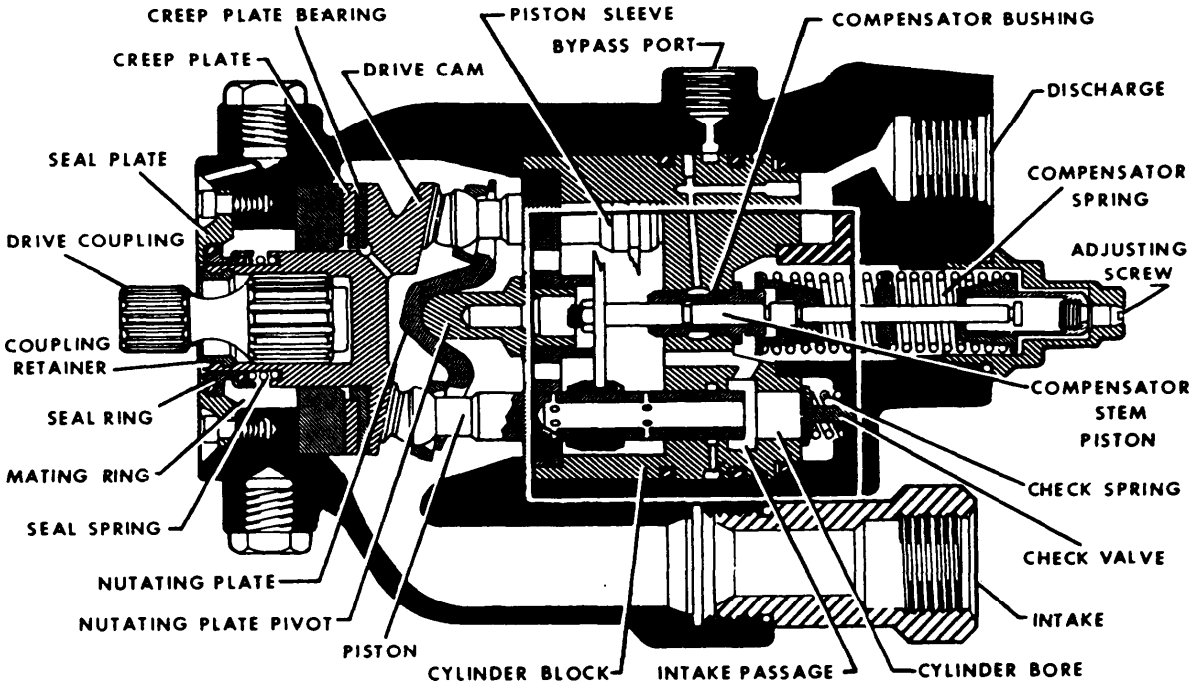

Figure 4-18.—Internal features of Stratopower variable-displacement pump.

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CHAPTER 5

FLUID LINES AND FITTINGS

The control and application of fluid power would be impossible without suitable means of transferring the fluid between the reservoir, the power source, and the points of application. Fluid lines are used to transfer the fluid, and fittings are used to connect the lines to the power source and the points of application.

This chapter is devoted to fluid lines and fittings. After studying this chapter, you should have the knowledge to identify the most commonly used lines and fittings, and be able to explain the procedure for fabricating, testing, and labeling the lines.

TYPES OF LINES

The three types of lines used in fluid power systems are pipe (rigid), tubing (semirigid), and hose (flexible). A number of factors are considered when the type of line is selected for a particular fluid system. These factors include the type of fluid, the required system pressure, and the location of the system. For example, heavy pipe might be used for a large stationary fluid power system, but comparatively lightweight tubing must be used in aircraft and missile systems because weight and space are critical factors. Flexible hose is required in installations where units must be free to move relative to each other.

PIPES AND TUBING

There are three important dimensions of any tubular product—outside diameter (OD), inside diameter (ID), and wall thickness. Sizes of pipe are listed by the nominal (or approximate) ID and the wall thickness. Sizes of tubing are listed by the actual OD and the wall thickness.

SELECTION OF PIPES AND TUBING

The material, ID, and wall thickness are the three primary considerations in the selection of lines for a particular fluid power system.

The ID of a line is important, since it determines how much fluid can pass through the line in a given time period (rate of flow) without loss of power due to excessive friction and heat. The velocity of a given flow is less through a large opening than through a small opening. If the ID of the line is too small for the amount of flow, excessive turbulence and friction heat cause unnecessary power loss and overheated fluid.

Sizing of Pipes and Tubing

Pipes are available in three different weights: standard (STD), or Schedule 40; extra strong (XS), or Schedule 80; and double extra strong (XXS). The schedule numbers range from 10 to 160 and cover 10 distinct sets of wall thickness. (See table 5-1.) Schedule 160 wall thickness is slightly thinner than the double extra strong.

As mentioned earlier, the size of pipes is determined by the nominal (approximate) ID. For example, the ID for a 1/4-inch Schedule 40 pipe is 0.364 inch, and the ID for a 1/2-inch Schedule 40 pipe is 0.622 inch.

It is important to note that the IDs of all pipes of the same nominal size are not equal. This is because the OD remains constant and the wall thickness increases as the schedule number increases. For example, a nominal size 1-inch Schedule 40 pipe has a 1.049 ID. The same size Schedule 80 pipe has a 0.957 ID, while Schedule

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Table 5-1.—Wall Thickness Schedule Designations for Pipe

Nominal size	Pipe OD	Inside diameter									
		Sched. 10	Sched. 20	Sched. 30	Sched. 40	Sched. 60	Sched. 80	Sched. 100	Sched. 120	Sched. 140	Sched. 160
1/8	0.405				0.269		0.215				
1/4	.540				.364		.302				
3/8	.675				.493		.423				
1/2	.840				.622		.546				0.466
3/4	1.050				.824		.742				.614
1	1.315				1.049		.957				.815
1 1/4	1.660				1.380		1.278				1.160
1 1/2	1.900				1.610		1.500				1.338
2	2.375				2.067		1.939				1.689

160 pipe has a 0.815 ID. In each case the OD is 1.315 (table 5-1) and the wall thicknesses are 0.133 $\left(\frac{1.315-1.049}{2}\right)$, 0.179 $\left(\frac{1.315-9.957}{2}\right)$, and 0.250 $\left(\frac{1.315-0.815}{2}\right)$ respectively. Note that the difference between the OD and ID includes two wall thicknesses and must be divided by 2 to obtain the wall thickness.

Tubing differs from pipe in its size classification. Tubing is designated by its actual OD. (See table 5-2.) Thus, 5/8-inch tubing has an OD of 5/8 inch. As indicated in the table, tubing is available in a variety of wall thicknesses. The diameter of tubing is often measured and indicated in 16ths. Thus, No. 6 tubing is 6/16 or 3/8 inch, No. 8 tubing is 8/16 or 1/2 inch, and so forth.

The wall thickness, material used, and ID determine the bursting pressure of a line or fitting. The greater the wall thickness in relation to the ID and the stronger the metal, the higher the bursting pressure. However, the greater the ID for a given wall thickness, the lower the bursting pressure, because force is the product of area and pressure.

Materials

The pipe and tubing used in fluid power systems are commonly made from steel, copper, brass, aluminum, and stainless steel. Each of these metals has its own distinct advantages or disadvantages in certain applications.

Steel pipe and tubing are relatively inexpensive and are used in many hydraulic and pneumatic systems. Steel is used because of its strength, suitability for bending and flanging, and adaptability to high pressures and temperatures. Its chief disadvantage is its comparatively low resistance to corrosion.

Copper pipe and tubing are sometimes used for fluid power lines. Copper has high resistance to corrosion and is easily drawn or bent. However, it is unsatisfactory for high temperatures and has a tendency to harden and break due to stress and vibration.

Aluminum has many of the characteristics and qualities required for fluid power lines. It has high resistance to corrosion and is easily drawn or bent. In addition, it has the outstanding characteristic of light weight. Since weight elimination is a vital factor in the design of aircraft, aluminum alloy tubing is used in the majority of aircraft fluid power systems.

Stainless-steel tubing is used in certain areas of many aircraft fluid power systems. As a general rule, exposed lines and lines subject to abrasion or intense heat are made of stainless steel.

An improperly piped system can lead to serious power loss and possible harmful fluid

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Table 5-2.—Tubing Size Designation

Tube OD	Wall thickness	Tube ID
1/8	0.028	0.069
	.032	.061
	.035	.055
3/16	0.032	0.1235
	.035	.1175
1/4	0.035	0.180
	.042	.166
	.049	.152
	.058	.134
	.065	.120
5/16	0.035	0.2425
	.042	.2285
	.049	.2145
	.058	.1965
	.065	.1825
3/8	0.035	0.305
	.042	.291
	.049	.277
	.058	.259
	.065	.245
1/2	0.035	0.430
	.042	.416
	.049	.402
	.058	.384
	.065	.370
	.072	.356
	.083	.334
	.095	.310
5/8	0.035	0.555
	.042	.541
	.049	.527
	.058	.509
	.065	.495
	.072	.481
	.083	.459
	.095	.435
3/4	0.049	0.652
	.058	.634
	.065	.620
	.072	.606
	.083	.584
	.095	.560
	.109	.532
7/8	0.049	0.777
	.058	.759
	.065	.745
	.072	.731
	.083	.709
	.095	.685
	.109	.657
1	0.049	0.902
	.058	.884
	.065	.870
	.072	.856
	.083	.834
	.095	.810
	.109	.782
	.120	.760
1 1/4	0.049	1.152
	.058	1.134
	.065	1.120
	.072	1.106
	.083	1.084
	.095	1.060
	.109	1.032
	.120	1.010
1 1/2	0.065	1.370
	.072	1.356
	.083	1.334
	.095	1.310
	.109	1.282
	.120	1.260
	.134	1.232
1 3/4	0.065	1.620
	.072	1.606
	.083	1.584
	.095	1.560
	.109	1.532
	.120	1.510
	.134	1.482
2	0.065	1.870
	.072	1.856
	.083	1.834
	.095	1.810
	.109	1.782
	.120	1.760
	.134	1.732

contamination. Therefore in maintenance and repair of fluid power system lines, the basic design requirements must be kept in mind. Two primary requirements are as follows:

1. The lines must have the correct ID to provide the required volume and velocity of flow with the least amount of turbulence during all demands on the system.

2. The lines must be made of the proper material and have the wall thickness to provide sufficient strength to both contain the fluid at the required pressure and withstand the surges of pressure that may develop in the system.

PREPARATION OF PIPES AND TUBING

Fluid power systems are designed as compactly as possible, to keep the connecting lines short. Every section of line should be anchored securely in one or more places so that neither the weight of the line nor the effects of vibration are carried on the joints. The aim is to minimize stress throughout the system.

Lines should normally be kept as short and free of bends as possible. However, tubing should not be assembled in a straight line, because a bend tends to eliminate strain by absorbing vibration and also compensates for thermal expansion and

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contraction. Bends are preferred to elbows, because bends cause less of a power loss. A few of the correct and incorrect methods of installing tubing are illustrated in figure 5-1.

Bends are described by their radius measurements. The ideal bend radius is 2 1/2 to 3 times the ID, as shown in figure 5-2. For example, if the ID of a line is 2 inches, the radius of the bend should be between 5 and 6 inches.

While friction increases markedly for sharper curves than this, it also tends to increase up to a certain point for gentler curves. The increases in friction in a bend with a radius of more than 3 pipe diameters result from increased turbulence near the outside edges of the flow. Particles of fluid must travel a longer distance in making the change in direction. When the radius of the bend is less than 2 1/2 pipe diameters, the increased pressure loss is due to the abrupt change in the direction of flow, especially for particles near the inside edge of the flow.

During your career in the Navy, you may be required to fabricate new tubing to replace damaged or failed lines. Fabrication of tubing consists of four basic operations: cutting, deburring, bending, and joint preparation.

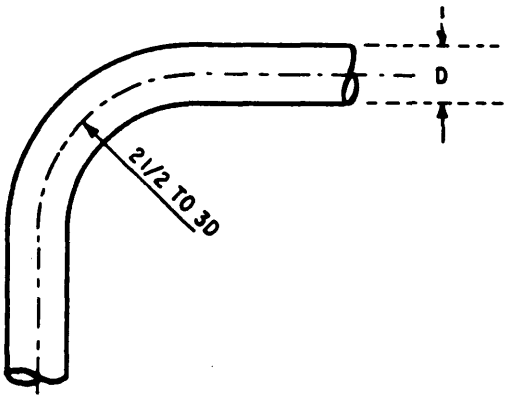

Figure 5-2.—Ideal bend radius.

Tube Cutting and Deburring

The objective of cutting tubing is to produce a square end that is free from burrs. Tubing may be cut using a standard tube cutter (fig. 5-3), a chipless cutter (fig. 5-4), or a fine-toothed hacksaw if a tube cutter is not available.

When you use the standard tube cutter, place the tube in the cutter with the cutting wheel at the point where the cut is to be made. Apply light pressure on the tube by tightening the adjusting knob. Too much pressure applied to the cutting wheel at onetime may deform the tubing or cause excessive burrs. Rotate the cutter toward its open side (fig. 5-3). As you rotate the cutter, adjust the tightening knob after each complete turn to maintain light pressure on the cutting wheel.

When you use the chipless cutter, take the following steps:

1. Select the chipless cutter according to tubing size.
2. Rotate the cutter head to accept the tubing in the cutting position. Check that the cutter ratchet is operating freely and that the cutter wheel is clear of the cutter head opening (fig. 5-4).
3. Center the tubing on two rollers and the cutting blade.
4. Use the hex key provided with the kit to turn the drive screw in until the cutter wheel touches the tube.

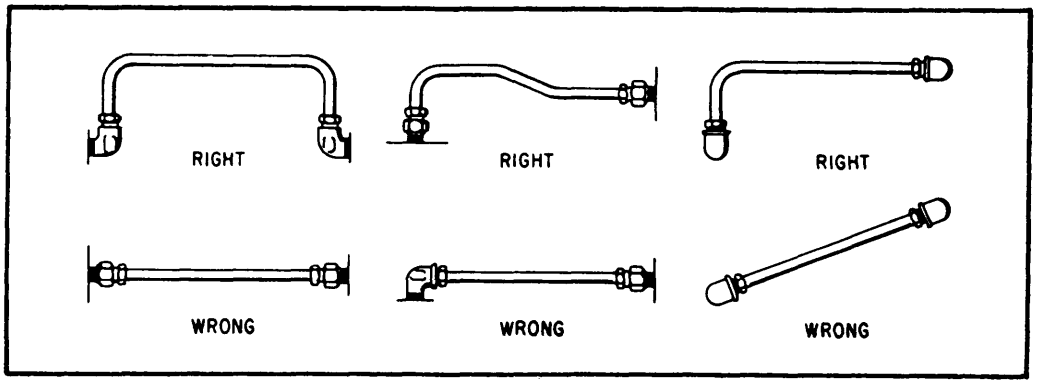

Figure 5-1.—Correct and incorrect methods of installing tubing.

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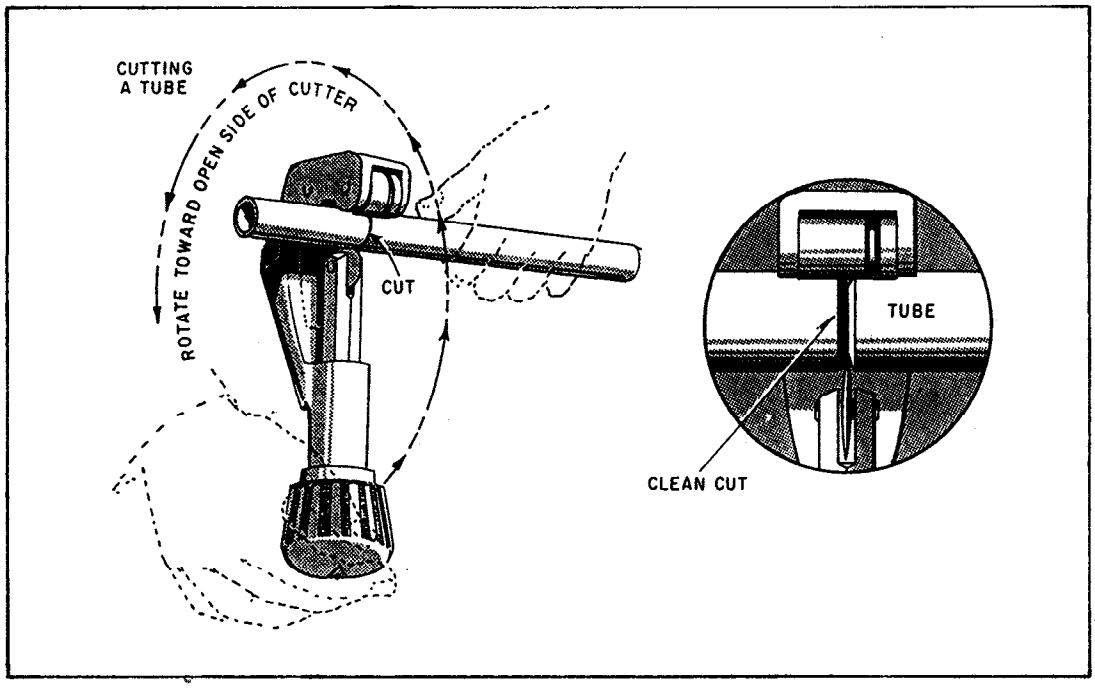

Figure 5-3.—Tube cutting.

5. Tighten the drive screw 1/8 to 1/4 turn. Do not overtighten the drive screw. Overtightening can damage soft tubing or cause excessive wear or breakage of the cutter wheel in hard tubing.

6. Swing the ratchet handle back and forth through the available clearance until there is a noticeable ease of rotation. Avoid putting side force on the cutter handle. Side force will cause the cutter wheel to break.

7. Tighten the drive screw an additional 1/8 to 1/4 turn and swing the ratchet handle back and forth, retightening the drive screw as needed until the cut is completed. The completed cut should be 1/2 degree square to the tube centerline.

After the tubing is cut, remove all burrs and sharp edges from inside and outside of the tube (fig. 5-5) with deburring tools. Clean out the tubing. Make sure no foreign particles remain.

A convenient method for cutting tubing with a hacksaw is to place the tube in a flaring block and clamp the block in a vice. After cutting the tubing with a hacksaw, remove all saw marks by filing.

Tube Bending

The objective in tube bending is to obtain a smooth bend without flattening the tube. Tube bending is usually done with either a hand tube bender or a mechanically operated bender.

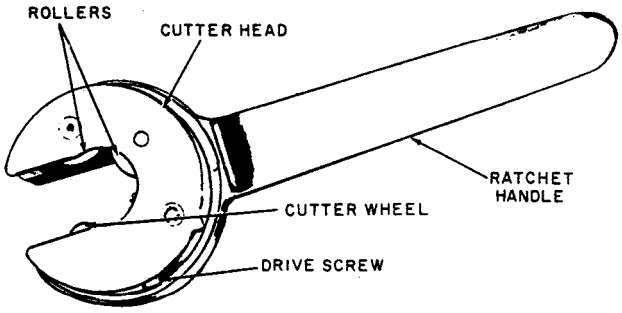

Figure 5-4.—Chipless cutter.

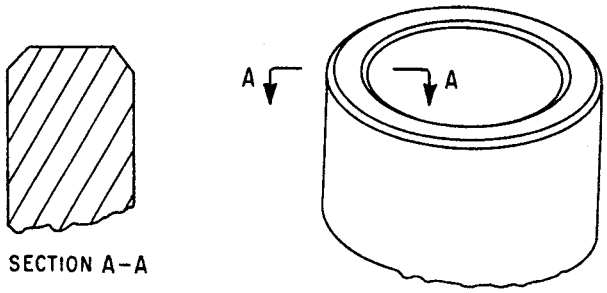

Figure 5-5.—Properly burred tubing.

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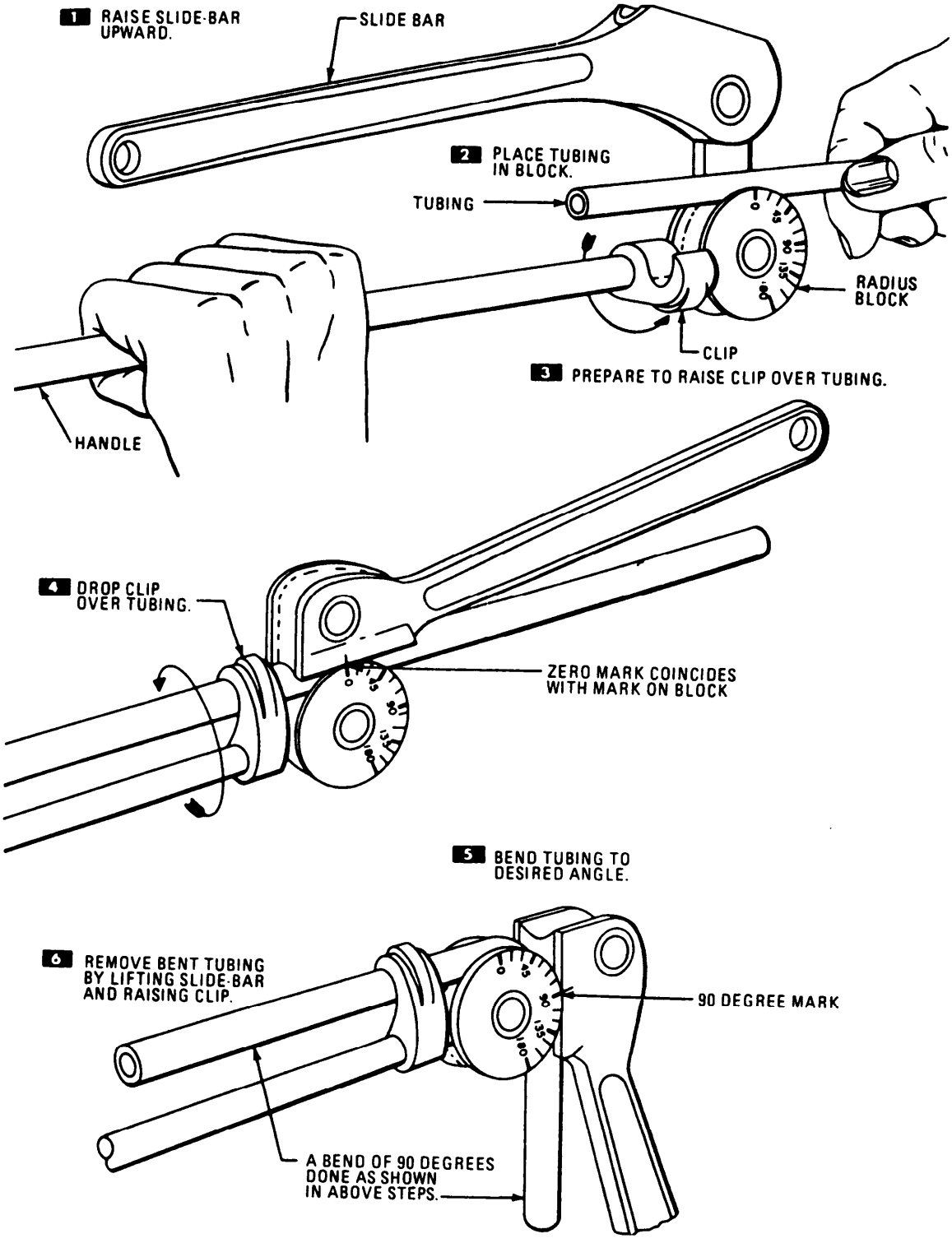

Figure 5-6.—Bending tubing with hand-operated tube bender.

HAND TUBE BENDER.— The hand tube bender shown in figure 5-6 consists of a handle, a radius block, a clip, and a slide bar. The handle and slide bar are used as levers to provide the mechanical advantage necessary to bend the tubing. The radius block is marked in degrees of bend ranging from 0 to 180 degrees. The slide bar has a mark which is lined up with the zero mark on the radius block. The tube is inserted in the tube bender, and after the marks are lined up, the

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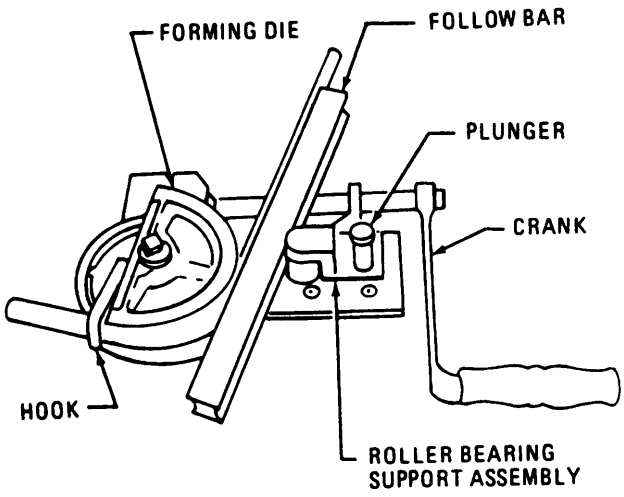

Figure 5-7.—Mechanically operated tube bender.

slide bar is moved around until the mark on the slide bar reaches the desired degree of bend on the radius block. See figure 5-6 for the six procedural steps in tube bending with the hand-operated tube bender.

MECHANICAL TUBE BENDER.— The tube bender shown in figure 5-7 is issued as a kit. The kit contains the equipment necessary for bending tubing from 1/4 inch to 3/4 inch in diameter.

This tube bender is designed for use with aircraft grade, high-strengths stainless-steel tubing, as well as all other metal tubing. It is designed to be fastened to a bench or tripod. The base is formed to provide a secure grip in a vise.

This type of tube bender uses a hand crank and gears. The forming die is keyed to the drive gear and is secured by a screw.

The forming die on the mechanical tube bender is calibrated in degrees, similarly to the radius block of the hand bender. A length of replacement tubing may be bent to a specified number of degrees or it may be bent to duplicate a bend either in a damaged tube or in a pattern. Duplicating a bend of a damaged tube or of a pattern is done by laying the sample or pattern on top of the tube being bent and slowly bending the new tube to the required bend.

Tube Flaring

Tube flaring is a method of forming the end of a tube into a funnel shape so it can be held by a threaded fitting. When a flared tube is prepared, a flare nut is slipped onto the tube and the end of the tube is flared. During tube installation, the flare is seated to a fitting with the inside of the flare against the cone-shaped end of the fitting, and the flare nut is screwed onto the fitting, pulling the inside of the flare against the seating surface of the fitting.

Either of two flaring tools (fig. 5-8) may be used. One gives a single flare and the other gives a double flare. The flaring tool consists of a split die block that has holes for various sizes of tubing,

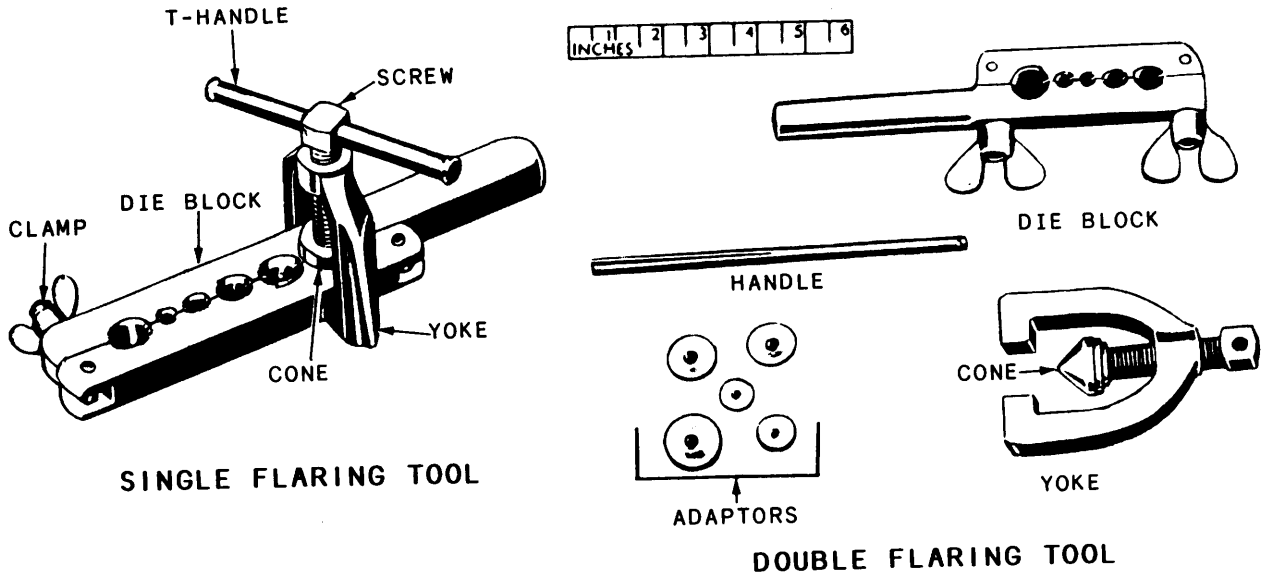

Figure 5-8.—Flaring tools.

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a clamp to lock the end of the tubing inside the die block, and a yoke with a compressor screw and cone that slips over the die block and forms the 45-degree flare on the end of the tube. The screw has a T-handle. A double flaring tube has adaptors that turn in the edge of the tube before a regular 45-degree double flare is made.

To use the single flaring tool, first check to see that the end of the tubing has been cut off squarely and has had the burrs removed from both inside and outside. Slip the flare nut onto the tube before you make the flare. Then, open the die block. Insert the end of the tubing into the hole corresponding to the OD of the tubing so that the end protrudes slightly above the top face of the die blocks. The amount by which the tubing extends above the blocks determines the finished diameter of the flare. The flare must be large enough to seat properly against the fitting, but small enough that the threads of the flare nut will slide over it. Close the die block and secure the tool with the wing nut. Use the handle of the yoke to tighten the wing nut. Then place the yoke over the end of the tubing and tighten the handle to force the cone into the end of the tubing. The completed flare should be slightly visible above the face of the die blocks.

FLEXIBLE HOSE

Shock-resistant, flexible hose assemblies are required to absorb the movements of mounted equipment under both normal operating conditions and extreme conditions. They are also used for their noise-attenuating properties and to connect moving parts of certain equipment. The two basic hose types are synthetic rubber and polytetrafluoroethylene (PTFE), such as Du Pont's Teflon® fluorocarbon resin.

Rubber hoses are designed for specific fluid, temperature, and pressure ranges and are provided in various specifications. Rubber hoses (fig. 5-9) consist of a minimum three layers; a seamless synthetic rubber tube reinforced with one or more layers of braided or spiraled cotton, wire, or synthetic fiber; and an outer cover. The inner tube is designed to withstand the attack of the fluid that passes through it. The braided or spiraled layers determine the strength of the hose. The greater the number of these layers, the greater is the pressure rating. Hoses are provided in three

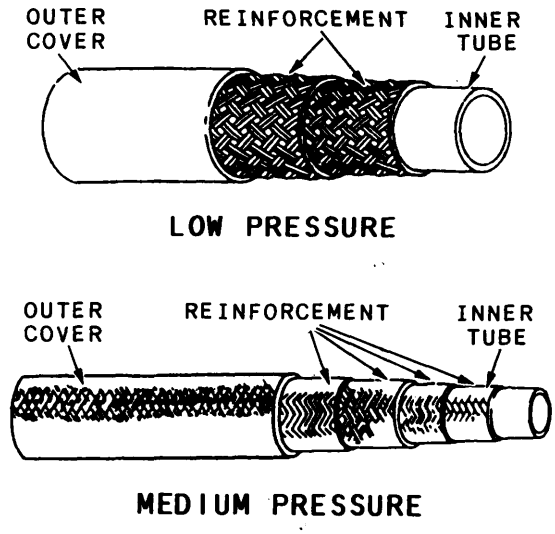

Figure 5-9.—Synthetic rubber hoses.

pressure ranges: low, medium, and high. The outer cover is designed to withstand external abuse and contains identification markings.

Synthetic rubber hoses with rubber covers are identified with the military specification number, the size by dash number, the quarter and year of cure or manufacture, and the manufacturer's code identification number or federal supply code number printed along their layline (fig. 5-10, view A). The layline is a legible marking parallel to the longitudinal axis of a hose used in determining the straightness or lay of the hose.

Synthetic rubber hoses with wire braid cover are identified by bands (fig. 5-10, view B) wrapped around the hose ends and at intervals along the length of the hose.

Sizing

The size of a flexible hose is identified by the dash (-) number, which is the ID of the hose expressed in 16ths of an inch. For example, the ID of a -64 hose is 4 inches. For a few hose styles this is the nominal and not the true ID.

Cure Date

Synthetic rubber hoses will deteriorate from aging. A cure date is used to ensure that they do not deteriorate beyond material and performance specifications. The cure date is the quarter and year the hose was manufactured. For example,

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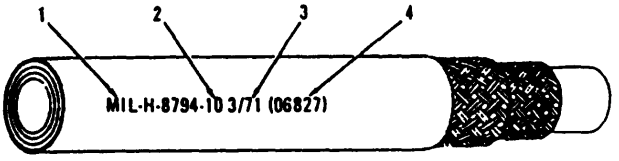

A. SYNTHETIC RUBBER HOSE

B. WIRE BRAID COVERED SYNTHETIC RUBBER HOSE

1. MILITARY SPECIFICATION OF HOSE
2. SIZE INDICATED BY A DASH (–) NO. OR FRACTION OF AN INCH FOR MIL-H-6000 AND MIL-H-7938 HOSES
3. CURE DATE FOR AGE CONTROL
4. MANUFACTURER'S FEDERAL SUPPLY CODE NO.

MANUFACTURER	AEROQUIP
MANUFACTURER'S CODE	00624
PART NO. WITH DASH (SIZE) NO.	AE206-10
LOT NO.	305496
OPERATING PRESSURE	3000 PSI
MILITARY SPECIFICATION	MIL-H-83298

C. WIRE BRAID COVERED PTFE HOSE LABEL

Figure 5-10.—Hose identification.

1Q89 or 1/89 means the hose was made during the first quarter (1 Jan to 31 Mar) of 1989.

The cure date limits the length of time a rubber hose can be stored, in bulk or as an assembly, prior to being placed into service. The storage or shelf life for rubber hose is 4 years. For the hose manufactured in 1Q89, the storage or shelf life will end on the 31st of March 1993. At this point, the hose is no longer considered usable and should be discarded or downgraded. The *Aviation Hose and Tube Manual,* NAVAIR 01-1A-20, and the *Technical Directive for Piping Devices and Flexible Hose Assemblies,* NAVSEA S6430-AE-TED-010. volume 1. provide detailed instructions on discarding and downgrading of rubber hoses exceeding their shelf life.

PFTE

PFTE hose is a flexible hose designed to meet the requirements of higher operating pressures and temperatures in present fluid power systems. This type of hose is made from a chemical resin, which is processed and extruded into a tube shaped to a desired size. It is reinforced with one or more layers of braided stainless-steel wire or with an even number of spiral wrap layers with an outer wire braid layer.

PTFE hose is unaffected by all fluids presently used in fluid power systems. It is inert to acids, both concentrated and diluted. Certain PFTE hose may be used in systems where operating temperatures range from –100°F to +500°F. PTFE is nonflammable; however, where the possibility of open flame exists, a special asbestos fire sleeve should be used.

PFTE hose will not absorb moisture. This, together with its chemical inertness and anti-adhesive characteristics, makes it ideal for missile fluid power systems where noncontamination and cleanliness are essential.

In lieu of layline marking, PTFE hoses are identified by metal or pliable plastic bands at their ends and at intervals along their length. Figure 5-10, view C, shows a hose label for a PTFE hose. Usually the only condition that will shorten the life of PTFE hose is excessive temperature. For this reason there is no manufacture date listed on the identification tag.

APPLICATION

As mentioned earlier, flexible hose is available in three pressure ranges: low, medium, and high. When replacing hoses, it is important to ensure that the replacement hose is a duplicate of the one removed in length, OD, material, type and contour, and associated markings. In selecting hose, several precautions must be observed. The selected hose must

1. be compatible with the system fluid,
2. have a rated pressure greater than the design pressure of the system,

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3. be designed to give adequate performance and service for infrequent transient pressure peaks up to 150 percent of the working pressure of the hose, and
4. have a safety factor with a burst pressure at a minimum of 4 times the rated working pressure.

There are temperature restrictions applied to the use of hoses. Rubber hose must not be used where the operating temperature exceeds 200°F. PTFE hoses in high-pressure air systems must not be used where the temperature exceeds 350°F. PTFE hoses in water and steam drain applications must not be used where the operating temperature exceeds 380°F.

FABRICATION AND TESTING

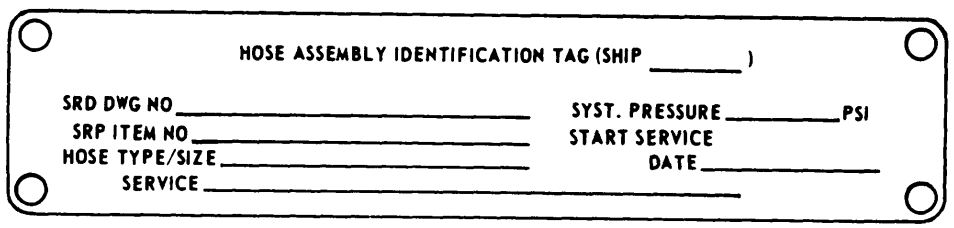
The fabrication of flexible hose assemblies is covered in applicable training manuals, technical publications, and NAVAIR 01-1A-20. After a hose assembly has been completely fabricated it must be cleaned, visually inspected for foreign materials, and proof tested.

A hose assembly is proof tested by the application of a nondestructive pressure for a minimum of 1 minute but not longer than 5 minutes to ensure that it will withstand normal working pressures. The test pressure, known as normal proof pressure, is twice the rated working pressure. While the test pressure is being applied, the hose must not burst, leak, or show signs of fitting separation. NAVAIR 01-1A-20 and NAVSEA S6430-AE-TED-010, volume 1, provide detailed instructions on cleaning of hoses, cleaning and test media, proof pressure and proof testing.

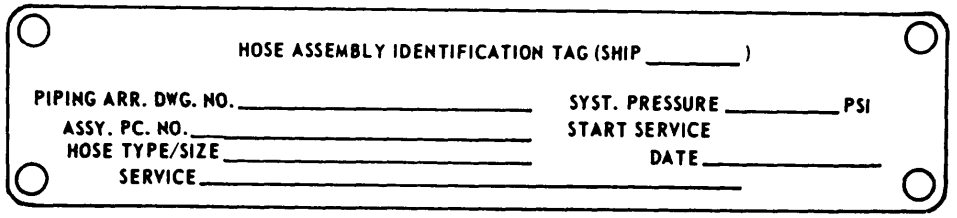
After proof testing is completed, the hose must be flushed and dried and the ends capped or plugged to keep dirt and other contaminants out of the hose.

IDENTIFICATION

The final step after fabrication and satisfactory testing of a hose assembly is the attachment of identification tags as shown in figure 5-11 (for ships) and in figure 5-12 (for aircraft). The tag shown in figure 5-12, view B, is used in areas where a tag maybe drawn into an engine intake. Hose assemblies to be installed in aircraft fuel and oil tanks are marked with an approved electric engraver on the socket-wrench flats with the required information.



ID TAG WHEN SELECTED RECORD DRAWING IS AVAILABLE



ID TAG WHEN SELECTED RECORD DRAWING DOES NOT EXIST

Figure 5-11.—Hose assembly identification tags (ships).

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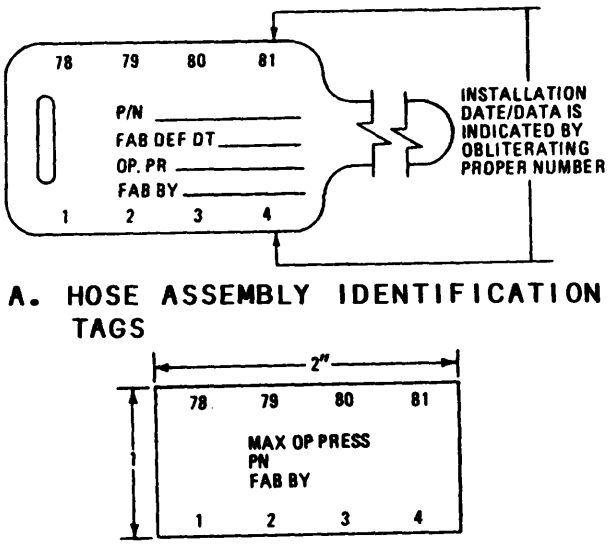

Figure 5-12.—Hose assembly identification tags (aircraft).

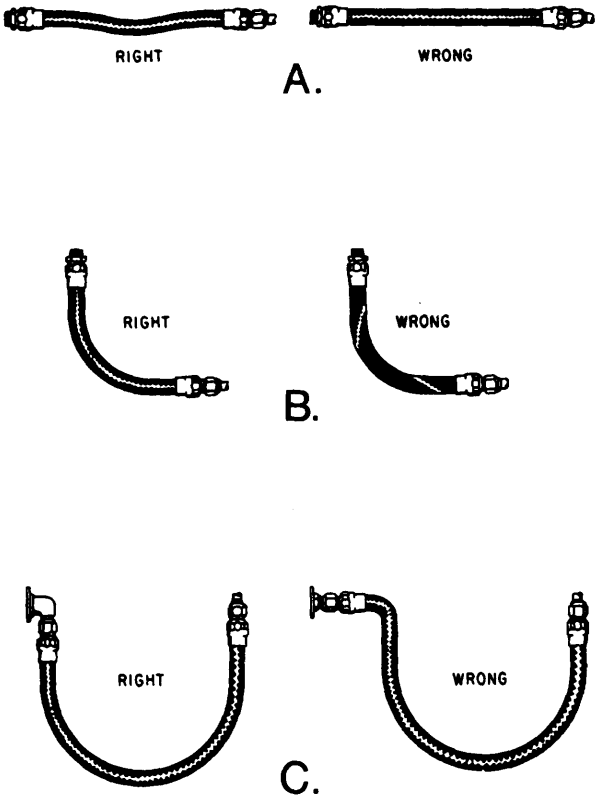

Figure 5-13.—Correct and incorrect installation of flexible hose.

INSTALLATION

Flexible hose must not be twisted during installation, since this reduces the life of the hose considerably and may cause the fittings to loosen as well. You can determine whether or not a hose is twisted by looking at the layline that runs along the length of the hose. If the layline does not spiral around the hose, the hose is not twisted. If the layline does spiral around the hose, the hose is twisted (fig. 5-13, view B) and must be untwisted.

Flexible hose should be protected from chafing by using a chafe-resistant covering wherever necessary.

The minimum bend radius for flexible hose varies according to the size and construction of the hose and the pressure under which the system operates. Current applicable technical publications contain tables and graphs showing minimum bend radii for the different types of installations. Bends that are too sharp will reduce the bursting pressure of flexible hose considerably below its rated value.

Flexible hose should be installed so that it will be subjected to a minimum of flexing during operation. Support clamps are not necessary with short installations; but for hose of considerable length (48 inches for example), clamps should be placed not more than 24 inches apart. Closer supports are desirable and in some cases may be required.

A flexible hose must never be stretched tightly between two fittings. About 5 to 8 percent of the total length must be allowed as slack to provide freedom of movement under pressure. When under pressure, flexible hose contracts in length and expands in diameter. Examples of correct and incorrect installations of flexible hose are illustrated in figure 5-13.

PFTE hose should be handled carefully during removal and installation. Some PFTE hose is preformed during fabrication. This type of hose tends to form itself to the installed position in the system. To ensure its satisfactory function and reduce the likelihood of failure, anyone who works with PFTE hose should observe the following rules:

1. Do not exceed recommended bend limits.
2. Do not exceed twisting limits.
3. Do not straighten a bent hose that has taken a permanent set.
4. Do not hang, lift, or support objects from PFTE hose.

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Once flexible hose assemblies are installed, there are no servicing or maintenance requirements other than periodic inspections. These inspections are conducted according to maintenance instruction manuals (MIMs), maintenance requirement cards (MRCs), and depot-level specifications.

TYPES OF FITTINGS AND CONNECTORS

Some type of connector or fitting must be provided to attach the lines to the components of the system and to connect sections of line to each other. There are many different types of connectors and fittings provided for this purpose. The type of connector or fitting required for a specific system depends on several factors. One determining factor, of course, is the type of fluid line (pipe, tubing, or flexible hose) used in the system. Other determining factors are the type of fluid medium and the maximum operating pressure of the system. Some of the most common types of fittings and connectors are described in the following paragraphs.

THREADED CONNECTORS

There are several different types of threaded connectors. In the type discussed in this section, both the connector and the end of the fluid line (pipe) are threaded. These connectors are used in some low-pressure fluid power systems and are usually made of steel, copper, or brass, and are available in a variety of designs.

Threaded connectors are made with standard pipe threads cut on the inside surface. The end of the pipe is threaded with outside threads. Standard pipe threads are tapered slightly to ensure tight connections. The amount of taper is approximately 3/4 inch in diameter per foot of thread.

Metal is removed when a pipe is threaded, thinning the pipe and exposing new and rough surfaces. Corrosion agents work more quickly at such points than elsewhere. If pipes are assembled with no protective compound on the threads, corrosion sets in at once and the two sections stick together so that the threads seize when disassembly is attempted. The result is damaged threads and pipes.

To prevent seizing, a suitable pipe thread compound is sometimes applied to the threads. The two end threads must be kept free of compound so that it will not contaminate the fluid. Pipe compound, when improperly applied, may get inside the lines and components and damage pumps and control equipment.

Another material used on pipe threads is sealant tape. This tape, which is made of PTFE, provides an effective means of sealing pipe connections and eliminates the necessity of torquing connections to excessively high values in order to prevent pressure leaks. It also provides for ease of maintenance whenever it is necessary to disconnect pipe joints. The tape is applied over the male threads, leaving the first thread exposed. After the tape is pressed firmly against the threads, the joint is connected.

FLANGE CONNECTORS

Bolted flange connectors (fig. 5-14) are suitable for most pressures now in use. The flanges are attached to the piping by welding, brazing, tapered threads (for some low-pressure systems), or rolling and bending into recesses. Those illustrated are the most common types of flange joints used. The same types of standard fitting shapes (tee, cross, elbow, and so forth) are manufactured for flange joints. Suitable gasket material must be used between the flanges.

WELDED CONNECTORS

The subassemblies of some fluid power systems are connected by welded joints, especially in high-pressure systems which use pipe for fluid lines. The welding is done according to standard

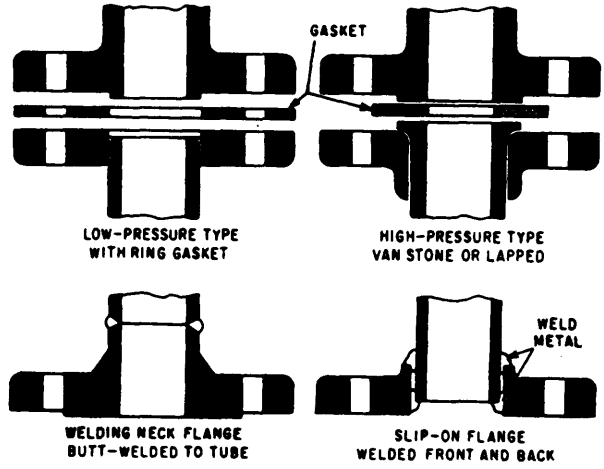

Figure 5-14.—Four types of bolted flange connectors.

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specifications which define the materials and techniques.

BRAZED CONNECTORS

Silver-brazed connectors are commonly used for joining nonferrous (copper, brass, and soon) piping in the pressure and temperature range where their use is practical. Use of this type of connector is limited to installations in which the piping temperature will not exceed 425°F and the pressure in cold lines will not-exceed 3,000 psi. The alloy is melted by heating the joint with an oxyacetylene torch. This causes the alloy insert to melt and fill the few thousandths of an inch annular space between the pipe and the fitting.

A fitting of this type which has been removed from a piping system can be rebrazed into a system, as in most cases sufficient alloy remains in the insert groove for a second joint. New alloy inserts may be obtained for fittings which do not have sufficient alloy remaining in the insert for making a new joint.

FLARED CONNECTORS

Flared connectors are commonly used in fluid power systems containing lines made of tubing. These connectors provide safe, strong, dependable connections without the need for threading, welding, or soldering the tubing. The connector consists of a fitting, a sleeve, and a nut (fig. 5-15).

The fittings are made of steel, aluminum alloy, or bronze. The fitting used in a connection should be made of the same material as that of the sleeve, the nut, and the tubing. For example, use steel connectors with steel tubing and aluminum alloy connectors with aluminum alloy tubing. Fittings are made in union, 45-degree and 90-degree elbow, tee, and various other shapes (fig. 5-16).

Tees, crosses, and elbows are self-explanatory. Universal and bulkhead fittings can be mounted solidly with one outlet of the fitting extending through a bulkhead and the other outlet(s) positioned at any angle. Universal means the fitting can assume the angle required for the specific installation. Bulkhead means the fitting is long enough to pass through a bulkhead and is designed so it can be secured solidly to the bulkhead.

For connecting to tubing, the ends of the fittings are threaded with straight machine threads to correspond with the female threads of the nut. In some cases, however, one end of the fitting may be threaded with tapered pipe threads to fit

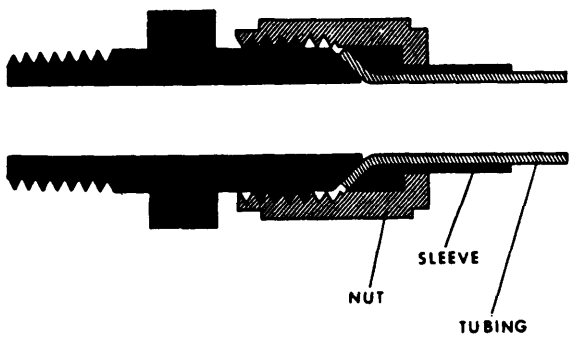

Figure 5-15.—Flared-tube fitting.

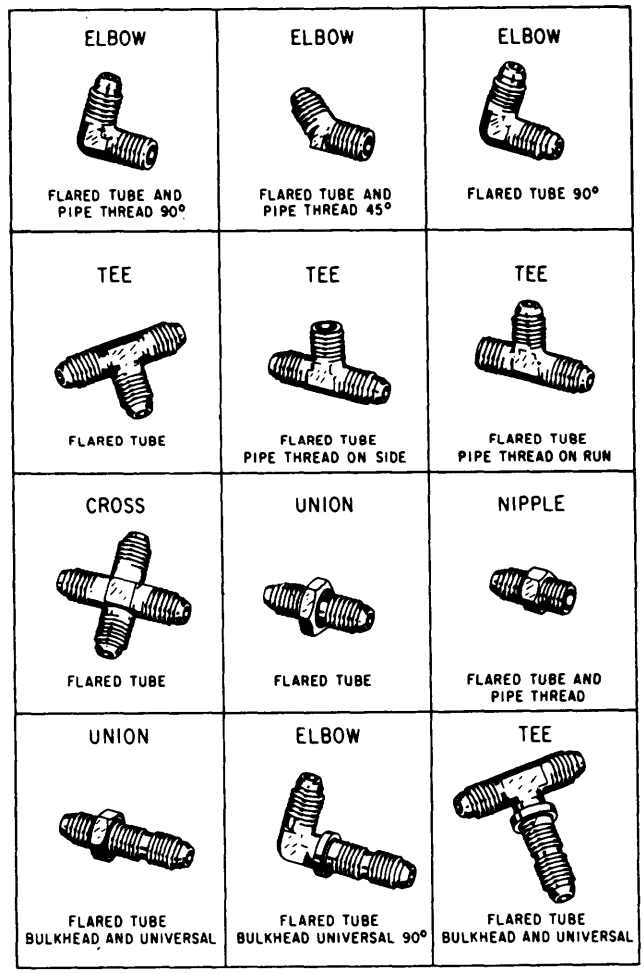

Figure 5-16.—Flared-tube fittings.

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threaded ports in pumps, valves, and other components. Several of these thread combinations are shown in figure 5-16.

Tubing used with flare connectors must be flared prior to assembly. The nut fits over the sleeve and when tightened, it draws the sleeve and tubing flare tightly against the male fitting to form a seal.

The male fitting has a cone-shaped surface with the same angle as the inside of the flare. The sleeve supports the tube so vibration does not concentrate at the edge of the flare, and distributes the shearing action over a wider area for added strength. Tube flaring is covered in *Tools and Their Uses*, NAVEDTRA 10085 (series), and other applicable training manuals.

Correct and incorrect methods of installing flared-tube connectors are illustrated in figure 5-17. Tubing nuts should be tightened with a torque wrench to the value specified in applicable technical publications.

If an aluminum alloy flared connector leaks after being tightened to the required torque, it must not be tightened further. Overtightening may severely damage or completely cut off the tubing flare or may result in damage to the sleeve or nut. The leaking connection must be disassembled and the fault corrected.

If a steel tube connection leaks, it may be tightened 1/6 turn beyond the specified torque in an attempt to stop the leakage; then if it still leaks, it must be disassembled and repaired.

Undertightening of connections may be serious, as this can allow the tubing to leak at the connector bemuse of insufficient grip on the flare by the sleeve. The use of a torque wrench will prevent undertightening.

CAUTION

A nut should never be tightened when there is pressure in the line, as this will tend to damage the connection without adding any appreciable torque to the connection.

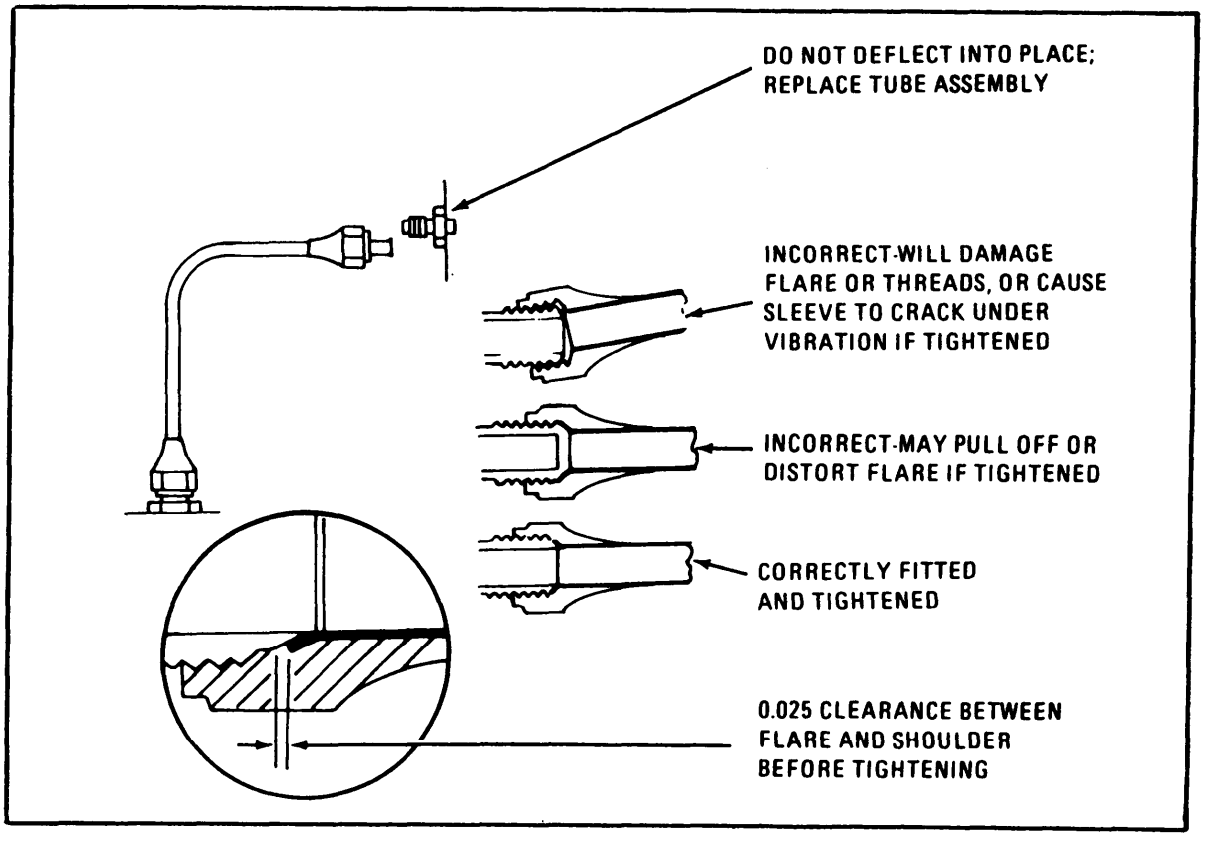

Figure 5-17.—Correct and incorrect methods of installing flared fittings.

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FLARELESS-TUBE CONNECTORS

This type of connector eliminates all tube flaring, yet provides a safe, strong, and dependable tube connection. This connector consists of a fitting, a sleeve or ferrule, and a nut. (See fig. 5-18.)

NOTE

Although the use of flareless tube connectors is widespread, NAVSEA policy is to reduce or eliminate use of flareless fittings in newly designed ships; the extent to which flareless fittings are approved for use in a particular ship is reflected in applicable ship drawings.

Flareless-tube fittings are available in many of the same shapes and thread combinations as flared-tube fittings. (See fig. 5-16.) The fitting has a counterbore shoulder for the end of the tubing to rest against. The angle of the counterbore causes the cutting edge of the sleeve or ferrule to cut into the outside surface of the tube when the two are assembled.

The nut presses on the bevel of the sleeve and causes it to clamp tightly to the tube. Resistance to vibration is concentrated at this point rather than at the sleeve cut. When fully tightened, the sleeve or ferrule is bowed slightly at the midsection and acts as a spring. This spring action of the sleeve or ferrule maintains a constant tension between the body and the nut and thus prevents the nut from loosening.

Prior to the installation of a new flareless-tube connector, the end of the tubing must be square, concentric, and free of burrs. For the connection to be effective, the cutting edge of the sleeve or ferrule must bite into the periphery of the tube (fig. 5-19). This is ensured by presetting the sleeve or ferrule on the tube.

Presetting

Presetting consists of deforming the ferrule to bite into the tube OD and deforming the end of the tube to form a shallow conical ring seating surface. The tube and ferrule assembly should be preset in a presetting tool that has an end section identical to a fitting body but which is made of specially hardened steel. This tool hardness is needed to ensure that all deformation at the tube end seat goes into the tube.

Presetting is done with a hydraulic presetting tool or a manual presetting tool, either in the shop or aboard ship. The tool vendor's instructions must be followed for the hydraulic presetting tool. If a presetting tool is not available, the fitting body intended for installation is used in the same manner as the manual presetting tool. (If an aluminum fitting is used, it should not be reused in the system.) The manual tool is used as follows:

WARNING

Failure to follow these instructions may result in improperly preset ferrules with insufficient bite into the tube. Improperly preset ferrules have resulted in joints that passed hydrostatic testing and operated for weeks or years, then failed catastrophically under shock, vibration, or normal operating loads. Flareless fitting failures have

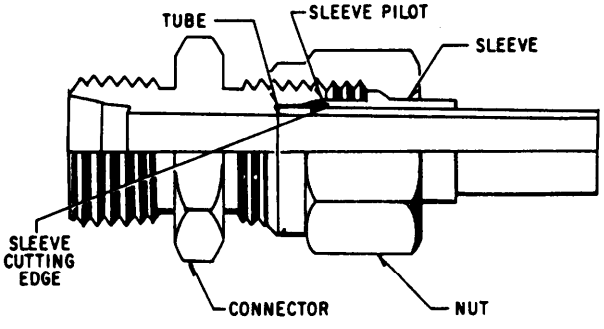

Figure 5-18.—Flareless-tube connector.

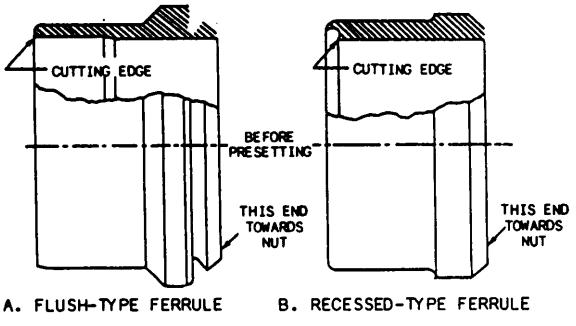

Figure 5-19.—Unused ferrules.

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caused personnel injury, damage to equipment, and unnecessary interruption of propulsion power.

1. Cut the tubing square and lightly deburr the inside and outside corners. For corrosion resisting steel (CRES) tubing, use a hacksaw rather than a tubing cutter to avoid work hardening the tube end. For CRES, and if necessary for other materials, dress the tube end smooth and square with a file. Tube ends with irregular cutting marks will not produce satisfactory seating surface impressions.

2. Test the hardness of the ferrule by making a light scratch on the tubing at least 1/2 inch back from the tube end, using a sharp corner on the ferrule. If the ferrule will not scratch the tube, no bite will be obtained. This test maybe omitted for flush-type ferrules where the bite will be visible. Moderate hand pressure is sufficient for producing the scratch.

3. Lubricate the nut threads, the ferrule leading and trailing edges, and the preset tool threads with a thread lubricant compatible with the system. Slide the nut onto the tubing so the threads face the tube end. Note whether the ferrule is a flush type or recessed type (fig. 5-19), and slide the ferrule onto the tube so the cutting edge is toward the tube end (large end toward the nut).

4. Bottom the end of the tubing in the presetting tool. Slide the ferrule up into the presetting tool, and confirm that the nut can be moved down the tube sufficiently to expose at least 1/8 inch of tubing past the ferrule after the presetting operation (fig. 5-20) to allow for inspection of the ferrule.

5. While keeping the tube bottomed in the presetting tool, tighten the nut onto the fitting body until the ferrule just grips the tube by friction. This ring grip point may be identified by lightly turning the tube or the presetting tool and slowly tightening the nut until the tube cannot be turned in the presetting tool by hand. Mark the nut and the presetting tool at this position.

6. Tighten the nut according to the number of turns given in table 5-3, depending on tube size.

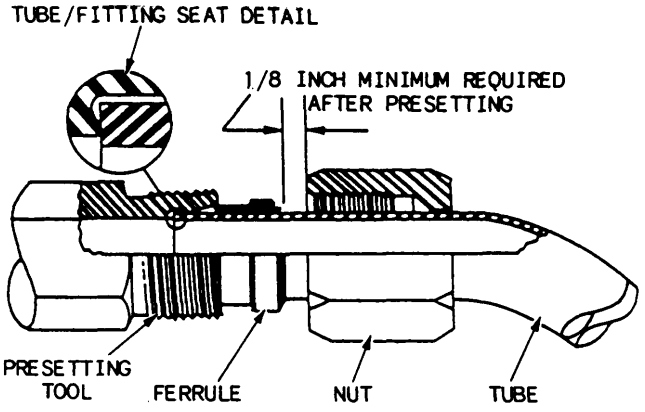

Figure 5-20.—Tube and ferrule assembled for presetting, showing nut position required for inspecting ferrule.

Inspection

Disassemble and inspect the fitting as follows (mandatory):

1. Ensure that the end of the tubing has an impression of the presetting tool seat surface (circular appearing ring) for 360 degrees. A partial circle, a visibly off-center circle, or a circle broken by the roughness of the tube end is unsatisfactory.

2. Check for proper bite:

a. For flush-type ferrules, a raised ridge (fig. 5-21) of tube metal must be visible completely around the tube at the leading edge of the ferrule. The best practice is to obtain a ridge about 50 percent of the ferrule edge thickness.

Table 5-3.—Number of Turns

Tube OD Inches	Number of Turns
1/8 to 1/2	1-1/6 (seven flats of the nut)
5/8 to 7/8	1 (six flats)
1	5/6 (five flats)
1-1/4 to 2	1 (six flats)

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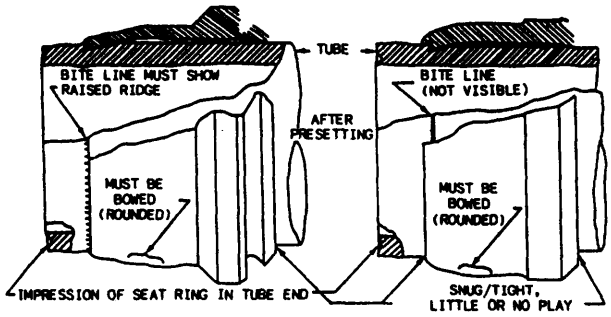

Figure 5-21.—Ferrules installed on tube, preset and removed for inspection.

b. For recessed-type ferrules, the leading edge must be snug against the tube OD. Determine this visually and by attempting to rock the ferrule on the tube.

3. Ensure that the nut end of the ferrule (both types) is collapsed around the tube to provide support against bending loads and vibration.

4. The ferrule (both types) must have little or no play along the direction of the tube run. Check this by trying to move the ferrule back and forth by hand. The ferrule will often be free to rotate on the tubing; this does not affect its function.

5. For flush-type ferrules, check that the gap between the raised metal ridge and the cutting end of the ferrule stays the same while the ferrule is rotated. (Omit this check for recessed-type ferrules or if the flush-type ferrule will not rotate on the tube).

6. Check that the middle portion of the ferrule (both types) is bowed or sprung into an arc. The leading edge of the ferrule may appear flattened into a cone shape; this is acceptable as long as there is a bowed section near the middle of the ferrule. If the whole leading section of the ferrule is flattened into a cone with no bowed section, the ferrule (and possibly the fitting body, if used) has been damaged by overtightening and will not seal reliably.

Final Assembly

When you make a final assembly in the system, use the following installation procedure:

1. Lubricate all threads with a liquid that is compatible with the fluid to be used in the system.

2. Place the tube assembly in position and check for alignment.

3. Tighten the nut by hand until you feel an increase in resistance to turning. This indicates that the sleeve or ferrule pilot has contacted the fitting.

4. If possible, use a torque wrench to tighten flareless tubing nuts. Torque values for specific installations are usually listed in the applicable technical publications. If it is not possible to use a torque wrench, use the following procedures for tightening the nuts:

After the nut is handtight, turn the nut 1/6 turn (one flat on a hex nut) with a wrench. Use a wrench on the connector to prevent it from turning while tightening the nut. After you install the tube assembly, have the system pressure tested. Should a connection leak, you may tighten the nut an additional 1/6 turn (making a total of 1/3 turn). If, after tightening the nut a total of 1/3 turn, leakage still exists, remove the assembly and inspect the components of the assembly for scores, cracks, presence of foreign material, or damage from overtightening.

NOTE: Overtightening a flareless-tube nut drives the cutting edge of the sleeve or ferrule deeply into the tube, causing the tube to be weakened to the point where normal vibration could cause the tube to shear. After you complete the inspection (if you do not find any discrepancies), reassemble the connection and repeat the pressure test procedures.

CAUTION: Do not in any case tighten the nut beyond 1/3 turn (two flats on the hex nut); this is the maximum the fitting may be tightened without the possibility of permanently damaging the sleeve or the tube.

CONNECTORS FOR FLEXIBLE HOSE

As stated previously, the fabrication of flexible hose assemblies is covered in applicable training manuals, technical publications, and NAVAIR 01-1A-20. There are various types of end fittings for both the piping connection side and the hose

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connection side of hose fittings. Figure 5-22 shows commonly used fittings.

Piping Connection Side of Hose Fitting

The piping side of an end fitting comes with several connecting variations: flange, JIC 37° flare, O-ring union, and split clamp, to name a few. Not all varieties are available for each hose. Therefore, installers must consult the military specification and manufacturer's data to determine the specific end fittings available.

Hose Connection Side of Hose Fitting

Hose fittings are attached to the hose by several methods. Each method is determined by the fitting manufacturer and takes into consideration such things as size, construction, wall thickness, and pressure rating. Hoses used for flexible connections use one of the following methods for attachment of the fitting to the hose.

ONE-PIECE REUSABLE SOCKET.— The socket component of the fitting is fabricated as a single piece. One-piece reusable sockets are screwed or rocked onto the hose OD, followed by insertion of the nipple component.

SEGMENTED, BOLTED SOCKET.— The segmented, bolted socket consists of two or more segments which are bolted together on the hose after insertion of the nipple component.

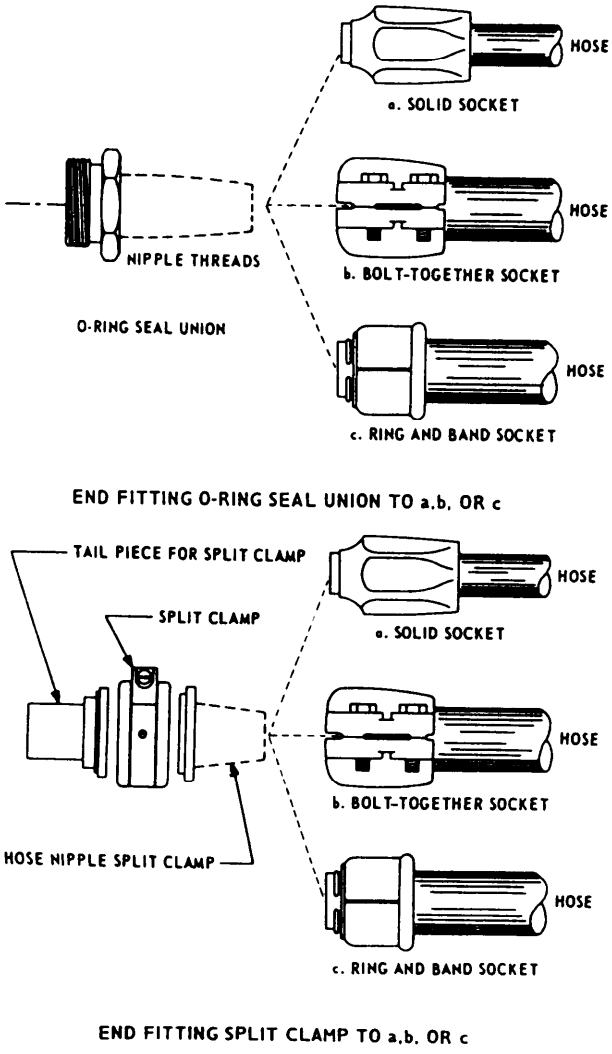

Figure 5-22.—End fittings and hose fittings.

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SEGMENTED SOCKET, RING AND BAND ATTACHED.— The segmented, ring and band attached socket consists of three or more segments. As with the bolt-together segments, the segments, ring and band are put on the hose after insertion of the nipple. A special tool is required to compress the segments.

SEGMENTED SOCKET, RING AND BOLT ATTACHED.— The segmented, ring and bolt attached socket consists of three or more segments. As with other segmented socket-type fittings, the segments, ring, and nuts and bolts are put on the hose after insertion of the nipple.

SOLID SOCKET, PERMANENTLY ATTACHED.— This type of socket is permanently attached to the hose by crimping or swaging. It is not reusable and is only found on hose assemblies where operating conditions preclude the use of other fitting types. Hose assemblies with this type of fitting attachment are purchased as complete hose assemblies from the manufacturer.

QUICK-DISCONNECT COUPLINGS

Self-sealing, quick-disconnect couplings are used at various points in many fluid power systems. These couplings are installed at locations where frequent uncoupling of the lines is required for inspection, test, and maintenance. Quick-disconnect couplings are also commonly used in pneumatic systems to connect sections of air hose and to connect tools to the air pressure lines. This provides a convenient method of attaching and detaching tools and sections of lines without losing pressure.

Quick-disconnect couplings provide a means for quickly disconnecting a line without the loss of fluid from the system or the entrance of foreign matter into the system. Several types of quick-disconnect couplings have been designed for use in fluid power systems. Figure 5-23 illustrates a coupling that is used with portable pneumatic tools. The male section is connected to the tool or to the line leading from the tool. The female section, which contains the shutoff valve, is installed in the pneumatic line leading from the pressure source. These connectors can be separated or connected by very little effort on the part of the operator.

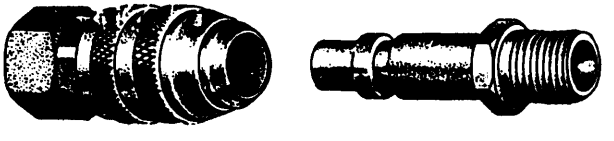

Figure 5-23.—Quick-disconnect coupling for air lines.

The most common quick-disconnect coupling for hydraulic systems consists of two parts, held together by a union nut. Each part contains a valve which is held open when the coupling is connected, allowing fluid to flow in either direction through the coupling. When the coupling is disconnected, a spring in each part closes the valve, preventing the loss of fluid and entrance of foreign matter.

MANIFOLDS

Some fluid power systems are equipped with manifolds in the pressure supply and/or return lines. A manifold is a fluid conductor that provides multiple connection ports. Manifolds eliminate piping, reduce joints, which are often a source of leakage, and conserve space. For example, manifolds may be used in systems that contain several subsystems. One common line connects the pump to the manifold. There are outlet ports in the manifold to provide connections to each subsystem. A similar manifold may be used in the return system. Lines from the control valves of the subsystem connect to the inlet ports of the manifold, where the fluid combines into one outlet line to the reservoir. Some manifolds are equipped with the check valves, relief valves, filters, and so on, required for the system. In some cases, the control valves are mounted on the manifold in such a manner that the ports of the valves are connected directly to the manifold.

Manifolds are usually one of three types—sandwich, cast, or drilled. The sandwich type is constructed of three or more flat plates. The center plate (or plates) is machined for passages, and the required inlet and outlet ports are drilled into the outer plates. The plates are then bonded together to provide a leakproof assembly. The cast type of manifold is designed with cast passages and drilled ports. The casting may be iron, steel, bronze, or aluminum, depending upon the type of system and fluid medium. In the drilled type of manifold, all ports and passages are drilled in a block of metal.

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A simple manifold is illustrated in figure 5-24. This manifold contains one pressure inlet port and several pressure outlet ports that can be blocked off with threaded plugs. This type of manifold can be adapted to systems containing various numbers of subsystems. A thermal relief valve may be incorporated in this manifold. In this case, the port labeled T is connected to the return line to provide a passage for the relieved fluid to flow to the reservoir.

Figure 5-25 shows a flow diagram in a manifold which provides both pressure and return passages. One common line provides pressurized fluid to the manifold, which distributes the fluid to any one of five outlet ports. The return side of the manifold is similar in design. This manifold is provided with a relief valve, which is connected to the pressure and return passages. In the event of excessive pressure, the relief valve opens and allows the fluid to flow from the pressure side of the manifold to the return side.

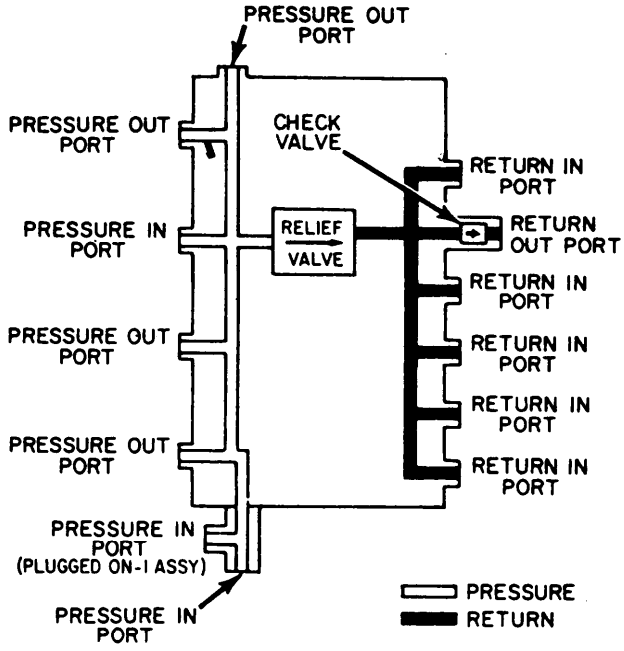

Figure 5-25.—Fluid manifold—flow diagram.

PRECAUTIONARY MEASURES

The fabrication, installation, and maintenance of all fluid lines and connectors are beyond the scope of this training manual. However, there are some general precautionary measures that apply to the maintenance of all fluid lines.

Regardless of the type of lines or connectors used to make up a fluid power system, make certain they are the correct size and strength and perfectly clean on the inside. All lines must be absolutely clean and free from scale and other foreign matter. Iron or steel pipes, tubing, and fittings can be cleaned with a boiler tube wire brush or with commercial pipe cleaning apparatus. Rust and scale can be removed from short, straight pieces by sandblasting, provided there is no danger that sand particles will remain lodged in blind holes or pockets after the piece

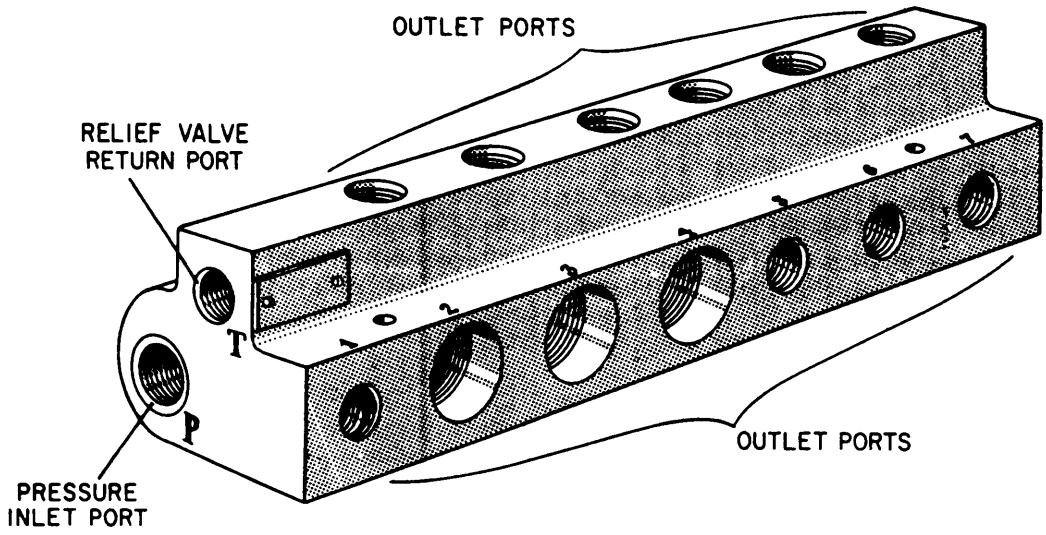

Figure 5-24.—Fluid manifold.

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is flushed. In the case of long pieces or pieces bent to complex shapes, rust and scale can be removed by pickling (cleaning metal in a chemical bath). Parts must be degreased prior to pickling. The manufacturer of the parts should provide complete pickling instructions.

Open ends of pipes, tubing, hose, and fittings should be capped or plugged when they are to be stored for any considerable period. Rags or waste must not be used for this purpose, because they deposit harmful lint which can cause severe damage to the fluid power system.

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CHAPTER 6

VALVES

It is all but impossible to design a practical fluid power system without some means of controlling the volume and pressure of the fluid and directing the flow of fluid to the operating units. This is accomplished by the incorporation of different types of valves. A valve is defined as any device by which the flow of fluid may be started, stopped, or regulated by a movable part that opens or obstructs passage. As applied in fluid power systems, valves are used for controlling the flow, the pressure, and the direction of the fluid flow.

Valves must be accurate in the control of fluid flow and pressure and the sequence of operation. Leakage between the valve element and the valve seat is reduced to a negligible quantity by precision-machined surfaces, resulting in carefully controlled clearances. This is one of the very important reasons for minimizing contamination in fluid power systems. Contamination causes valves to stick, plugs small orifices, and causes abrasions of the valve seating surfaces, which results in leakage between the valve element and valve seat when the valve is in the closed position. Any of these can result in inefficient operation or complete stoppage of the equipment.

Valves may be controlled manually, electrically, pneumatically, mechanically, hydraulically, or by combinations of two or more of these methods. Factors that determine the method of control include the purpose of the valve, the design and purpose of the system, the location of the valve within the system, and the availability of the source of power.

The different types of valves used in fluid power systems, their classification, and their application are discussed in this chapter.

CLASSIFICATIONS

Valves are classified according to their use: flow control, pressure control, and directional control. Some valves have multiple functions that fall into more than one classification.

FLOW CONTROL VALVES

Flow control valves are used to regulate the flow of fluids in fluid-power systems. Control of flow in fluid-power systems is important because the rate of movement of fluid-powered machines depends on the rate of flow of the pressurized fluid. These valves may be manually, hydraulically, electrically, or pneumatically operated.

Some of the different types of flow control valves are discussed in the following paragraphs.

BALL VALVES

Ball valves, as the name implies, are stop valves that use a ball to stop or start a flow of fluid. The ball, shown in figure 6-1, performs the

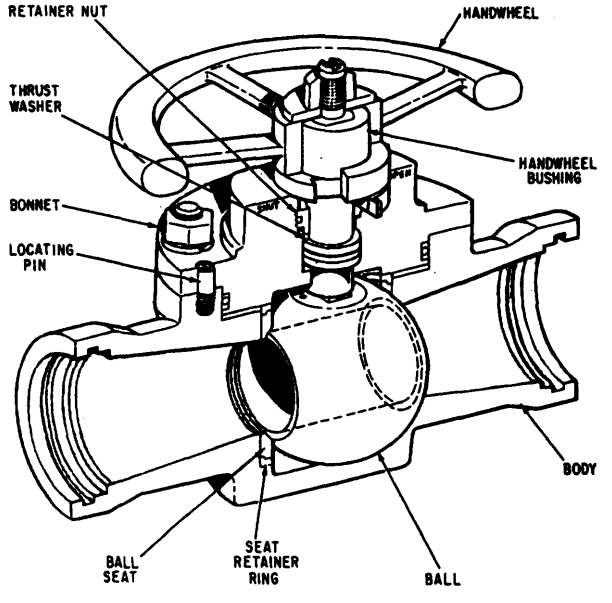

Figure 6-1.—Typical ball valve.

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same function as the disk in other valves. As the valve handle is turned to open the valve, the ball rotates to a point where part or all of the hole through the ball is in line with the valve body inlet and outlet, allowing fluid to flow through the valve. When the ball is rotated so the hole is perpendicular to the flow openings of the valve body, the flow of fluid stops.

Most ball valves are the quick-acting type. They require only a 90-degree turn to either completely open or close the valve. However, many are operated by planetary gears. This type of gearing allows the use of a relatively small handwheel and operating force to operate a fairly large valve. The gearing does, however, increase the operating time for the valve. Some ball valves also contain a swing check located within the ball to give the valve a check valve feature. Figure 6-2 shows a ball-stop, swing-check valve with a planetary gear operation.

In addition to the ball valves shown in figures 6-1 and 6-2, there are three-way ball valves that are used to supply fluid from a single source to one component or the other in a two-component system (fig. 6-3).

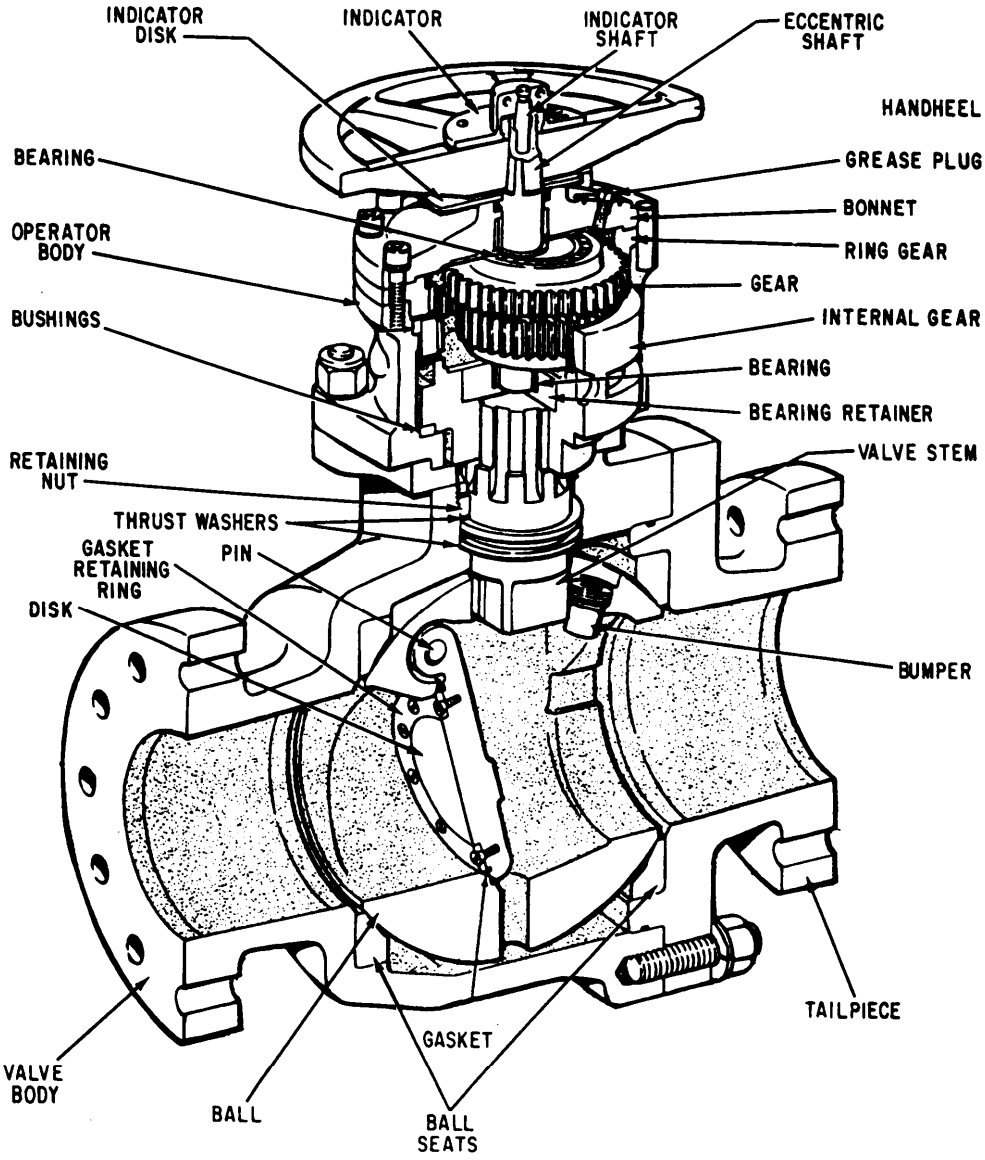

Figure 6-2.—Typical ball-stop, swing-check valve.

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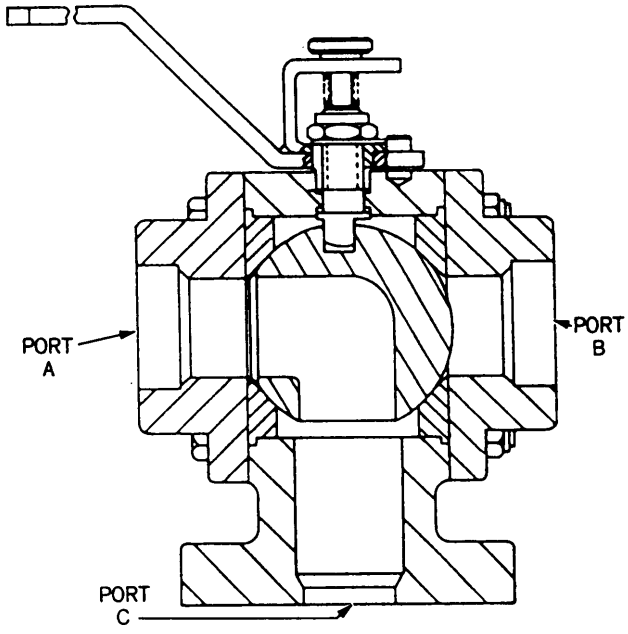

Figure 6-3.—Three-way ball valve.

GATE VALVES

Gate valves are used when a straight-line flow of fluid and minimum flow restriction are needed. Gate valves are so-named because the part that either stops or allows flow through the valve acts somewhat like a gate. The gate is usually wedge-shaped. When the valve is wide open the gate is fully drawn up into the valve bonnet. This leaves an opening for flow through the valve the same size as the pipe in which the valve is installed (fig. 6-4). Therefore, there is little pressure drop or flow restriction through the valve.

Gate valves are not suitable for throttling purposes. The control of flow is difficult because of the valve's design, and the flow of fluid slapping against a partially open gate can cause extensive damage to the valve. Except as specifically authorized, gate valves should not be used for throttling.

Gate valves are classified as either rising-stem or nonrising-stem valves. The nonrising-stem valve is shown in figure 6-4. The stem is threaded into the gate. As the handwheel on the stem is rotated, the gate travels up or down the stem on the threads while the stem remains vertically stationary. This type of valve will almost always have a pointer indicator threaded onto the upper end of the stem to indicate the position of the gate.

Valves with rising stems (fig. 6-5) are used when it is important to know by immediate inspection whether the valve is open or closed and when the threads (stem and gate) exposed to the fluid could become damaged by fluid contaminants. In this valve, the stem rises out of the valve when the valve is opened.

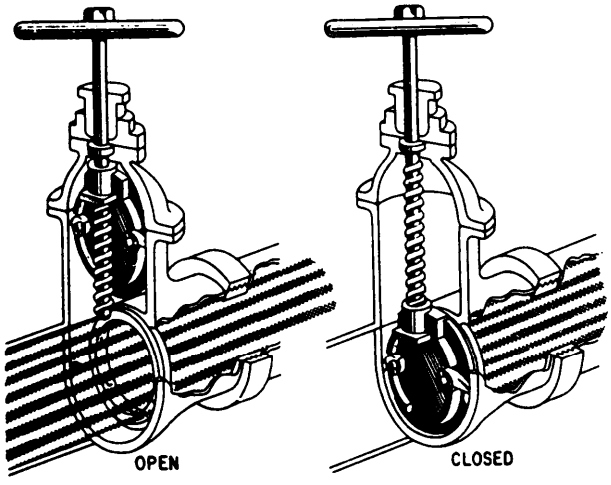

Figure 6-4.—Operation of a gate valve.

GLOBE VALVES

Globe valves are probably the most common valves in existence. The globe valve gets its name

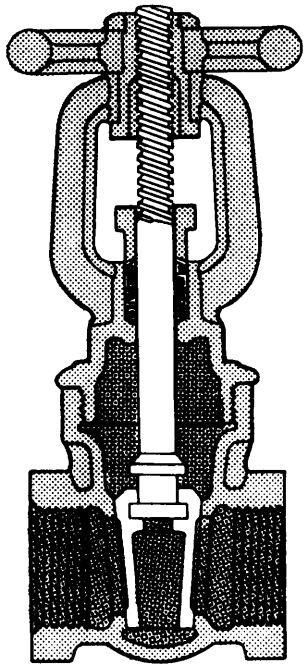

Figure 6-5.—Rising stem gate valve.

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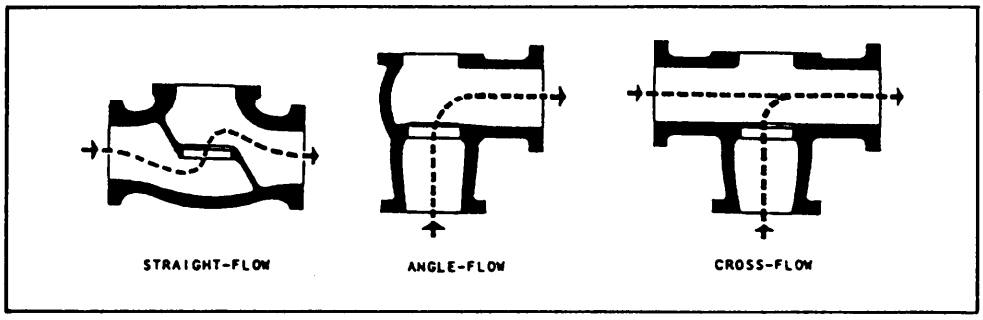

Figure 6-6.—Types of globe valve bodies.

from the globular shape of the valve body. Other types of valves may also have globular-shaped bodies. Thus, it is the internal structure of the valve that identifies the type of valve.

The inlet and outlet openings for globe valves are arranged in a way to satisfy the flow requirements. Figure 6-6 shows straight-, angle-, and cross-flow valves.

The moving parts of a globe valve consist of the disk, the valve stem, and the handwheel. The stem connects the handwheel and the disk. It is threaded and fits into the threads in the valve bonnet.

The part of the globe valve that controls flow is the disk, which is attached to the valve stem. (Disks are available in various designs.) The valve is closed by turning the valve stem in until the disk is seated into the valve seat. This prevents fluid from flowing through the valve (fig. 6-7, view A). The edge of the disk and the seat are very accurately machined so that they forma tight seal when the valve is closed. When the valve is open (fig. 6-7, view B), the fluid flows through the space between the edge of the disk and the seat. Since the fluid flows equally on all sides of the center of support when the valve is open, there is no unbalanced pressure on the disk to cause uneven wear. The rate at which fluid flows through the valve is regulated by the position of the disk in relation to the seat. The valve is commonly used as a fully open or fully closed valve, but it may be used as a throttle valve. However, since the seating surface is a relatively large area, it is not suitable as a throttle valve, where fine adjustments are required in controlling the rate of flow.

The globe valve should never be jammed in the open position. After a valve is fully opened, the handwheel should be turned toward the closed position approximately one-half turn. Unless this is done, the valve is likely to seize in the open position, making it difficult, if not impossible, to close the valve. Many valves are damaged in this

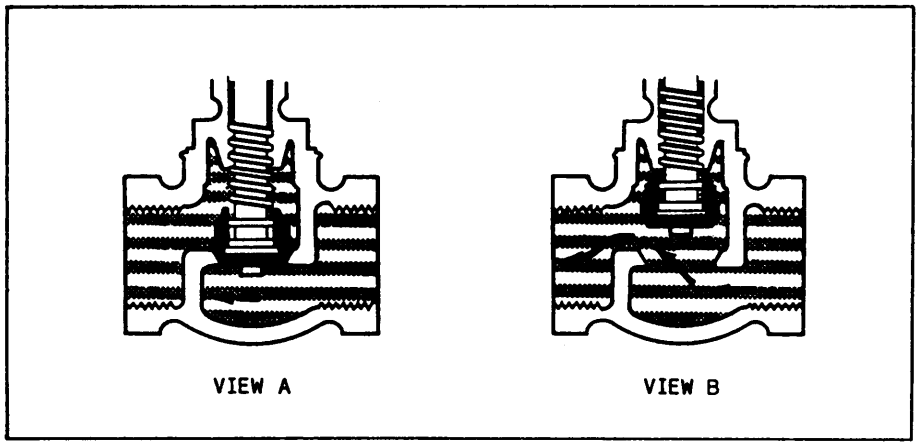

Figure 6-7.—Operation of a globe valve.

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manner. Another reason for not leaving globe valves in the fully open position is that it is sometimes difficult to determine if the valve is open or closed. If the valve is jammed in the open position, the stem may be damaged or broken by someone who thinks the valve is closed, and attempts to open it.

It is important that globe valves be installed with the pressure against the face of the disk to keep the system pressure away from the stem packing when the valve is shut.

NEEDLE VALVES

Needle valves are similar in design and operation to the globe valve. Instead of a disk, a needle valve has a long tapered point at the end of the valve stem. A cross-sectional view of a needle valve is illustrated in figure 6-8.

The long taper of the valve element permits a much smaller seating surface area than that of the globe valve; therefore, the needle valve is more suitable as a throttle valve. Needle valves are used to control flow into delicate gauges, which might be damaged by sudden surges of fluid under pressure. Needle valves are also used to control the end of a work cycle, where it is desirable for motion to be brought slowly to a halt, and at other points where precise adjustments of flow are necessary and where a small rate of flow is desired.

Although many of the needle valves used in fluid power systems are the manually operated type (fig. 6-8), modifications of this type of valve are often used as variable restrictors. This valve is constructed without a handwheel and is adjusted to provide a specific rate of flow. This rate of flow will provide a desired time of operation for a particular subsystem. Since this type of valve can be adjusted to conform to the requirements of a particular system, it can be used in a variety of systems. Figure 6-9 illustrates a needle valve that was modified as a variable restrictor.

HYDRAULIC AND PNEUMATIC GLOBE VALVES

The valve consists of a valve body and a stem cartridge assembly. The stem cartridge assembly includes the bonnet, gland nut, packing, packing retainer, handle, stem, and seat. On small valves (1/8 and 1/4 inch) the stem is made in one piece, but on larger sizes it is made of a stem, guide, and stem retainer. The valve disk is made of nylon and is swaged into either the stem, for 1/8- and 1/4-inch valves, or the guide, for larger valves. The bonnet screws into the valve body with left-hand threads and is sealed by an O-ring (including a back-up ring).

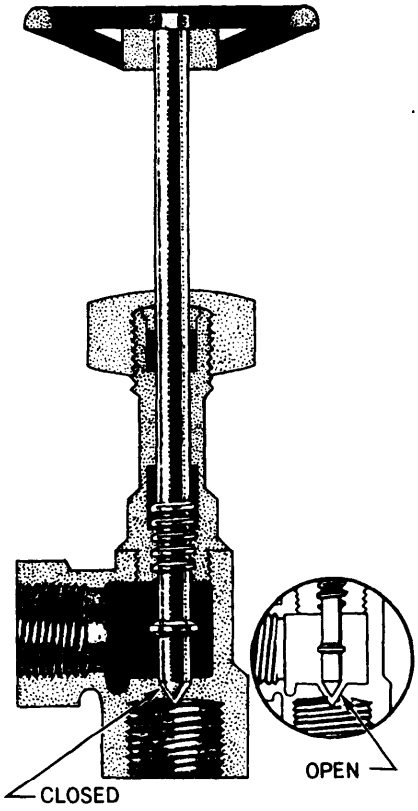

Figure 6-8.—Cross-sectional view of a needle valve.

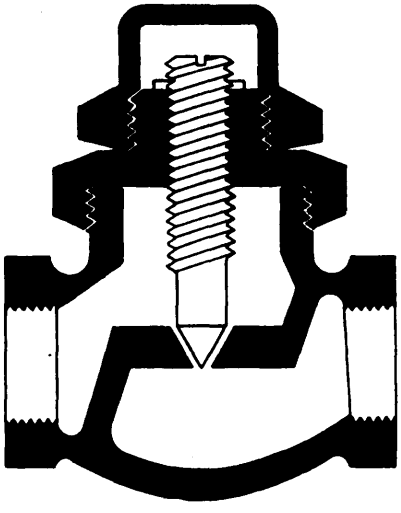

Figure 6-9.—Variable restrictor.

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The valve is available with either a rising stem or a non-rising stem. The rising stem valve uses the same port body design as does the non-rising stem valve. The stem is threaded into the gland nut and screws outward as the valve is opened.

This valve does not incorporate provisions for tightening the stem packing nor replacing the packing while the valve is in service; therefore, complete valve disassembly is required for maintenance. Figure 6-10 illustrates a rising stem hydraulic and pneumatic globe valve. Additional information on this valve is available in *Standard Navy Valves,* NAVSHIPS 0948-012-5000.

PRESSURE CONTROL VALVES

The safe and efficient operation of fluid power systems, system components, and related equipment requires a means of controlling pressure. There are many types of automatic pressure control valves. Some of them merely provide an escape for pressure that exceeds a set pressure; some only reduce the pressure to a lower pressure system or subsystem; and some keep the pressure in a system within a required range.

RELIEF VALVES

Some fluid power systems, even when operating normally, may temporarily develop excessive pressure; for example, when an unusually strong work resistance is encountered. Relief valves are used to control this excess pressure.

Relief valves are automatic valves used on system lines and equipment to prevent overpressurization. Most relief valves simply lift (open) at a preset pressure and reset (shut) when the pressure drops slightly below the lifting pressure. They do not maintain flow or pressure at a given amount, but prevent pressure from rising above a specific level when the system is temporarily overloaded.

Main system relief valves are generally installed between the pump or pressure source and the first system isolation valve. The valve must be large enough to allow the full output of the hydraulic pump to be delivered back to the reservoir. In a pneumatic system, the relief valve controls excess pressure by discharging the excess gas to the atmosphere.

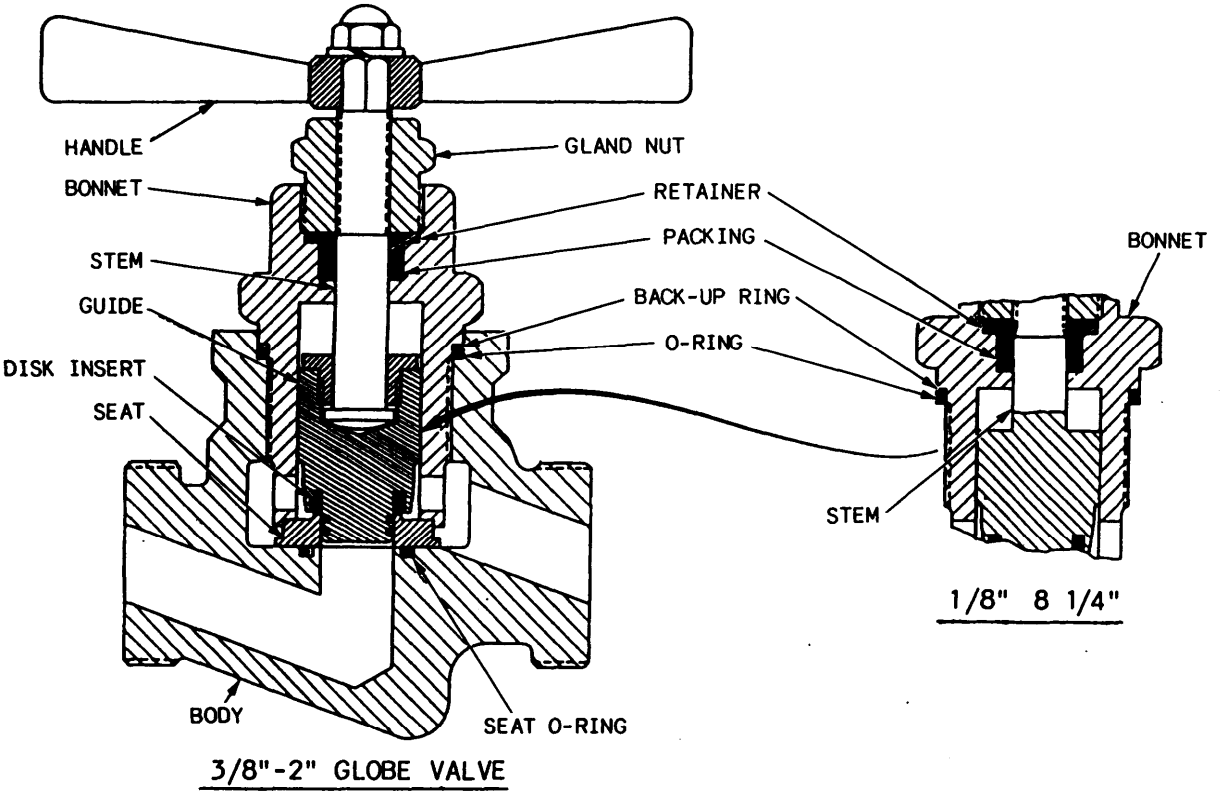

Figure 6-10.—Hydraulic and pneumatic globe valve (rising stem).

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Smaller relief valves, similar in design and operation to the main system relief valve, are often used in isolated parts of the system where a check valve or directional control valve prevents pressure from being relieved through the main system relief valve and where pressures must be relieved at a set point lower than that provided by the main system relief. These small relief valves are also used to relieve pressures caused by thermal expansion (see glossary) of the fluids.

Figure 6-11 shows a typical relief valve. System pressure simply acts under the valve disk at the inlet to the valve. When the system pressure exceeds the force exerted by the valve spring, the valve disk lifts off of its seat, allowing some of the system fluid to escape through the valve outlet until the system pressure is reduced to just below the relief set point of the valve.

All relief valves have an adjustment for increasing or decreasing the set relief pressure. Some relief valves are equipped with an adjusting screw for this purpose. This adjusting screw is usually covered with a cap, which must be removed before an adjustment can be made. Some type of locking device, such as a lock nut, is usually provided to prevent the adjustment from changing through vibration. Other types of relief valves are equipped with a handwheel for making adjustments to the valve. Either the adjusting screw or the handwheel is turned clockwise to increase the pressure at which the valve will open. In addition, most relief valves are also provided with an operating lever or some type of device to allow manual cycling or gagging the valve open for certain tasks.

Various modifications of the relief valve shown in figure 6-11 are used to efficiently serve the requirements of some fluid power systems; however, this relief valve is unsatisfactory for some applications. To give you a better understanding of the operation of relief valves, we will discuss some of the undesirable characteristics of this valve.

A simple relief valve, such as the one illustrated in figure 6-11, with a suitable spring adjustment can be set so that it will open when the system pressure reaches a certain level, 500 psi for example. When the valve does open, the volume of flow to be handled may be greater than the capacity of the valve; therefore, pressure in the system may increase to several hundred psi above the set pressure before the valve brings the pressure under control. A simple relief valve will be effective under these conditions only if it is very large. In this case, it would operate stiffly and the valve element would chatter back and forth. In addition, the valve will not close until the system pressure decreases to a point somewhat below the opening pressure.

The surface area of the valve element must be larger than that of the pressure opening if the valve is to seat satisfactorily as shown in figure 6-12. The pressure in the system acts on the valve element open to it. In each case in figure 6-12, the force exerted directly upward by system pressure when the valve is closed depends on the area (A) across the valve element where the element seats against the pressure tube. The moment the valve opens, however, the upward force exerted depends on the horizontal area (B) of the entire valve element, which is greater than area A. This causes an upward jump of the valve element immediately after it opens, because the

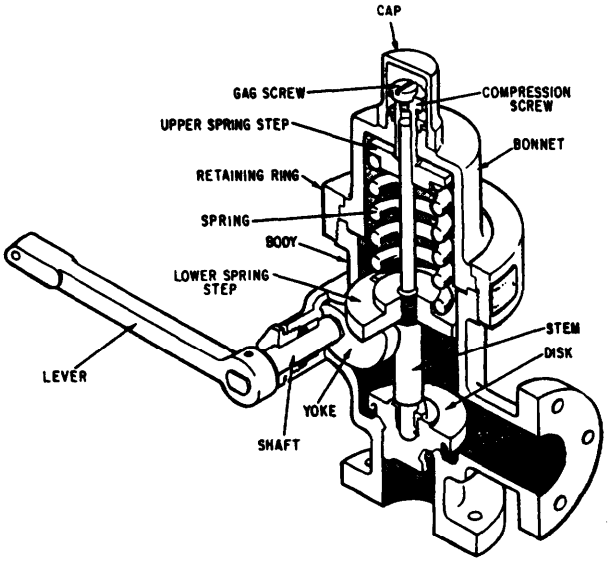

Figure 6-11.—Relief valve.

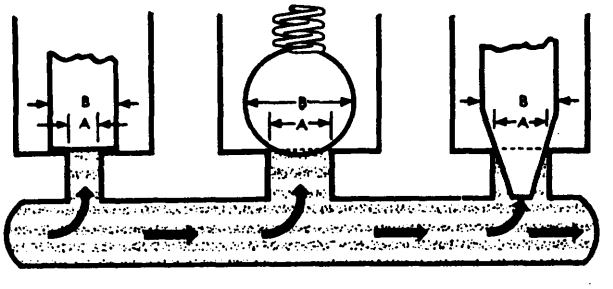

Figure 6-12.—Pressure acting on different areas.

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same pressure acting over different areas produces forces proportional to the areas. It also requires a greater force to close the valve than was required to open it. As a result, the valve will not close until the system pressure has decreased to a certain point below the pressure required to open it.

Let us assume that a valve of this type is set to open at 500 psi. (Refer to fig. 6-12.) When the valve is closed, the pressure acts on area A. If this area is 0.5 square inch, an upward force of 250 pounds (500 × 0.5) will be exerted on the valve at the moment of opening. With the valve open, however, the pressure acts on area B. If area B is 1 square inch, the upward force is 500 pounds, or double the force at which the valve actually opened. For the valve to close, pressure in the system would have to decrease well below the point at which the valve opened. The exact pressure would depend on the shape of the valve element.

In some hydraulic systems, there is a pressure in the return line. This back pressure is caused by restrictions in the return line and will vary in relation to the amount of fluid flowing in the return line. This pressure creates a force on the back of the valve element and will increase the force necessary to open the valve and relieve system pressure.

It follows that simple relief valves have a tendency to open and close rapidly as they "hunt" above and below the set pressure, causing pressure pulsations and undesirable vibrations and producing a noisy chatter. Because of the unsatisfactory performance of the simple relief valve in some applications, compound relief valves were developed.

Compound relief valves use the principles of operation of simple relief valves for one stage of their action—that of the pilot valve. Provision is made to limit the amount of fluid that the pilot valve must handle, and thereby avoid the weaknesses of simple relief valves. (A pilot valve is a small valve used for operating another valve.)

The operation of a compound relief valve is illustrated in figure 6-13. In view A, the main valve, which consists of a piston, stem, and spring, is closed, blocking flow from the high-pressure line to the reservoir. Fluid in the high-pressure line flows around the stem of the main valves as it flows to the actuating unit. The stem of the main valve is hollow (the stem passage) and contains the main valve spring, which forces the main valve against its seat. When the pilot valve is open the stem passage allows fluid to flow from the pilot

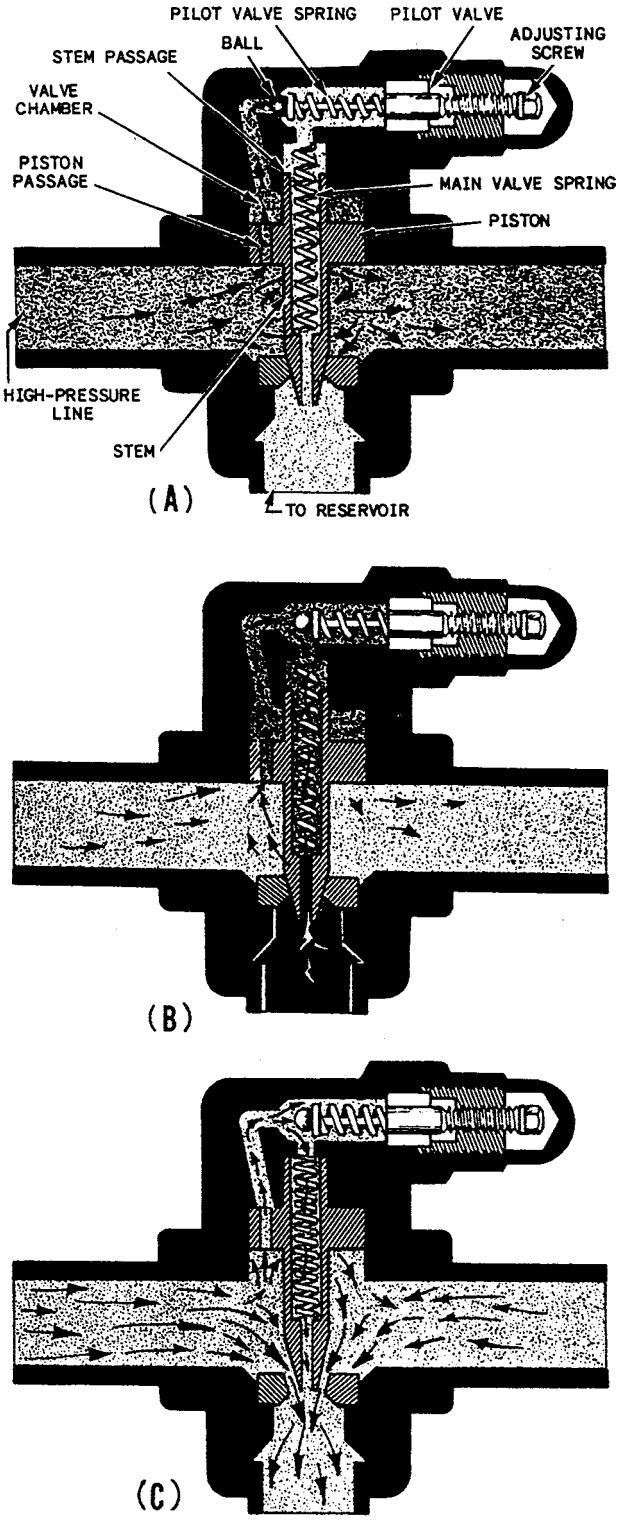

Figure 6-13.—Operation of compound relief valve,

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valve, around the main valve spring, and down to the return line.

There is also a narrow passage (piston passage) through the main valve piston. This passage connects the high-pressure line to the valve chamber.

The pilot valve is a small, ball-type, spring-loaded check valve, which connects the top of the passage from the valve chamber with the passage through the main valve stem. The pilot valve is the control unit of the relief valve because the pressure at which the relief valve will open depends on the tension of the pilot valve spring. The pilot valve spring tension is adjusted by turning the adjusting screw so that the ball will unseat when system pressure reaches the preset limit.

Fluid at line pressure flows through the narrow piston passage to fill the chamber. Because the line and the chamber are connected, the pressure in both are equal. The top and bottom of the main piston have equal areas; therefore, the hydraulic forces acting upward and downward are equal, and there is no tendency for the piston to move in either direction. The only other force acting on the main valve is that of the main valve spring, which holds it closed.

When the pressure in the high-pressure line increases to the point at which the pilot valve is set, the ball unseats (fig. 6-13, view B). This opens the valve chamber through the valve stem passage to the low-pressure return line. Fluid immediately begins to flow out of the chamber, much faster than it can flow through the narrow piston passage. As a result the chamber pressure immediately drops, and the pilot valve begins to close again, restricting the outward flow of fluid. Chamber pressure therefore increases, the valve opens, and the cycle repeats.

So far, the only part of the valve that has moved appreciably is the pilot, which functions just like any other simple spring-loaded relief valve. Because of the small size of the piston passage, there is a severe limit on the amount of overpressure protection the pilot can provide the system. All the pilot valve can do is limit fluid pressure in the valve chamber above the main piston to a preset maximum pressure, by allowing excess fluid to flow through the piston passage, through the stem passage, and into the return line. When pressure in the system increases to a value that is above the flow capacity of the pilot valve, the main valve opens, permitting excess fluid to flow directly to the return line. This is accomplished in the following manner.

As system pressure increases, the upward force on the main piston overcomes the downward force, which consists of the tension of the main piston spring and the pressure of the fluid in the valve chamber (fig. 6-13, view C). The piston then rises, unseating the stem, and allows the fluid to flow from the system pressure line directly into the return line. This causes system pressure to decrease rapidly, since the main valve is designed to handle the complete output of the pump. When the pressure returns to normal, the pilot spring forces the ball onto the seat. Pressures are equal above and below the main piston, and the main spring forces the valve to seat.

As you can see, the compound valve overcomes the greatest limitation of a simple relief valve by limiting the flow through the pilot valve to the quantity it can satisfactorily handle. This limits the pressure above the main valve and enables the main line pressure to open the main valve. In this way, the system is relieved when an overload exists.

PRESSURE REGULATORS

Pressure regulators, often referred to as unloading valves, are used in fluid power systems to regulate pressure. In pneumatic systems, the valve, commonly referred to as a pressure regulator, simply reduces pressure. This type of valve is discussed later in this chapter under pressure-reducing valves. In hydraulic systems the pressure regulator is used to unload the pump and to maintain and regulate system pressure at the desired values. All hydraulic systems do not require pressure regulators. The open-center system (discussed in chapter 12) does not require a pressure regulator. Many systems are equipped with variable-displacement pumps (discussed in chapter 4), which contain a pressure-regulating device.

Pressure regulators are made in a variety of types and by various manufacturers; however, the

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basic operating principles of all regulators are similar to the one illustrated in figure 6-14.

A regulator is open when it is directing fluid under pressure into the system (fig. 6-14, view A). In the closed position (fig. 6-14, view B), the fluid in the part of the system beyond the regulator is trapped at the desired pressure, and the fluid from the pump is bypassed into the return line and back to the reservoir. To prevent constant opening and closing (chatter), the regulator is designed to open at a pressure somewhat lower than the closing pressure. This difference is known as differential or operating range. For example, assume that a pressure regulator is set to open when the system pressure drops below 600 psi, and close when the pressure rises above 800 psi. The differential or operating range is 200 psi.

Referring to figure 6-14, assume that the piston has an area of 1 square inch, the pilot valve has a cross-sectional area of one-fourth square inch, and the piston spring provides 600 pounds of force pushing the piston down. When the pressure in the system is less than 600 psi, fluid from the pump will enter the inlet port, flow to the top of the regulator, and then to the pilot valve. When the pressure of the fluid at the inlet increases to the point where the force it creates against the front of the check valve exceeds the force created against the back of the check valve by system pressure and the check valve spring, the check valve opens. This allows fluid to flow into the system and to the bottom of the regulator against the piston. When the force created by the system pressure exceeds the force exerted by the spring, the piston moves up, causing the pilot valve to unseat. Since the fluid will take the path of least resistance, it will pass through the regulator and back to the reservoir through the return line.

When the fluid from the pump is suddenly allowed a free path to return, the pressure on the input side of the check valve drops and the check valve closes. The fluid in the system is then trapped under pressure. This fluid will remain pressurized until a power unit is actuated, or until pressure is slowly lost through normal internal leakage within the system.

When the system pressure decreases to a point slightly below 600 psi, the spring forces the piston down and closes the pilot valve. When the pilot valve is closed, the fluid cannot flow directly to the return line. This causes the pressure to increase in the line between the pump and the regulator. This pressure opens the check valve, causing the fluid to enter the system.

In summary, when the system pressure decreases a certain amount, the pressure regulator will open, sending fluid to the system. When the system pressure increases sufficiently, the regulator will close, allowing the fluid from the pump to flow through the regulator and back to the reservoir. The pressure regulator takes the load off of the pump and regulates system pressure.

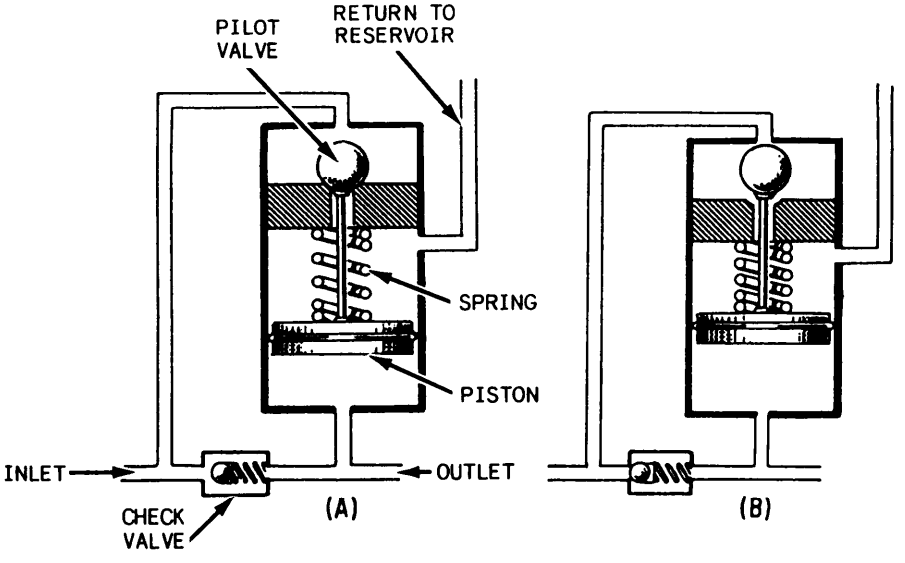

Figure 6-14.—Hydraulic pressure regulator.

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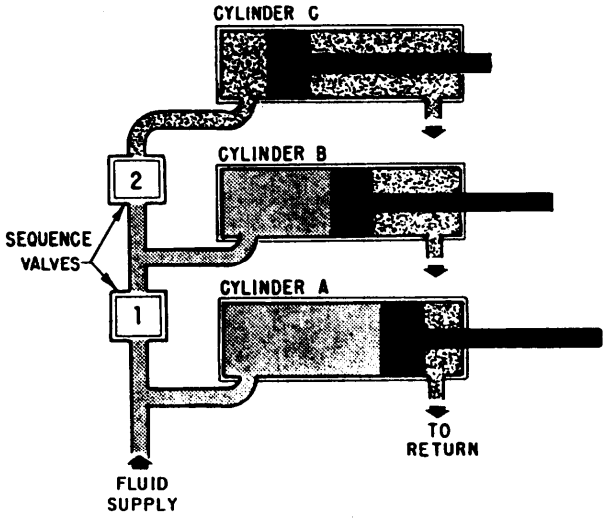

Figure 6-15.—Installation of sequence valves.

SEQUENCE VALVES

Sequence valves control the sequence of operation between two branches in a circuit; that is, they enable one unit to automatically set another unit into motion. An example of the use of a sequence valve is in an aircraft landing gear actuating system.

In a landing gear actuating system, the landing gear doors must open before the landing gear starts to extend. Conversely, the landing gear must be completely retracted before the doors close. A sequence valve installed in each landing gear actuating line performs this function.

A sequence valve is somewhat similar to a relief valve except that, after the set pressure has been reached, the sequence valve diverts the fluid to a second actuator or motor to do work in another part of the system. Figure 6-15 shows an installation of two sequence valves that control the sequence of operation of three actuating cylinders. Fluid is free to flow into cylinder A. The first sequence valve (1) blocks the passage of fluid until the piston in cylinder A moves to the end of its stroke. At this time, sequence valve 1 opens, allowing fluid to enter cylinder B. This action continues until all three pistons complete their strokes.

There are various types of sequence valves. Some are controlled by pressure and some are controlled mechanically.

Pressure-Controlled Sequence Valve

The operation of a typical pressure-controlled sequence valve is illustrated in figure 6-16. The opening pressure is obtained by adjusting the tension of the spring that normally holds the piston in the closed position. (Note that the top part of the piston has a larger diameter than the lower part.) Fluid enters the valve through the inlet port, flows around the lower part of the piston and exits the outlet port, where it flows to the primary (first) unit to be operated (fig. 6-16, view A). This fluid pressure also acts against the lower surface of the piston.

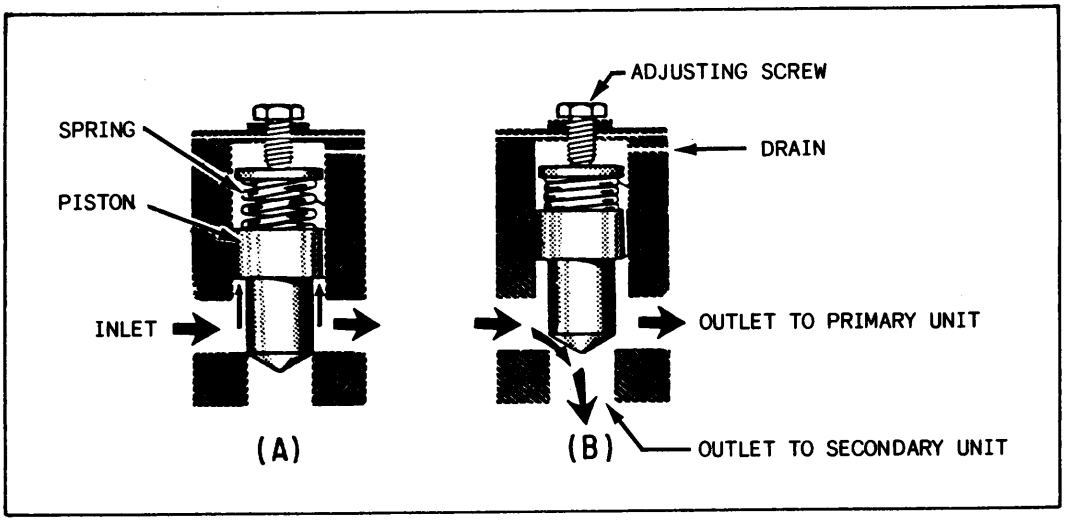

Figure 6-16.—Operation of a pressure-controlled sequence valve.

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When the primary actuating unit completes its operation, pressure in the line to the actuating unit increases sufficiently to overcome the force of the spring, and the piston rises. The valve is then in the open position (fig. 6-16, view B). The fluid entering the valve takes the path of least resistance and flows to the secondary unit.

A drain passage is provided to allow any fluid leaking past the piston to flow from the top of the valve. In hydraulic systems, this drain line is usually connected to the main return line.

Mechanically Operated Sequence Valve

The mechanically operated sequence valve (fig. 6-17) is operated by a plunger that extends through the body of the valve. The valve is mounted so that the plunger will be operated by the primary unit.

A check valve, either a ball or a poppet, is installed between the fluid ports in the body. It can be unseated by either the plunger or fluid pressure.

Port A (fig. 6-17) and the actuator of the primary unit are connected by a common line. Port B is connected by a line to the actuator of the secondary unit. When fluid under pressure flows to the primary unit, it also flows into the sequence valve through port A to the seated check valve in the sequence valve. In order to operate the secondary unit, the fluid must flow through the sequence valve. The valve is located so that the primary unit depresses the plunger as it completes its operation. The plunger unseats the check valve and allows the fluid to flow through the valve, out port B, and to the secondary unit.

This type of sequence valve permits flow in the opposite direction. Fluid enters port B and flows to the check valve. Although this is return flow from the actuating unit, the fluid overcomes spring tension, unseats the check valve, and flows out through port A.

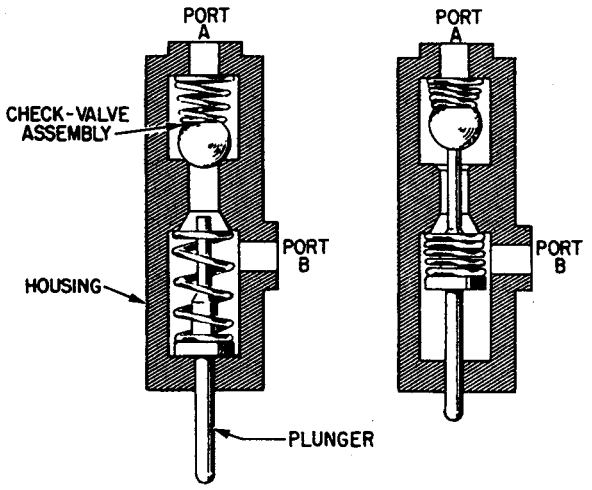

Figure 6-17.—Mechanically operated sequence valve.

PRESSURE-REDUCING VALVES

Pressure-reducing valves provide a steady pressure into a system that operates at a lower pressure than the supply system. A reducing valve can normally be set for any desired downstream pressure within the design limits of the valve. Once the valve is set, the reduced pressure will be maintained regardless of changes in supply pressure (as long as the supply pressure is at least as high as the reduced pressure desired) and regardless of the system load, providing the load does not exceed the design capacity of the reducer.

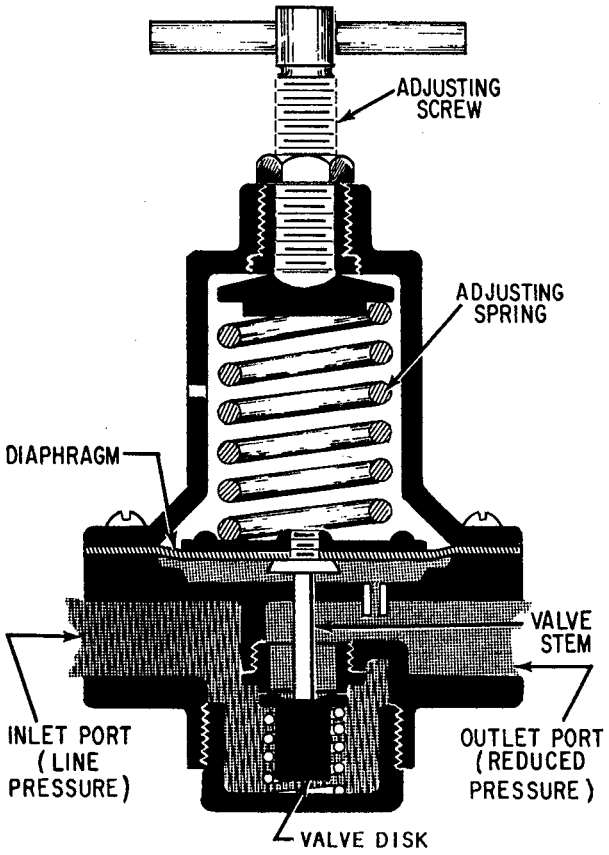

Figure 6-18.—Spring-loaded pressure-reducing valve.

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There are various designs and types of pressure-reducing valves. The spring-loaded reducer and the pilot-controlled valve are discussed in this text.

Spring-Loaded Reducer

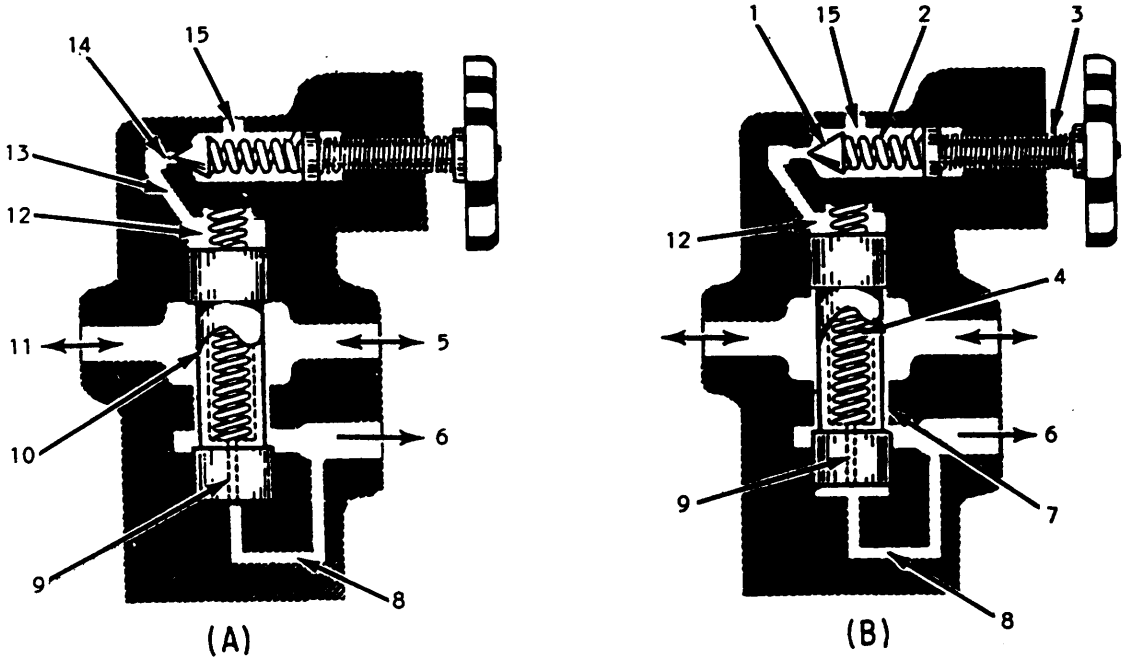
The spring-loaded pressure-reducing valve (fig. 6-18) is commonly used in pneumatic systems. It is often referred to as a pressure regulator.

The valve simply uses spring pressure against a diaphragm to open the valve. On the bottom of the diaphragm, the outlet pressure (the pressure in the reduced-pressure system) of the valve forces the diaphragm upward to shut the valve. When the outlet pressure drops below the set point of the valve, the spring pressure overcomes the outlet pressure and forces the valve stem downward, opening the valve. As the outlet pressure increases, approaching the desired value, the pressure under the diaphragm begins to overcome spring pressure, forcing the valve stem upwards, shutting the valve. You can adjust the downstream pressure by turning the adjusting screw, which varies the spring pressure against the diaphragm. This particular spring-loaded valve will fail in the open position if a diaphragm rupture occurs.

Pilot-Controlled Pressure-Reducing Valve

Figure 6-19 illustrates the operation of a pilot-controlled pressure-reducing valve. This valve consists of an adjustable pilot valve, which controls the operating pressure of the valve, and a spool valve, which reacts to the action of the pilot valve.

The pilot valve consists of a poppet (1), a spring (2), and an adjusting screw (3). The valve



1. POPPET VALVE
2. PILOT VALVE SPRING
3. ADJUSTING SCREW
4. SPOOL VALVE SPRING
5. HIGH-PRESSURE OUTLET PORT
6. REDUCED PRESSURE OUTLET PORT
7. OPENING
8. FLUID PASSAGE
9. FLUID PASSAGE
10. VALVE SPOOL
11. HIGH-PRESSURE INLET PORT
12. FLUID CHAMBER
13. FLUID PASSAGE
14. FLUID CHAMBER
15. DRAIN

Figure 6-19.—Pilot-controlled pressure-reducing valve.

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spool assembly consists of a valve spool (10) and a spring (4).

Fluid under main pressure enters the inlet port (11) and under all conditions is free to flow through the valve and the outlet port (5). (Either port 5 or port 11 maybe used as the high-pressure port.)

Figure 6-19, view A, shows the valve in the open position. In this position, the pressure in the reduced-pressure outlet port (6) has not reached the preset operating pressure of the valve. The fluid also flows through passage 8, through smaller passage 9 in the center of the valve spool, and into chamber 12. The fluid pressure at outlet port 6 is therefore distributed to both ends of the spool. When these pressures are equal the spool is hydraulically balanced. Spring 4 is a low-tension spring and applies only a slight downward force on the spool. Its main purpose is to position the spool and to maintain opening 7 at its maximum size.

As the pressure increases in outlet port 6 (fig. 16, view B), this pressure is transmitted through passages 8 and 9 to chamber 12. This pressure also acts on the pilot valve poppet (1). When this pressure increases above the preset operating pressure of the valve, it overcomes the force of pilot valve spring 2 and unseats the poppet. This allows fluid to flow through the drain port (15). Because the small passage (9) restricts flow into chamber 12, the fluid pressure in the chamber drops. This causes a momentary difference in pressure across the valve spool (10) which allows fluid pressure acting against the bottom area of the valve spool to overcome the downward force of spring 4. The spool is then forced upward until the pressures across its ends are equalized. As the spool moves upward, it restricts the flow through opening 7 and causes the pressure to decrease in the reduced pressure outlet port 6. If the pressure in the outlet port continues to increase to a value above the preset pressure, the pilot valve will open again and the cycle will repeat. This allows the spool valve to move up higher into chamber 12; thus further reducing the size of opening 7. These cycles repeat until the desired pressure is maintained in outlet 6.

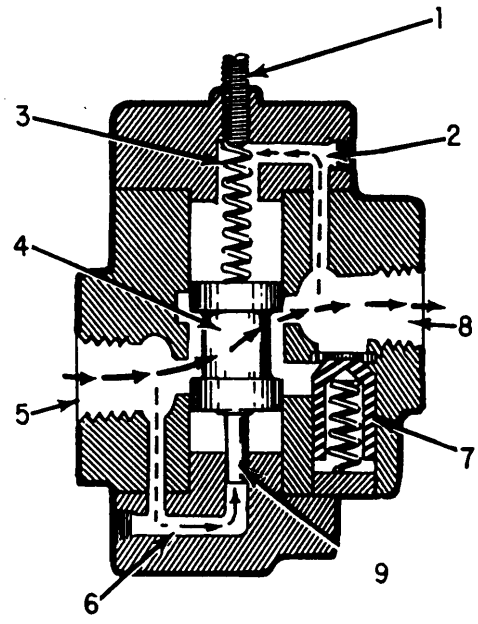
When the pressure in outlet 6 decreases to a value below the preset pressure, spring 4 forces the spool downward, allowing more fluid to flow through opening 7.

COUNTERBALANCE VALVE

The counterbalance valve is normally located in the line between a directional control valve and the outlet of a vertically mounted actuating cylinder which supports weight or must be held in position for a period of time. This valve serves as a hydraulic resistance to the actuating cylinder. For example, counterbalance valves are used in some hydraulically operated forklifts. The valve offers a resistance to the flow from the actuating cylinder when the fork is lowered. It also helps to support the fork in the UP position.

Counterbalance valves are also used in air-launched weapons loaders. In this case the valve is located in the top of the lift cylinder. The valve requires a specific pressure to lower the load. If adequate pressure is not available, the load cannot be lowered. This prevents collapse of the load due to any malfunction of the hydraulic system.

One type of counterbalance valve is illustrated in figure 6-20. The valve element is a balanced spool (4). The spool consists of two pistons permanently fixed on either end of a shaft. The inner surface areas of the pistons are equal; therefore, pressure acts equally on both areas regardless of the position of the valve and has no effect on the movement of the valve—hence, the term *balanced.* The shaft area between the two pistons provides the area for the fluid to flow



1. ADJUSTMENT SCREW
2. INTERNAL DRAIN
3. SPRING
4. SPOOL
5. PRESSURE INLET OR REVERSE FREE FLOW OUTLET
6. PILOT PASSAGE
7. CHECK VALVE
8. DISCHARGE OUTLE OR REVERSE FREE FLOW INLET
9. PISTON

Figure 6-20.—Counterbalance valve.

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when the valve is open. A small piston (9) is attached to the bottom of the spool valve.

When the valve is in the closed position, the top piston of the spool valve blocks the discharge port (8). With the valve in this position, fluid flowing from the actuating unit enters the inlet port (5). The fluid cannot flow through the valve because discharge port 8 is blocked. However, fluid will flow through the pilot passage (6) to the small pilot piston. As the pressure increases, it acts on the pilot piston until it overcomes the preset pressure of spring 3. This forces the valve spool (4) up and allows the fluid to flow around the shaft of the valve spool and out discharge port 8. Figure 6-20 shows the valve in this position. During reverse flow, the fluid enters port 8. The spring (3) forces valve spool 4 to the closed position. The fluid pressure overcomes the spring tension of the check valve (7). The check valve opens and allows free flow around the shaft of the valve spool and out through port 5.

The operating pressure of the valve can be adjusted by turning the adjustment screw (1), which increases or decreases the tension of the spring. This adjustment depends on the weight that the valve must support.

It is normal for a small amount of fluid to leak around the top piston of the spool valve and into the area around the spring. An accumulation would cause additional pressure on top of the spool valve. This would require additional pressure to open the valve. The drain (2) provides a passage for this fluid to flow to port 8.

DIRECTIONAL CONTROL VALVES

Directional control valves are designed to direct the flow of fluid, at the desired time, to the point in a fluid power system where it will do work. The driving of a ram back and forth in its cylinder is an example of when a directional control valve is used. Various other terms are used to identify directional valves, such as selector valve, transfer valve, and control valve. This manual will use the term directional control valve to identify these valves.

Directional control valves for hydraulic and pneumatic systems are similar in design and operation. However, there is one major difference. The return port of a hydraulic valve is ported through a return line to the reservoir, while the similar port of a pneumatic valve, commonly referred to as the exhaust port, is usually vented to the atmosphere. Any other differences are pointed out in the discussion of the valves.

Directional control valves may be operated by differences in pressure acting on opposite sides of the valving element, or they maybe positioned manually, mechanically, or electrically. Often two or more methods of operating the same valve will be used in different phases of its action.

CLASSIFICATION

Directional control valves may be classified in several ways. Some of the different ways are by the type of control, the number of ports in the valve housing, and the specific function of the valve. The most common method is by the type of valving element used in the construction of the valve. The most common types of valving elements are the ball, cone or sleeve, poppet, rotary spool, and sliding spool. The basic operating principles of the poppet, rotary spool, and sliding spool valving elements are discussed in this text.

Poppet

The poppet fits into the center bore of the seat (fig. 6-21). The seating surfaces of the poppet and the seat are lapped or closely machined so that the center bore will be sealed when the poppet is

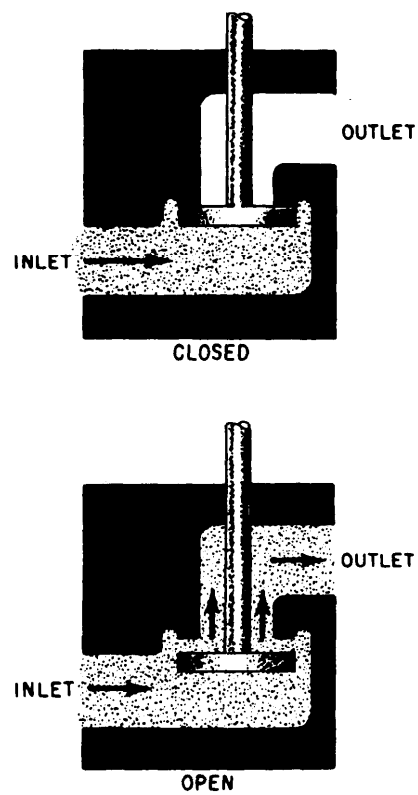

Figure 6-21.—Operation of a simple poppet valve.

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seated (shut). The action of the poppet is similar to that of the valves in an automobile engine. In most valves the poppet is held in the seated position by a spring.

The valve consists primarily of a movable poppet which closes against the valve seat. In the closed position, fluid pressure on the inlet side tends to hold the valve tightly closed. A small amount of movement from a force applied to the top of the poppet stem opens the poppet and allows fluid to flow through the valve.

The use of the poppet as a-valving element is not limited to directional control valves.

Rotary Spool

The rotary spool directional control valve (fig. 6-22) has a round core with one or more passages or recesses in it. The core is mounted within a stationary sleeve. As the core is rotated within the stationary sleeve, the passages or recesses connect or block the ports in the sleeve. The ports in the sleeve are connected to the appropriate lines of the fluid system.

Sliding spool

The operation of a simple sliding spool directional control valve is shown in figure 6-23. The valve is so-named because of the shape of the valving element that slides back and forth to block and uncover ports in the housing. (The sliding element is also referred to as a piston.) The inner piston areas (lands) are equal. Thus fluid under pressure which enters the valve from the inlet ports acts equally on both inner piston areas regardless of the position of the spool. Sealing is usually accomplished by a very closely machined fit between the spool and the valve body or sleeve. For valves with more ports, the spool is designed with more pistons or lands on a common shaft. The sliding spool is the most commonly used type of valving element used in directional control valves.

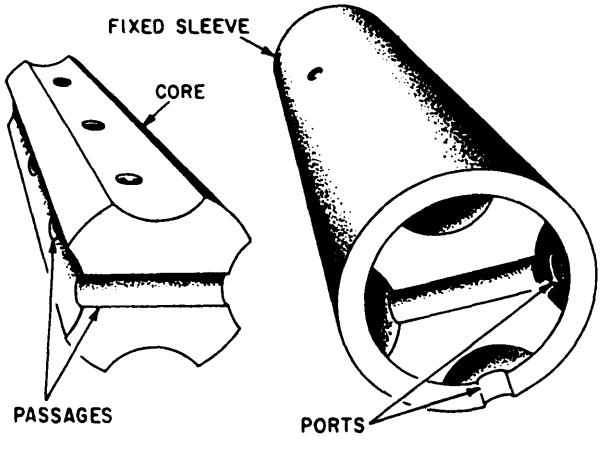

Figure 6-22.—Parts of a rotary spool directional control valve.

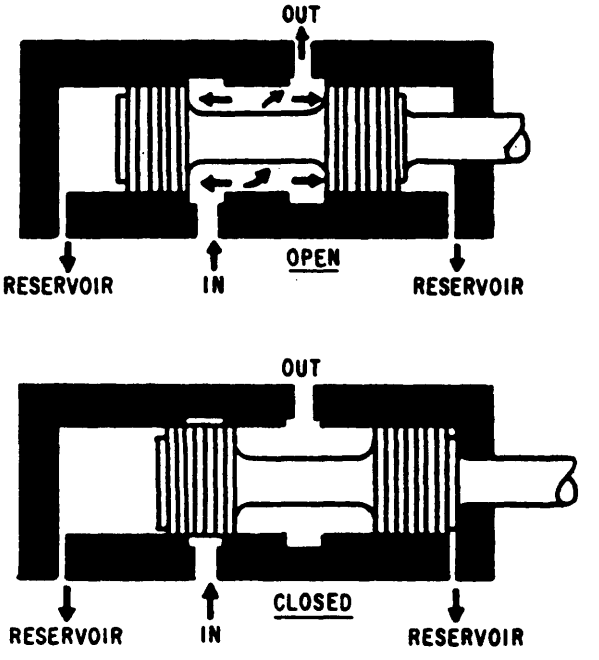

Figure 6-23.—Two-way, sliding spool directional control valve.

CHECK VALVE

Check valves are used in fluid systems to permit flow in one direction and to prevent flow in the other direction. They are classified as one-way directional control valves.

The check valve may be installed independently in a line to allow flow in one direction only, or it may be used as an integral part of globe, sequence, counterbalance, and pressure-reducing valves.

Check valves are available in various designs. They are opened by the force of fluid in motion flowing in one direction, and are closed by fluid attempting to flow in the opposite direction. The force of gravity or the action of a spring aids in closing the valve.

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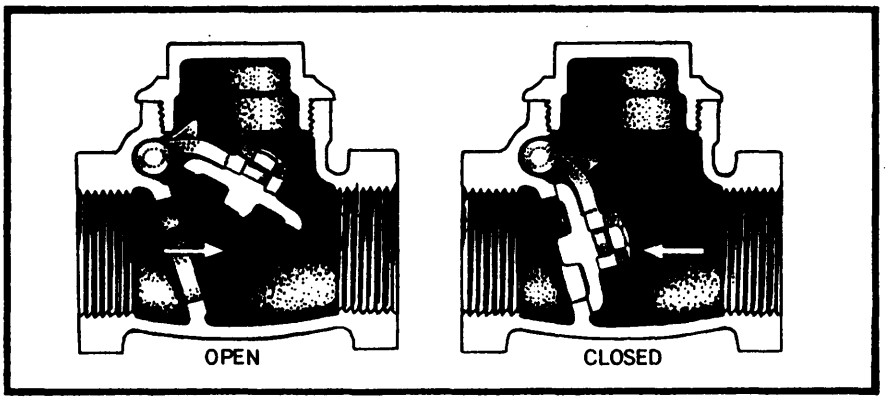

Figure 6-24.—Swing check valve.

Figure 6-24 shows a swing check valve. In the open position, the flow of fluid forces the hinged disk up and allows free flow through the valve. Flow in the opposite direction with the aid of gravity, forces the hinged disk to close the passage and blocks the flow. This type of valve is sometimes designed with a spring to assist in closing the valve.

The most common type of check valve, installed in fluid-power systems, uses either a ball or cone for the sealing element (fig. 6-25). As fluid pressure is applied in the direction of the arrow, the cone (view A) or ball (view B) is forced off its seat, allowing fluid to flow freely through the valve. This valve is known as a spring-loaded check valve.

The spring is installed in the valve to hold the cone or ball on its seat whenever fluid is not flowing. The spring also helps to force the cone or ball on its seat when the fluid attempts to flow in the opposite direction. Since the opening and closing of this type of valve is not dependent on gravity, its location in a system is not limited to the vertical position.

A modification of the spring-loaded check valve is the orifice check valve (fig. 6-26). This

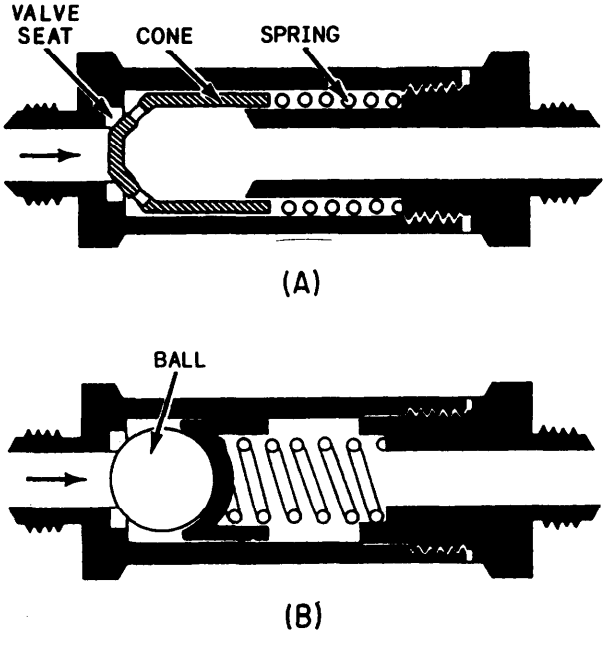

Figure 6-25.—Spring-loaded check valves.

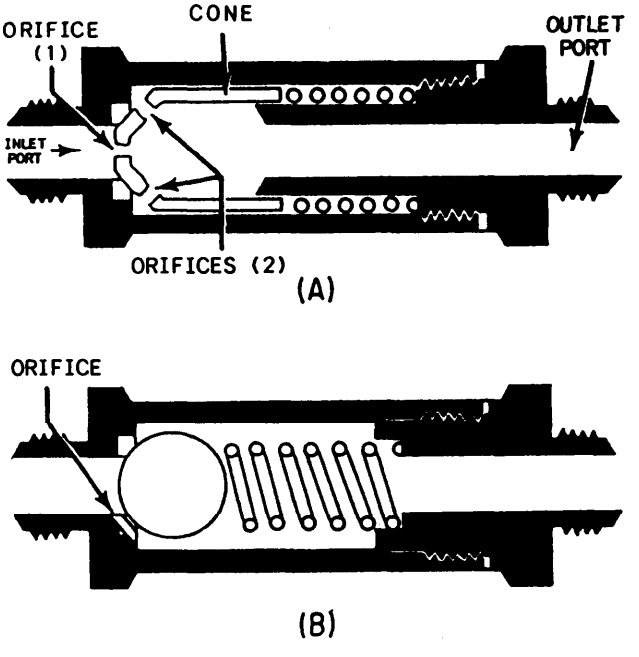

Figure 6-26.—Typical orifice check valves.

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valve allows normal flow in one direction and restricted flow in the other. It is often referred to as a one-way restrictor.

Figure 6-26, view A, shows a cone-type orifice check valve. When sufficient fluid pressure is applied at the inlet port, it overcomes spring tension and moves the cone off of its seat. The two orifices (2) in the illustration represent several openings located around the slanted circumference of the cone. These orifices allow free flow of fluid through the valve while the cone is off of its seat. When fluid pressure is applied through the outlet port, the force of the fluid and spring tension move the cone to the left and onto its seat. This action blocks the flow of fluid through the valve, except through the orifice (1) in the center of the cone. The size of the orifice (in the center of the cone) determines the rate of flow through the valve as the fluid flows from right to left.

Figure 6-26, view B, shows a ball-type orifice check valve. Fluid flow through the valve from left to right forces the ball off of its seat and allows normal flow. Fluid flow through the valve in the opposite direction forces the ball onto its seat. Thus, the flow is restricted by the size of the orifice located in the housing of the valve.

NOTE: The direction of free flow through the orifice check valve is indicated by an arrow stamped on the housing.

SHUTTLE VALVE

In certain fluid power systems, the supply of fluid to a subsystem must be from more than one source to meet system requirements. In some systems an emergency system is provided as a source of pressure in the event of normal system failure. The emergency system will usually actuate only essential components.

The main purpose of the shuttle valve is to isolate the normal system from an alternate or emergency system. It is small and simple; yet, it is a very important component.

Figure 6-27 is a cutaway view of a typical shuttle valve. The housing contains three ports—normal system inlet, alternate or emergency system inlet, and outlet. A shuttle valve used to operate more than one actuating unit may contain additional unit outlet ports. Enclosed in the housing is a sliding part called the shuttle. Its purpose is to seal off either one or the other inlet ports. There is a shuttle seat at each inlet port.

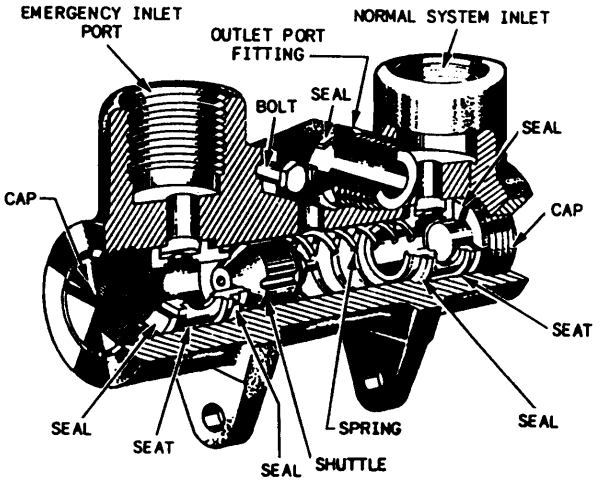

Figure 6-27.—Shuttle valve.

When a shuttle valve is in the normal operation position, fluid has a free flow from the normal system inlet port, through the valve, and out through the outlet port to the actuating unit. The shuttle is seated against the alternate system inlet port and held there by normal system pressure and by the shuttle valve spring. The shuttle remains in this position until the alternate system is activated. This action directs fluid under pressure from the alternate system to the shuttle valve and forces the shuttle from the alternate system inlet port to the normal system inlet port. Fluid from the alternate system then has a free flow to the outlet port, but is prevented from entering the normal system by the shuttle, which seals off the normal system port.

The shuttle may be one of four types: (1) sliding plunger, (2) spring-loaded piston, (3) spring-loaded ball, or (4) spring-loaded poppet. In shuttle valves that are designed with a spring, the shuttle is normally held against the alternate system inlet port by the spring.

TWO-WAY VALVES

The term *two-way* indicates that the valve contains and controls two functional flow control ports-an inlet and an outlet. A two-way, sliding spool directional control valve is shown in figure 6-23. As the spool is moved back and forth, it either allows fluid to flow through the valve or prevents flow. In the open position, the fluid enters the inlet port, flows around the shaft of the spool, and through the outlet port. The spool cannot move back and forth by difference of

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forces set up within the cylinder, since the forces there are equal. As indicated by the arrows against the pistons of the spool, the same pressure acts on equal areas on their inside surfaces. In the closed position, one of the pistons of the spool simply blocks the inlet port, thus preventing flow through the valve.

A number of features common to most sliding spool valves are shown in figure 6-23. The small ports at either end of the valve housing provide a path for any fluid that leaks past the spool to flow to the reservoir. This prevents pressure from building up against the ends of the pistons, which would hinder the movement of the spool. When spool valves become worn, they may lose balance because of greater leakage on one side of the spool than on the other. In that event, the spool would tend to stick when it is moved back and forth. Small grooves are therefore machined around the sliding surface of the piston; and in hydraulic valves, leaking liquid will encircle the pistons and keep the contacting surfaces lubricated and centered.

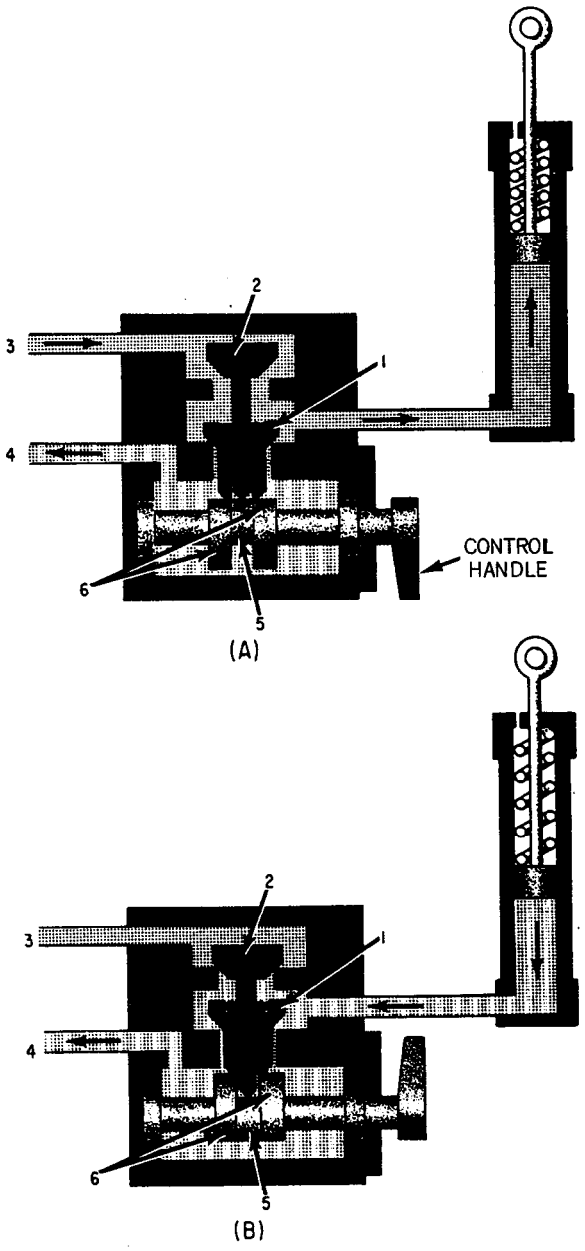
THREE-WAY VALVES

Three-way valves contain a pressure port, a cylinder port, and a return or exhaust port. The three-way directional control valve is designed to operate an actuating unit in one direction; it permits either the load on the actuating unit or a spring to return the unit to its original position.

Cam-Operated Three-Way Valves

Figure 6-28 shows the operation of a cam-operated, three-way, poppet-type directional control valve. View A shows fluid under pressure forcing the piston outward against a load. The upper poppet (2) is unseated by the inside cam (5), permitting fluid to flow from the line (3) into the cylinder to actuate the piston. The lower poppet (1) is seated, sealing off the flow into the return line (4). As the force of the pressurized fluid extends the piston rod, it also compresses the spring in the cylinder.

View B shows the valve with the control handle turned to the opposite position. In this position, the upper poppet (2) is seated, blocking the flow of fluid from the pressure line (3). The lower poppet (1) is unseated by the outside cam (6). This releases the pressure in the cylinder and allows the spring to expand, which forces the piston rod to retract. The fluid from the cylinder flows through the control valve and out the return



1. LOWER (RETURN OR EXHAUST) POPPET
2. UPPER (PRESSURE) POPPET
3. PRESSURE LINE
4. RETURN OR EXHAUST PORT
5. INSIDE CAM
6. OUTSIDE CAM

Figure 6-28.—Three-way, poppet-type directional control valve (cam-operated).

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port (4). In hydraulic systems, the return port is connected by a line to the reservoir. In pneumatic systems, the return port is usually open to the atmosphere.

Pilot-Operated Three-Way Valves

A pilot-operated, poppet-type, three-way directional control valve is shown in figure 6-29. Valves of this design are often used in pneumatic systems. This valve is normally closed and is forced open by fluid pressure entering the pilot chamber. The valve contains two poppets connected to each other by a common stem. The poppets are connected to diaphragms which hold them in a centered position.

The movement of the poppet is controlled by the pressure in the pilot port and the chamber above the upper diaphragm. When the pilot chamber is not pressurized, the lower poppet is seated against the lower valve seat. Fluid can flow from the supply line through the inlet port and through the holes in the lower diaphragm to fill the bottom chamber. This pressure holds the lower poppet tightly against its seat and blocks flow from the inlet port through the valve. At the same time, due to the common stem, the upper poppet is forced off of its seat. Fluid from the actuating unit flows through the open passage, around the stem, and through the exhaust port to the atmosphere.

When the pilot chamber is pressurized, the force acting against the diaphragm forces the poppet down. The upper poppet closes against its seat, blocking the flow of fluid from the cylinder to the exhaust port. The lower poppet opens, and the passage from the supply inlet port to the cylinder port is open so that the fluid can flow to the actuating unit.

The valve in figure 6-29 is a normally closed valve. Normally open valves are similar in design. When no pressure is applied to the pilot chamber, the upper poppet is forced off of its seat and the lower poppet is closed. Fluid is free to flow from the inlet port through the cylinder to the actuating unit. When pilot pressure is applied, the poppets are forced downward, closing the upper poppet and opening the lower poppet. Fluid can now flow from the cylinder through the valve and out the exhaust port to the atmosphere.

FOUR-WAY VALVES

Most actuating devices require system pressure for operation in either direction. The four-way directional control valve, which contains four ports, is used to control the operation of such devices. The four-way valve is also used in some systems to control the operation of other valves. It is one of the most widely used directional control valves in fluid power systems.

The typical four-way directional control valve has four ports: a pressure port, a return or exhaust port, and two cylinder or working ports. The pressure port is connected to the main system pressure line and the return line is connected to the reservoir in hydraulic systems. In pneumatic systems the return port is usually vented to the atmosphere. The two cylinder ports are connected by lines to the actuating units.

Poppet-Type Four-Way Valves

Figure 6-30 shows atypical four-way, poppet-type directional control valve. This is a manually operated valve and consists of a group of conventional spring-loaded poppets. The poppets are enclosed in a common housing and are interconnected by ducts to direct the flow of fluid in the desired direction.

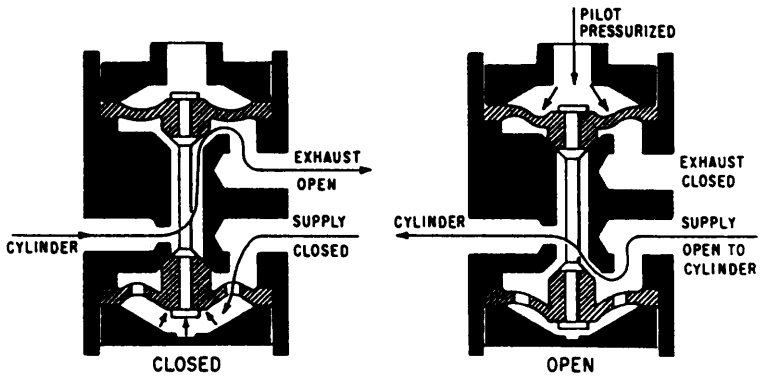

Figure 6-29.—Three-way, poppet-type, normally closed directional control valve (pilot-operated).

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The poppets are actuated by cams on a camshaft (fig. 6-30). The camshaft is controlled by the movement of the handle. The valve may be operated by manually moving the handle, or, in some cases, the handle may be connected by mechanical linkage to a control handle which is located in a convenient place for the operator some distance from the valve.

The camshaft may be rotated to any one of three positions (neutral and two working positions). In the neutral position the camshaft lobes are not contacting any of the poppets. This assures that the poppet springs will hold all four poppets firmly seated. With all poppets seated, there is no fluid flow through the valve. This also blocks the two cylinder ports; so when the valve is in neutral, the fluid in the actuating unit is trapped. Relief valves are installed in both working lines to prevent overpressurization caused by thermal expansion.

NOTE: In some versions of this type of valve, the cam lobes are designed so that the two return/exhaust poppets are open when the valve is in the neutral position. This compensates for thermal expansion, because both working lines are open to the return/exhaust when the valve is in the neutral position.

The poppets are arranged so that rotation of the camshaft will open the proper combination of poppets to direct the flow of fluid through the desired working line to an actuating unit. At the same time, fluid will be directed from the actuating unit through the opposite working line, through the valve, and back to the reservoir (hydraulic) or exhausted to the atmosphere (pneumatic).

To stop rotation of the camshaft at an exact position, a stop pin is secured to the body and extends through a cutout section of the camshaft flange. This stop pin prevents overtravel by ensuring that the camshaft stops rotating at the point where the cam lobes have moved the poppets the greatest distance from their seats and where any further rotation would allow the poppets to start returning to their seats.

O-rings are spaced at intervals along the length of the shaft to prevent external leakage around the ends of the shaft and internal leakage from one of the valve chambers to another. The camshaft has two lobes, or raised portions. The shape of these lobes is such that when the shaft is placed in the neutral position the lobes will not contact any of the poppets.

When the handle is moved in either direction from neutral, the camshaft is rotated. This rotates

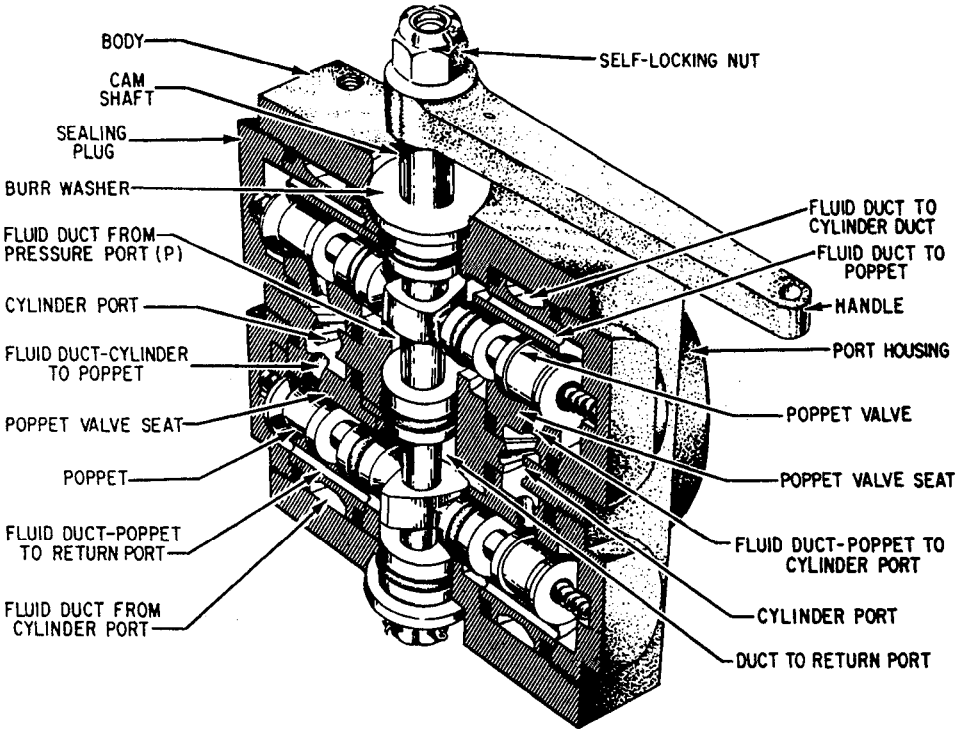

Figure 6-30.—Cutaway view of poppet-type, four-way directional control valve.

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the lobes, which unseat one pressure poppet and one return/exhaust poppet (fig. 6-31). The valve is now in the working position. Fluid under pressure, entering the pressure port, flows through the vertical fluid passages in both pressure poppets seats. Since only one pressure poppet, IN (2), is unseated by the cam lobe, the fluid flows past the open poppet to the inside of the poppet seat. From there it flows through the diagonal passages, out one cylinder port, C2, and to the actuating unit.

Return fluid from the actuating unit enters the other cylinder port, C1. It then flows through the corresponding fluid passage, past the unseated return poppet, OUT (1), through the vertical fluid passages, and out the return/exhaust port. When the camshaft is rotated in the opposite direction to the neutral position, the two poppets seat and the flow stops. When the camshaft is further rotated in this direction until the stop pins hits, the opposite pressure and return poppets are unseated. This reverses the flow in the working lines, causing the actuating unit to move in the opposite direction.

Rotary Spool Valve

Four-way directional control valves of this type are frequently used as pilot valves to direct flow to and from other valves (fig. 6-32). Fluid is directed from one source of supply through the rotary valve to another directional control valve, where it positions the valve to direct flow from another source to one side of an actuating unit. Fluid from the other end of the main valve flows through a return line, through the rotary valve to the return or exhaust port.

The principal parts of a rotary spool directional control valve are shown in figure 6-22.

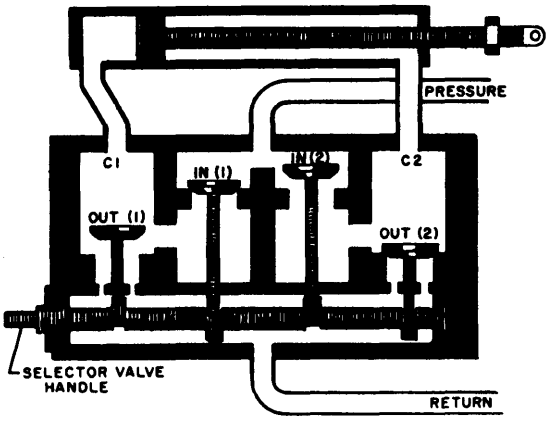

Figure 6-31.—Working view of a poppet-type, four-way directional control valve.

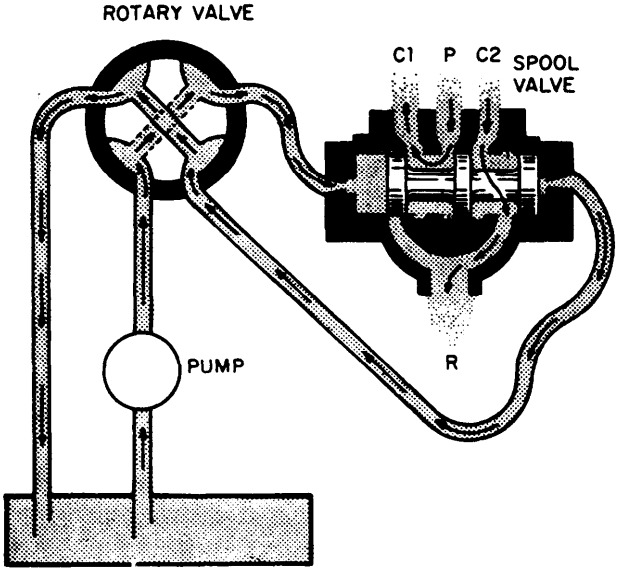

Figure 6-32.—Sliding spool valve controlled by a rotary spool valve.

Figure 6-33 shows the operation of a rotary spool valve. Views A and C show the valve in a position to deliver fluid to another valve, while view B shows the valve in the neutral position, with all passages through the valve blocked.

Rotary spool valves can be operated manually, electrically, or by fluid pressure.

Sliding Spool Valve

The sliding spool four-way directional control valve is similar in operation to the two-way valve previously described in this chapter. It is simple in its principle of operation and is the most durable and trouble-free of all four-way directional control valves.

The valve described in the following paragraphs is a manually operated type. The same principle is used in many remotely controlled directional control valves.

The valve (fig. 6-34) consists of a valve body containing four fluid ports—pressure (P),

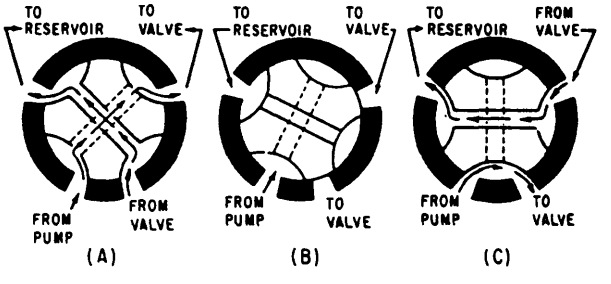

Figure 6-33.—Operation of a rotary spool, four-way directional control valve.

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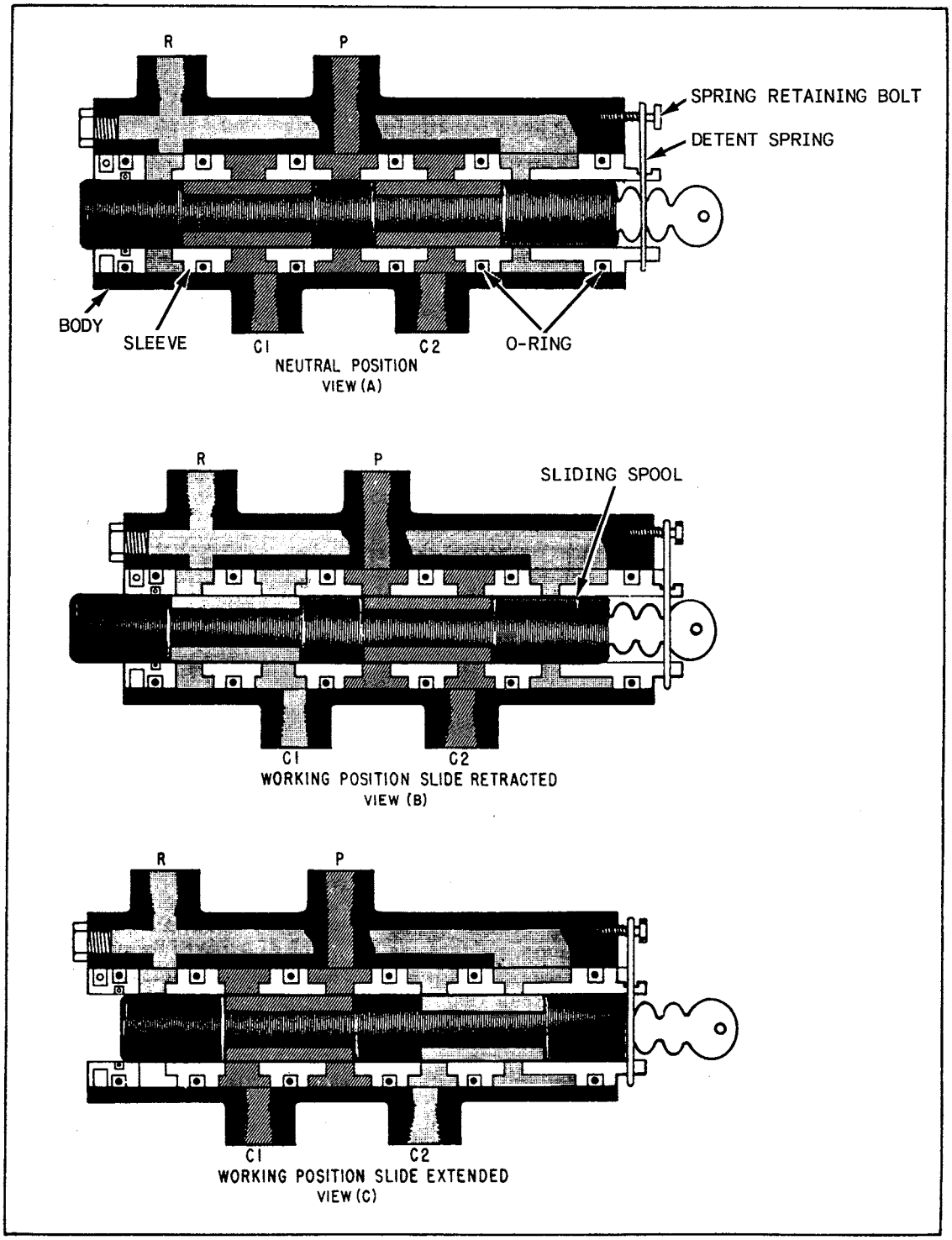

Figure 6-34.—Operation of a sliding spool, four-way directional control valve.

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return/exhaust (R), and two cylinder ports (C/1 and C2). A hollow sleeve fits into the main bore of the body. There are O-rings placed at intervals around the outside diameter of the sleeve. These O-rings form a seal between the sleeve and the body, creating chambers around the sleeve. Each of the chambers is lined up with one of the fluid ports in the body. The drilled passage in the body accounts for a fifth chamber which results in having the two outboard chambers connected to the return/exhaust port. The sleeve has a pattern of holes drilled through it to allow fluid to flow from one port to another. A series of holes are drilled into the hollow center sleeve in each chamber.

The sleeve is prevented from turning by a sleeve retainer bolt or pin which secures it to the valve body.

The sliding spool fits into the hollow center sleeve. This spool is similar to the spool in the two-way valve, except that this spool has three pistons or lands. These lands are lapped or machine fitted to the inside of the sleeve.

One end of the sliding spool is connected to a handle either directly or by mechanical linkage to a more desirable location. When the control handle is moved, it will position the spool within the sleeve. The lands of the spool then line up different combinations of fluid ports thus directing a flow of fluid through the valve.

The detent spring is a clothespin-type spring, secured to the end of the body by a spring retaining bolt. The two legs of the spring extend down through slots in the sleeve and fit into the detents. The spool is gripped between the two legs of the spring. To move the spool, enough force must be applied to spread the two spring legs and allow them to snap back into the next detent, which would be for another position.

Figure 6-34, view A, shows a manually operated sliding spool valve in the neutral position. The detent spring is in the center detent of the sliding spool. The center land is lined up with the pressure port (P) preventing fluid from flowing into the valve through this port. The return/exhaust port is also blocked, preventing flow through that port. With both the pressure and return ports blocked, fluid in the actuating lines is trapped. For this reason, a relief valve is usually installed in each actuating line when this type of valve is used.

Figure 6-34, view B, shows the valve in the working position with the end of the sliding spool retracted. The detent spring is in the outboard detent, locking the sliding spool in this position. The lands have shifted inside the sleeve, and the ports are opened. Fluid under pressure enters the sleeve, passes through it by way of the drilled holes, and leaves through cylinder port C2. Return fluid, flowing from the actuator enters port C1, flows through the sleeve, and is directed out the return port back to the reservoir or exhausted to the atmosphere. Fluid cannot flow past the spool lands because of the lapped surfaces.

Figure 6-34, view C, shows the valve in the opposite working position with the sliding spool extended. The detent spring is in the inboard detent. The center land of the sliding spool is now on the other side of the pressure port, and the fluid under pressure is directed through the sleeve and out port C1. Return fluid flowing in the other cylinder port is directed to the drilled passage in the body. It flows along this passage to the other end of the sleeve where it is directed out of the return/exhaust port.

The directional control valves previously discussed are for use in closed-center fluid power systems. Figure 6-35 shows the operation of

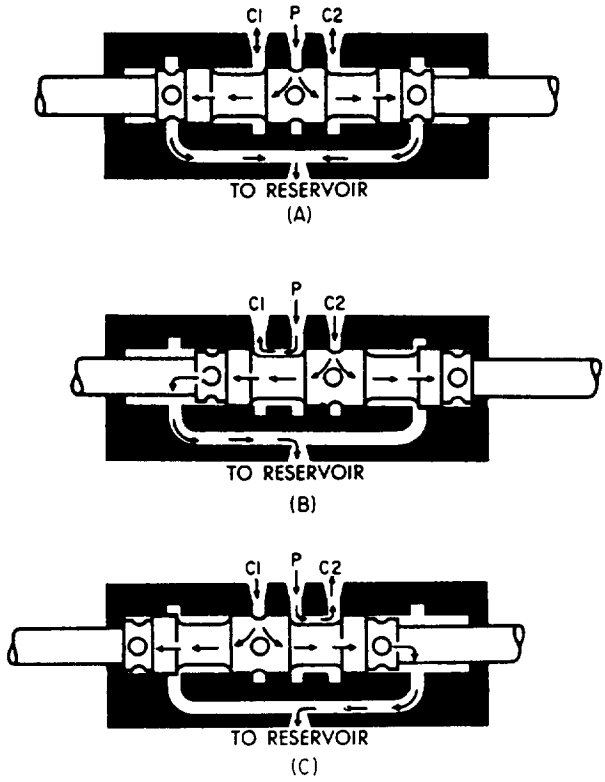

Figure 6-35.—Open center, sliding spool directional control valve.

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a representative open-center, sliding spool directional control valve.

When this type of valve is in the neutral position (fig. 6-35, view A), fluid flows into the valve through the pressure port (P) through the hollow spool, and return to the reservoir.

When the spool is moved to the right of the neutral position, view B, one working line (C1) is aligned to system pressure and the other working line (C2) is open through the hollow spool to the return port. View C shows the flow of fluid through the valve with the spool moved to the left of neutral.

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CHAPTER 7

SEALING DEVICES AND MATERIALS

Recall from chapter 1 that Pascal's theorem, from which the fundamental law for the science of hydraulics evolved, was proposed in the seventeenth century. One stipulation to make the law effective for practical applications was a piston that would "fit" the opening in the vessel "exactly." However, it was not until the late eighteenth century that Joseph Brahmah invented an effective piston seal, the cup packing. This led to Brahmah's development of the hydraulic press.

The packing was probably the most important invention in the development of hydraulics as a leading method of transmitting power. The development of machines to cut and shape closely fitted parts was also very important in the development of hydraulics. However, regardless of how precise the machining process is, some type of packing is usually required to make the piston, and many other parts of hydraulic components, "fit exactly." This also applies to the components of pneumatic systems.

Through years of research and experiments, many different materials and designs have been created in attempts to develop suitable packing devices. Suitable materials must be durable, must provide effective sealing, and must be compatible with the fluid used in the system.

The packing materials are commonly referred to as seals or sealing devices. The seals used in fluid power systems and components are divided into two general classes-static seals and dynamic seals.

The static seal is usually referred to as a gasket. The function of a gasket is to provide a material that can flow into the surface irregularities of mating areas that require sealing. To do this, the gasket material must be under pressure. This requires that the joint be tightly bolted or otherwise held together.

The dynamic seal, commonly referred to as a packing, is used to provide a seal between two parts that move in relation to each other.

These two classifications of seals—gaskets and packing—apply in most cases; however, deviations are found in some technical publications. Certain types of seals (for example, the O-ring, which is discussed later) may be used either as a gasket or a packing.

Many of the seals in fluid power systems prevent external leakage. These seals serve two purposes—to seal the fluid in the system and to keep foreign matter out of the system. Other seals simply prevent internal leakage within a system.

NOTE: Although leakage of any kind results in a loss of efficiency, some leakage, especially internal leakage, is desired in hydraulic systems to provide lubrication of moving parts. This also applies to some pneumatic systems in which drops of oil are introduced into the flow of air in the system.

The first part of this chapter deals primarily with the different types of materials used in the construction of seals. The next section is devoted to the different shapes and designs of seals and their application as gaskets and/or packings in fluid power systems. Also included in this chapter are sections concerning the functions of wipers and backup washers in fluid power systems and the selection, storage, and handling of sealing devices.

SEAL MATERIALS

As mentioned previously, many different materials have been used in the development of sealing devices. The material used for a particular application depends on several factors: fluid compatibility, resistance to heat, pressure, wear resistance, hardness, and type of motion.

The selection of the correct packings and gaskets and their proper installation are important factors in maintaining an efficient fluid power system. The types of seals to be used in a particular piece of equipment is specified by the equipment manufacturer.

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Often the selection of seals is limited to seals covered by military specifications. However, there are occasions when nonstandard or proprietary seals reflecting the advancing state of the art may be approved. Thus, it is important to follow the manufacturer's instructions when you replace seals. If the proper seal is not available, you should give careful consideration in the selection of a suitable substitute. Consult the *Naval Ships' Technical Manual*, military standards, military standardization handbooks, and other applicable technical manuals if you have any doubts in selecting the proper seal.

Seals are made of materials that have been carefully chosen or developed for specific applications. These materials include tetrafluoroethylene (TFE), commonly called Teflon; synthetic rubber (elastomers); cork; leather; metal; and asbestos. Some of the most common materials used to make seals for fluid power systems are discussed in the following paragraphs.

CORK

Cork has several of the required properties, which makes it ideally suited as a sealing material in certain applications. The compressibility of cork seals makes them well suited for confined applications in which little or no spread of the material is allowed. The compressibility of cork also makes a good seal that can be cut to any desired thickness and shape to fit any surface and still provide an excellent seal.

One of the undesirable characteristics of cork is its tendency to crumble. If cork is used as packing or in areas where there is a high fluid pressure and/or high flow velocity, small particles will be cast off into the system. Cork use in fluid power systems is therefore limited. It is sometimes used as gasket materials for inspection plates of hydraulic reservoirs.

Cork is generally recommended for use where sustained temperatures do not exceed 275°F.

CORK AND RUBBER

Cork and rubber seals are made by combining synthetic rubber and cork. This combination has the properties of both of the two materials. This means that seals can be made with the compressibility of cork, but with a resistance to fluid comparable to the synthetic rubber on which they are based. Cork and rubber composition is sometimes used to make gaskets for applications similar to those described for cork gaskets.

LEATHER

Leather is a closely knit material that is generally tough, pliable, and relatively resistant to abrasion, wear, stress, and the effects of temperature changes. Because it is porous, it is able to absorb lubricating fluids. This porosity makes it necessary to impregnate leather for most uses. In general, leather must be tanned and treated in order to make it useful as a gasket material. The tanning processes are those normally used in the leather industry.

Leather is generally resistant to abrasion regardless of whether the grain side or the flesh side is exposed to abrasive action. Leather remains flexible at low temperatures and can be forced with comparative ease into contact with metal flanges. When properly impregnated, it is impermeable to most liquids and some gases, and capable of withstanding the effects of temperatures ranging from –70°F to +220°F.

Leather has four basic limitations. First, the size of the typical hide limits the size of the seals that can be made from leather. A second limitation is the number of seals that are acceptable. Another limitation is that under heavy mechanical pressures leather tends to extrude. Finally, many of the properties (such as impermeability, tensile strength, high- and low-temperature resistance, pliability, and compatibility with environment) depend upon the type of leather and impregnation. Leathers not tanned and impregnated for specific conditions and properties will become brittle, dry, and completely degreased by exposure to particular chemicals. Leather is never used with steam pressure of any type, nor with acid or alkali solutions.

Leather may be used as packing. When molded into *V's* and *U's*, and cups, and other shapes, it can be applied as dynamic packing, while in its flat form it can be used as straight compression packing.

METAL

One of the most common metal seals used in Navy equipment is copper. Flat copper rings are sometimes used as gaskets under adjusting screws to provide a fluid seal. Molded copper rings are sometimes used as packing with speed gears operating under high pressures. Either type is

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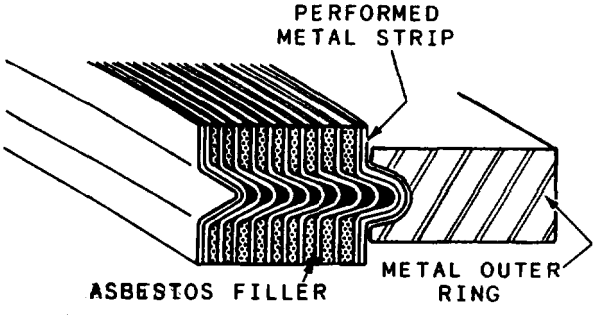

Figure 7-1.—Spiral-wouna metallic-asbestos gasket.

easily bent and requires careful handling. In addition, copper becomes hard when used over long periods and when subjected to compression. Whenever a unit or component is disassembled, the copper sealing rings should be replaced. However, if new rings are not available and the part must be repaired, the old ring should be softened by annealing. (Annealing is the process of heating a metal, then cooling it, to make it more pliable and less brittle.)

Metallic piston rings are used as packing in some fluid power actuating cylinders. These rings are similar in design to the piston rings in automobile engines.

Metal is also used with asbestos to form spiral-wound metallic-asbestos gaskets (fig. 7-1). These gaskets are composed of interlocked plies of preformed corrugated metal and asbestos strips, called a filler.

The filler may or may not be encased in a solid metal outer ring. These gaskets are used in flanged connections and for connecting the body to the bonnet in some valves, and are usually required in specific high-pressure, high-temperature applications.

RUBBER

The term *rubber* covers many natural and synthetic rubbers, each of which can be compounded into numerous varieties. The characteristics of these varieties have a wide range, as shown in table 7-1. The table shows, with the exception of a few basic similarities, that rubbers have diverse properties and limitations; therefore, specific applications require careful study before the sealing material is selected.

Natural rubbers have many of the characteristics required in an effective seal. However, their very poor resistance to petroleum fluids and rapid aging when exposed to oxygen or ozone limit their use. Today their use has almost ceased.

There are two general classes of synthetic rubber seals. One class is made entirely of a certain synthetic rubber. The term homogeneous, which means having uniform structure or composition throughout, is frequently used to describe this class of seal. The other class of seal is made by impregnating woven cotton duck or fine-weave asbestos with synthetic rubber. This class is sometimes referred to as fabricated seals.

Additional information on sealing materials is provided in the *Military Handbook, Gasket Materials (Nonmetalic),* MIL-HDBK-212; and the *Naval Ships' Technical Manual,* chapter 078.

TYPES OF SEALS

Fluid power seals are usually typed according to their shape or design. These types include T-seals, V-rings, O-rings, U-cups and so on. Some of the most commonly used seals are discussed in the remainder of this chapter.

T-SEALS

The T-seal has an elastomeric bidirectional sealing element resembling an inverted letter *T*. This sealing element is always paired with two special extrusion-resisting backup rings, one on each side of the *T*. The basic T-seal configuration is shown in figure 7-2, view A. The backup rings

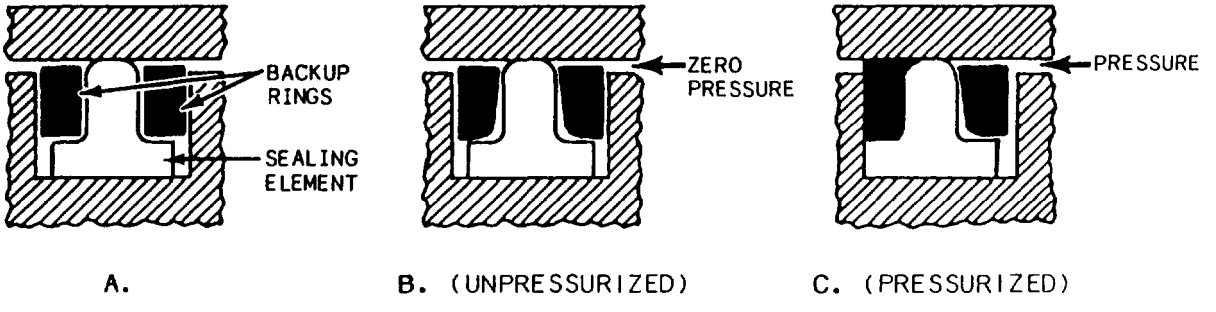

Figure 7-2.–T-seals.

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Table 7-l.—Comparison of Physical Properties for Some Hydraulic Fluid Seal Materials

Seal Materials / Application	Nitrile (Buna N)[3]	Styrene Butadiene (SBR, Buna S)	Butyl Rubber	Chloroprene (Neoprene)	Ethylene Propylene Rubber	Fluorocarbon Rubber[2]	Natural Rubber	Silicone	TFE (Plastic)[4]
Alkyl Aryl Phosphate (Skydrol 500)	P	P	F	P	E	P	P	F	G
Tri Aryl Phosphate (MIL-H-19457)	P	P	E	P	E	E	P	PF	E
Petroleum Oil[1]	G	P	P	FG	P	E	P	PG	E
Synthetic Hydrocarbon (MIL-H-83282)	E	P	P	F	P	E	P	PG	E
Impermeability	G	F	E	G	G	G	F	P	G
Cold Resistance	G	F	G	G	GE	F	G	E	E
Tear Resistance	FG	PF	FG	FG	GE	F	GE	PF	E
Abrasion Resistance	G	G	FG	GE	GE	G	E	PF	E
Set Resistance	GE	F	FG	F	GE	GE	G	GE	P
Dynamic Properties	GE	G	F	F	GE	GE	G	P	P
Tensile Strength	GE	GE	G	G	GE	GE	E	P	E
Water/Steam Resistance	FG	FG	G	F	E	FG	FG	F	E
Heat Resistance	G	F	GE	F	E	E	F	E	E

LEGEND: E—Excellent; G—Good; F—Fair; P—Poor

NOTES:
1. Includes MIL-L-17672, MIL-L-17331, and MIL-H-5606. Compatibility with specific petroleum fluid should be checked.
2. The material is supplied under various trade names such as VITON (Du Pont) or FLUOREL (Minnesota Mining and Manufacturing Co.) MIL-R-25897, MIL-R-83248, and MIL-G-23652 apply.
3. Includes MIL-P-5516, MIL-P-25732, and MIL-P-5510.
4. Polytetrafluoroethylene, commonly identified as TEFLON (Du Pont trade name).

Figure 7-3.—V-rings.

are single turn, bias cut, and usually made of TFE, molybdenum-disulfide-impregnated nylon, or a combination of TFE and nylon. Nylon is widely used for T-seal backup rings because it provides excellent resistance to extrusion and has low friction characteristics.

The special T-ring configuration adds stability to the seal, eliminating spiraling and rolling.

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T-seals are used in applications where large clearances could occur as a result of the expansion of the thin-walled hydraulic cylinder. The T-ring is installed under radial compression and provides a positive seal at zero or low pressure. Backup rings, one on each side, ride free of T-ring flanges and the rod or cylinder wall (fig. 7-2, view B). These clearances keep seal friction to a minimum at low pressure. When pressure is applied (fig. 7-2, view C), the T-ring acts to provide positive sealing action as fluid pressure increases. One frequently used T-ring, manufactured by Greene, Tweed and Company, (called a G-Tring®[1]), incorporates a unique, patented backup ring feature. One corner on the ID of each radius-styled backup ring on the G-Tring® set has been rounded to mate with the inside corner of the rubber *T*. Figure 7-2, views B and C, shows the G-Tring®.

There is no military standard part numbering system by which T-seals can be identified. In general, each manufacturer issues proprietary part numbers to identify seals. However, it is common practice to identify T-seal sizes by the same dash numbers used for equivalent O-ring sizes (discussed later in this chapter) as defined by AS568 and MS28775 dimension standards. Typically, an O-ring groove that accepts a certain O-ring dash number will accept the same dash number T-seal.

In the absence of an existing military standard for identifying T-seals, a new and simple numbering system was created to identify T-seals required for hydraulic actuators (piston seals only) without reference to a particular manufacturer's part number. The Navy number is composed of the letters *G-T* followed by a dash number of three digits and one letter, *R*, *S*, or *T* (for example, G-T-217T). The three digits are the appropriate O-ring size dash number according to AS568 or MS28775. The letters *R*, *S*, and *T* designate the number of backup rings that the groove of the T-seal is designed to accommodate: none, one, or two, respectively.

V-RINGS

The V-ring is one of the most frequently used dynamic seals in ship service although its identification, installation, and performance are probably most misunderstood. Properly selected and installed, V-rings can provide excellent service life; otherwise, problems associated with friction, rod and seal wear, noise, and leakage can be expected.

The V-ring is the part of the packing set that does the sealing. It has a cross section resembling the letter *V*, (fig. 7-3) from which its name is derived. To achieve a seal, the V-ring must be installed as part of a packing set or stack, which includes one male adapter, one female adapter, and several V-rings (fig. 7-4). The male adapter is the first ring on the pressure end of the packing stack and is flat on one side and wedge-shaped on the other to contain the *V* of the adjacent V-ring. The female adapter, the last ring of the

[1]**G-Tring® is a Greene, Tweed Trademark,**

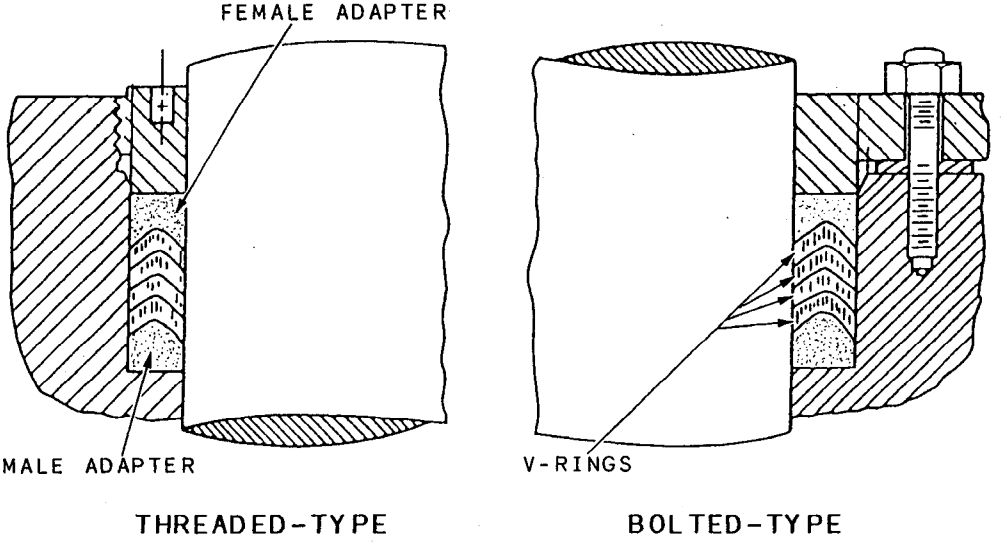

Figure 7-4.—Outside packed V-ring installations.

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packing stack, is flat on one side and V-shaped on the other to properly support the adjacent V-ring. Proper design and installation of the female adapter has significant impact on the service life and performance of the V-rings because the female adapter bridges the clearance gap between the moving surfaces and resists extrusion.

The packing set is installed in a cavity that is slightly deeper than the free stack height (the nominal overall height of a V-ring packing set, including the male and female adapters as measured before installation) and as wide as the nominal cross section of the V-rings. This cavity, called a packing gland or stuffing box, contains and supports the packing around the shaft, rod, or piston. Adjustment of the packing gland depth through the use of shims or spacers is usually necessary to obtain the correct squeeze or clearance on the packing stack for good service life.

Two basic installations apply to V-ring packings. The more common is referred to as an outside packed installation, in which the packing seals against a shaft or rod, as shown in figure 7-4. The inside packed installation, is shown as a piston seal in figure 7-5. When V-ring packing is to be used in an inside packed installation, only endless ring packing should be used. Where pressures exist in both directions, as on a double-acting piston, opposing sets of packing should always be installed so the sealing lips face away from each other as in figure 7-5. This prevents trapping pressure between the sets of packings. The female adapters in inside packed installations should always be located adjacent to a fixed or rigid part of the piston.

O-RINGS

An O-ring is doughnut-shaped. O-rings are usually molded from rubber compounds; however, they can be molded or machined from plastic materials. The O-ring is usually fitted into a rectangular groove (usually called a gland) machined into the mechanism to be sealed. An O-ring seal consists of an O-ring mounted in the gland so that the O-ring's cross section is compressed (squeezed) when the gland is assembled (fig. 7-6).

An O-ring sealing system is often one of the first sealing systems considered when a fluid closure is designed because of the following advantages of such a system:

1. Simplicity
2. Ruggedness
3. Low cost
4. Ease of installation
5. Ease of maintenance
6. No adjustment required
7. No critical torque in clamping

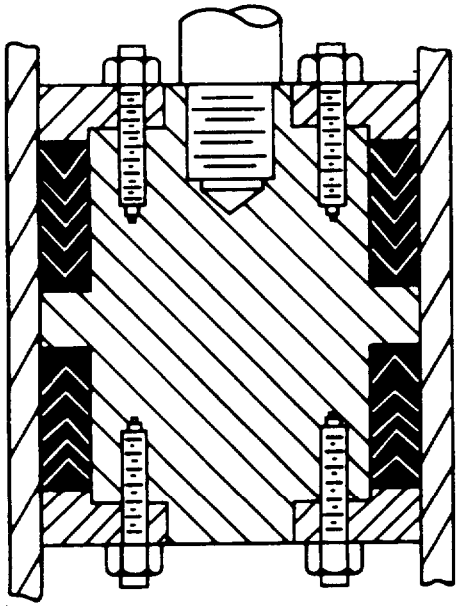

Figure 7-5.—Inside packed V-ring installation.

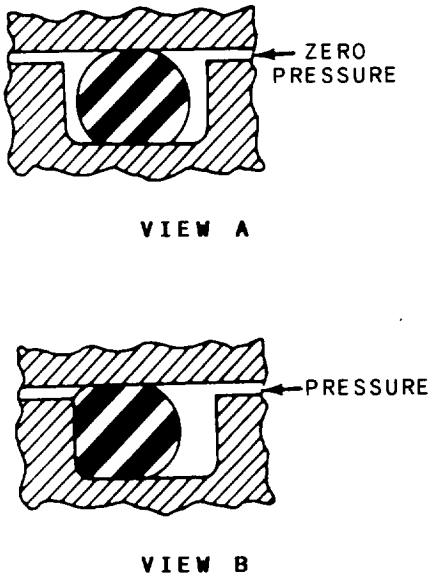

Figure 7-6.—O-ring installed in a gland.

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8. Low distortion of structure
9. Small space requirement
10. Reliability
11. Effectiveness over wide pressure and temperature ranges

As stated previously, O-rings are used in both static (as gaskets) and dynamic (as packing) applications. An O-ring will almost always be the most satisfactory choice of seals in static applications if the fluids, temperatures, pressure, and geometry permit.

Standard O-ring packings are not specifically designed to be used as rotary seals. When infrequent rotary motion or low peripheral velocity is involved standard O-ring packings may be used, provided consistent surface finishes over the entire gland are used and eccentricities are accurately controlled. O-rings cannot compensate for out-of-round or eccentrically rotating shafts.

As rotary seals, O-rings perform satisfactorily in two application areas:

1. In low-speed applications where the surface speed of the shaft does not exceed 200 ft/min

2. In high-speed moderate-pressure applications, between 50 and 800 psi

The use of low-friction extrusion-resistant devices is helpful in prolonging the life and improving the performance of O-rings used as rotary seals.

O-rings are often used as reciprocating seals in hydraulic and pneumatic systems. While best suited for short-stroke, relatively small diameter applications, O-rings have been used successfully in long-stroke, large diameter applications. Glands for O-rings used as reciprocating seals are usually designed according to MIL-G-5514 to provide a squeeze that varies from 8 to 10 percent minimum and 13.5 to 16 percent maximum. A squeeze of 20 percent is allowed on O-rings with a cross section of 0.070-inch or less. In some reciprocating pneumatic applications, a floating O-ring design may simultaneously reduce friction and wear by maintaining no squeeze by the gland on the O-ring. When air pressure enters the cylinder, the air pressure flattens the O-ring, causing sufficient squeeze to seal during the stroke. If the return stroke does not use pneumatic power, the O-ring returns to its round cross section, minimizing drag and wear on the return stroke.

Identification

As a maintenance person or supervisor working with fluid power systems, you must be able to positively identify, inspect, and install the correct size and type of O-ring to ensure the best possible service. These tasks can be difficult since part numbers cannot be put directly on the seals and because of the continual introduction of new types of seals and obsolescence of others. *(Naval Ships' Technical Manual,* chapter 078, contains a table that cross-references obsolete and current O-ring specifications for ship applications.)

O-rings are packaged in individually sealed envelopes. O-ring seals manufactured to government specifications are marked according to the requirements of the specific military specification and standard. The required marking for each package is as follows:

1. National stock number (NSN)
2. Nomenclature
3. Military part number
4. Material specification
5. Manufacturer's name
6. Manufacturer's compound number
7. Manufacturer's batch number
8. Contract number
9. Cure date

NOTE: Keep preformed packings in their original envelopes, which provide preservation, protection, identification, and cure date.

When you select an O-ring for installation, carefully observe the information on the package. If you cannot positively identify an O-ring, discard it. The part number on the sealed package provides the most reliable and complete identification.

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Sizes

A standardized dash number system for O-ring sizes is used in many military and industrial specifications. The O-ring size is identified by a dash number rather than the actual dimensions for convenience. The basis for the dash numbers is contained in Aerospace Standard AS568. For nongasket O-rings (packing), the dash numbers are divided into groups of one hundred. Each hundred group identifies the cross section size of the O-rings within the group (table 7-2).

The 900 series dash numbers contained in AS568 identify all the presently standardized straight thread tube fitting boss gaskets. With the exception of -901, the last two digits of the dash designate the tube size in 16ths of an inch. For example, the -904 size is for a 1/4-inch tube.

Dimensions

The critical dimensions of an O-ring are its ID, its cross sectional diameter (W), and the height and width of the residual molding flash (see fig. 7-7).

Nominal dimensions have been used to describe O-ring sizes, although this practice is rapidly being replaced by the use of dash numbers. The actual inside diameter of a seal will be slightly less than the nominal ID, but the actual OD will be slightly larger than the nominal OD. For example, an AS568-429 O-ring is described in nominal dimensions as 5 inches ID by 5-1/2 inches OD by 1/4-inch W. Actual dimensions are 4.975 inches ID by 5.525 inches OD by 0.275 inches W.

Table 7-2.—O-Ring Dash Numbers Versus Cross Section Sizes

Dash Number	Cross Section (inches)
-001 thru -099	0.070 and smaller
-100 thru -199	0.103
-200 thru -299	0.139
-300 thru -399	0.210
-400 thru -499	0.275

Specifications

Material and performance requirements for O-rings are often identified in military specifications. The dimensions of these O-rings will usually be found in accompanying slash sheets (which bear the specification number and are a part of the specification) or will be identified by various drawings and standards that relate to the specification. Included among the specifications are Air Force-Navy Standards (AN), Military Standards (MS), and National Aerospace Standards (NAS). If the specification does not identify sizes, the sizes should be identified by the AS568 dash number. Usually, you can use drawings, technical manuals, and allowance parts lists (APLs) to identify replacement O-rings. (Notes 2 and 3 of table 7-1 list some of the frequently used military specifications).

Cure Date

A cure date is as applicable to natural or synthetic O-rings as it is to rubber hoses. This date is the basis for determining the age of O-rings. It is extremely important that the cure date be noted on all packages.

Shelf Life and Expiration Date

All elastomers change gradually with age; some change more rapidly than others. The shelf life for rubber products is contained in MIL-HDBK-695.

Check the age of natural or synthetic rubber preformed packings before installation to determine whether they are acceptable for use. Make a positive identification, indicating the source, cure date, and expiration date. Ensure that this information is available for all packing used. Shelf life requirements do not apply once the packing is installed in a component.

The expiration date is the date after which packing should not be installed. The expiration date of all packings can be determined by adding the shelf life to the cure date.

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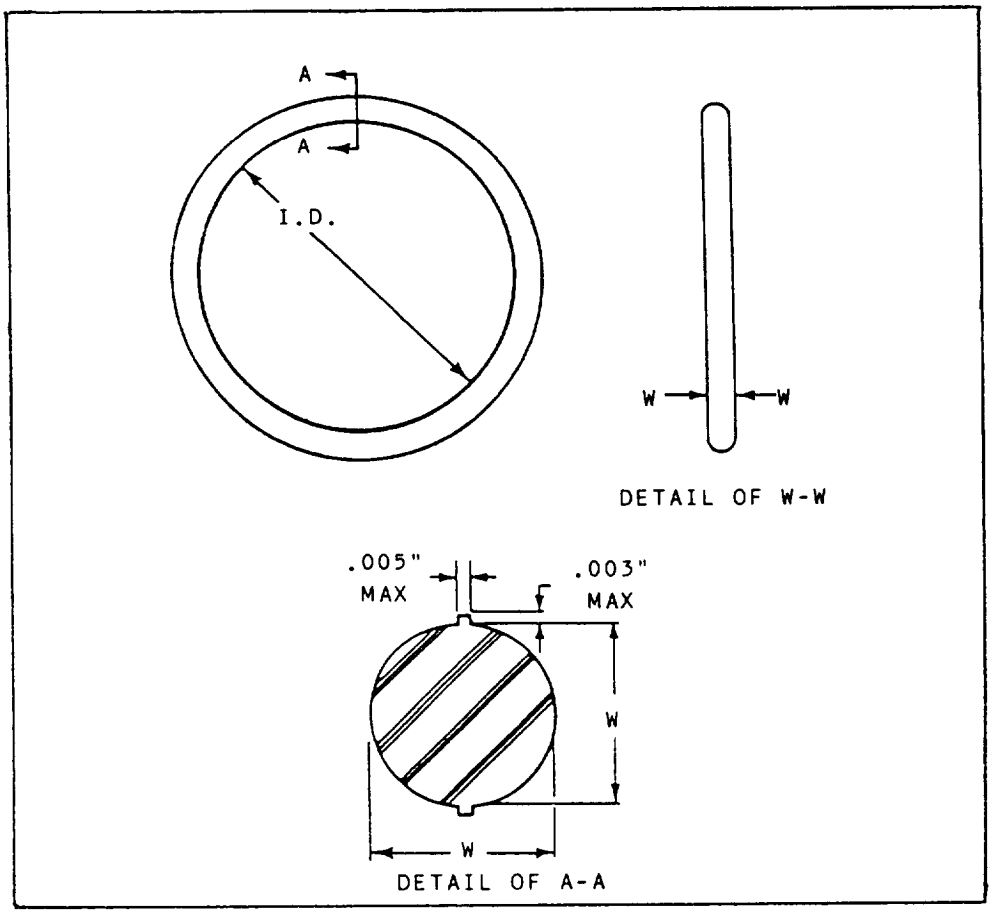

Figure 7-7.—Critical dimensions of an O-ring.

Replacement

Figure 7-8 shows a typical O-ring installation. When such an installation shows signs of internal or external leakage, the component must be disassembled and the seals replaced. Sometimes components must be resealed because of the age limitations of the seals. The O-ring should also be replaced whenever a gland that has been in service is disassembled and reassembled.

Often a poor O-ring installation begins when an old seal is removed. O-ring removal involves working with parts that have critical surface finishes. If hardened-steel, pointed, or sharp-edged tools are used for removal of O-rings or backup rings, scratches, abrasions, dents, and other deformities on critical sealing surfaces can result in seal failure which, in turn, can result in functional failure of the equipment.

When removing or installing O-rings, do **not** use pointed or sharp-edged tools which might scratch or mar component surfaces or damage the O-ring. An O-ring tool kit is available in the supply system for O-ring installation or removal. If these tools are not on hand, special tools can be made for this purpose. A few examples of tools used in the removal and installation of O-rings are illustrated in

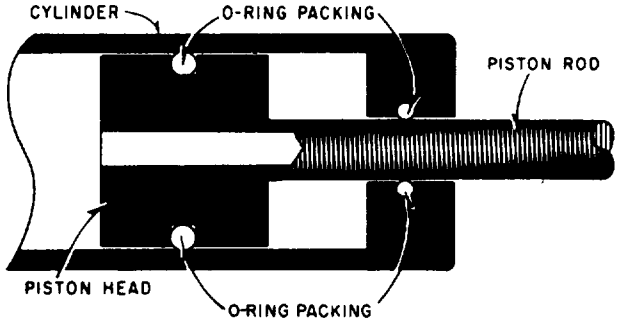

Figure 7-8.–Typical O-ring instalation.

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figure 7-9. These tools should be fabricated from soft metal such as brass or aluminum; however, tools made from phenolic rod, wood, or plastic may also be used.

Tool surfaces must be well rounded, polished, and free of burrs. Check the tools often, especially the surfaces that come in contact with O-ring grooves and critical polished surfaces.

Notice in figure 7-9, view A, how the hook-type removal tool is positioned under the O-ring and then lifted to allow the extractor tool, as well as the removal tool, to pull the O-ring from its cavity. View B shows the use of another type of extractor tool in the removal of internally installed O-rings.

In view C, notice the extractor tool positioned under both O-rings at the same time. This method of manipulating the tool positions both O-rings, which allows the hook-type removal tool to extract both O-rings with minimum effort. View D shows practically the same removal as view C, except for the use of a different type of extractor tool.

The removal of external O-rings is less difficult than the removal of internally installed O-rings. Views E and F show the use of a spoon-type extractor, which is positioned under the seal. After the O-ring is dislodged from its cavity, the spoon is held stationary while the piston is simultaneously rotated and withdrawn. View F is similar to view E, except that only one O-ring is installed, and a different type of extractor tool is used. The wedge-type extractor tool is inserted beneath the O-ring; the hook-type removal tool hooks the O-ring. A slight pull on the latter tool removes the O-ring from its cavity.

After removing all O-rings, cleaning of the affected parts that will receive new O-rings is

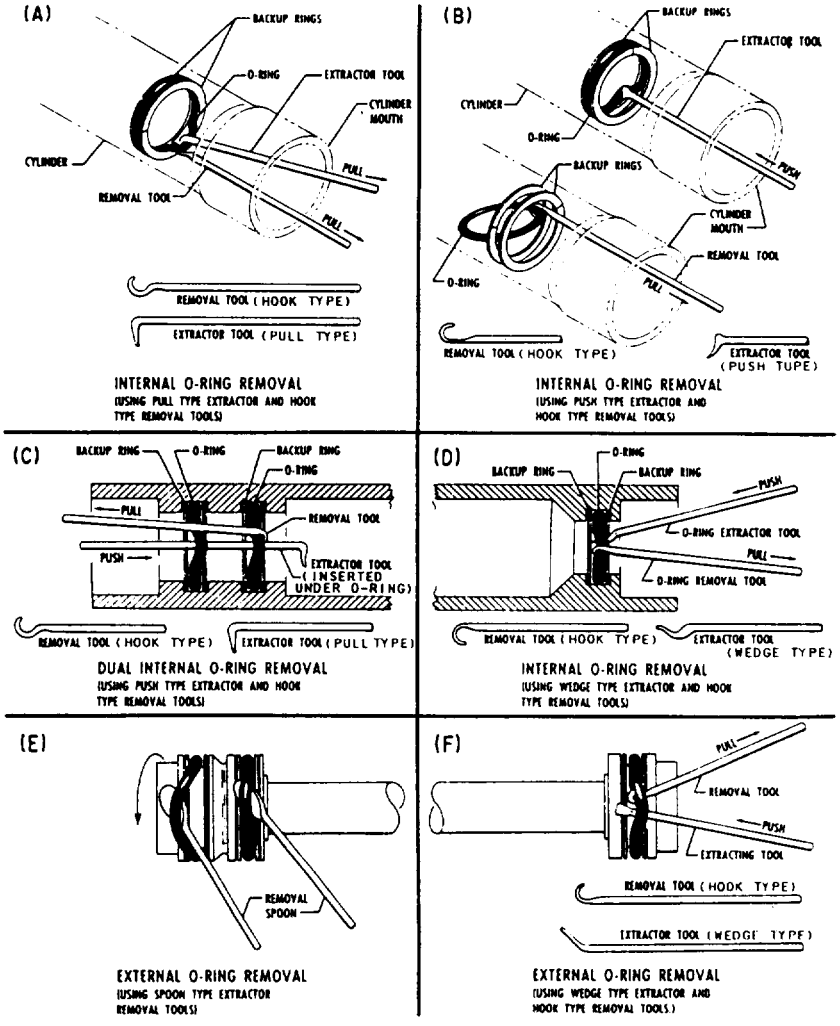

Figure 7-9.—O-ring tools and O-ring removal.

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mandatory. Ensure that the area used for such installations is clean and free from all contamination.

Remove each O-ring that is to be installed from its sealed package and inspect it for defects such as blemishes, abrasions, cuts, or punctures. Although an O-ring may appear perfect at first glance, slight surface flaws may exist. These are often capable of preventing satisfactory O-ring performance. O-rings should be rejected for flaws that will affect their performance.

By rolling the ring on an inspection cone or dowel, the inner diameter surface can be checked for small cracks, particles of foreign material, and other irregularities that will cause leakage or shorten its life. The slight stretching of the ring when it is rolled inside out will help to reveal some defects not otherwise visible. A further check of each O-ring should be made by stretching it between the fingers, but care must be taken not to exceed the elastic limits of the rubber. Following these inspection practices will prove to be a maintenance economy. It is far more desirable to take care identifying and inspecting O-rings than to repeatedly overhaul components with faulty seals.

After inspection and prior to installation, lubricate the O-ring, and all the surfaces that it must slide over with a light coat of the system fluid or a lubricant approved for use in the system. Consult the applicable technical instruction or *Naval Ships' Technical Manual* for the correct lubricant for pneumatic systems.

Assembly must be made with care so that the O-ring is properly placed in the groove and not damaged as the gland is closed. During some installations, such as on a piston, it will be necessary to stretch the O-ring. Stretch the O-ring as little and as uniformly as possible. Avoid rolling or twisting the O-ring when maneuvering it into place. Keep the position of the O-ring mold line constant. O-rings should not be left in a twisted condition after installation.

If the O-ring installation requires spanning or inserting through sharp-threaded areas, ridges, slots, and edges, use protective measures, such as the O-ring entering sleeve (fig. 7-10, view A). If

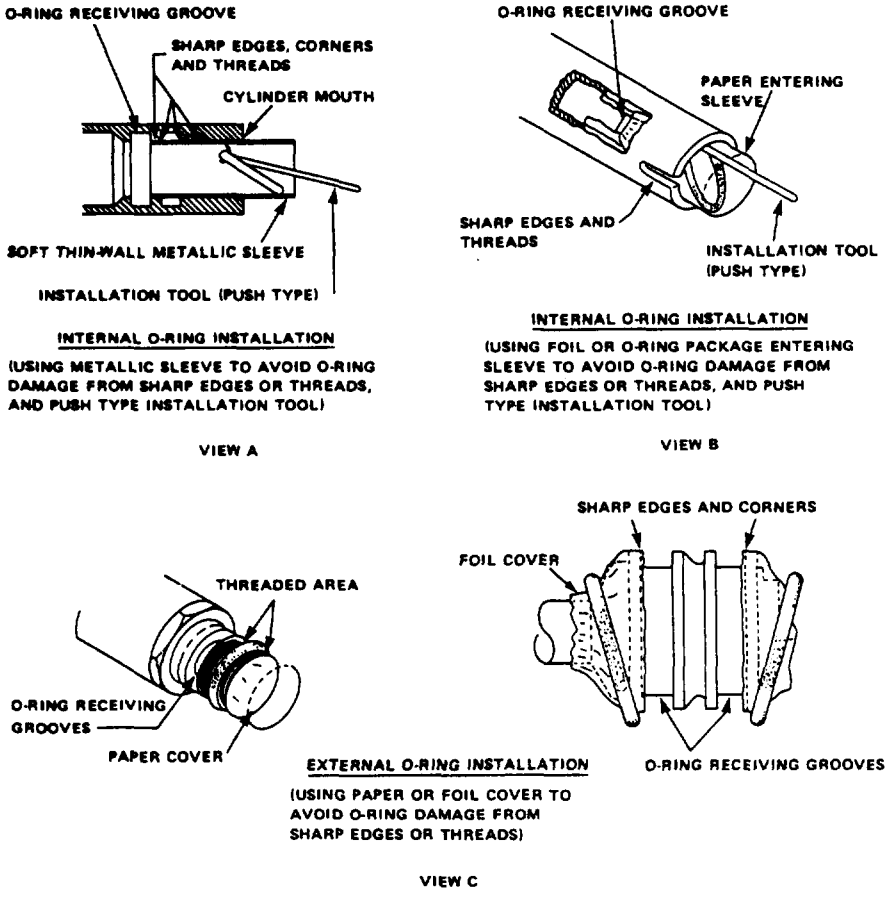

Figure 7-10.–O-ring installation.

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the recommended O-ring entering sleeve (a soft, thin wall, metallic sleeve) is not available, paper sleeves and covers may be fabricated by using the seal package (glossy side out) or lint-free bond paper (see views B and C of fig. 7-10).

After you place the O-ring in the cavity provided, gently roll the O-ring with your fingers to remove any twist that might have occurred during the installation. After installation, an O-ring should seat snugly but freely in its groove. If backup rings are installed in the groove, be certain the backup rings are installed on the correct side of the ring.

BACKUP RINGS

Backup rings, also referred to as retainer rings, antiextrusion devices, and nonextrusion rings, are washer-like devices that are installed on the low-pressure side of packing to prevent extrusion of the packing material. Backup rings in dynamic seals minimize erosion of the packing materials and subsequent failure of the seal. At lower pressures, backup rings will prolong the normal wear life of the packing. At higher pressures, backup rings permit greater clearances between the moving parts. Normally, backup rings are required for operating pressures over 1500 psi.

Backup rings can be made of polytetrafluoroethylene, hard rubber, leather, and other materials. The most common material currently used is tetrafluoroethylene (TFE). Backup rings are available as single-turn continuous (uncut or solid), single-turn (bias) cut, and spiral cut. See figure 7-11. Leather rings are always furnished in solid ring form (unsplit). Rings of TFE are available in all three types.

Packaging and Storing

Backup rings are not color-coded or otherwise marked and must be identified from the packaging labels. The dash number following the military standard number found on the package indicates the size, and usually relates directly to the dash number of the O-rings for which the backup ring is dimensionally suited. Backup rings made of TFE do not deteriorate with age and do not have shelf life limitations. TFE backup rings are provided by manufacturer either in individually sealed packages or on mandrels. If unpackaged rings are stored for a long time without the use of mandrels, a condition of overlap may develop. Overlap occurs when the ID of the backup ring becomes smaller and its ends overlap each other. To correct this overlap condition, stack TFE rings on a mandrel of the correct diameter, and clamp the rings with their coils flat and parallel. Place the rings in an oven at a maximum temperature of 177°C (350°F) for approximately 10 minutes. Do not overheat them because fumes from decomposing TFE are toxic. Remove and water-quench the rings. Store the rings at room temperature before you use them (preferably for 48 hours).

Installation

Care must be taken in handling and installing backup rings. Do not insert them with sharp tools. Backup rings must be inspected prior to using them for evidence of compression damage, scratches, cuts, nicks, or frayed conditions. If O-rings are to be replaced where backup rings are installed in the same groove, never replace the O-ring without replacing the backup rings, or vice versa. Many seals use two backup rings, one on either side of the O-ring (fig. 7-12). Two backup rings are used primarily in situations (such as a reciprocating piston seal) where alternating pressure direction can cause packing to be extruded on both sides of the gland.

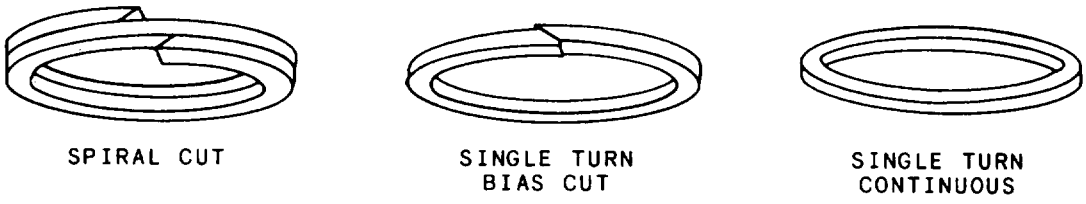

Figure 7-11.—Types of backup rings.

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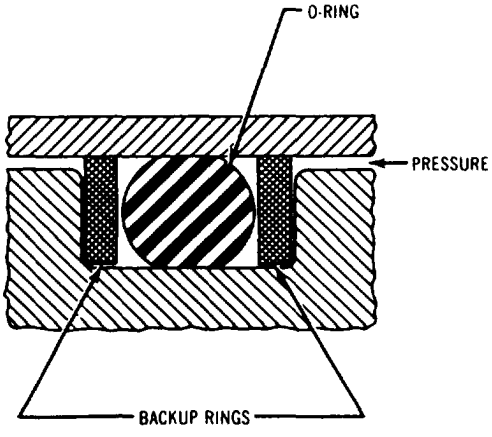

Figure 7-12.—Backup ring configuration.

If only one backup ring is used, place the backup ring on the low-pressure side of the packing (fig. 7-13, view A). When a backup ring is placed on the high-pressure side of the packing, the pressure against the relatively hard surface of the backup ring forces the softer packing against the low-pressure side of the gland, resulting in a rapid failure due to extrusion (fig. 7-13, view B).

When dual backup rings are installed, stagger the split scarfed ends as shown in figure 7-14. When installing a spiral cut backup ring (MS28782 or MS28783), be sure to wind the ring correctly to ease installation and ensure optimum performance.

When TFE spiral rings are being installed in internal grooves, the ring must have a right-hand

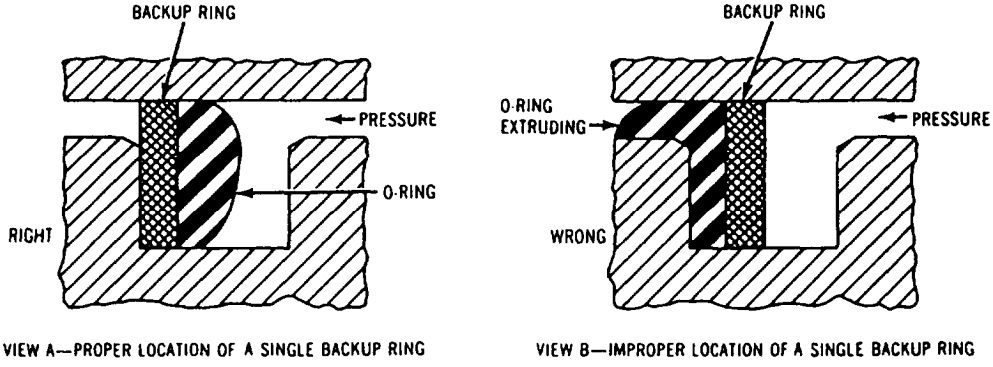

Figure 7-13.–Location of a single backup ring.

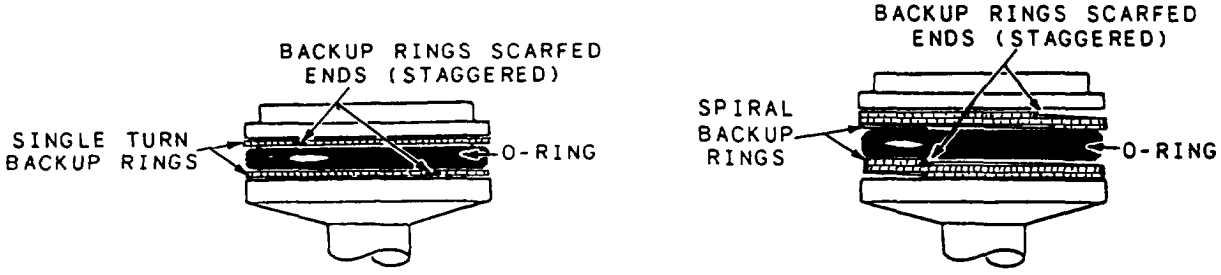

Figure 7-14.—Installation of cut dual backup rings.

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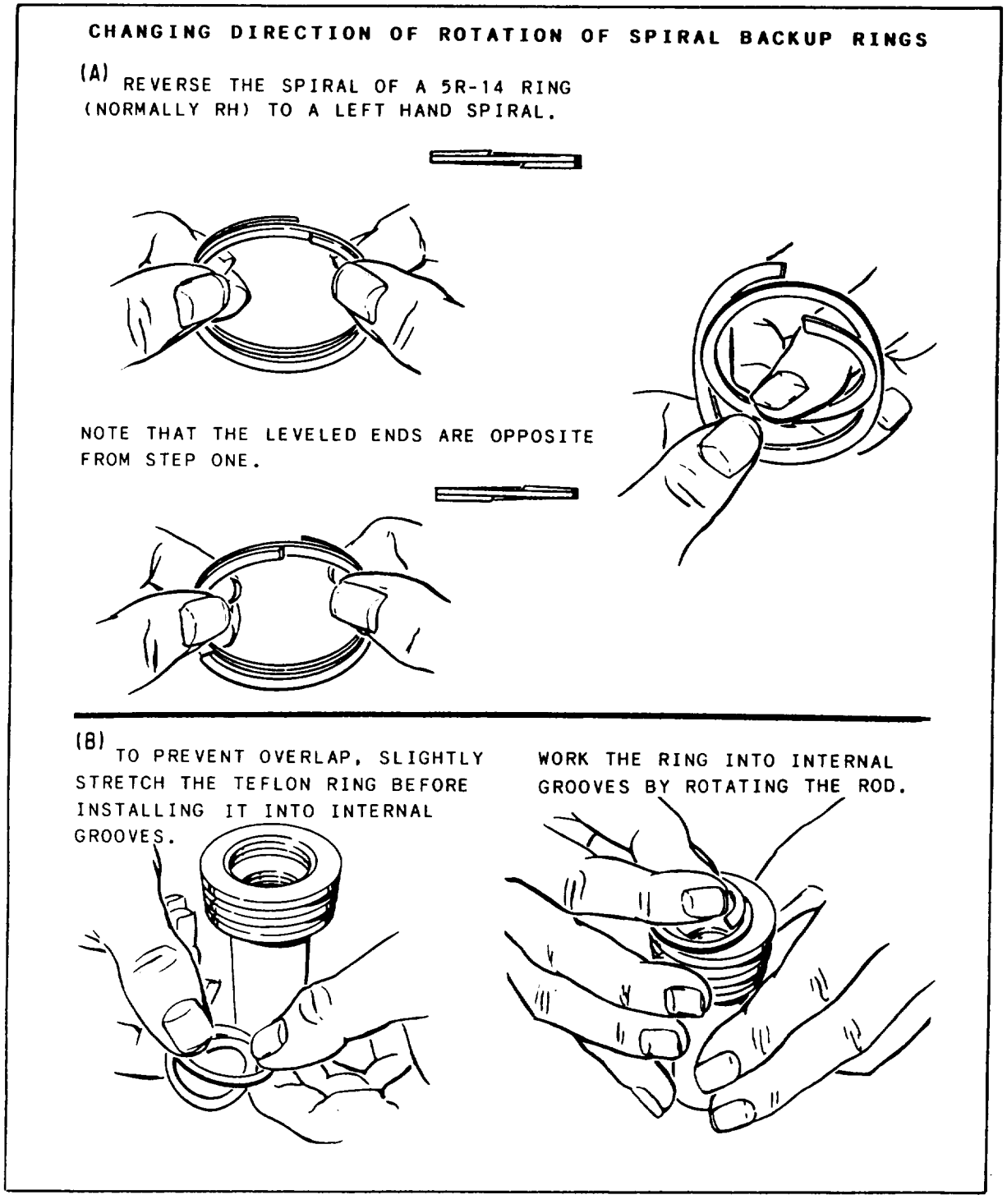

Figure 7-15.–Installation of TFE back up rings (internal).

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spiral. Figure 7-15, view A, shows how to change the direction of the spiral. The ring is then stretched slightly, as shown in view B prior to installation into the groove. While the TFE ring is being inserted into the groove, rotate the component in a clockwise direction. This will tend to expand the ring diameter and reduce the possibility of damaging the ring.

When TFE spiral rings are being installed in external grooves, the ring should have a left-hand spiral. As the ring is being inserted into the groove, rotate the component in a clockwise direction. This action will tend to contract the ring diameter and reduce the possibility of damaging the ring.

In applications where a leather backup ring is called for, place the smooth-grained side of the leather next to the ring. Do not cut leather backup rings. Use a leather backup ring as one continuous ring and lubricate the ring prior to installing it, particularly the smaller sizes. If stretching is necessary for proper installation, soak the backup ring in the system fluid or in an acceptable lubricant at room temperature for at least 30 minutes.

QUAD-RINGS

The Quad-Ring® seal is a special configuration ring packing, manufactured by the Minnesota Rubber. As opposed to an O-ring, a Quad-Ring® seal has a more square cross-sectional shape with rounded corners (fig. 7-16). The Quad-Ring® seal design offers more stability than the O-ring design and practically eliminates the spiraling or twisting that is sometimes encountered with the O-ring.

Quad-Rings® seals are completely interchangeable with O-rings in the sizes offered by the manufacturer. They may be installed with one or two backup rings, depending upon the specific seal groove application and width. The Quad-Ring® seal works well in, both hydraulic and pneumatic systems.

Many Quad-Ring® seal sizes have been assigned NSNs and are stocked in the Federal Supply System. Quad-Ring® seals in manufacturer's sizes designated as Q1 through Q88 are interchangeable with O-rings conforming to AN6227. Likewise, Quad-Ring® seals in commercial sizes Q101 through Q152 are interchangeable with O-rings conforming to AN6230 in the respective dash sizes from –1 through–52. Therefore, the Quad-Ring® seal stock part number uses the AN standard O-ring designations AN6227 and AN6230 and the commercial Q dash number designation. For example, NSNs are found under such reference part numbers as AN6227Q10 and AN6230Q103. If the letter Q does not follow AN6227 or AN6230, the part number is an O-ring not a Quad-Ring® seal.

If Quad-Ring® seals are not available for maintenance actions, appropriate sized O-rings can be installed and they work satisfactorily.

QUAD-O-DYN® SEALS

The Quad-O-Dyn®, also manufactured by Minnesota Rubber, is a special form of the Quad-Ring. The Quad-O-Dyn differs from the Quad-Ring in configuration (fig. 7-17), is harder, is subject to greater squeeze, and is made of a different material. The Quad-O-Dyn® seal also works well in O-rings glands.

The Quad-O-Dyn® is used in relatively few applications. However, for difficult dynamic sealing applications, the Quad-O-Dyn® can perform better than the Quad-Ring. Quad-O-Dyn® rings are installed in submarine hydraulic systems plant accumulators.

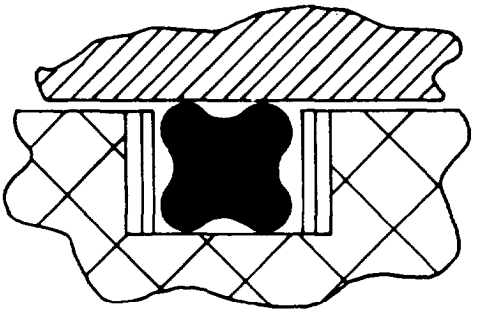

Figure 7-16.—Quad-Ring.

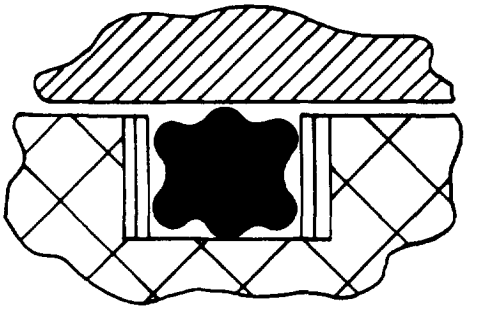

Figure 7-17.—Quad-O-Dyn® seal.

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U-CUPS AND U-PACKINGS

The distinction between U-cups and U-packings results from the difference in materials used in their fabrication. The U-cup is usually made of homogeneous synthetic rubber; U-packings are usually made of leather or fabric-reinforced rubber. Special aspects of each type will be discussed separately. However, all U-cups and U-packings have cross sections resembling the letter *U*. Both types are balanced packings, both seal on the ID and the OD, and both are applied individually, not in stacks like V-rings. Size differences between U-cups and U-packings are usually substantial enough to prevent interchangeability. There are a few sizes with smaller diameters and cross sections that may appear to be dimensionally equivalent but are not. Therefore, U-packings should not be substituted for U-cups (or vice versa) in any installation.

U-CUPS

The U-cup (fig. 7-18) has been a popular packing in the past because of installation ease and low friction. U-cups are used primarily for pressures below 1500 psi, but higher pressures are possible with the use of antiextrusion rings. For double-acting pistons, two U-cups are installed in separate grooves, back-to-back or heel-to-heel. Two U-cups are never used in the same groove. This heel-to-heel type of installation is common for single-acting (monodirectional) seals, such as U-cups and V-rings, and is necessary to prevent a pressure trap (hydraulic lock) between two packings. Installation of two U-cups with sealing lips facing each other can result in hydraulic lock and must be avoided.

Leather U-Packings

As a rule, leather U-packings are made with straight side walls (no flared sealing lips). See figure 7-19. The leather may be chemically treated or otherwise impregnated to improve its performance. Leather U-packings are available in standard sizes conforming to industrial specifications. For support, the cavity of the U-packing should contain a metal pedestal ring or should be filled with a suitable material. Leather U-packings with an integral pedestal support have been installed in some submarine steering and diving ram piston seals.

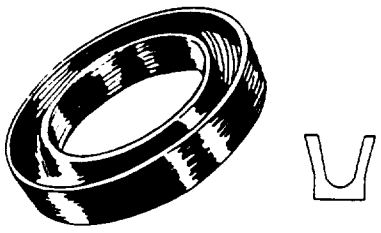

Figure 7-18.—Typical U-cup seal.

CUP PACKINGS

Cup packings resemble a cup or deep dish with a hole in the center for mounting (fig. 7-20). Cup seals are used exclusively to seal pistons in both low- and high-pressure hydraulic and pneumatic service. They are produced in leather, homogeneous synthetic rubber, and fabric-reinforced synthetic rubber. Although the cup packing lip flares outward, the rubbing contact is made at the lip only when the fluid pressure is low. As the fluid pressure increases, the cup heel expands outward until it contacts the cylinder wall, at which point high-pressure sealing is in effect. As the pressure loading shifts the sealing line to the cup heel, the lip is actually pulled into the cup and away from the cylinder wall. On the return stroke when the pressure is relaxed, the heel will shrink slightly, leaving only the lip in contact with the wall, avoiding unnecessary wear at the heel.

For reciprocating pistons, two cups installed back-to-back in separate glands are required.

FLANGE PACKINGS

Flange packings are used exclusively in low-pressure, outside-packed installations, such as rod

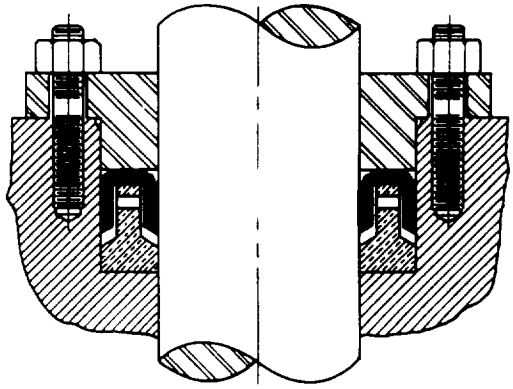

Figure 7-19.—U-packing.

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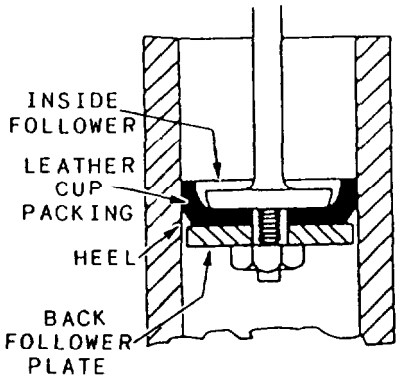

Figure 7-20.–Cup packing.

seals. The flange (sometimes called the hat) is made of leather, fabric-reinforced rubber, or homogeneous rubber. Lip sealing occurs only on the packing ID (fig. 7-21). Flange packings are generally used only for rod seals when other packings such as V-rings or U-seals cannot be used.

DIRT EXCLUSION SEALS (WIPERS AND SCRAPERS)

Dirt exclusion devices are essential if a satisfactory life is to be obtained from most rod seals. The smooth finished moving rod surface, if not enclosed or protected by some sort of covering, will accumulate a coating of dust or abrasive material that will be dragged or carried into the packing assembly area on the return rod stroke. Exclusion devices called wipers or scrapers are designed to remove this coating. While the terms *wiper* and *scraper* are often used interchangeably, it is useful to reserve scraper for metal lip-type devices that remove heavily encrusted deposits of dirt or other abrasive material that would merely deflect a softer lip and be carried into the cylinder. Sometimes a rod will have both a scraper and a wiper, the former to remove heavy deposits and the latter to exclude any dust particles that remain. Whenever metallic scrapers are used with felt wipers in the same groove, the felt wiper must not be compressed nor restricted in any way that affects its function as a lubricator. A wiper installed in a seal assembly in a pneumatic application may remove too much oil from the rod, requiring some method of replacing the oil. A common remedy is to provide a periodically oiled felt ring between the wiper and the seal. Felt wipers provide lubrication to extended operating rods, thus increasing component wear life. These wipers are only used to provide lubrication to parts.

Much longer life could be obtained from most seals if proper attention were given to wipers and scrapers. Often, wiper or scraper failure is not noticed when a seal packing fails. As a result, only the packing is replaced, and the same worn wiper or scraper is reinstalled to destroy another packing. Check the wiper or scraper condition upon its removal. If the wiper is worn, dirty, or embedded with metallic particles, replace it with a new one. It is usually good practice to replace the wiper every time you replace the seal and even more frequently if the wiper is readily accessible without component disassembly. If replacements are not available, wash dirty wipers that are still in good condition with suitable solvent and reinstall them. Remember that a wiper or scraper is deliberately installed as a sacrificial part to protect and preserve the sealing packing. Therefore, from a user's standpoint, wipers and scrapers should be inspected and replaced as necessary.

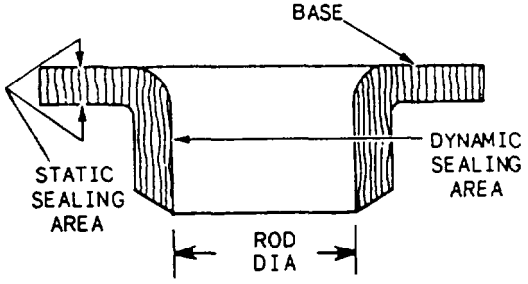

Figure 7-21.—Typical flange packing cross section.

STORAGE OF SEALS

Proper storage practices must be observed to prevent deformation and deterioration of seals. Most synthetic rubbers are not damaged by storage under ideal conditions. However, most synthetic rubbers will deteriorate when exposed to heat, light, oil, grease, fuels, solvents, thinners, moisture, strong drafts, or ozone (form of oxygen formed from an electrical discharge). Damage by exposure is magnified when rubber is under tension, compression, or stress. There are several

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conditions to be avoided, which include the following:

1. Deformation as a result of improper stacking of parts and storage containers.
2. Creasing caused by force applied to corners and edges, and by squeezing between boxes and storage containers.
3. Compression and flattening, as a result of storage under heavy parts.
4. Punctures caused by staples used to attach identification.
5. Deformation and contamination due to hanging the seals from nails or pegs. Seals should be kept in their original envelopes, which provide preservation, protection, identification, and cure date.
6. Contamination by piercing the sealed envelope to store O-rings on rods, nails, or wire hanging devices.
7. Contamination by fluids leaking from parts stored above and adjacent to the seal surfaces.
8. Contamination caused by adhesive tapes applied to seal surfaces. A torn seal package should be secured with a pressure-sensitive moistureproof tape, but the tape must not contact the seal surfaces.
9. Retention of overage parts as a result of improper storage arrangement or illegible identification. Seals should be arranged so the older seals are used first.

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CHAPTER 8

MEASUREMENT AND PRESSURE CONTROL DEVICES

For safe and efficient operation, fluid power systems are designed to operate at a specific pressure and/or temperature, or within a pressure and/or temperature range.

You have learned that the lubricating power of hydraulic fluids varies with temperature and that excessively high temperatures reduce the life of hydraulic fluids. Additionally, you have learned that the materials, dimensions, and method of fabrication of fluid power components limit the pressure and temperature at which a system operates. You have also learned of means of automatically controlling pressure in both hydraulic and pneumatic systems.

Most fluid power systems are provided with pressure gauges and thermometers for measuring and indicating the pressure and/or the temperature in the system. Additionally, various temperature and pressure switches are used to warn of an adverse pressure or temperature condition. Some switches will even shut the system off when an adverse condition occurs. These devices will be discussed in this chapter.

PRESSURE GAUGES

Many pressure-measuring instruments are called gauges. However, this section will be restricted to two mechanical instruments that contain elastic elements that respond to pressures found in fluid power systems—the Bourdon-tube and bellows gauges.

BOURDON TUBE GAUGES

The majority of pressure gauges in use have a Bourdon-tube as a measuring element. (The gauge is named for its inventor, Eugene Bourdon, a French engineer.) The Bourdon tube is a device that senses pressure and converts the pressure to displacement. Since the Bourdon-tube displacement is a function of the pressure applied, it may be mechanically amplified and indicated by a pointer. Thus, the pointer position indirectly indicates pressure.

The Bourdon-tube gauge is available in various tube shapes: curved or C-shaped, helical, and spiral. The size, shape, and material of the tube depend on the pressure range and the type of gauge desired. Low-pressure Bourdon tubes (pressures up to 2000 psi) are often made of phosphor bronze. High-pressure Bourdon tubes (pressures above 2000 psi) are made of stainless steel or other high-strength materials. High-pressure Bourdon tubes tend to have more circular cross sections than their lower-range counterparts, which tend to have oval cross sections. The Bourdon tube most commonly used is the C-shaped metal tube that is sealed at one end and open at the other (fig. 8-1).

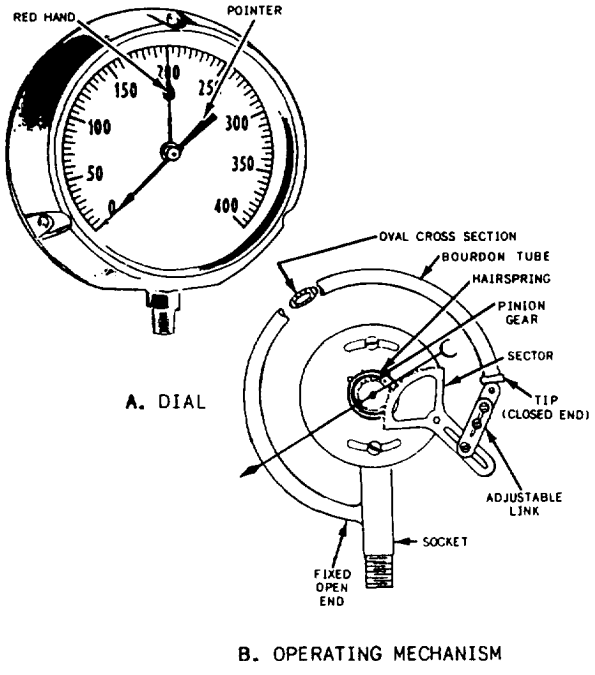

Figure 8-1.—Simplex Bourdon-tube pressure gauge.

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C-shaped Bourdon Tube

The C-shaped Bourdon tube has a hollow, elliptical cross section. It is closed at one end and is connected to the fluid pressure at the other end. When pressure is applied, its cross section becomes more circular, causing the tube to straighten out, like a garden hose when the water is first turned on, until the force of the fluid pressure is balanced by the elastic resistance of the tube material. Since the open end of the tube is anchored in a fixed position, changes in pressure move the closed end. A pointer is attached to the closed end of the tube through a linkage arm and a gear and pinion assembly, which rotates the pointer around a graduated scale.

Bourdon-tube pressure gauges are often classified as simplex or duplex, depending upon whether they measure one pressure or two pressures. A simplex gauge has only one Bourdon tube and measures only one pressure. The pressure gauge shown in figure 8-1 is a simplex gauge. A red hand is available on some gauges. This hand is manually positioned at the maximum operating pressure of the system or portion of the system in which the gauge is installed.

When two Bourdon tubes are mounted in a single case, with each mechanism acting independently but with the two pointers mounted on a common dial, the assembly is called a duplex gauge. Figure 8-2 shows a duplex gauge with views of the dial and the operating mechanism. Note that each Bourdon tube has its own pressure connection and its own pointer. Duplex gauges are used to give a simultaneous indication of the pressure from two different locations. For example, it may be used to measure the inlet and outlet pressures of a strainer to obtain the differential pressure across it.

Differential pressure may also be measured with Bourdon-tube gauges. One kind of Bourdon-tube differential pressure gauge is shown in figure 8-3. This gauge has two Bourdon tubes but only one pointer. The Bourdon tubes are connected in such a way that they indicate the pressure difference, rather than either of two actual pressures.

As mentioned earlier, Bourdon-tube pressure gauges are used in many hydraulic systems. In this application they are usually referred to as hydraulic gauges. Bourdon-tube hydraulic gauges are not particularly different from other types of Bourdon-tube gauges in how they operate; however, they do sometimes have special design features because of the extremely high system pressures to which they may be exposed. For

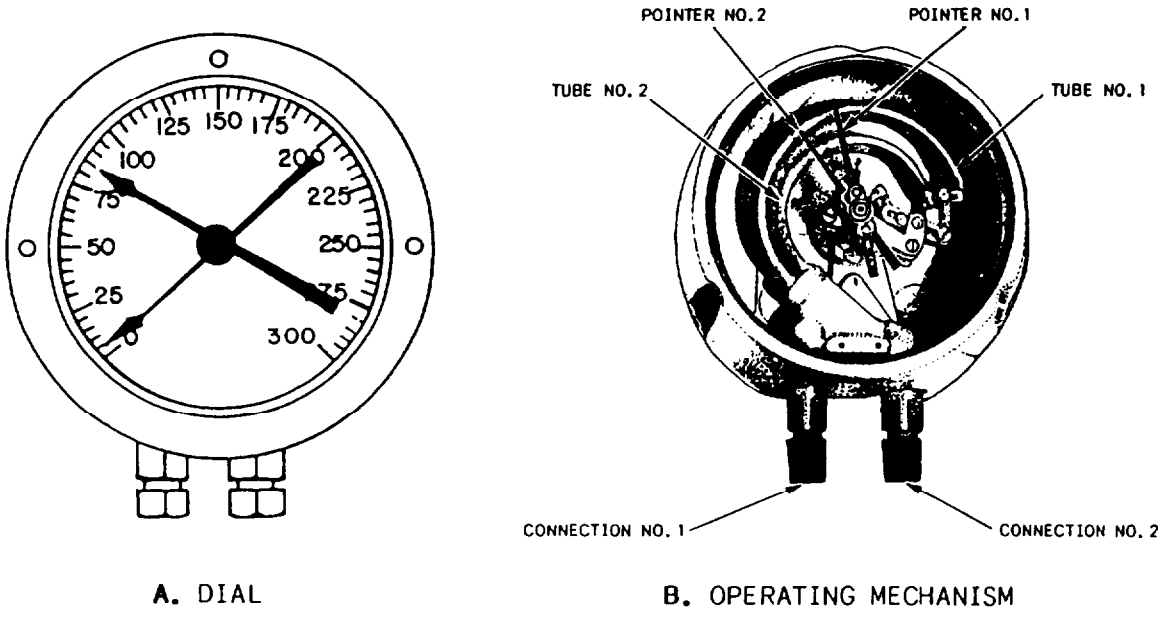

Figure 8-2.—Duplex Bourdon-tube pressure gauge.

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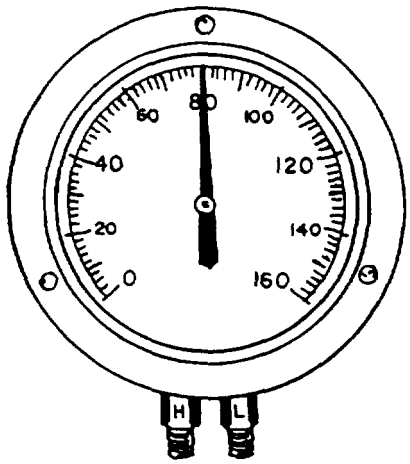
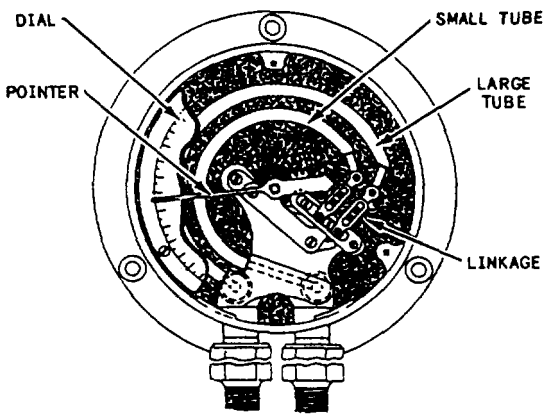


A. DIAL



B. OPERATING MECHANISM

Figure 8-3.—Bourdon-tube differential pressure gauge.

example, some hydraulic gauges have a special type of spring-loaded linkage that is capable of taking overpressure and underpressure without damage to the movement and that keeps the pointer from slamming back to zero when the pressure is suddenly changed. A hydraulic gauge that does not have such a device must be protected by a suitable check valve. Some hydraulic gauges may also have special dials that indicate both the pressure (in psi) and the corresponding total force being applied, for example tons of force produced by a hydraulic press.

Spiral and Helical Bourdon Tubes

Spiral and helical Bourdon tubes (figs. 8-4 and 8-5) are made from tubing with a flattened cross

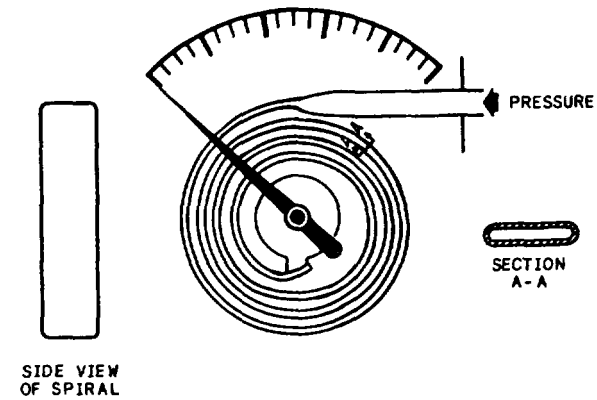


Figure 8-4.—Spiral Bourdon tube.

section. Both were designed to provide more travel of the tube tip, primarily for moving the recording pen of pressure recorders.

BELLOWS ELASTIC ELEMENTS

A bellows elastic element is a convoluted unit that expands and contracts axially with changes in pressure. The pressure to be measured can be applied to either the outside or the inside of the bellows; in practice, most bellows measuring

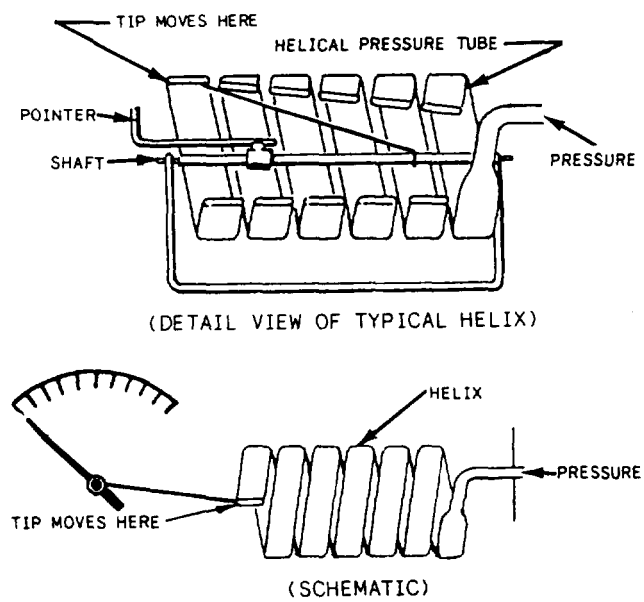


Figure 8-5.—Helical Bourdon tube.

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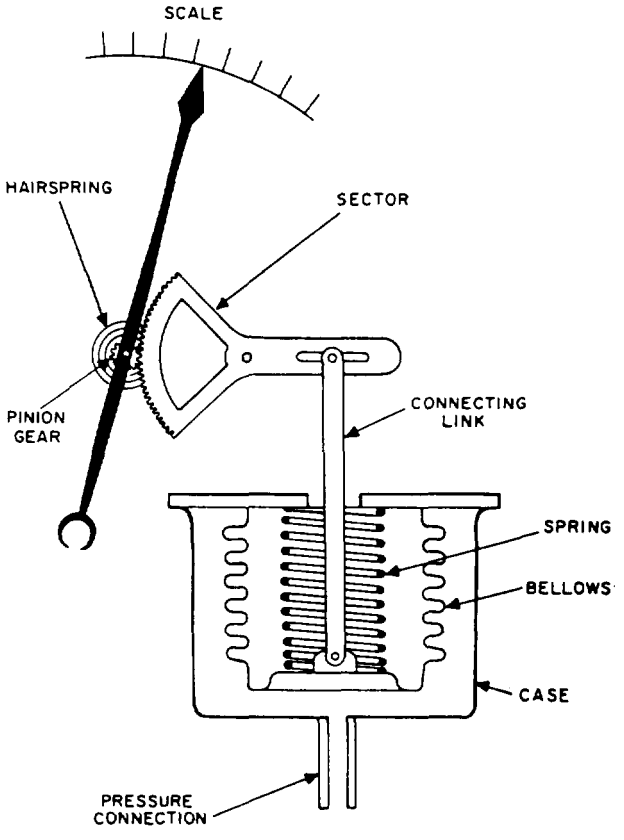

Figure 8-6.—Simple bellows gauge.

devices have the pressure applied to the outside of the bellows (fig. 8-6).

Simple Bellows Elements

Bellows elastic elements are made of brass, phosphor bronze, stainless steel, beryllium-copper, or other metal suitable for the intended service of the gauge. Motion of the element (bellows) is transmitted by suitable linkage and gears to a dial pointer. Most bellows gauges are spring-loaded—that is, a spring opposes the bellows and thus prevents full expansion of the bellows. Limiting the expansion of the bellows in this way protects the bellows and prolongs its life. Because of the elasticity in both the bellows and the spring in a spring-loaded bellows element, the relationship between the applied pressure and bellows movement is linear.

Dual Bellows Indicators

Another type of bellows element is the dual-bellows element. Figure 8-7 is a schematic diagram of this indicator. Dual-bellows element pressure indicators are used throughout the Navy as flow-measuring, level-indicating, or pressure-indicating devices.

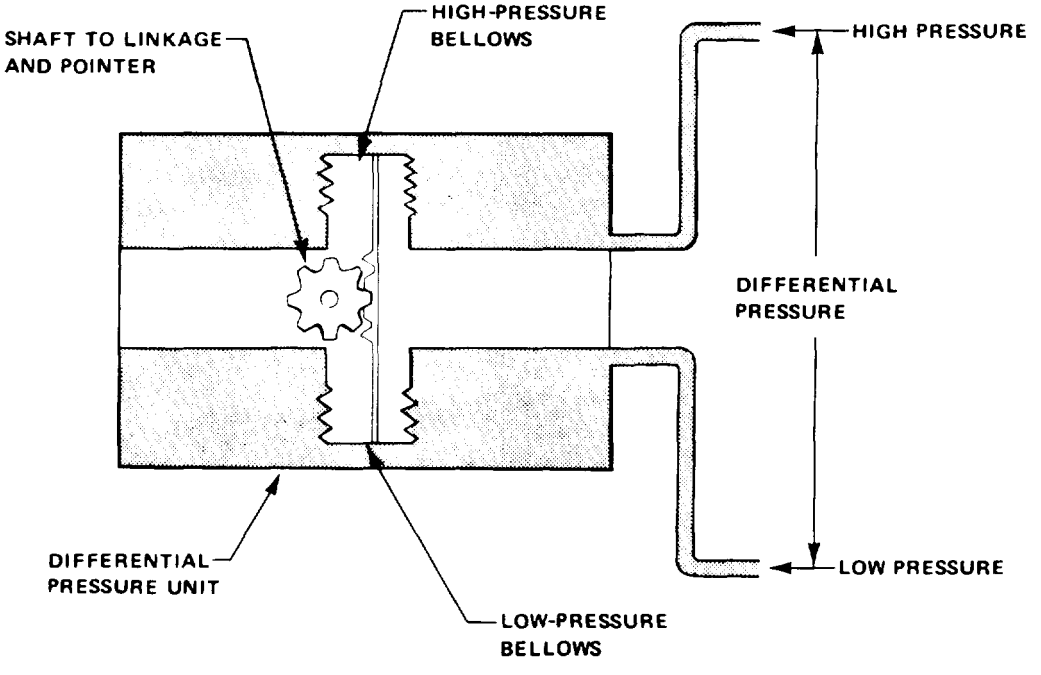

Figure 8-7.–Differential pressure sensor dual bellows.

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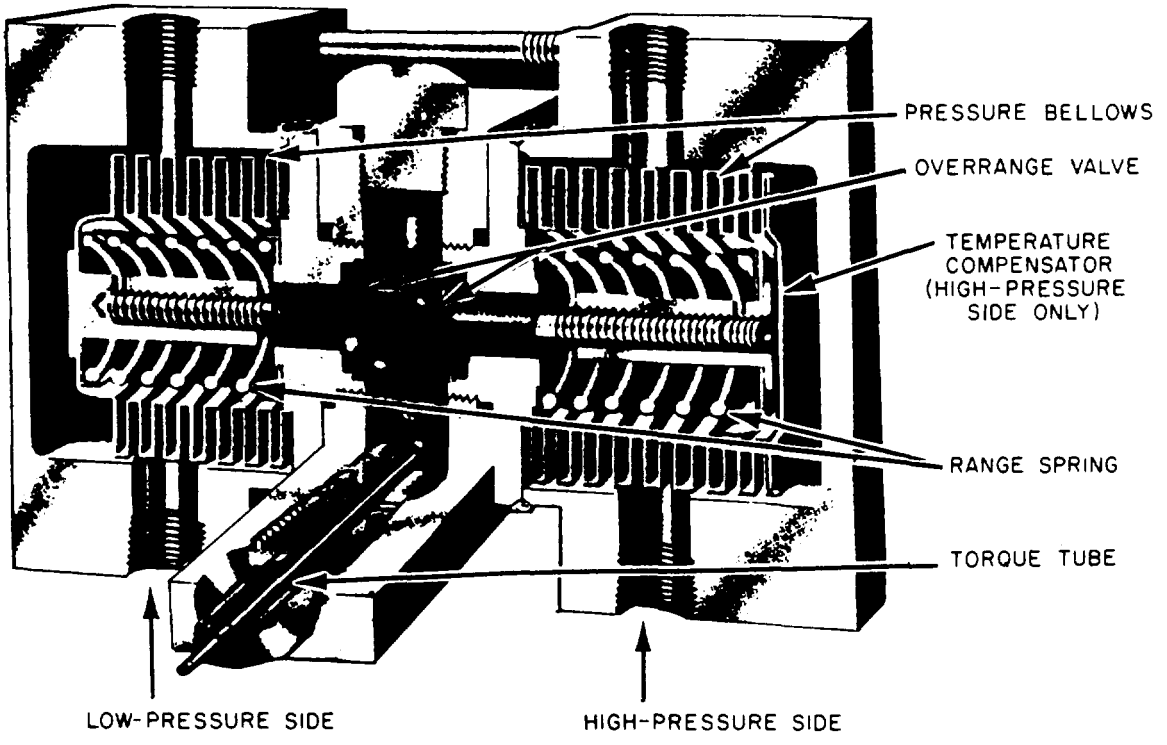

Figure 8-8.–Dual bellows assembly.

When in operation, the bellows will move in proportion to the difference in pressure applied across the bellows unit assembly. The linear motion of the bellows is picked up by a drive arm and transmitted as a rotary motion through a torque tube assembly (fig. 8-8). The indicating mechanism multiplies rotation of the torque tube through a gear and pinion to the indicating pointer.

Bellows elements are used in various applications where the pressure-sensitive device must be powerful enough to operate not only the indicating pointer but also some type of recording device.

PRESSURE SWITCHES

Often when a measured pressure reaches a certain maximum or minimum value, it is desirable to have an alarm sound a warning, a light to give a signal, or an auxiliary control system to energize or de-energize. A pressure switch is the device commonly used for this purpose.

One of the simplest pressure switches is the single-pole, single-throw, quick-acting type shown in figure 8-9. This switch is contained in a metal

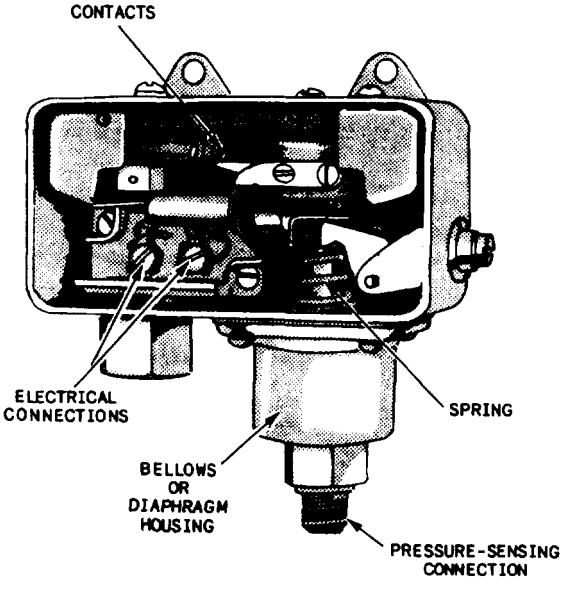

Figure 8-9.—Typical pressure switch.

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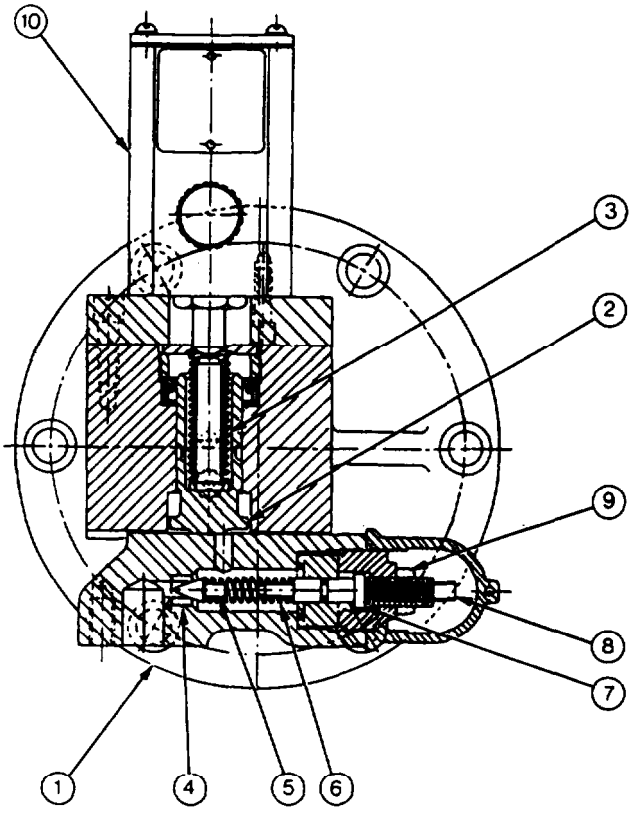
case that has a removable cover, an electrical connection, and a pressure-sensing connection. The switch contains a seamless metallic bellows located in its housing. Changes in the measured pressure causes the bellows to work against an adjustable spring. This spring determines the pressure required to actuate the switch. Through suitable linkage, the spring causes the contacts to open or close the electrical circuit automatically when the operating pressure falls below or rises above a specified value. A permanent magnet in the switch mechanism provides a positive snap on both the opening and closing of the contacts. The switch is constantly energized. However, it is the closing of the contacts that energizes the entire electrical circuit.

Another pressure switch is an electric-hydraulic assembly that is used for shutting off the pump's motor whenever the system pressure exceeds a pre-determined maximum value (fig. 8-10). The switch is mounted on the pump housing so that the former's low pressure ports drain directly into the pump housing.

This pressure switch principally consists of a flange-mounted hydraulic valve to which is fixed a normally closed electrical limit switch.

The valve consists of two hydraulically interconnected components, the pilot valve sub-assembly, which bolts on the bottom of the body (l), functions to sense system pressure continuously and initiates pressure switch action whenever this pressure exceeds the adjusted setting of the pilot adjustment. System pressure is directed into the bottom port and is applied against the exposed tip of the pilot piston (5). This piston is held on its seat by compression from the piston spring (6) which is dependent on the position of the adjusting screw (8). Whenever the pressure causes a force sufficiently large enough to raise the pilot piston from its seat, fluid flows through an interconnecting passage to the actuating piston (2) chamber. The accompanying fluid force raises the actuating piston against the force of spring 3 and causes depression of the extended switch plunger. This, in turn, disconnects the contained electrical switch, which may be connected into the pump motor's electric supply system.

Pressure switches come in many sizes and configurations depending on how they will be used.



1. Body
2. Actuating Piston
3. Spring
4. Seat
5. Pilot Piston
6. Piston Spring
7. Plunger-Adjustment Screw
8. Adjusting Screw
9. Adjusting Screw Lock Nut
10. Limit Switch

Figure 8-10.—Electric-hydraulic pressure switch.

TEMPERATURE-MEASURING INSTRUMENTS

Temperature is the degree of hotness or coldness of a substance measured on a definite scale. Temperature is measured when a measuring instrument, such as a thermometer, is brought into contact with the medium being measured.

All temperature-measuring instruments use some change in a material to indicate temperature. Some of the effects that are used to indicate temperature are changes in physical properties and altered physical dimensions. One of the more important physical properties used in temperature-measuring instruments is the change in the length of a material in the form of expansion and contraction.

Consider the uniform homogeneous bar illustrated in figure 8-11. If the bar has a given

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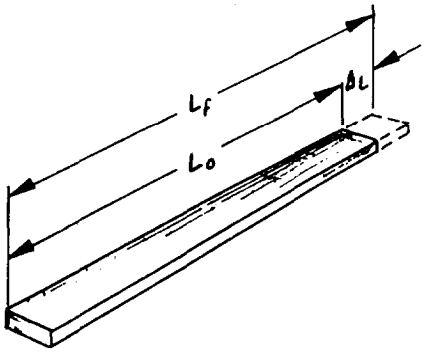

Figure 8-11.—Expansion of a bar.

length (L_o) at some temperature and is heated, it will expand (L_f). The amount of expansion (ΔL) is a function of the original length and the temperature increase. The amount a material changes in length with temperature is called the linear coefficient of expansion.

The linear coefficient of expansion for a material is a physical property of that material and describes its behavior with respect to temperature.

BIMETALLIC EXPANSION THERMOMETER

If two materials with different linear coefficients are bonded together, as the temperature changes their rate of expansion will be different. This will cause the entire assembly to bend in an arc as shown in figure 8-12.

When the temperature is raised, an arc is formed around the material with the smaller expansion coefficient. Since this assembly is formed by joining two dissimilar materials, it is known as a bimetallic element.

A modification of this bimetallic strip serves as the basis for one of the simplest and most commonly encountered temperature-measuring instruments, the bimetallic thermometer.

Figure 8-13 shows a bimetallic thermometer. In it, a bimetallic strip is wound in the form of a long helix. One end of the helix is held rigid. As the temperature varies, the helix tries to wind or unwind. This causes the free end to rotate. The

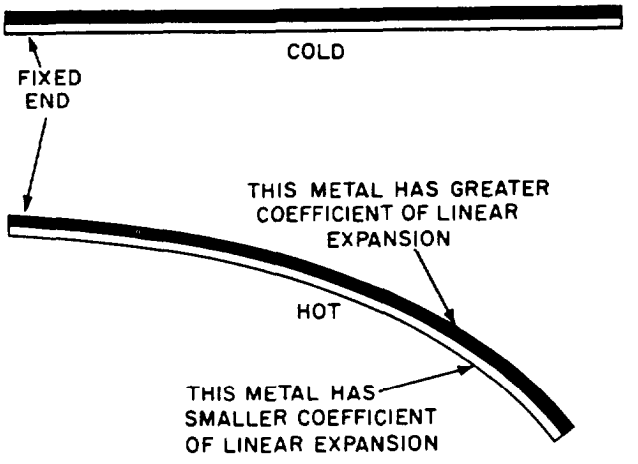

Figure 8-12.—Effect of unequal expansion of a bimetallic strip.

free end is connected to a pointer. The pointer actually indicates angular rotation of the helix; however, since the rotation is linear and a function of temperature, the scale is marked in units of temperature.

DISTANT-READING THERMOMETERS

Distant-reading dial thermometers are used when the indicating portion of the instrument must be placed at a distance from where the temperature is being measured. The distant-reading thermometer has a long capillary, some

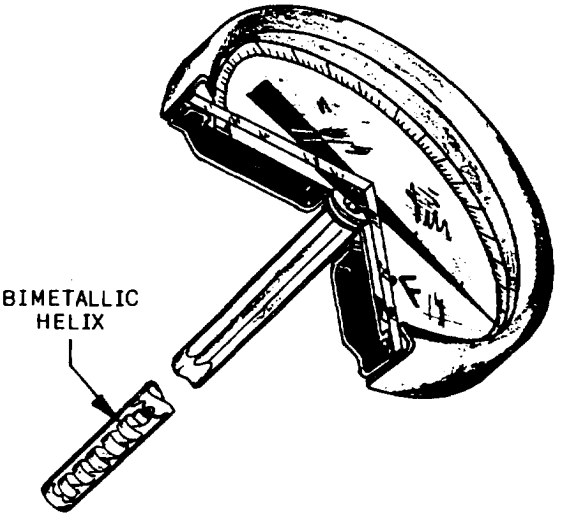

Figure 8-13.—Bimetallic thermometer.

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as long as 125 feet, which separates the sensing bulb from the Bourdon tube and dial (fig. 8-14).

There are three basic types of distant-reading thermometers: the liquid filled, the gas filled, and the combination liquid-vapor filled. The thermometers are filled with fluid (liquid or gas) at some temperature and sealed. Almost the entire volume of the fluid is in the sensing bulb.

As the temperature of the bulb changes, the volume of the fluid tries to change. Since the volume of the thermometer (sensing bulb, capillary, and Bourdon tube) is constant, a pressure change occurs within the thermometer. This pressure change causes the Bourdon tube to straighten out (with an increase in pressure), working a system of levers and gears, which causes the thermometer pointer to move over the dial and register temperature.

TEMPERATURE SWITCHES

Temperature switches operate from temperature changes occurring in an enclosure, or in the air surrounding the temperature-sensing element. The operation of the temperature switch is similar to the operation of the pressure switch shown in figure 8-9; both switches are operated by changes in pressure. The temperature element is arranged so a change in temperature causes a change in the internal pressure of a sealed-gas or air-filled bulb or helix, which is connected to the actuating device by a small tube or pipe. Figure 8-15 shows a temperature switch and two types of sensing elements.

A temperature change causes a change in the volume of the sealed-in gas, which causes movement of a bellows. The movement is transmitted by a plunger to the switch arm. The moving contact is on the arm. A fixed contact may be arranged so the switch will open or close on a temperature rise. This allows the switch contacts to be arranged to close when the temperature drops to a predetermined value and to open when the temperature rises to the desired value. The reverse action can be obtained by a change in the contact positions.

GAUGE SNUBBERS

The irregularity of impulses applied to the fluid power system by some pumps or air compressors causes the gauge pointer to oscillate violently. This makes reading of the gauge not only difficult but often impossible. Pressure oscillations and other sudden pressure changes existing in fluid power systems will also affect the delicate internal mechanism of gauges and cause either damage to or complete destruction of the

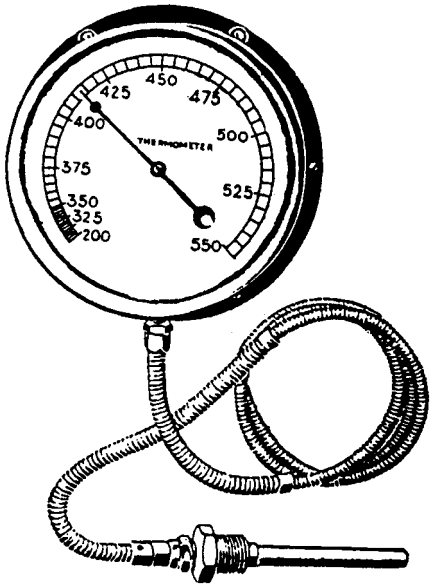

Figure 8-14.—Distant-reading, Bourdon-tube thermometers.

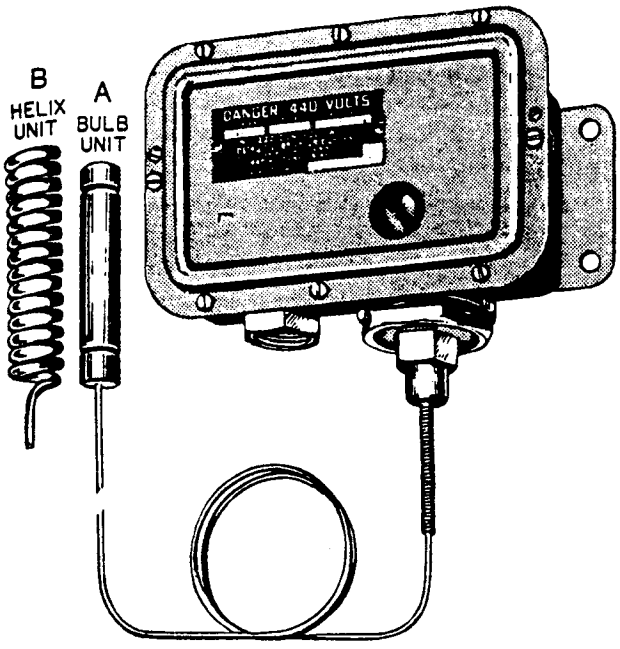

Figure 8-15.—Temperature switch with two types of sensing elements. A. Bulb unit. B. Helix unit.

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gauge. A pressure gauge snubber is therefore installed in the line that leads to the pressure gauge.

The purpose of the snubber is to dampen the oscillations and thus provide a steady reading and protection for the gauge. The basic components of a snubber are the housing, fitting assembly with a fixed orifice diameter, and a pin and plunger assembly (fig. 8-16). The snubbing action is obtained by metering fluid through the snubber. The fitting assembly orifice restricts the amount of fluid that flows to the gauge, thereby snubbing the force of a pressure surge. The pin is pushed and pulled through the orifice of the fitting assembly by the plunger, keeping it clean and at a uniform size.

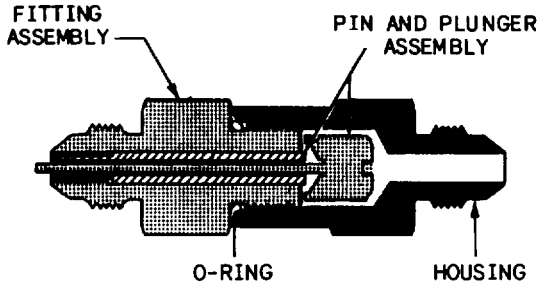

Figure 8-16.—Pressure gauge snubber.

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CHAPTER 9

RESERVOIRS, STRAINERS, FILTERS, AND ACCUMULATORS

Fluid power systems must have a sufficient and continuous supply of uncontaminated fluid to operate efficiently. As stated in chapter 3 and emphasized throughout this manual, the fluid must be kept free of all foreign matter.

This chapter covers hydraulic reservoirs, various types of strainers and filters, and accumulators installed in fluid power systems.

RESERVOIRS

A hydraulic system must have a reserve of fluid in addition to that contained in the pumps, actuators, pipes, and other components of the system. This reserve fluid must be readily available to make up losses of fluid from the system, to make up for compression of the fluid under pressure, and to compensate for the loss of volume as the fluid cools. This extra fluid is contained in a tank usually called a reservoir. A reservoir may sometimes be referred to as a sump tank, service tank, operating tank, supply tank, or base tank.

In addition to providing storage for the reserve fluid needed for the system, the reservoir acts as a radiator for dissipating heat from the fluid and as a settling tank where heavy particles of contamination may settle out of the fluid and remain harmlessly on the bottom until removed by cleaning or flushing of the reservoir. Also, the reservoir allows entrained air to separate from the fluid.

Most reservoirs have a capped opening for filling, an air vent, an oil level indicator or dip stick, a return line connection, a pump inlet or suction line connection, a drain line connection, and a drain plug (fig. 9-1). The inside of the reservoir generally will have baffles to prevent excessive sloshing of the fluid and to put a partition between the fluid return line and the pump suction or inlet line. The partition forces the returning fluid to travel farther around the tank before being drawn back into the active system through the pump inlet line. This aids in settling the contamination and separating the air from the fluid.

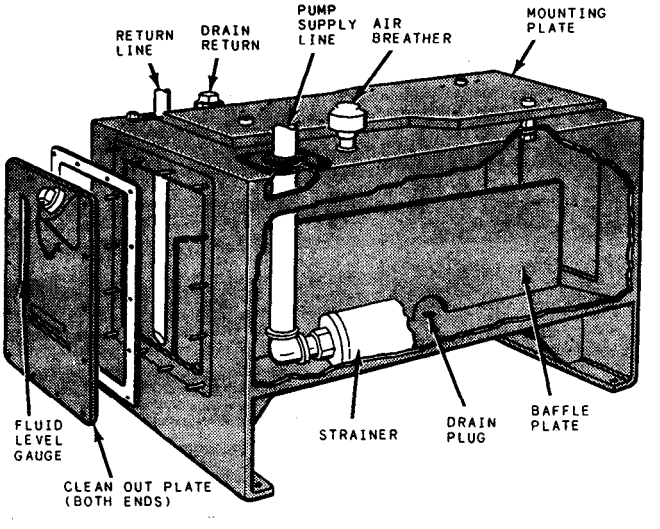

Figure 9-1.—Nonpressurized reservoir (ground or ship installation).

Large reservoirs are desirable for cooling. A large reservoir also reduces recirculation which helps settle contamination and separate air. As a "thumb rule," the ideal reservoir should be two to three times the pump output per minute. However, due to space limitations in mobile and aerospace systems, the benefits of a large reservoir may have to be sacrificed. But, they must be large enough to accommodate thermal expansion of the fluid and changes in fluid level due to system operation. Reservoirs are of two general types—nonpressurized and pressurized.

NONPRESSURIZED RESERVOIRS

Hydraulic systems designed to operate equipment at or near sea level are normally equipped with nonpressurized reservoirs. This includes the hydraulic systems of ground and ship

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installations. A typical reservoir for use with ground and ship installations is shown in figure 9-1. This type of reservoir is made of hot rolled steel plates and has welded seams. The ends extend below the bottom of the reservoir and serve as supports. The bottom of the reservoir is convex, and a drain plug is incorporated at the lowest point.

Nonpressurized reservoirs are also used in several transport-, patrol-, and utility-type aircraft. These aircraft are not designed for violent maneuvers and, in some cases, do not fly at high altitude. Those aircraft that have nonpressurized reservoirs installed and that fly at high altitudes have the reservoirs installed within a pressurized area. (High altitude in this situation means an altitude where atmospheric pressure is inadequate to maintain sufficient flow of fluid to the hydraulic pumps.)

Most nonpressurized aircraft reservoirs are constructed in a cylindrical shape (fig. 9-2). The outer housing is manufactured from a strong corrosion-resistant metal. Filter elements are normally installed internally within the reservoir to clean returning system hydraulic fluid. Some of the older aircraft have a filter bypass valve installed to allow fluid to bypass the filter if the filter becomes clogged. Reservoirs that are filled by pouring fluid directly into them have a filler (finger) strainer assembly installed in the filler well to strain out impurities as the fluid enters the reservoir.

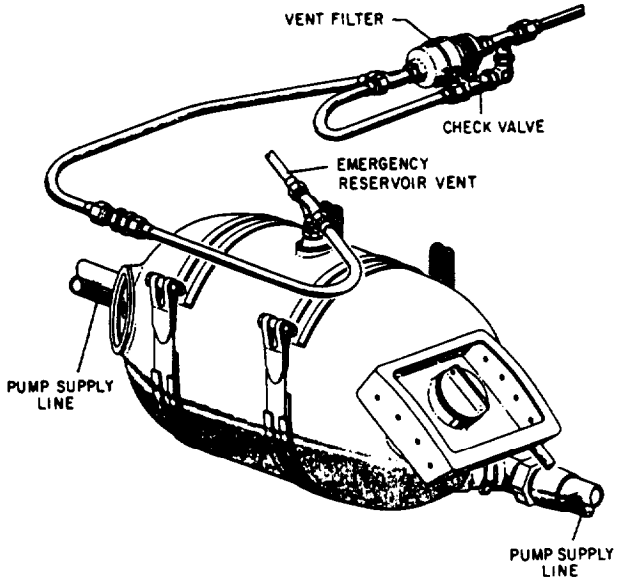

Figure 9-2.—Nonpressurized aircraft reservoir.

The quantity of fluid in the reservoir is indicated by either a glass tube, a directing gauge, or a float-type rod, which is visible through a transparent dome installed on the reservoir.

PRESSURIZED RESERVOIRS

A pressurized reservoir is required in hydraulic systems where atmospheric pressure is insufficient to maintain a net positive suction head (NPSH) to the pump. There are two common types of pressurized reservoirs—fluid-pressurized and air-pressurized.

Fluid-Pressurized Reservoir

Some aircraft hydraulic systems use fluid pressure for pressurizing the reservoir. The reservoir shown in figure 9-3 is of this type. This reservoir is divided into two chambers by a floating piston. The piston is forced downward in the reservoir by a compression spring within the pressurizing cylinder and by system pressure entering the pressurizing port of the cylinder.

The pressurizing port is connected directly to the pressure line. When the system is pressurized, pressure enters the pressure port, thus pressurizing the reservoir. This pressurizes the pump suction line and the reservoir return line to the same pressure.

The reservoir shown in figure 9-3 has five ports—pump suction, return, pressurizing, overboard drain, and bleed. Fluid is supplied to the pump through the pump suction port. Fluid returns to the reservoir from the system through the return port. Pressure from the pump enters the pressurizing cylinder in the top of the reservoir through the pressurizing port. The overboard drain port is used to drain the reservoir while performing maintenance, and the bleed port is used as an aid when servicing the reservoir.

Air-Pressurized Reservoirs

Air-pressurized reservoirs, such as the one shown in figure 9-4, are currently used in many high-performance naval aircraft. The reservoir is cylindrical in shape and has a piston installed internally to separate the air and fluid chambers. Air pressure is usually provided by engine bleed air. The piston rod end protrudes through the reservoir end cap and indicates the fluid quantity. The quantity indication may be seen by inspecting the distance the piston rod protrudes from the reservoir end cap. The reservoir is provided with

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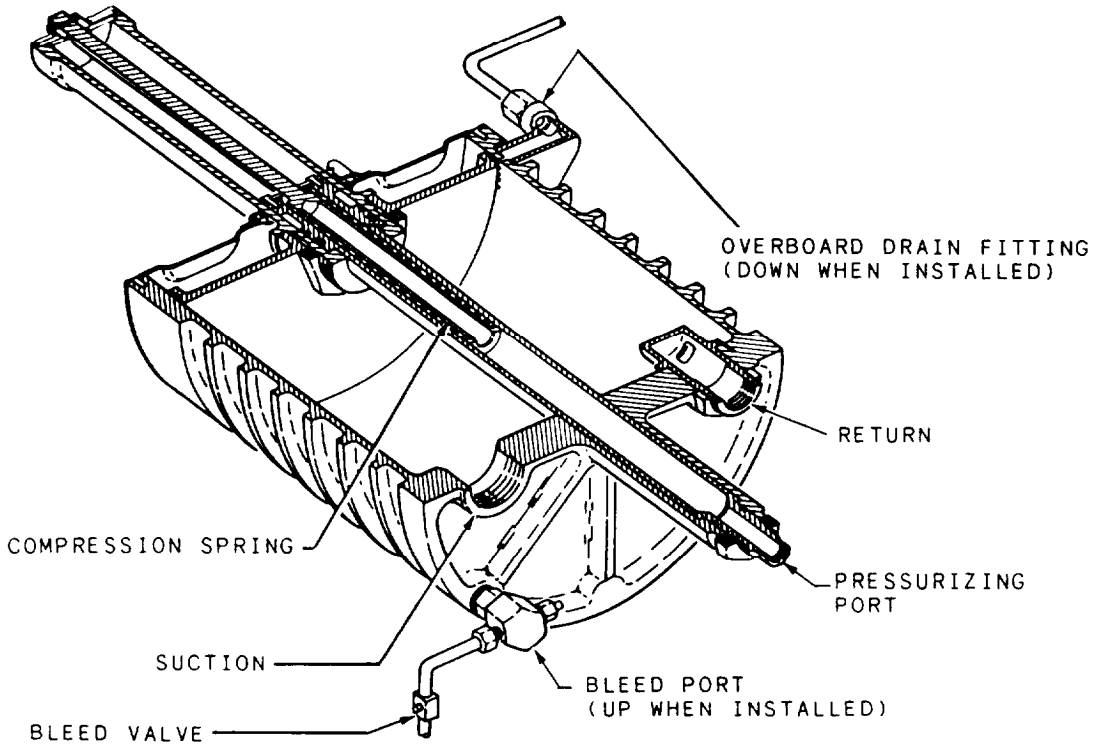

Figure 9-3.—Typical fluid-pressurized reservoir.

threaded openings for connecting fittings and components. Figure 9-4 shows several components installed in lines leading to and from the reservoir; however, this may not be the case in actual installation. The air relief valve, bleeder valve, and soon, may reinstalled directly on the reservoir.

Because the reservoir is pressurized, it can normally be installed at any altitude and still maintain a positive flow of fluid to the pump.

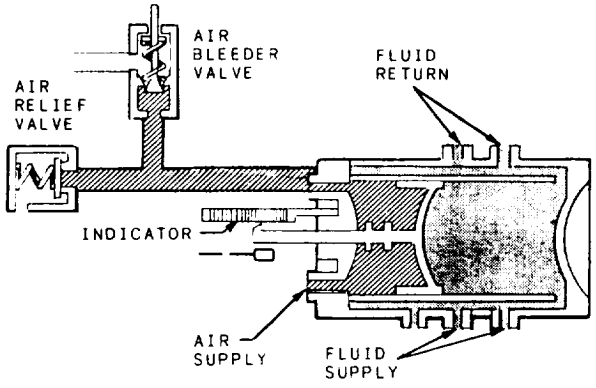

Figure 9-4.—Air-pressurized reservoir.

Some air-pressurized reservoirs also have direct contact of fluid to gas. These reservoirs are installed in large systems and may be cylindrical or rectangular in shape. They contain an oil level indicator, a pump inlet or suction line connection, a return line, a gas pressurization and venting connection, and a drain line connection or a drain plug. These reservoirs are pressurized by air from the ship's service air system or nitrogen banks. These reservoirs are found on board aircraft carriers and submarines.

ACCUMULATORS

An accumulator is a pressure storage reservoir in which hydraulic fluid is stored under pressure from an external source. The storage of fluid under pressure serves several purposes in hydraulic systems.

In some hydraulic systems it is necessary to maintain the system pressure within a specific pressure range for long periods of time. It is very difficult to maintain a closed system without some leakage, either external or internal. Even a small leak can cause a decrease in pressure. By using an accumulator, leakage can be compensated for

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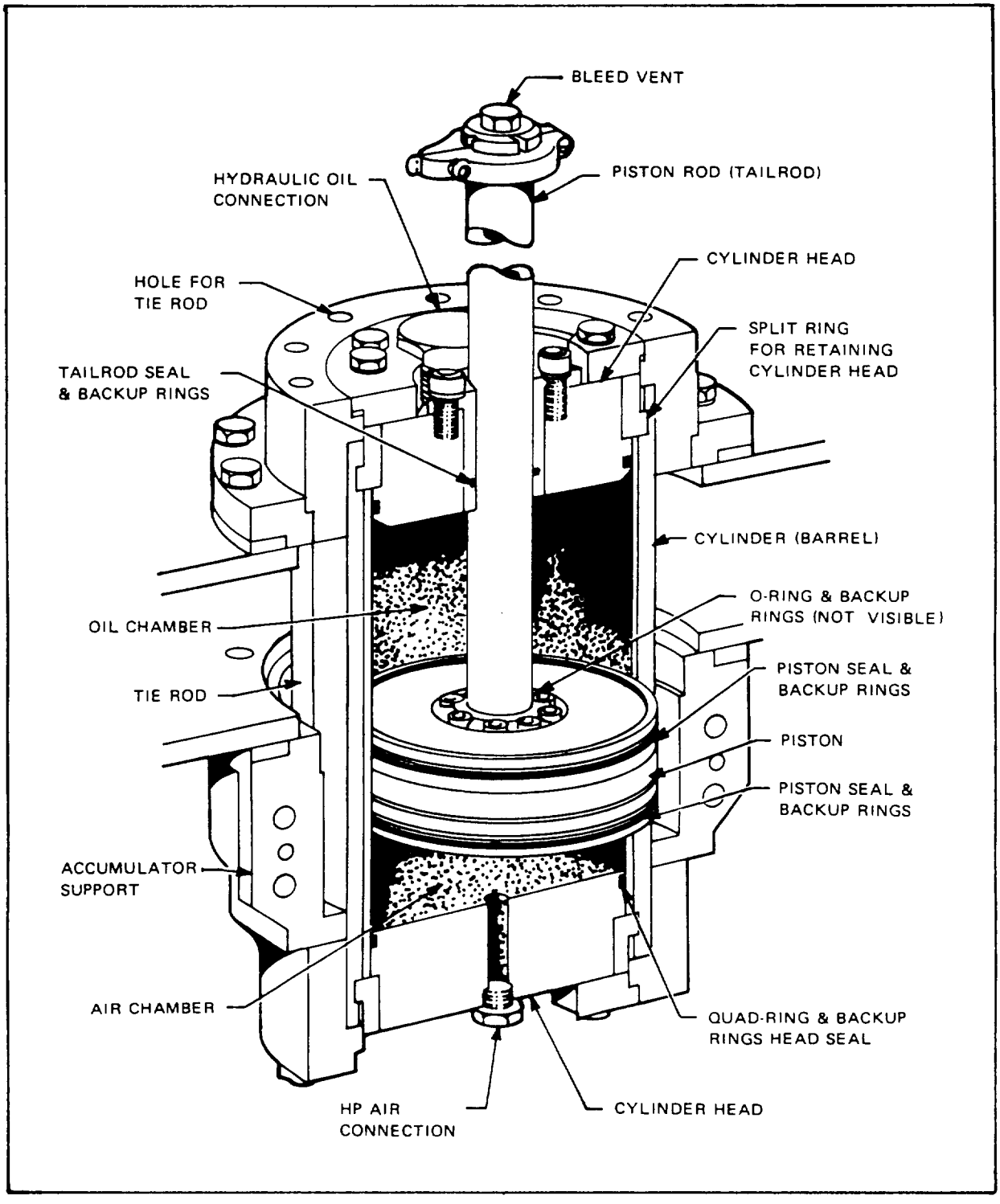

Figure 9-5.–Cross-section view of a piston-type accumulator with a tailrod.

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and the system pressure can be maintained within an acceptable range for long periods of time. Accumulators also compensate for thermal expansion and contraction of the liquid due to variations in temperature.

A liquid, flowing at a high velocity in a pipe will create a backward surge when stopped suddenly by the closing of a valve. This sudden stoppage causes instantaneous pressures two to three times the operating pressure of the system. These pressures, or shocks, produce objectional noise and vibrations which can cause considerable damage to piping, fittings, and components. The incorporation of an accumulator enables such shocks and surges to be absorbed or cushioned by the entrapped gas, thereby reducing their effects. The accumulator also dampens pressure surges caused by pulsating delivery from the pump.

There are times when hydraulic systems require large volumes of liquid for short periods of time. This is due to either the operation of large cylinders or the necessity of operating two or more circuits simultaneously. It is not economical to install a pump of such large capacity in the system for only intermittent usage, particularly if there is sufficient time during the working cycle for an accumulator to store up enough liquid to aid the pump during these peak demands.

The energy stored in accumulators maybe also used to actuate hydraulically operated units if normal hydraulic system failure occurs.

Four types of accumulators used in Navy hydraulic systems are as follows:

1. Piston type
2. Bag or bladder type
3. Direct-contact gas-to-fluid type
4. Diaphragm type

PISTON-TYPE ACCUMULATORS

Piston-type accumulators consist of a cylindrical body called a barrel, closures on each end called heads, and an internal piston. The piston may be fitted with a tailrod, which extends through one end of the cylinder (fig. 9-5), or it may not have a tailrod at all (fig. 9-6). In the latter case, it is referred to as a floating piston. Hydraulic fluid is pumped into one end of the cylinder and the piston is forced toward the opposite end of the cylinder against a captive

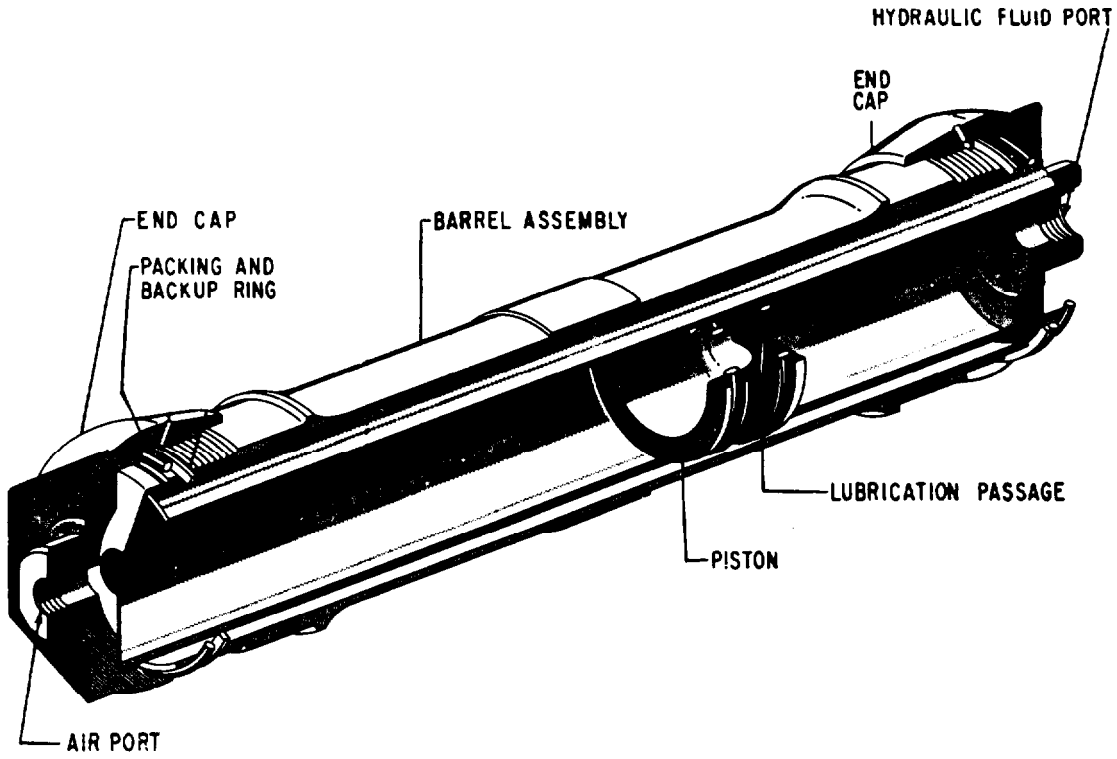

Figure 9-6.—Floating piston-type accumulator.

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charge of air or an inert gas such as nitrogen. Sometimes the amount of air charge is limited to the volume within the accumulator; other installations may use separate air flasks which are piped to the air side of the accumulator. Piston accumulators may be mounted in any position.

The gas portion of the accumulator may be located on either side of the piston. For example, in submarine hydraulic systems with tailrod pistons, the gas is usually on the bottom and the fluid on top; in surface ships with floating pistons, the gas is usually on the top. The orientation of the accumulator and the type of accumulator are based upon such criteria as available space, maintenance accessibility, size, need for external monitoring of the piston's location (tailrod indication), contamination tolerance, seal life, and safety. The purpose of the piston seals is to keep the fluid and the gas separate.

Usually, tailrod accumulators use two piston seals, one for the air side and one for the oil side, with the space between them vented to the atmosphere through a hole drilled the length of the tailrod. When the piston seals fail in this type of accumulator, air or oil leakage is apparent. However, seal failure in floating piston or nonvented tailrod accumulators will not be as obvious. Therefore, more frequent attention to venting or draining the air side is necessary. An indication of worn and leaking seals can be detected by the presence of significant amounts of oil in the air side.

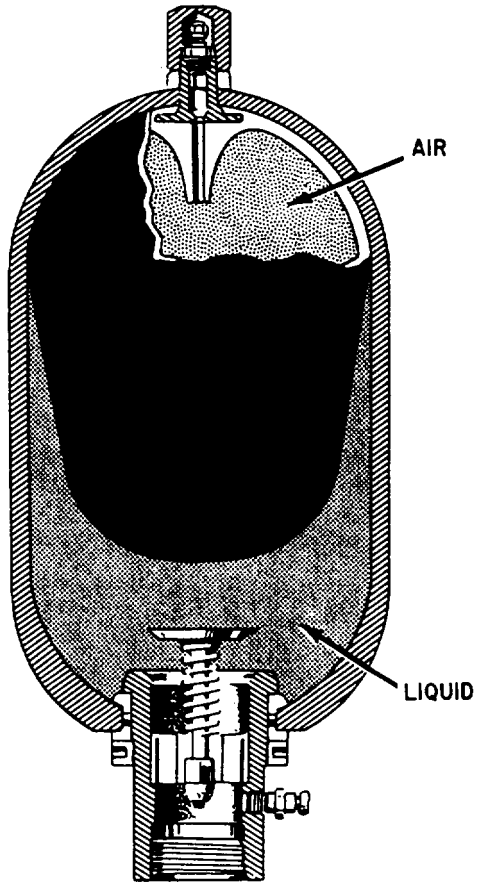

Figure 9-7.—Bladder-type accumulator.

BLADDER-TYPE ACCUMULATORS

Bladder- or bag-type accumulators consist of a shell or case with a flexible bladder inside the shell. See figure 9-7. The bladder is larger in diameter at the top (near the air valve) and gradually tapers to a smaller diameter at the bottom. The synthetic rubber is thinner at the top of the bladder than at the bottom. The operation of the accumulator is based on Barlow's formula for hoop stress, which states: "The stress in a circle is directly proportional to its diameter and wall thickness." This means that for a certain thickness, a large diameter circle will stretch faster than a small diameter circle; or for a certain diameter, a thin wall hoop will stretch faster than a thick wall hoop. Thus, the bladder will stretch around the top at its largest diameter and thinnest wall thickness, and then will gradually stretch downward and push itself outward against the walls of the shell. As a result, the bladder is capable of squeezing out all the liquid from the accumulator. Consequently, the bladder accumulator has a very high volumetric efficiency. In other words, this type of accumulator is capable of supplying a large percentage of the stored fluid to do work.

The bladder is precharged with air or inert gas to a specified pressure. Fluid is then forced into the area around the bladder, further compressing the gas in the bladder. This type of accumulator has the advantage that as long as the bladder is intact there is no exposure of fluid to the gas charge and therefore less danger of an explosion.

DIRECT-CONTACT GAS-TO-FLUID ACCUMULATORS

Direct-contact gas-to-fluid accumulators generally are used in very large installations where it would be very expensive to require a piston- or bladder-type accumulator. This type of accumulator consists of a fully enclosed cylinder, mounted in a vertical position, containing a liquid

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port on the bottom and a pneumatic charging port at the top (fig. 9-8). This type of accumulator is used in some airplane elevator hydraulic systems where several thousand gallons of fluid are needed to supplement the output of the hydraulic pumps for raising the elevator platform. The direct contact between the air or gas and the hydraulic fluid tends to entrain excessive amounts of gas in the fluid. For this reason, direct contact accumulators are generally not used for pressures over 1200 psi. The use of this type of accumulator with flammable fluid is dangerous because there is a possibility of explosion if any oxygen is present in the gas, and pressure surges generate excessive heat. For this reason, safety fluids are used in this type of installation.

DIAPHRAGM ACCUMULATORS

The diaphragm-type accumulator is constructed in two halves which are either screwed or bolted together. A synthetic rubber diaphragm is installed between both halves, making two chambers. Two threaded openings exist in the assembled component. The opening at the top, as shown in figure 9-9, contains a screen disc which prevents the diaphragm from extruding through the threaded opening when system pressure is depleted, thus rupturing the diaphragm. On some designs the screen is replaced by a button-type protector fastened to the center of the diaphragm. An air valve for pressurizing the accumulator is located in the gas chamber end of the sphere, and the liquid port to the hydraulic system is located on the opposite end of the sphere. This accumulator operates in a manner similar to that of the bladder-type accumulator.

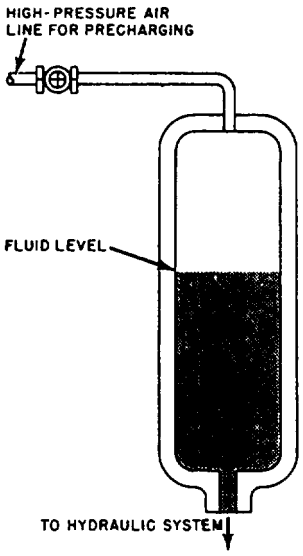

Figure 9-8.—Direct-contact gas-to-fluid accumulator.

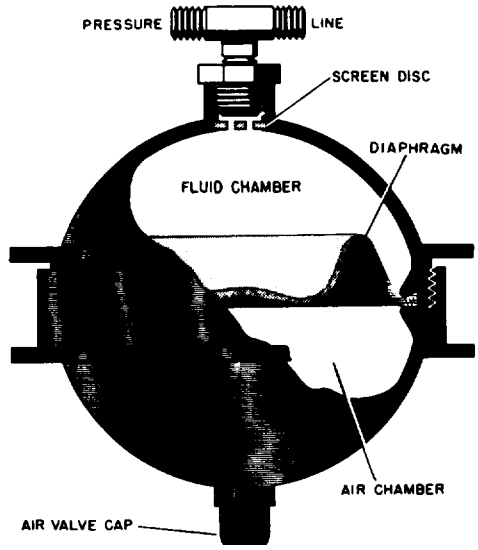

Figure 9-9.—Diaphragm accumulator.

FILTRATION

You have learned that maintaining hydraulic fluids within allowable limits is crucial to the care and protection of hydraulic equipment. While every effort must be made to prevent contaminants from entering the system, contaminants which *do* find their way into the system must be removed. Filtration devices are installed at key points in fluid power systems to remove the contaminants that enter the system along with those that are generated during normal operations.

Filtration devices for hydraulic systems differ somewhat from those of pneumatic systems. Therefore, they will be discussed separately.

The filtering devices used in hydraulic systems are commonly referred to as strainers and filters. Since they share a common function, the terms *strainer* and *filter* are often used interchangeably. As a general rule, devices used to remove large particles of foreign matter from hydraulic fluids are referred to as strainers, while those used to remove the smallest particles are referred to as filters.

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STRAINERS

Strainers are used primarily to catch only very large particles and will be found in applications where this type of protection is required. Most hydraulic systems have a strainer in the reservoir at the inlet to the suction line of the pump. A strainer is used in lieu of a filter to reduce its chance of being clogged and starving the pump. However, since this strainer is located in the reservoir, its maintenance is frequently neglected. When heavy dirt and sludge accumulate on the suction strainer, the pump soon begins to cavitate. Pump failure follows quickly.

FILTERS

The most common device installed in hydraulic systems to prevent foreign particles and contamination from remaining in the system are referred to as filters. They may be located in the reservoir, in the return line, in the pressure line, or in any other location in the system where the designer of the system decides they are needed to safeguard the system against impurities.

Filters are classified as full flow and proportional or partial flow. In the full-flow type of filter, all the fluid that enters the unit passes through the filtering element, while in the proportional-flow type, only a portion of the fluid passes through the element.

Full-Flow Filter

The full-flow filter provides a positive filtering action; however, it offers resistance to flow, particularly when the element becomes dirty. Hydraulic fluid enters the filter through the inlet port in the body and flows around the filter element inside the filter bowl. Filtering takes place as the fluid passes through the filtering element and into the hollow core, leaving the dirt and impurities on the outside of the filter element. The filtered fluid then flows from the hollow core through the outlet port and into the system (fig. 9-10).

Some full-flow filters are equipped with a contamination indicator (fig. 9-11). These indicators, also known as differential pressure indicators, are available in three types—gauge indicators, mechanical pop-up indicators, and electrical with mechanical pop-up indicators. As contaminating particles collect on the filter element, the differential pressure across the element increases. In some installations using

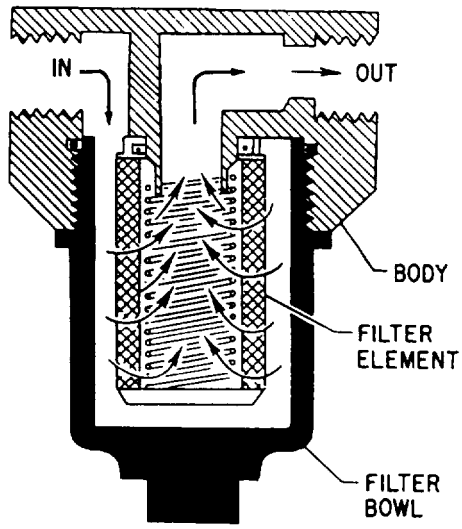

Figure 9-10.—Full-flow hydraulic filter.

gauges as indicators, the differential pressure must be obtained by subtracting the readings of two gauges located somewhere along the filter inlet and outlet piping. For pop-up indicators, when the increase in pressure reaches a specific value, an indicator (usually in the filter head) pops out, signifying that the filter must be cleaned or replaced. A low-temperature lockout feature is installed in most pop-up types of contamination indicators to eliminate the possibility of false indications due to cold weather because the pressure differential may be much higher with a cold fluid due to increased viscosity.

Filter elements used in filters that have a contamination indicator are not normally removed or replaced until the indicator is actuated. This decreases the possibility of system contamination from outside sources due to unnecessary handling.

The use of the nonbypassing type of filter eliminates the possibility of contaminated fluid bypassing the filter element and contaminating the entire system. This type of filter will minimize the necessity for flushing the entire system and lessen the possibility of failure of pumps and other components in the system.

A bypass relief valve is installed in some filters. The bypass relief valve allows the fluid to bypass the filter element and pass directly through the outlet port in the event that the filter element becomes clogged. These filters may or may not be equipped with the contamination indicator. Figure 9-11 shows a full-flow bypass-type

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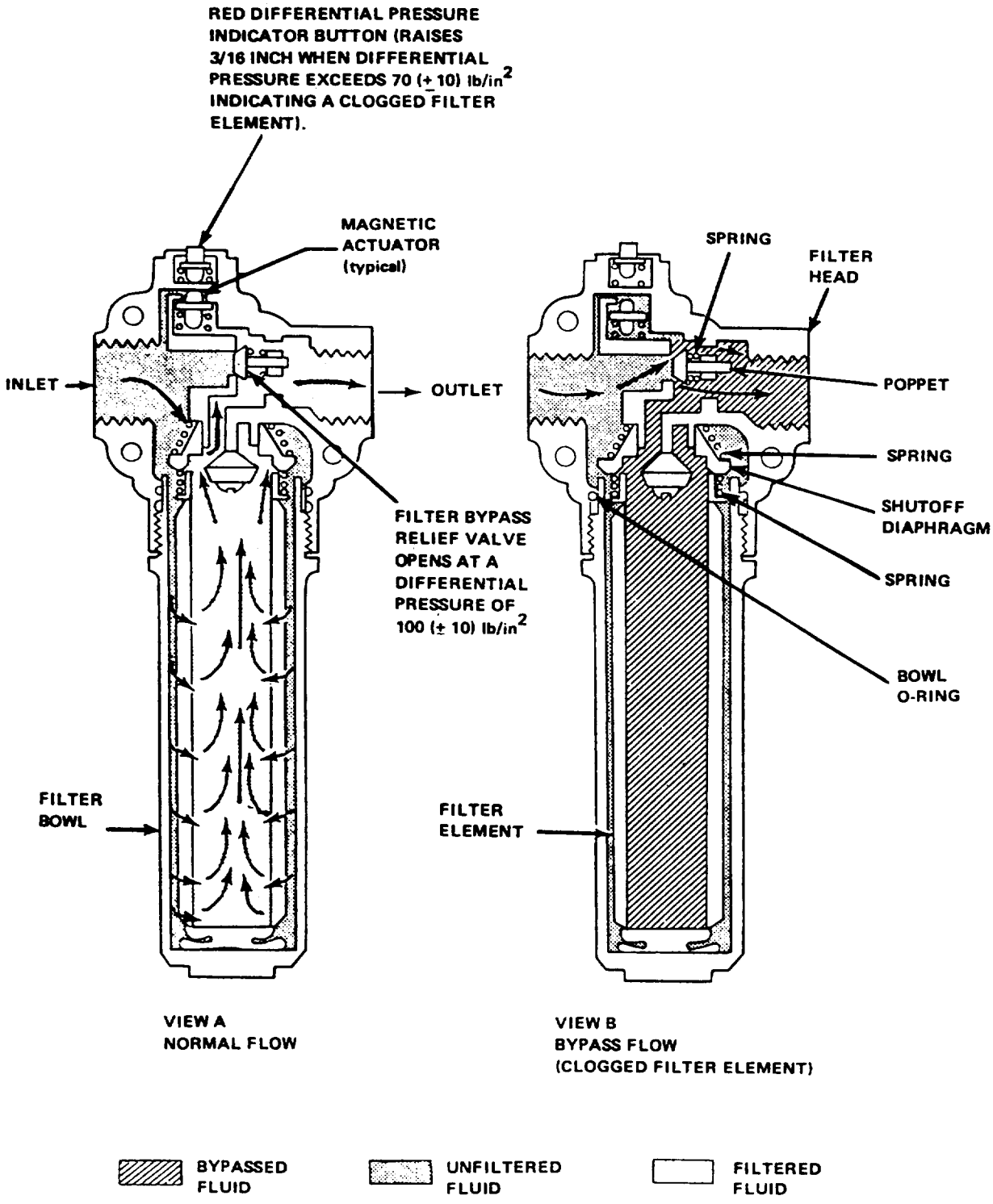

Figure 9-11.—Full-flow bypass-type hydraulic filter (with contamination indicator).

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hydraulic filter with a contamination indicator. Figure 9-12 shows a full-flow bypass-type hydraulic filter without a contamination indicator.

A filter bypass indicator provides a positive indication, when activated, that fluid is bypassing the filter element by flowing through the bypass relief valve. This indicator should not be confused with the pop-up differential pressure indicator previously discussed which simply monitors the pressure across the element. With the bypass indicator, a similar pop-up button is often used to signal that maintenance is needed. However, the bypass indicators further signal that, as a result of the high differential pressures across the element, an internal bypass relief valve has lifted and some of the fluid is bypassing the element.

Identification of the type of installed indicator can be obtained from filter manifold drawings or related equipment manuals. Both a fluid bypass indicator and a differential pressure indicator or gauge may be installed on the same filter assembly.

As with differential pressure indicators, bypass relief indicators can be activated by pressure surges, such as may develop during cold starts or rapid system pressurization. On some relief indicators, the pop-up button, or whatever signal device is used, will return to a normal position when the surge passes and pressure is reduced. Other relief indicators may continue to indicate a bypass condition until they are manually reset. Before corrective action is taken based on indicator readings, the bypass condition should be verified at normal operating temperature and flow conditions by attempting to reset the indicator.

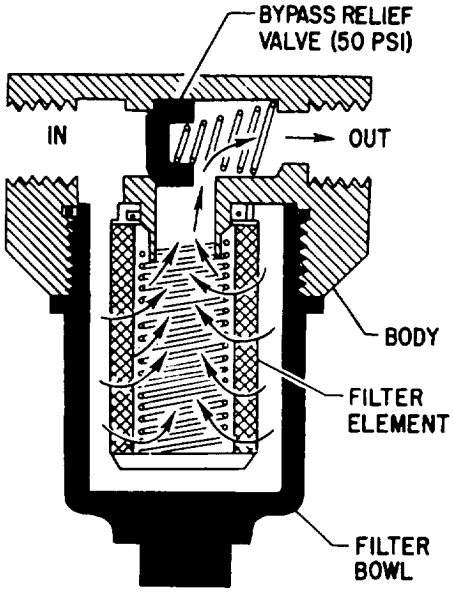

Figure 9-12.—Full-flow bypass-type hydraulic filter.

Proportional-Flow Filter

This type of filter operates on the venturi principle. (See glossary.) As the fluid passes through the venturi throat a drop in pressure is created at the narrowest point. See figure 9-13. A portion of the fluid flowing toward and away from the throat of the venturi flows through the passages into the body of the filter. A fluid passage connects the hollow core of the filter with the throat of the venturi. Thus, the low-pressure area at the throat of the venturi causes the fluid under pressure in the body of the filter to flow through the filter element, through the hollow core, into the low-pressure area, and then return to the system. Although only a portion of the fluid is filtered during each cycle, constant recirculation through the system will eventually cause all the fluid to pass through the filter element.

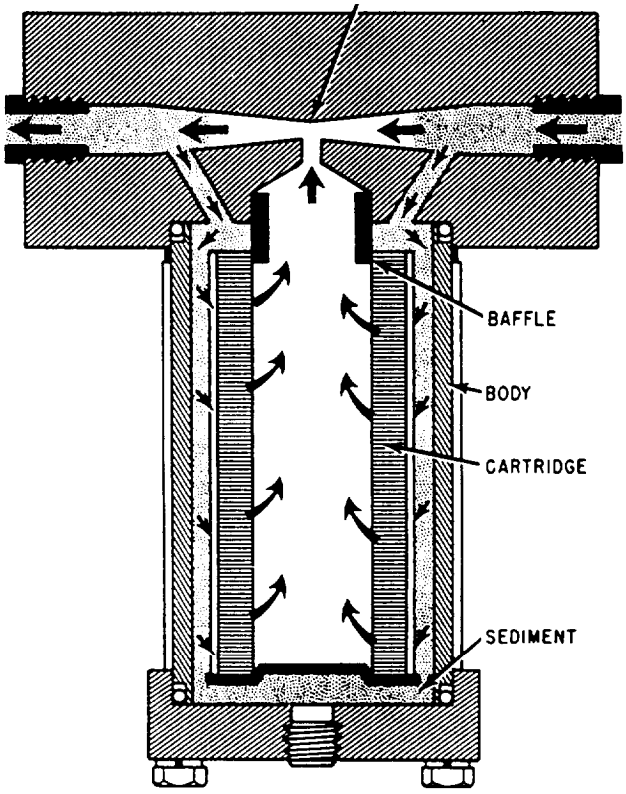

Figure 9-13.Proportional-flow filter.

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Filter Rating

Filters are rated in several ways—absolute, mean, and nominal. The absolute filtration rating is the diameter in microns of the largest spherical particle that will pass through the filter under a certain test condition. This rating is an indication of the largest opening in the filter element. The mean filtration rating is the measurement of the average size of the openings in the filter element. The nominal filtration rating is usually interpreted to mean the size of the smallest particles of which 90 percent will be trapped in the filter at each pass through the filter.

Filter Elements

Filter elements generally may be divided into two classes—surface and depth. Surface filters are made of closely woven fabric or treated paper with a uniform pore size. Fluid flows through the pores of the filter material and contaminants are stopped on the filter's surface. This type of element is designed to prevent the passage of a high percentage of solids of a specific size. Depth filters, on the other hand, are composed of layers of fabric or fibers which provide many tortuous paths for the fluid to flow through. The pores or passages must be larger than the rated size of the filter if particles are to be retained in the depth of the media rather than on the surface. Consequently, there is a statistical probability that a rather large particle may pass through a depth-type filter.

Filter elements may be of the 5-micron, woven mesh, micronic, porous metal, or magnetic type. The micronic and 5-micron elements have noncleanable filter media and are disposed of when they are removed. Porous metal, woven mesh, and magnetic filter elements are usually designed to be cleaned and reused.

5-MICRON NONCLEANABLE FILTER ELEMENTS.— The most common 5-micron filter medium is composed of organic and inorganic fibers integrally bonded by epoxy resin and faced with a metallic mesh upstream and downstream for protection and added mechanical strength. Filters of this type are not to be cleaned under any circumstances and will be marked Disposable or Noncleanable.

Another 5-micron filter medium uses layers of very fine stainless-steel fibers drawn into a random but controlled matrix. Filter elements of this material may be either cleanable or noncleanable, depending upon their construction.

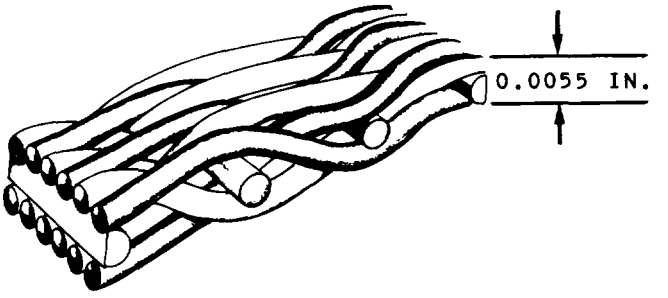

Figure 9-14.—Cross-section of a stainless steel hydraulic filter element.

WOVEN WIRE-MESH FILTER ELEMENTS.— Filters of this type are made of stainless steel and are generally rated as 15 or 25 micron (absolute). Figure 9-14 shows a magnified cross section of a woven wire-mesh filter element. This type of filter is reusable.

MICRONIC HYDRAULIC FILTER ELEMENT.— The term *micronic is* derived from the word *micron*. It could be used to describe any filter element; however, through usage, this term has become associated with a specific filter with a filtering element made of a specially treated cellulose paper (fig. 9-15). The filter shown in figure 9-10 is a typical micronic hydraulic filter. This filter is designed to remove 99 percent of all particles 10 to 20 microns in diameter or larger.

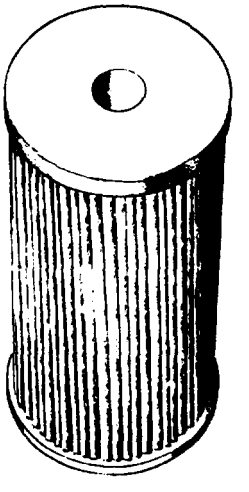

Figure 9-15.—Micronic filter element.

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The replaceable element is made of specially treated convolutions (wrinkles) to increase its dirt-holding capacity. The element is noncleanable and should be replaced with a new filter element during maintenance inspections.

MAGNETIC FILTERS.— Some hydraulic systems have magnetic filters installed at strategic points. Filters of this type are designed primarily to trap any ferrous particles that may be in the system.

PNEUMATIC GASES

Clean, dry gas is required for the efficient operation of pneumatic systems. Due to the normal conditions of the atmosphere, free air seldom satisfies these requirements adequately. The atmosphere contains both dust and impurities in various amounts and a substantial amount of moisture in vapor form.

Solids, such as dust, rust, or pipe scale in pneumatic systems, may lead to excessive wear and failure of components and, in some cases, may prevent the pneumatic devices from operating. Moisture is also very harmful to the system. It washes lubrication from moving parts, thereby aiding corrosion and causing excessive wear of components. Moisture will also settle in low spots in the system and freeze during cold weather, causing a stoppage of the system or ruptured lines.

An ideal filter would remove all dirt and moisture from a pneumatic system without causing a pressure drop in the process. Obviously, such a condition can only be approached; it cannot be attained.

Removal of Solids

The removal of solids from the gas of pneumatic systems is generally done by screening (filtering), centrifugal force, or a combination of the two. In some cases, the removal of moisture is done in conjunction with the removal of solids.

Some types of air filters are similar in design and operation to the hydraulic filters discussed earlier. Some materials used in the construction of elements for air filters are woven screen wire, steel wool, fiber glass, and felt fabrics. Elements made of these materials are often used in the unit that filters the air as it enters the compressor.

Porous metal and ceramic elements are commonly used in filters that are installed in the compressed air supply lines. These filters also use a controlled air path to provide some filtration. Internal design causes the air to flow in a circular path within the bowl (fig. 9-16). Heavy particles and water droplets are thrown out of the airstream and drop to the bottom of the bowl. The air then flows through the filter element, which filters out most of the smaller particles. This type of filter is designed with a drain valve at the bottom of the bowl. When the valve is opened with air pressure in the system, the accumulation of solids and water will be blown out of the bowl.

An air filter that uses moving mechanical devices as an element is illustrated in figure 9-17. As compressed air passes through the filter the force revolves a number of multi-blade rotors at high speed. Moisture and dirt are caught on the blades of the rotors. The whirling blades hurl the impurities by centrifugal force to the outer rims of the rotors and to the inner walls of the filter housing. Here, contaminating matter is out of the airstream and falls to the bottom of the bowl where it must be drained at periodic intervals.

Removal of Moisture

The removal of moisture from compressed air is important for a compressed air system. If air at atmospheric pressure, even at a very low relative humidity, is compressed to 3000 or 4500 psi, it becomes saturated with water vapor. Some moisture is removed by the intercoolers and aftercoolers (see glossary). Also, air flasks, receivers, and banks are provided with low point drains to allow periodic draining of any collected moisture. However, many uses of air require air with an even smaller moisture content than can be obtained through these methods. Moisture in

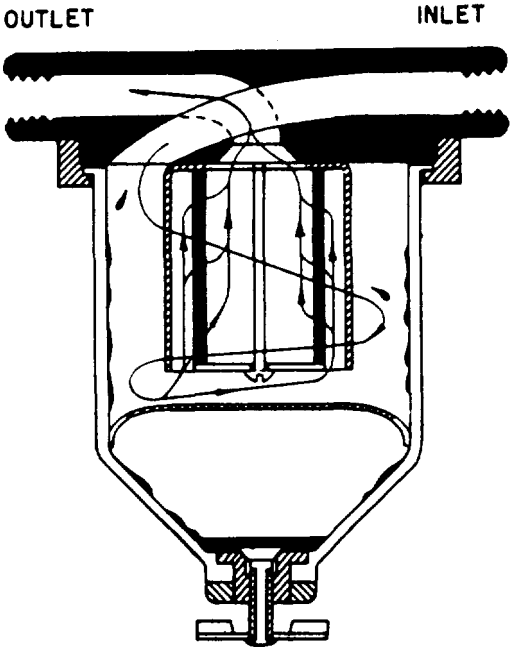

Figure 9-16.—Air filter.

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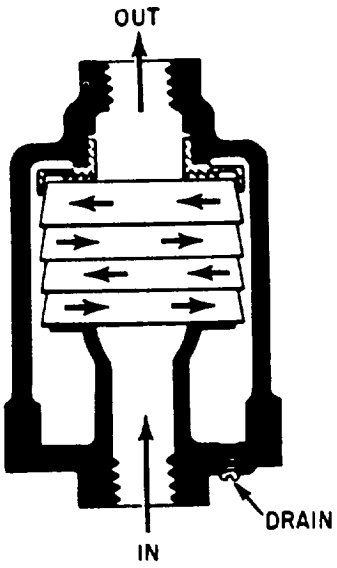

Figure 9-17.—Air filter using rotating blades as element.

air lines can create problems which are potentially hazardous, such as the freezing of valves and controls. This can occur, for example, if very high pressure air is throttled to a very low pressure at a high flow rate. The venturi effect of the throttled air produces very low temperatures which will cause any moisture in the air to freeze into ice. This makes the valve (especially an automatic valve) either very difficult or impossible to operate. Also, droplets of water can cause serious water hammer in an air system which has high pressure and a high flow rate and can cause corrosion, rust, and dilution of lubricants within the system. For these reasons, air driers (*dehydrator, air purifier,* and *desiccator* are all terms used by different manufacturers to identify these components) are used to dry the compressed air. Some water removal devices are similar in design and operation to the filters shown in figures 9-16 and 9-17. Two basic types of air dehydrators are the refrigerated-type and the desiccant-type.

REFRIGERATED-TYPE DEHYDRATORS.– In refrigerated-type dehydrators, compressed air is passed over a set of refrigerated cooling coils. Oil and moisture vapors condense from the air and can be collected and removed via a low point drain.

DESICCANT-TYPE DEHYDRATORS.– A desiccant is a chemical substance with a high capacity to absorb water or moisture. It also has the capacity to give off that moisture so that the desiccant can be reused.

Some compressed air system dehydrators use a pair of desiccant towers (flasks full of desiccant). One is kept in service dehydrating the compressed air, while the other one is being reactivated. A desiccant tower is normally reactivated by passing dry, heated air through it in the direction opposite the normal dehydration airflow.

Another type of chemical drier is shown in figure 9-18. This unit consists of the housing, a cartridge containing a chemical agent, a filter (sintered bronze), and a spring. Various types of absorbent chemicals are used by the different manufacturers in the construction of the cartridges. To ensure proper filtering, the air must pass through the drier in the proper direction. The correct direction of flow is indicated by an arrow and the word *FLOW* printed on the side of the cartridge.

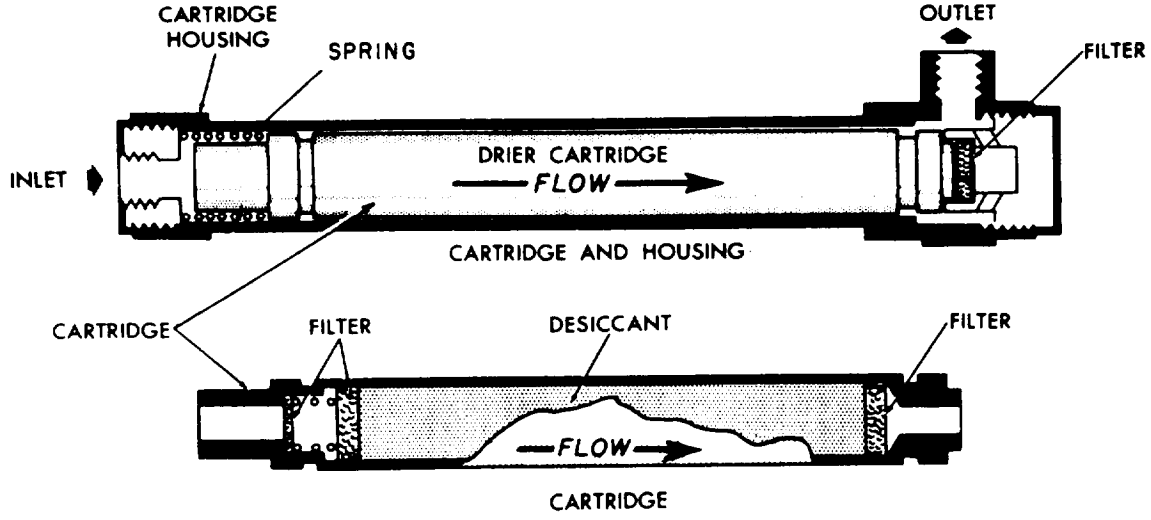

Figure 9-18.–Chemical drier.

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CHAPTER 10

ACTUATORS

One of the outstanding features of fluid power systems is that force, generated by the power supply, controlled and directed by suitable valving, and transported by lines, can be converted with ease to almost any kind of mechanical motion desired at the very place it is needed. Either linear (straight line) or rotary motion can be obtained by using a suitable actuating device.

An actuator is a device that converts fluid power into mechanical force and motion. Cylinders, motors, and turbines are the most common types of actuating devices used in fluid power systems.

This chapter describes various types of actuating cylinders and their applications, different types of fluid motors, and turbines used in fluid power systems.

CYLINDERS

An actuating cylinder is a device that converts fluid power to linear, or straight line, force and motion. Since linear motion is a back-and-forth motion along a straight line, this type of actuator is sometimes referred to as a reciprocating, or linear, motor. The cylinder consists of a ram or piston operating within a cylindrical bore. Actuating cylinders may be installed so that the cylinder is anchored to a stationary structure and the ram or piston is attached to the mechanism to be operated, or the piston or ram may be anchored to the stationary structure and the cylinder attached to the mechanism to be operated.

Actuating cylinders for pneumatic and hydraulic systems are similar in design and operation. Some of the variations of ram- and piston-type actuating cylinders are described in the following paragraphs.

RAM-TYPE CYLINDERS

The terms *ram* and *piston* are often used interchangeably. However, a ram-type cylinder is usually considered one in which the cross-sectional area of the piston rod is more than one-half the cross-sectional area of the movable element. In most actuating cylinders of this type, the rod and the movable element have equal areas. This type of movable element is frequently referred to as a plunger.

The ram-type actuator is used primarily to push rather than to pull. Some applications require simply a flat surface on the external part of the ram for pushing or lifting the unit to be operated. Other applications require some mechanical means of attachment, such as a clevis or eyebolt. The design of ram-type cylinders varies in many other respects to satisfy the requirements of different applications.

Single-Acting Ram

The single-acting ram (fig. 10-1) applies force in only one direction. The fluid that is directed into the cylinder displaces the ram and forces it outward, lifting the object placed on it. Since there

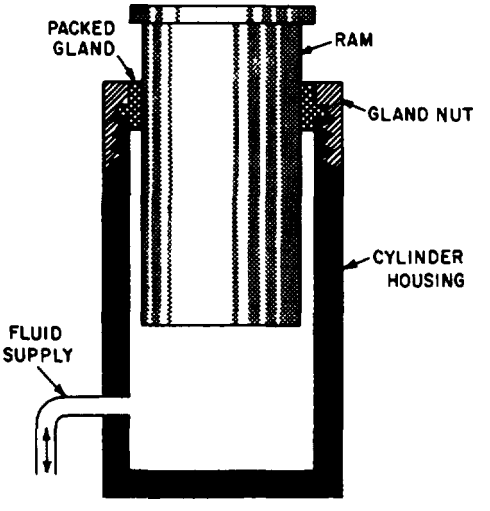

Figure 10-1.—Single-acting ram-type actuating cylinder.

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is no provision for retracting the ram by fluid power, when fluid pressure is released, either the weight of the object or some mechanical means, such as a spring, forces the ram back into the cylinder. This forces the fluid back to the reservoir.

The single-acting ram-type actuating cylinder is often used in the hydraulic jack. The elevators used to move aircraft to and from the flight deck and hangar deck on aircraft carriers also use cylinders of this type. In these elevators, the cylinders are installed horizontally and operate the elevator through a series of cables and sheaves. Fluid pressure forces the ram outward and lifts the elevator. When fluid pressure is released from the ram, the weight of the elevator forces the ram back into the cylinder. This, in turn, forces the fluid back into the reservoir.

Double-Acting Ram

A double-acting ram-type cylinder is illustrated in figure 10-2. In this cylinder, both strokes of the ram are produced by pressurized fluid. There are two fluid ports, one at or near each end of the cylinder. Fluid under pressure is directed to the closed end of the cylinder to extend the ram and apply force. To retract the ram and reduce the force, fluid is directed to the opposite end of the cylinder.

A four-way directional control valve is normally used to control the double-acting ram. When the valve is positioned to extend the ram, pressurized fluid enters port A (fig. 10-2), acts on the bottom surface of the ram, and forces the ram outward. Fluid above the ram lip is free to flow out of port B, through the control valve, and to the return line in hydraulic systems or to the atmosphere in pneumatic systems.

Normally, the pressure of the fluid is the same for either stroke of the ram. Recall from chapter 2 that force is equal to pressure times area ($F = PA$). Notice the difference of the areas upon which the pressure acts in figure 10-2. The pressure acts against the large surface area on the bottom of the ram during the extension stroke, during which time the ram applies force. Since the ram does not require a large force during the retraction stroke, pressure acting on the small area on the top surface of the ram lip provides the necessary force to retract the ram.

Telescoping Rams

Figure 10-3 shows a telescoping ram-type actuating cylinder. A series of rams is nested in the telescoping assembly. With the exception of the smallest ram, each ram is hollow and serves as the cylinder housing for the next smaller ram. The ram assembly is contained in the main cylinder assembly, which also provides the fluid ports. Although the assembly requires a small space with all the rams retracted, the telescoping action of the assembly provides a relatively long stroke when the rams are extended.

An excellent example of the application of this type of cylinder is in the dump truck. It is used to lift the forward end of the truck bed and dump the load. During the lifting operation, the greatest force is required for the initial lifting of the load.

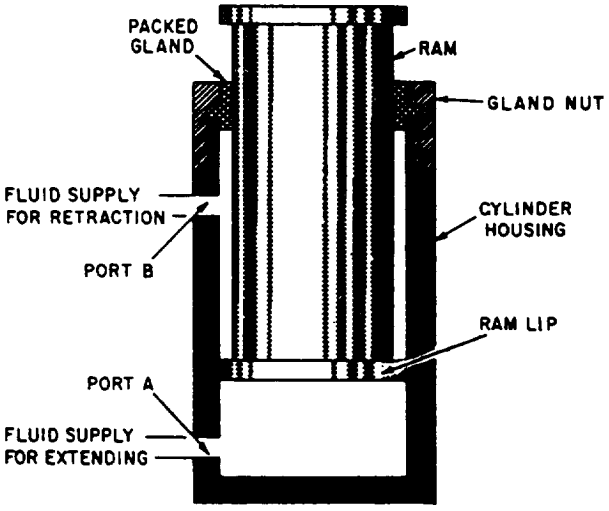

Figure 10-2.—Double-acting ram-type actuating cylinder.

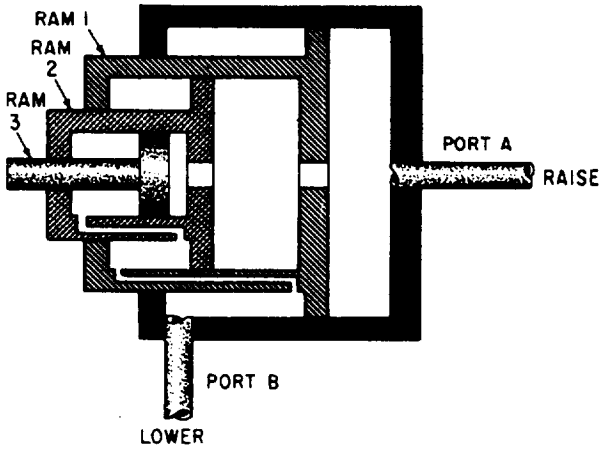

Figure 10-3.—Telescoping ram-type actuating cylinder.

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As the load is lifted and begins to dump, the required force becomes less and less until the load is completely dumped. During the raise cycle, pressurized fluid enters the cylinder through port A (fig. 10-3) and acts on the bottom surface of all three rams. Ram 1 has a larger surface area and, therefore, provides the greater force for the initial load, As ram 1 reaches the end of its stroke and the required force is decreased, ram 2 moves, providing the smaller force needed to continue raising the load. When ram 2 completes its stroke, a still smaller force is required. Ram 3 then moves outward to finish raising and dumping the load.

Some telescoping ram-type cylinders are of the single-acting type. Like the single-acting ram discussed previously, these telescoping ram-type cylinders are retracted by gravity or mechanical force. Some hydraulic jacks are equipped with telescoping rams. Such jacks are used to lift vehicles with low clearances to the required height.

Other types of telescoping cylinders, like the one illustrated in figure 10-3, are of the double-acting type. In this type, fluid pressure is used for both the extension and retraction strokes. A four-way directional control valve is commonly used to control the operation of the double-acting type. Note the small passages in the walls of rams 1 and 2. They provide a path for fluid to flow to and from the chambers above the lips of rams 2 and 3. During the extension stroke, return fluid flows through these passages and out of the cylinder through port B. It then flows through the directional control valve to the return line or reservoir.

To retract the rams, fluid under pressure is directed into the cylinder through port B and acts against the top surface areas of all three ram lips. This forces the rams to the retracted position. The displaced fluid from the opposite side of the rams flows out of the cylinder through port A, through the directional control valve to the return line or reservoir.

Dual Rams

A dual ram assembly consists of a single ram with a cylinder at either end (fig. 10-4). Fluid can be directed to either cylinder, forcing the ram to move in the opposite direction. The ram is connected through mechanical linkage to the unit to be operated. A four-way directional control valve is commonly used to operate the dual ram. When the control valve is positioned to direct fluid under pressure to one of the cylinders (let's say the left one), the ram is forced to the right. This action displaces the fluid in the opposite cylinder. The displaced fluid flows back through the directional control valve to the return line or reservoir in hydraulic systems or to the atmosphere in pneumatic systems.

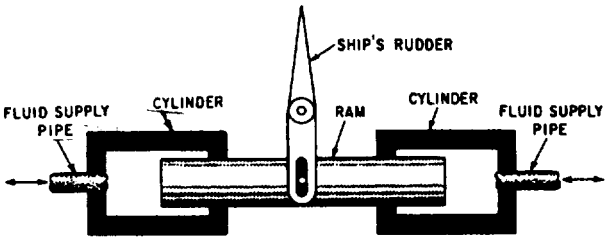

Figure 10-4.-Dual ram actuating assembly.

Dual ram actuating assemblies are used in steering systems of most ships. In some systems, one assembly is used to actuate the rudder in either direction; while in other systems, two assemblies are used for the same purpose.

PISTON-TYPE CYLINDERS

An actuating cylinder in which the cross-sectional area of the piston is less than one-half the cross-sectional area of the movable element is referred to as a piston-type cylinder. This type of cylinder is normally used for applications that require both push and pull functions. The piston-type cylinder is the most common type used in fluid power systems.

The essential parts of a piston-type cylinder are a cylindrical barrel, a piston and rod, end caps, and suitable seals. The end caps are attached to the ends of the barrel. These end caps usually contain the fluid ports. The end cap on the rod end contains a hole for the piston rod to pass through. Suitable seals are used between the hole and the piston rod to keep fluid from leaking out and to keep dirt and other contaminants from entering the barrel. The opposite end cap of most cylinders is provided with a fitting for securing the actuating cylinder to some structure. This end cap is referred to as the anchor end cap.

The piston rod may extend through either or both ends of the cylinder. The extended end of the rod is normally threaded so that some type of mechanical connector, such as an eyebolt or a clevis, and a locknut can be attached. This threaded connection of the rod and mechanical connector provides for adjustment between the rod and the unit to be actuated. After the correct

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adjustment is made, the locknut is tightened against the connector to prevent the connector from turning. The other end of the connector is attached, either directly or through additional mechanical linkage, to the unit to be actuated.

In order to satisfy the many requirements of fluid power systems, piston-type cylinders are available in various designs.

Single-Acting Cylinder

The single-acting piston-type cylinder is similar in design and operation to the single-acting ram-type cylinder. The single-acting piston-type cylinder uses fluid pressure to provide the force in one direction, and spring tension, gravity, compressed air, or nitrogen is used to provide the force in the opposite direction. Figure 10-5 shows a single-acting, spring-loaded, piston-type actuating cylinder. In this cylinder the spring is located on the rod side of the piston. In some spring-loaded cylinders the spring is located on the blank side, and the fluid port is on the rod side of the cylinder.

A three-way directional control valve is normally used to control the operation of the single-acting piston-type cylinder. To extend the piston rod, fluid under pressure is directed through the port into the cylinder (fig. 10-5). This pressure acts on the surface area of the blank side of the piston and forces the piston to the right. This action moves the rod to the right, through the end of the cylinder, thus moving the actuated unit in one direction. During this action, the spring is compressed between the rod side of the piston and the end of the cylinder. The length of the stroke depends upon the physical limits within the cylinder and the required movement of the actuated unit.

To retract the piston rod, the directional control valve is moved to the opposite working position, which releases the pressure in the cylinder. The spring tension forces the piston to the left, retracting the piston rod and moving the actuated unit in the opposite direction. The fluid is free to flow from the cylinder through the port, back through the control valve to the return line in hydraulic systems or to the atmosphere in pneumatic systems.

The end of the cylinder opposite the fluid port is vented to the atmosphere. This prevents air from being trapped in this area. Any trapped air would compress during the extension stroke, creating excess pressure on the rod side of the piston. This would cause sluggish movement of the piston and could eventually cause a complete lock, preventing the fluid pressure from moving the piston.

The spring-loaded cylinder is used in arresting gear systems on some models of carrier aircraft. To raise (retract) the arresting hook, fluid pressure is directed through the arresting hook control valve to the rod side of the cylinder. This force moves the piston, which, through the rod and mechanical linkage, retracts the arresting hook. The arresting hook extends when fluid pressure is released from the rod side of the cylinder, allowing the spring to expand.

Leakage between the cylinder wall and piston is prevented by adequate seals. The piston in figure 10-5 contains V-ring seals.

Double-Acting Cylinder

Most piston-type actuating cylinders are double-acting, which means that fluid under pressure can be applied to either side of the piston to apply force and provide movement.

One design of the double-acting cylinder is shown in figure 10-6. This cylinder contains one piston and piston rod assembly. The stroke of the piston and piston rod assembly in either direction is produced by fluid pressure. The two fluid ports, one near each end of the cylinder, alternate as inlet and outlet ports, depending on the direction of

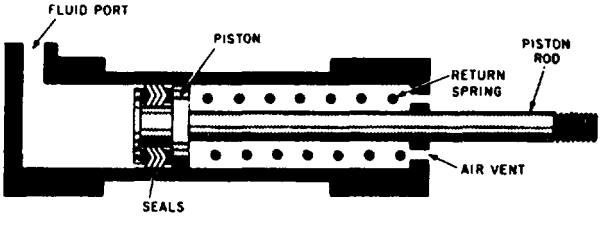

Figure 10-5.—Single-acting, spring-loaded, piston-type actuating cylinder.

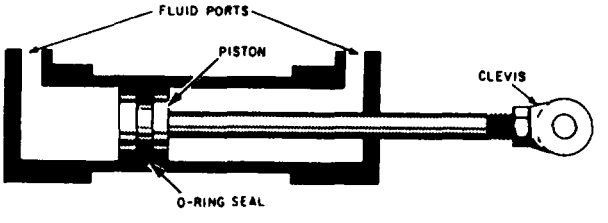

Figure 10-6.-Doub1e-acting piston-type actuating cylinder.

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flow from the directional control valve. This actuator (fig. 10-6) is referred to as an unbalanced actuating cylinder because there is a difference in the effective working areas on the two sides of the piston. Therefore, this type of cylinder is normally installed so that the blank side of the piston carries the greater load; that is, the cylinder carries the greater load during the piston rod extension stroke.

A four-way directional control valve is normally used to control the operation of this type of cylinder. The valve can be positioned to direct fluid under pressure to either end of the cylinder and allow the displaced fluid to flow from the opposite end of the cylinder through the control valve to the return line in hydraulic systems or to the atmosphere in pneumatic systems.

There are applications where it is necessary to move two mechanisms at the same time. In this case, double-acting piston-type actuating cylinders of different designs are required. See figures 10-7 and 10-8.

Figure 10-7 shows a three-port, double-acting piston-type actuating cylinder. This actuator contains two pistons and piston rod assemblies. Fluid is directed through port A by a four-way directional control valve and moves the pistons outward, thus moving the mechanisms attached to the pistons' rods. The fluid on the rod side of each piston is forced out of the cylinder through ports B and C, which are connected by a common line to the directional control valve. The displaced fluid then flows through the control valve to the return line or to the atmosphere.

When fluid under pressure is directed into the cylinder through ports B and C, the two pistons move inward, also moving the mechanisms attached to them. Fluid between the two pistons is free to flow from the cylinder through port A and through the control valve to the return line or to the atmosphere.

The actuating cylinder shown in figure 10-8 is a double-acting balanced type. The piston rod extends through the piston and out through both ends of the cylinder. One or both ends of the

Figure 10-7.—Three-port, double-acting actuating cylinder.

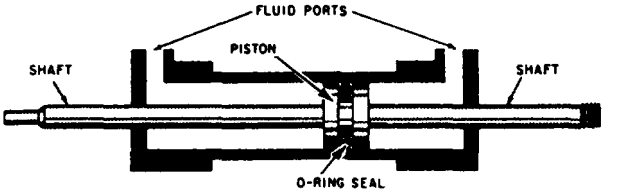

Figure 10-8.-Balanced, double-acting piston-type actuating cylinder.

piston rod may be attached to a mechanism to be operated. In either case, the cylinder provides equal areas on each side of the piston. Therefore, the same amount of fluid and force is used to move the piston a certain distance in either direction.

Tandem Cylinders

A tandem actuating cylinder consists of two or more cylinders arranged one behind the other but designed as a single unit (fig. 10-9). This type of actuating cylinder is used in applications that require two or more independent systems; for example, power-operated flight control systems in naval aircraft.

The flow of fluid to and from the two chambers of the tandem actuating cylinder is provided from two independent hydraulic systems and is controlled by two sliding spool directional control valves. In some applications, the control valves and the actuating cylinder are two separate units. In some units, the pistons (lands) of the two sliding spools are machined on one common shaft. In other applications, the valves and the actuator are directly connected in one compact unit. Although the two control valves are hydraulically independent, they are interconnected mechanically. In other units, the two sliding spools are connected through mechanical linkages with a synchronizing rod. In either case, the movement of the two sliding spools is synchronized, thus equalizing the

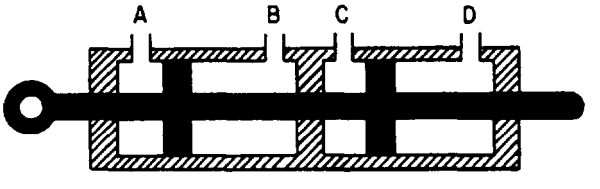

Figure 10-9.—Tandem actuating cylinder.

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flow of fluid to and from the two chambers of the actuating cylinder.

Since the two control valves operate independently of each other as far as hydraulic pressure is concerned, failure of either hydraulic system does not render the actuator inoperative. Failure of one system does reduce the output force by one-half; however, this force is sufficient to permit operation of the actuator.

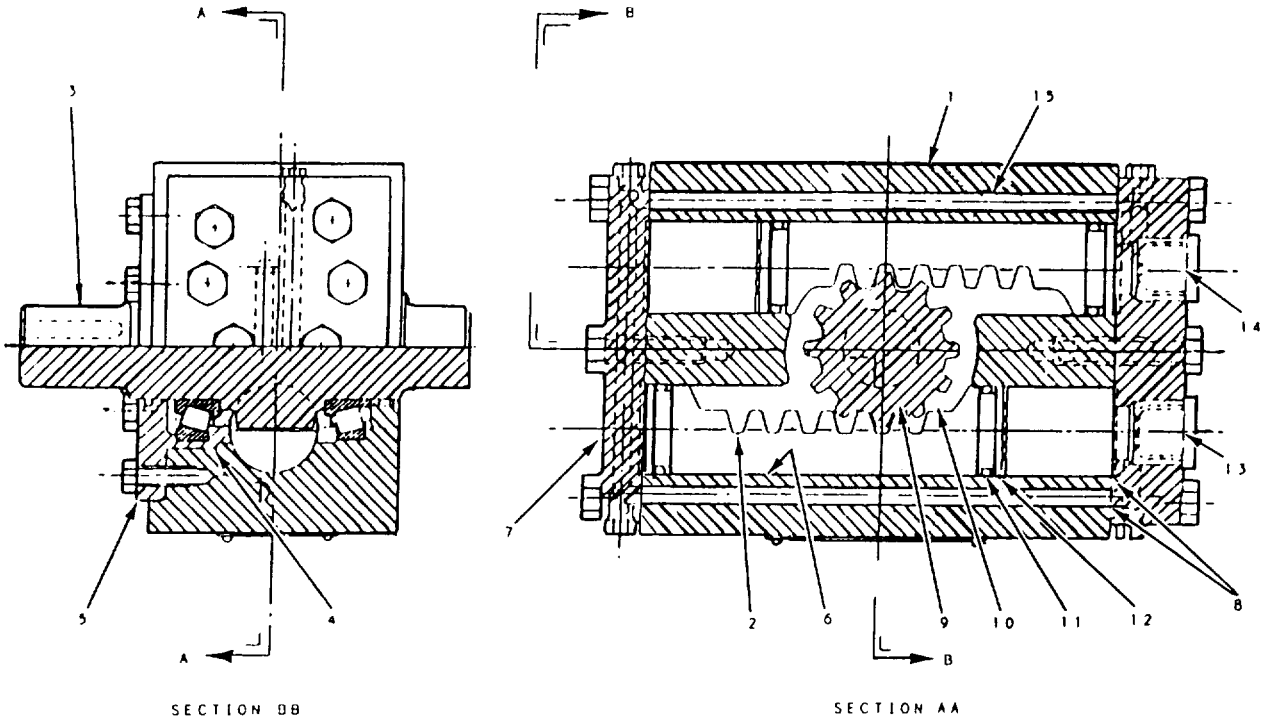
RACK-AND-PINION PISTON-TYPE ROTARY ACTUATORS

The rack-and-pinion-type actuators, also referred to as limited rotation cylinders, of the single or multiple, bidirectional piston are used for turning, positioning, steering, opening and closing, swinging, or any other mechanical function involving restricted rotation. Figure 10-10 shows a typical rack-and-pinion double-piston actuator.

The actuator consists of a body and two reciprocating pistons with an integral rack for rotating the shaft mounted in roller or journal bearings. The shaft and bearings are located in a central position and are enclosed with a bearing cap. The pistons, one on each side of the rack, are enclosed in cylinders machined or sleeved into the body. The body is enclosed with end caps and static seals to prevent external leakage of pressurized fluid.

Only a few of the many applications of actuating cylinders were discussed in the preceding paragraphs. Figure 10-11 shows additional types of force and motion applications.

In addition to its versatility, the cylinder-type actuator is probably the most trouble-free component of fluid power systems. However, it is very important that the cylinder, mechanical linkage, and actuating unit are correctly aligned. Any misalignment will cause excessive wear of the piston, piston rod, and seals. Also, proper adjustment between the piston rod and the actuating unit must be maintained.



1. Body
2. Pistons and Racks
3. Shaft (Splined typical)
4. Bearing
5. Bearing Cap
6. Cylinder
7. End Cap
8. Seals Static
9. Pinion Gear
10. Gear Case Chamber
11. Seals Dynamic
12. Backup Rings
13. Port Inlet
14. Port Outlet
15. Conduit Internal

Figure 10-10.—Rack-and-pinion double-piston rotary actuator.

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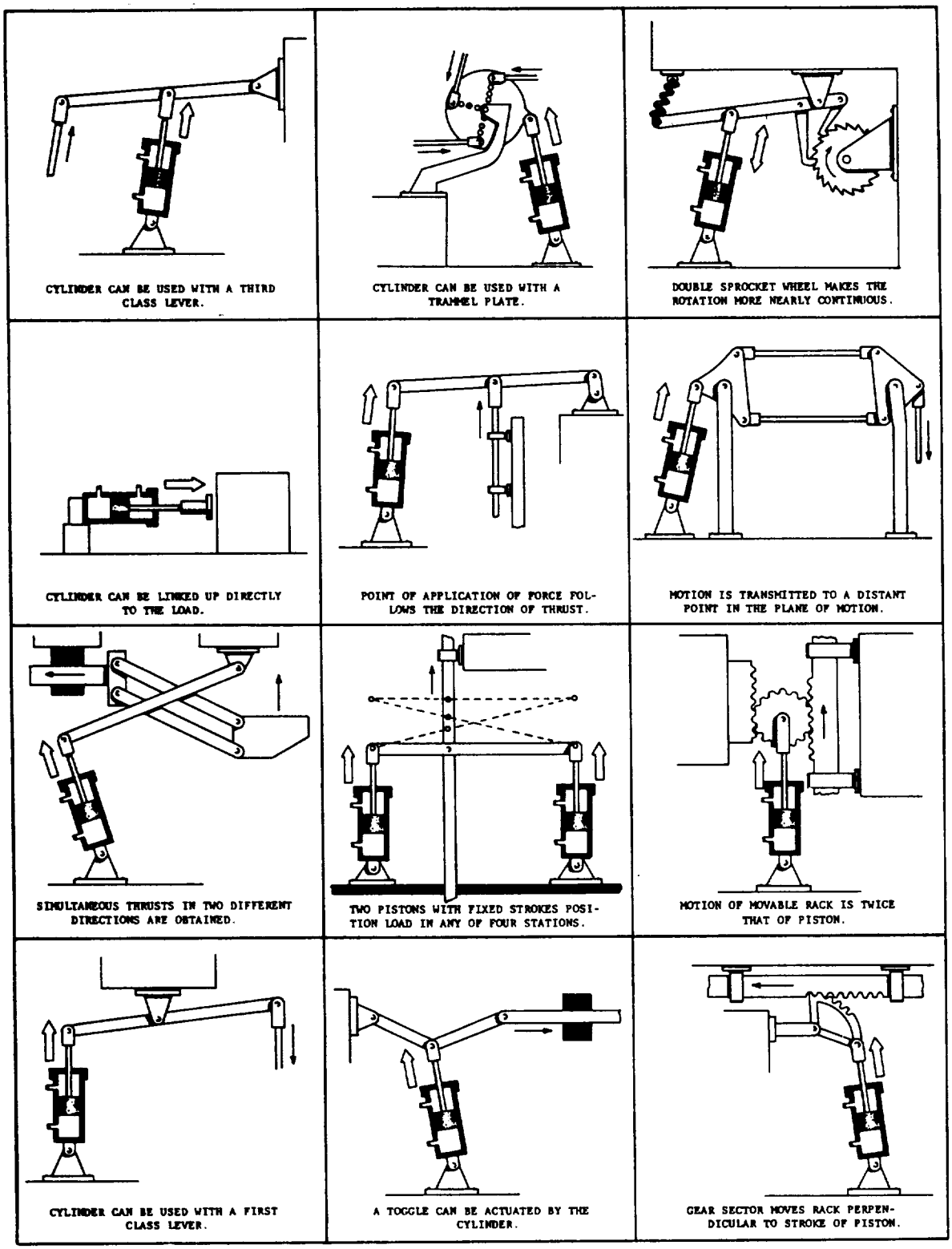

Figure 10-11.—Applications of actuating cylinders.

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MOTORS

A fluid power motor is a device that converts fluid power energy to rotary motion and force. The function of a motor is opposite that of a pump. However, the design and operation of fluid power motors are very similar to pumps. Therefore, a thorough knowledge of the pumps described in chapter 4 will help you understand the operation of fluid power motors.

Motors have many uses in fluid power systems. In hydraulic power drives, pumps and motors are combined with suitable lines and valves to form hydraulic transmissions. The pump, commonly referred to as the A-end, is driven by some outside source, such as an electric motor. The pump delivers fluid to the motor. The motor, referred to as the B-end, is actuated by this flow, and through mechanical linkage conveys rotary motion and force to the work. This type of power drive is used to operate (train and elevate) many of the Navy's guns and rocket launchers. Hydraulic motors are commonly used to operate the wing flaps, radomes, and radar equipment in aircraft. Air motors are used to drive pneumatic tools. Air motors are also used in missiles to convert the kinetic energy of compressed gas into electrical power, or to drive the pump of a hydraulic system.

Fluid motors may be either fixed or variable displacement. Fixed-displacement motors provide constant torque and variable speed. The speed is varied by controlling the amount of input flow. Variable-displacement motors are constructed so that the working relationship of the internal parts can be varied to change displacement. The majority of the motors used in fluid power systems are the fixed-displacement type.

Although most fluid power motors are capable of providing rotary motion in either direction, some applications require rotation in only one direction. In these applications, one port of the motor is connnected to the system pressure line and the other port to the return line or exhausted to the atmosphere. The flow of fluid to the motor is controlled by a flow control valve, a two-way directional control valve, or by starting and stopping the power supply. The speed of the motor may be controlled by varying the rate of fluid flow to it.

In most fluid power systems, the motor is required to provide actuation power in either direction. In these applications the ports are referred to as working ports, alternating as inlet and outlet ports. The flow to the motor is usually controlled by either a four-way directional control valve or a variable-displacement pump.

Fluid motors are usually classified according to the type of internal element, which is directly actuated by the flow. The most common types of elements are the gear, the vane, and the piston, AU three of these types are adaptable for hydraulic systems, while only the vane type is used in pneumatic systems.

GEAR-TYPE MOTORS

The spur, helical, and herringbone design gears are used in gear-type motors. The motors use external-type gears, as discussed in chapter 4.

The operation of a gear-type motor is shown in figure 10-12. Both gears are driven gears; however, only one is connected to the output shaft. As fluid under pressure enters chamber A, it takes the path of least resistance and flows around the inside surface of the housing, forcing the gears to rotate as indicated. The flow continues through the outlet port to the return. This rotary motion of the gears is transmitted through the attached shaft to the work unit.

The motor shown in figure 10-12 is operating in one direction; however, the gear-type motor is capable of providing rotary motion in either direction. To reverse the direction of rotation, the ports may be alternated as inlet and outlet. When fluid is directed through the outlet port (fig. 10-12) into chamber B, the gears rotate in the opposite direction.

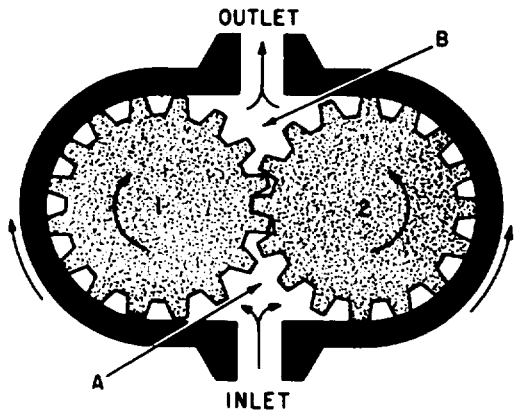

Figure 10-12.—Gear-type motor.

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VANE-TYPE MOTORS

A typical vane-type air motor is shown in figure 10-13. This particular motor provides rotation in only one direction. The rotating element is a slotted rotor which is mounted on a drive shaft. Each slot of the rotor is fitted with a freely sliding rectangular vane. The rotor and vanes are enclosed in the housing, the inner surface of which is offset from the drive shaft axis. When the rotor is in motion, the vanes tend to slide outward due to centrifugal force. The distance the vanes slide is limited by the shape of the rotor housing.

This motor operates on the principle of differential areas. When compressed air is directed into the inlet port, its pressure is exerted equally in all directions. Since area A (fig. 10-13) is greater than area B, the rotor will turn counterclockwise. Each vane, in turn, assumes the No. 1 and No. 2 positions and the rotor turns continuously. The potential energy of the compressed air is thus converted into kinetic energy in the form of rotary motion and force. The air at reduced pressure is exhausted to the atmosphere. The shaft of the motor is connected to the unit to be actuated.

Many vane-type motors are capable of providing rotation in either direction. A motor of this design is shown in figure 10-14. This motor operates on the same principle as the vane motor shown in figure 10-13. The two ports may be alternately used as inlet and outlet, thus providing rotation in either direction. Note the springs in the slots of the rotor. Their purpose is to hold the vanes against the housing during the initial starting of the motor, since centrifugal force does not exist until the rotor begins to rotate.

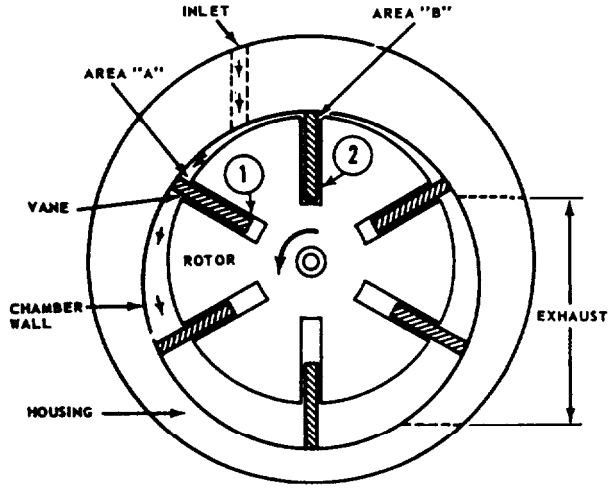

Figure 10-13.—Vane-type air motor.

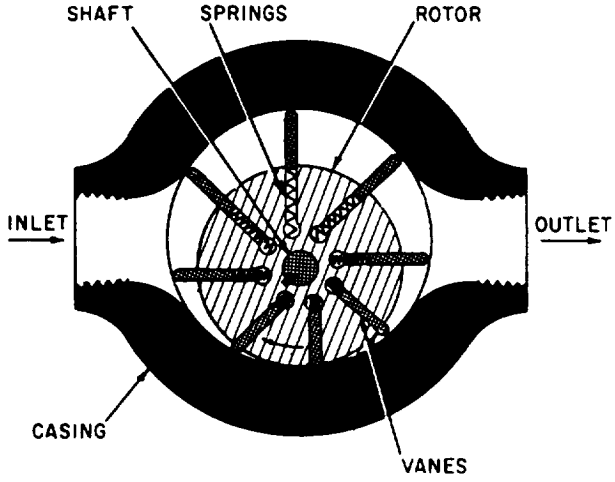

Figure 10-14.—Vane-type motor.

PISTON-TYPE MOTORS

Piston-type motors are the most commonly used in hydraulic systems. They are basically the same as hydraulic pumps except they are used to convert hydraulic energy into mechanical (rotary) energy.

The most commonly used hydraulic motor is the fixed-displacement piston type. Some equipment uses a variable-displacement piston motor where very wide speed ranges are desired.

Although some piston-type motors are controlled by directional control valves, they are often used in combination with variable-displacement pumps. This pump-motor combination is used to provide a transfer of power between a driving element and a driven element. Some applications for which hydraulic transmissions may be used are speed reducers, variable speed drives, constant speed or constant torque drives, and torque converters. Some advantages of hydraulic transmission of power over mechanical transmission of power are as follows:

1. Quick, easy speed adjustment over a wide range while the power source is operating at a constant (most efficient) speed. Rapid, smooth acceleration or deceleration.
2. Control over maximum torque and power.
3. Cushioning effect to reduce shock loads.
4. Smoother reversal of motion.

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Radial-Piston Motor

The radial-piston motor operates in reverse of the radial-piston pump. In the radial-piston pump, as the cylinder block rotates, the pistons press against the rotor and are forced in and out of the cylinders, thereby receiving fluid and pushing it out into the system. In the radial motor, fluid is forced into the cylinders and drives the pistons outward. The pistons pushing against the rotor cause the cylinder block to rotate.

The operation of a radial-piston motor is shown in figure 10-15. This motor is shown with three pistons for simplicity. Normally it contains seven or nine pistons. When liquid is forced into the cylinder bore containing piston 1, the piston moves outward since the liquid cannot be compressed. This causes the cylinder to rotate in a clockwise direction. As the force acting on piston 1 causes the cylinder block to rotate, piston 2 starts to rotate and approach the position of piston 3. (Note that the distance between the cylinder block and the reaction ring of the rotor gets progressively shorter on the top and right half of the rotor.)

As piston 2 rotates, it is forced inward and, in turn, forces the fluid out of the cylinder. Since there is little or no pressure on this side of the pintle valve, the piston is easily moved in by its contact with the reaction ring of the rotor. The fluid is easily forced out of the cylinder and back to the reservoir or to the inlet side of the pump. As the piston moves past the midpoint, or past the shortest distance between the cylinder block and the rotor, it enters the pressure side of the pintle valve and fluid is forced into the cylinder. Piston 3 then becomes the pushing piston and in turn rotates the cylinder block. This action continues as long as fluid under pressure enters the cylinders.

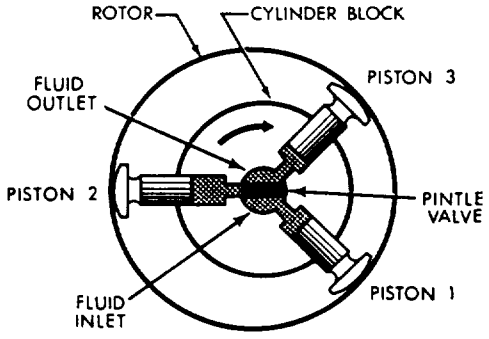

Figure 10-15.—Operation of a radial-piston motor.

The direction of rotation of the motor (fig. 10-15) is changed by reversing the flow of fluid to it. Admitting fluid under pressure on the top side of the pintle valve forces piston 3 out of the cylinder block. This causes the cylinder to rotate in the counterclockwise direction.

Axial-Piston Motor

The variable-stroke axial-piston pump is often used as a part of variable speed gear, such as electrohydraulic anchor windlasses, cranes, winches, and the power transmitting unit in electrohydraulic steering engines. In those cases, the tilting box is arranged so that it maybe tilted in either direction. Thus it maybe used to transmit bidirectional power hydraulically to pistons or rams, or it may be used to drive a hydraulic motor. In the latter use, the pump is the A-end of the variable speed gear and the hydraulic motor is the B-end.

The B-end of the hydraulic unit of the hydraulic speed gear is exactly the same as the A-end of the variable-stroke pump mentioned previously. However, it generally does not have a variable-stroke feature. The tilting box is installed at a permanently fixed angle. Thus, the B-end becomes a fixed-stroke axial-piston motor. Figure 10-16 illustrates an axial-piston hydraulic speed gear with the A-end and B-end as a single unit. It is used in turrets for train and elevation driving units. For electrohydraulic winches and cranes, the A-end and B-end are in separate housings connected by hydraulic piping.

Hydraulic fluid introduced under pressure to a cylinder (B-end) tries to push the piston out of the cylinder. In being pushed out, the piston, through its piston rod, will seek the point of greatest distance between the top of the cylinder and the socket ring. The resultant pressure of the piston against the socket ring will cause the cylinder barrel and the socket ring to rotate. This action occurs during the half revolution while the piston is passing the intake port of the motor, which is connected to the pressure port of the pump. After the piston of the motor has taken all the hydraulic fluid it can from the pump, the piston passes the valve plate land and starts to discharge oil through the outlet ports of the motor

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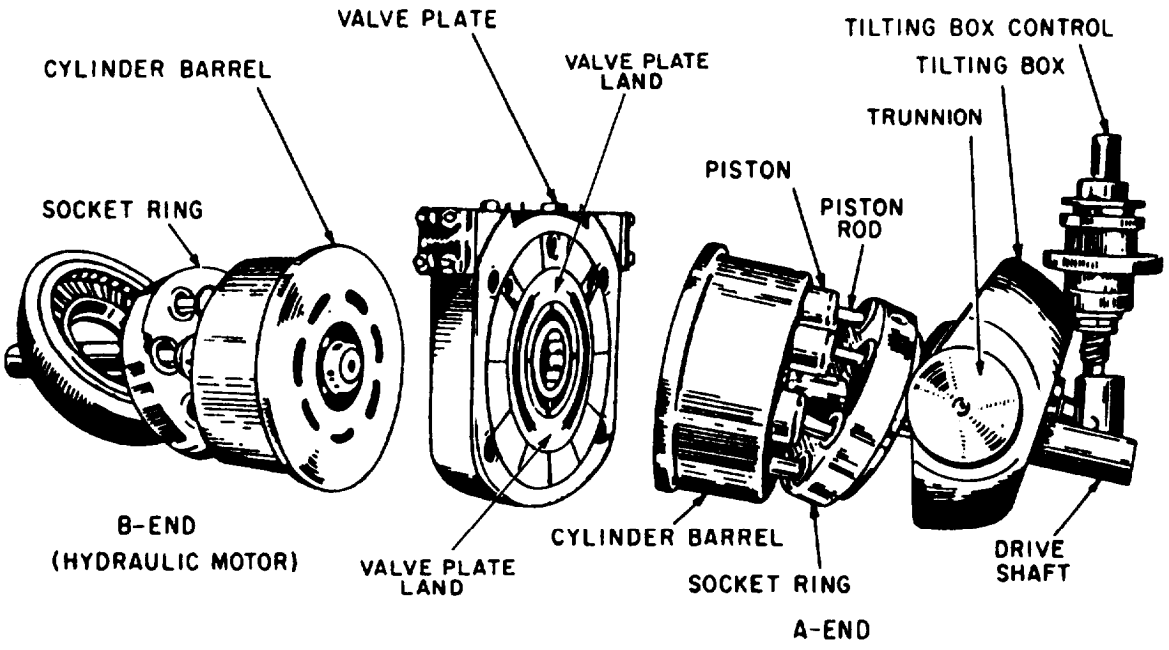

Figure 10-16.—Exploded view of a axial-piston hydraulic speed gear.

to the suction pistons of the pump. The pump is constantly putting pressure on one side of the motor and receiving hydraulic fluid from the other side. The fluid is merely circulated from pump to motor and back again.

Both of the axial-piston motors described in this section may be operated in either direction. The direction of rotation is controlled by the direction of fluid flow to the valve plate. The direction of flow may be instantly reversed without damage to the motor.

TURBINES

Turbines are used in pneumatic systems to convert kinetic energy of gases to mechanical energy. Turbines are used to drive electric generators, to convert mechanical energy into electrical energy, and to drive pumps to supply fluid flow in hydraulic systems.

The basic parts of a turbine are the rotor, which has blades projecting radially from its periphery; and nozzles, through which the gas is expanded and directed. The conversion of kinetic energy to mechanical energy occurs on the blades.

The basic distinction between types of turbines is the manner in which the gas causes the turbine rotor to move. When the rotor is moved by a direct push or “impulse” from the gas impinging upon the blades, the turbine is said to be an impulse turbine. When the rotor is moved by force of reaction, the turbine is said to be a reaction turbine.

Although the distinction between impulse turbines and reaction turbines is a useful one, it should not be considered as an absolute distinction in real turbines. An impulse turbine uses both the impulse of the gas jet and, to a lesser extent, the reactive force that results when the curved blades cause the gas to change direction. A reaction turbine is moved primarily by reactive force, but some motion of the rotor is caused by the impact of the gas against the blades.

IMPULSE TURBINE

The impulse turbine consists essentially of a rotor mounted on a shaft that is free to rotate in a set of bearings. The outer rim of the rotor carries a set of curved blades, and the whole assembly is enclosed in an airtight case. Nozzles direct the

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rapidly moving fluid against the blades and turn the rotor (fig. 10-17).

REACTION TURBINE

The reaction turbine, as the name implies, is turned by reactive force rather than by a direct push or impulse. In reaction turbines, there are no nozzles as such. Instead, the blades that project radially from the periphery of the rotor are formed and mounted so that the spaces between the blades have, in cross section, the shape of nozzles. Since these blades are mounted on the revolving rotor, they are called moving blades.

Fixed or stationary blades of the same shape as the moving blades (fig. 10-18) are fastened to the stator (casing) in which the rotor revolves. The fixed blades guide the gas into the moving blade system and, since they are also shaped and mounted to provide nozzle-shaped spaces between the blades, the freed blades also act as nozzles.

A reaction turbine is moved by three main forces: (1) the reactive force produced on the moving blades as the gas increases in velocity as it expands through the nozzle-shaped spaces between the blades; (2) the reactive force produced on the moving blades when the gas changes direction; and (3) the push or impulse of the gas impinging upon the blades. Thus, as previously noted, a reaction turbine is moved primarily by reactive force but also to some extent by direct impulse.

Impulse and reaction blades can be combined to form an impulse-reaction turbine. This turbine combines the rotational forces of the previously described turbines; that is, it derives its rotation from both the impulse of the gas striking the turbine blades and the reactive force of the gas changing direction.

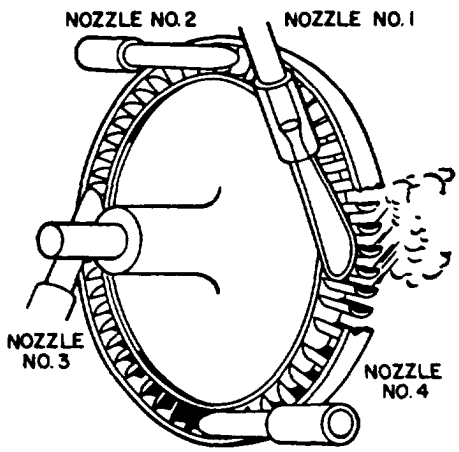

Figure 10-17.—Impulse turbine.

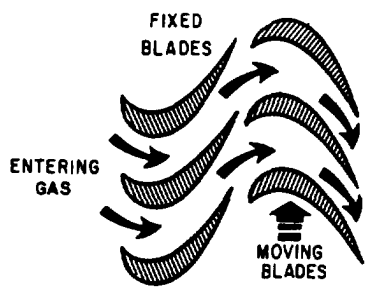

Figure 10-18.—Reaction turbine blading.

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CHAPTER 11

PNEUMATICS

The word *pneumatics* is a derivative of the Greek word *pneuma,* which means air, wind, or breath. It can be defined as that branch of engineering science that pertains to gaseous pressure and flow. As used in this manual, pneumatics is the portion of fluid power in which compressed air, or other gas, is used to transmit and control power to actuating mechanisms.

This chapter discusses the origin of pneumatics. It discusses the characteristics of gases and compares them with those of liquids. It also explains factors which affect the properties of gases, identifies and explains the gas laws, and identifies gases commonly used in pneumatics and their pressure ranges. It also discusses hazards of pneumatic gases, methods of controlling contamination, and safety precautions associated with compressed gases.

DEVELOPMENT OF PNEUMATICS

There is no record of man's first uses of air to do work. Probably the earliest uses were to separate chaff from grain and to move ships. One of the first pneumatic devices was the blow gun used by primitive man. In the latter part of the eighteenth century, heated air was used to carry the first balloon aloft. The heated air, being lighter than the surrounding air, caused the balloon to rise.

Every age of man has witnessed the development of devices which used air to do work. However, man used air to do work long before he understood it.

Many of the principles of hydraulics apply to pneumatics. For example, Pascal's law applies to gases as well as liquids. Also, like hydraulics, the development of pneumatics depended on closely fitted parts and the development of gaskets and packings. Since the invention of the air compressor, pneumatics has become a very reliable way to transmit power.

Probably one of the most common uses of pneumatic power is in the operation of pneumatic tools. However, you should understand that pneumatics is also of great importance in large and complex systems such as the controls of vital propulsion and weapon systems.

CHARACTERISTICS OF GASES

Recall from chapter 1 that gas is one of the three states of matter. It has characteristics similar to those of liquids in that it has no definite shape but conforms to the shape of its container and readily transmits pressure.

Gases differ from liquids in that they have no definite volume. That is, regardless of the size or shape of the containing vessel, a gas will completely fill it. Gases are highly compressible, while liquids are only slightly so. Also, gases are lighter than equal volumes of liquids, making gases less dense than liquids.

DENSITY

Early experiments were conducted concerning the behavior of air and similar gases. These experiments were conducted by scientists such as Boyle and Charles (discussed later in this chapter). The results of their experiments indicated that the gases' behavior follows the law known as the *ideal-gas law.* It states as follows: *For a given weight of any gas, the product of the absolute pressure and the volume occupied, divided by the absolute temperature, is constant.* In equation form, it is expressed as follows:

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2} = \frac{pV}{T}. \qquad \text{Equation 11-1}$$

For 1 pound of gas,

$$\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2} = \frac{pv}{T} = R \text{ (a constant)}, \qquad \text{Equation 11-2}$$

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also,

$$pv = RT, \qquad \text{Equation 11-}$$

where:

p = absolute pressure

V = total volume

v = specific volume

T = absolute temperature

R = gas constant.

The specific volume (v) is expressed in cubic feet per pound.

For any weight of a gas this equation maybe modified as follows:

W = weight of the gas in pounds,

V = volume of W pounds of the gas in cubic feet.

The volume of 1 pound would then be V/W. If we substitute this for v in equation 11-3, it then becomes

$$p\frac{V}{W} = RT \text{ or } pV = WRT. \qquad \text{Equation 11-4}$$

Solving equation 11-4 for pressure,

$$p = \frac{WRT}{V}. \qquad \text{Equation 11-5}$$

In chapter 2 we defined density as the mass per unit volume. In equation 11-5,

$$\frac{W}{V}$$

represents density. (Notice that this is the reverse of the specific volume.) We can now say that pressure is equal to the density of the gas times the gas constant times the absolute temperature of the gas. (The gas constant varies for different gases.) From this equation we can show how density varies with changes in pressure and temperature. Decreasing the volume, with the weight of the gas and the temperature held constant, causes the pressure to increase.

NOTE: During the compression of the gas, the temperature will actually increase; however, the explanation is beyond the scope of this text.

Also, since density equals

$$\frac{W}{V},$$

a decrease in volume with the weight held constant will cause density to increase.

TEMPERATURE

As indicated previously, temperature is a dominant factor affecting the physical properties of gases. It is of particular concern in calculating changes in the states of gases.

Three temperature scales are used extensively in gas calculations. They are the Celsius (C), the Fahrenheit (F), and the Kelvin (K) scales. The Celsius (or centigrade) scale is constructed by identifying the freezing and boiling points of water, under standard conditions, as fixed points of 0° and 100°, respectively, with 100 equal divisions between. The Fahrenheit scale identifies 32° as the freezing point of water and 212° as the boiling point, and has 180 equal divisions between. The Kelvin scale has its zero point equal to –273°C, or –460°F.

Absolute zero, one of the fundamental constants of physics, is commonly used in the study of gases. It is usually expressed in terms of the Celsius scale. If the heat energy of a gas sample could be progressively reduced, some temperature should be reached at which the motion of the molecules would cease entirely. If accurately determined, this temperature could then be taken as a natural reference, or as a true absolute zero value.

Experiments with hydrogen indicated that if a gas were cooled to –273.16°C (–273° for most calculations), all molecular motion would cease and no additional heat could be extracted. Since this is the coldest temperature to which an ideal gas can be cooled, it is considered to be absolute zero. Absolute zero may be expressed as 0°K, –273°C, or –459.69°F (–460°F for most calculations).

When you work with temperatures, always be sure which system of measurement is being used and how to convert from one to another. The conversion formulas are shown in figure 11-1. For purposes of calculations, the Rankine (R) scale illustrated in figure 11-1 is commonly used to

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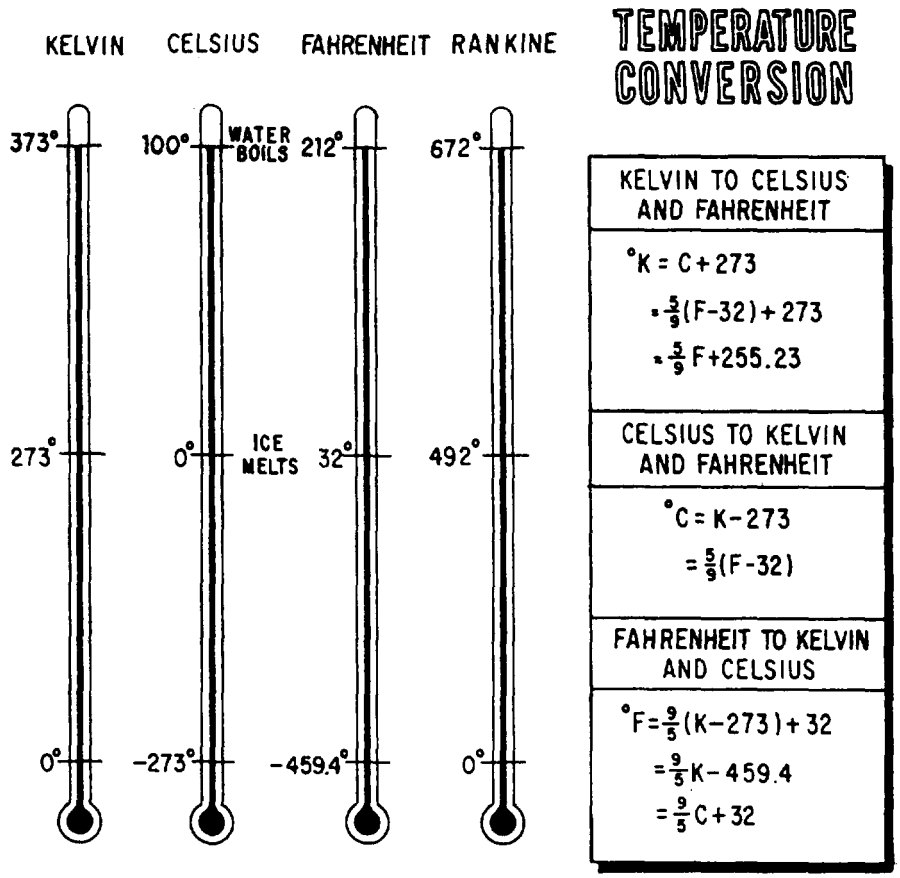

Figure 11-1.-Comparison of Kelvin, Celsius, Fahrenheit, and Rankine temperature.

convert Fahrenheit to absolute. For Fahrenheit readings above zero, 460° is added. Thus, 72°F equals 460° plus 72°, or 532° absolute (532°R). If the Fahrenheit reading is below zero, it is subtracted from 460°. Thus, -40°F equals 460° minus 40°, or 420° absolute (420°R).

The Kelvin and Celsius scales are used internationally in scientific measurements; therefore, some technical manuals may use these scales in directions and operating instructions. The Fahrenheit scale is commonly used in the United States; therefore, it is used in most areas of this manual.

PRESSURE

We defined pressure in chapter 2 as force per unit area. Remember, liquids exert pressure on all surfaces with which they come in contact. Gases, because of their ability to completely fill containers, exert pressure on all sides of a container.

In practice, we maybe interested in either of two pressure readings. We may desire either the gauge pressure or the absolute pressure.

Absolute pressure is measured from absolute zero pressure rather than from normal or atmospheric pressure (approximately 14.7 psi). Gauge pressure is used on all ordinary gauges, and indicates pressure in excess of atmospheric pressure. Therefore, absolute pressure is equal to atmospheric pressure plus gauge pressure. For example, 100 psi gauge pressure (psig) equals 100 psi plus 14.7 psi or 114.7 psi absolute pressure (psia). Whenever gas laws are applied, absolute pressures are required.

COMPRESSIBILITY AND EXPANSION OF GASES

Gases can be readily compressed and are assumed to be perfectly elastic. This combination of properties gives a gas the ability to yield to a

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force and return promptly to its original condition when the force is removed. These are the properties of air that is used in pneumatic tires, tennis balls and other deformable objects whose shapes are maintained by compressed air.

KINETIC THEORY OF GASES

In an attempt to explain the compressibility of gases, Bernoulli proposed the hypothesis that is accepted as the *kinetic theory of gases*. According to this theory, the pressure exerted by a gas on the walls of a closed container is caused by continual bombardment of the walls by molecules of the gas.

Consider the container shown in figure 11-2 as containing a gas. At any given time, some molecules are moving in one direction, some are traveling in other directions; some are traveling fast, some slow, and some may even be in a state of rest. The average effect of the molecules bombarding each container wall corresponds to the pressure of the gas.

As more gas is pumped into the container, more molecules are available to bombard the walls; thus the pressure in the container increases.

The gas pressure in a container can also be increased by increasing the speed with which the molecules hit the walls. If the temperature of the gas is raised, the molecules move faster causing an increase in pressure. This can be shown by considering the automobile tire. When you take a long drive on a hot day, the pressure in the tires increases and a tire which appeared to be somewhat "soft" in cool morning temperature may appear normal at a higher midday temperature.

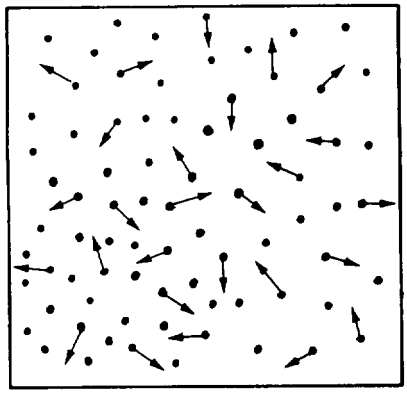

Figure 11-2.—Molecular bombardment creating pressure.

BOYLE'S LAW

When the automobile tire is initially inflated, air which normally occupies a specific volume is compressed into a smaller volume inside the tire. This increases the pressure on the inside of the tire.

Charles Boyle, an English scientist, was among the first to experiment with the pressure-volume relationship of gas. During an experiment when he compressed a volume of air he found that the volume decreased as the pressure increased, and by doubling the force exerted on the air he could decrease the volume of the air by half. See figure 11-3. Recall from the example of the automobile tire that changes in temperature of a gas also change the pressure and volume. Therefore, the experiment must be performed at a constant temperature. The relationship between pressure and volume is known as *Boyle's law*. It states: *When the temperature of a gas is kept constant, the volume of an enclosed gas varies inversely with its pressure.*

In equation form, this relationship may be expressed as either

$$V_1P_1 = V_2P_2$$

or Equation 11-6

$$\frac{V_1}{V_2} = \frac{P_2}{P_1},$$

where V_1 and P_1 are the original volume and pressure, and V_2 and P_2 are the final volume and pressure (P_1 and P_2 are absolute pressures).

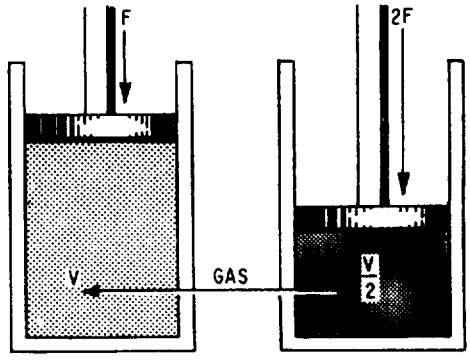

Figure 11-3.-Gas compressed to half its original volume by a doubled force.

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Example of Boyle's law: 4 cubic feet of nitrogen are under a pressure of 100 psi (gauge). The nitrogen is allowed to expand to a volume of 6 cubic feet. What is the new gauge pressure? Remember to convert gauge pressure to absolute pressure by adding 14.7.

Using equation 11-6, $V_1P_1 = V_2P_2$, where V_1 is 4 ft^3, V_2 is 6 ft, and P_1 is 100 psig:

Solution:

$$V_1P_1 = V_2P_2$$

$$P_2 = \frac{V_1P_1}{V_2}$$

Substituting:

$$P_2 = \frac{4(100 + 14.7)}{6}$$

$$= 76.47 \text{ psia}$$

Converting absolute pressure to gauge pressure:

$$\begin{array}{r} 76.47 \\ -14.7 \\ \hline 61.77 \text{ psi gauge pressure} \end{array}$$

CHARLES'S LAW

Boyle's law assumes conditions of constant temperature. In actual situations this is rarely the case. Temperature changes continually and affects the volume of a given mass of gas.

Jacques Charles, a French physicist, provided much of the foundation for the modern kinetic theory of gases. Through experiments, he found that all gases expand and contract proportionally to the change in the absolute temperature, providing the pressure remains constant. The relationship between volume and temperature is known as *Charles's law*. It states: *The volume of a gas is proportional to its absolute temperature, if constant pressure is maintained.* In equation form, this relationship may be expressed as

Equation 11-7

or

$$\frac{V_1}{V_2} = \frac{T_1}{T_2},$$

where V_1 and V_2 are the original and final volumes, and T_1 and T_2 are the original and final absolute temperatures.

Since an increase in the temperature of a gas causes it to expand if the pressure is kept constant, it is reasonable to expect that if a given sample is heated within a closed container and its volume remains constant, the pressure of the gas will increase. Experiments have proven this to be true. In equation form, this becomes

$P_1T_2 = P_2T_1$ Equation 11-8

or

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}.$$

This equation states that for a constant volume, the absolute pressure of a gas varies directly with the absolute temperature.

Example: A cylinder of gas under a pressure of 1800 psig at 70°F is left out in the sun in the tropics and heats up to a temperature of 130°F. What is the new pressure within the cylinder? (Remember that both pressure and temperature must be converted to absolute pressure and absolute temperature.)

Using equation 11-8, $P_1T_2 = P_2T_1$, when P_1 is 1800 pisg, T_1 is 70°F, and T_2 = 130°F:

Solution:

$$P_1T_2 = P_2T_1; \quad P_2 = \frac{P_1T_2}{T_1}$$

Substituting:

$$P_2 = \frac{(1800 + 14.7) \times (130° + 460°)}{70° + 460°}$$

$$= \frac{1814.7 \times 590}{530}$$

$$P_2 = 2020 \text{ psia}$$

Converting absolute pressure to gauge pressure:

$$\begin{array}{r} 2020.0 \\ -14.7 \\ \hline 2005.3 \text{ psig} \end{array}$$

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GENERAL GAS LAW

We have learned that Boyle's law pertains to situations in which the temperature remains constant (fig. 11-4), and that Charles's law pertains to situations in which pressure remains constant (fig. 11-4). It is usually not possible to control pressure or temperature in tanks or bottles of gas subject to the weather and shipboard demands. Boyle's and Charles's laws are combined to form the *general gas law*. This law states: *The product of the initial pressure, initial volume, and new temperature (absolute scale) of an enclosed gas is equal to the product of the new pressure, new volume, and initial temperature.* It is a mathematical statement which allows many gas problems to be solved by using the principles of Boyle's law and/or Charles's law. The equation is expressed as

$$P_1V_1T_2 = P_2V_2T_1 \qquad \text{Equation 11-9}$$

or

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}.$$

(P and T represent absolute pressure and absolute temperature, respectively.)

You can see by examining figure 11-4 that the three equations are special cases of the general equation. Thus, if the temperature remains constant, T_1 equals T_2 and both can be eliminated from the general formula, which then reduces to the form shown in part A. When the volume remains constant, V_1 equals V_2, thereby reducing the general equation to the form given in part B. Similarly, P_1 is equated to P_2 for constant pressure, and the equation then takes the form given in part C.

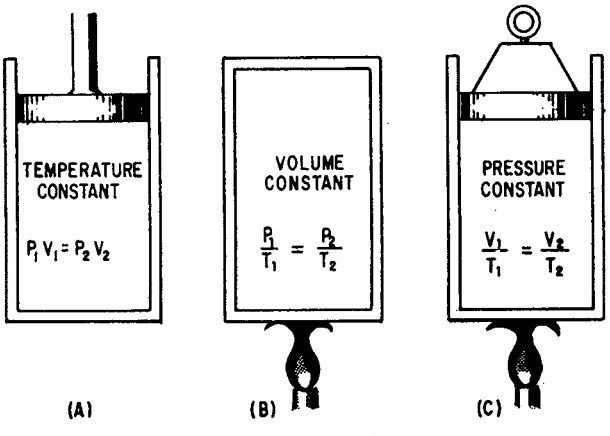

Figure 11-4.—The general gas law.

The general gas law applies with exactness only to "ideal" gases in which the molecules are assumed to be perfectly elastic. However, it describes the behavior of actual gases with sufficient accuracy for most practical purposes.

Two examples of the general equation follow:

1. Two cubic feet of a gas at 75 psig and 80°F are compressed to a volume of 1 cubic foot and then heated to a temperature of 300°F. What is the new gauge pressure?

Using equation 11-9, $P_1V_1T_2 = P_2V_2T_1$, where V_1 is 2 ft^3, P_1 is 75 psig, T_1 is 80°F, V_2 is 1 ft^3 and T_2 is 300°F:

Solution:

$$P_1V_1T_2 = P_2V_2T_1$$

$$P_2 = \frac{P_1V_1T_2}{V_2T_1}$$

Substituting:

$$P_2 = \frac{(75 + 14.7)\ (2)\ (300 + 460)}{(1)\ (80 + 460)}$$

$$= \frac{(89.7)\ (2)\ (760)}{(1)\ (540)}$$

$$= \frac{(179.4)\ (760)}{(540)}$$

$$P_2 = 252.5 \text{ psia}$$

Converting absolute pressure to gauge pressure:

$$\begin{array}{r} 252.5 \\ -14.7 \\ \hline 237.8 \text{ psig} \end{array}$$

2. Four cubic feet of a gas at 75 psig and 80°F are compressed to 237.8 psig and heated to a temperature of 300°F. What is the volume of the gas resulting from these changes? Using equation 11-9, $P_1V_1T_2 = P_2V_2T_1$, where V_1 is 4 ft^3, P_2 is

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75 psig, T_1 is 800, P_1 is 237.8 psig, and T_2 is 300°F:

Solution:

$$P_1V_1T_2 = P_2V_2T_1$$

$$V_2 = \frac{P_1V_1T_2}{P_2T_1}$$

Substituting:

$$V_2 = \frac{(75 + 14.7)\ (4)\ (300 + 460)}{(237.8 + 14.7)\ (80 + 460)}$$

$$= \frac{(89.7)\ (4)\ (760)}{(252.5)\ (540)}$$

$$V_2 = 2 \text{ cubic feet}$$

PNEUMATIC GASES

In chapter 1, you learned that many factors are considered in determining whether to use hydraulics or pneumatics as a power source in a fluid power system. Once it is determined that pneumatics will be used as the source of power, some of the same factors are considered in selecting the pneumatic gas.

QUALITIES

The ideal fluid medium for a pneumatic system is a readily available gas that is nonpoisonous (nontoxic), chemically stable, free from any acids that cause corrosion of system components, and nonflammable. It also will not support combustion of other elements.

Gases that have these desired qualities may not have the required lubricating power. Therefore, lubrication of the components of some pneumatic systems must be arranged by other means. For example, some air compressors are provided with a lubricating system, some components are lubricated upon installation or, in some cases, lubrication is introduced into the air supply line.

Two gases meeting these qualities and most commonly used in pneumatic systems are compressed air and nitrogen.

COMPRESSED AIR

Compressed air is a mixture of all gases contained in the atmosphere. In this manual, compressed air is referred to as a gas when it is used as a fluid medium.

The unlimited supply of air and the ease of compression make compressed air the most widely used fluid for pneumatic systems. Although moisture and solid particles must be removed from the air, it does not require the extensive distillation or separation process required in the production of other gases.

Compressed air has most of the desired properties and characteristics of a gas for pneumatic systems. It is nonpoisonous and nonflammable but does contain oxygen, which supports combustion. One of the most undesirable qualities of compressed air as a fluid medium for pneumatic systems is moisture content. The atmosphere contains varying amounts of moisture in vapor form. Changes in the temperature of compressed air will cause condensation of moisture in the pneumatic system. This condensed moisture can be very harmful to the system, as it increases corrosion, dilutes lubricants, and may freeze in lines and components during cold weather. Moisture separators and air driers (dehydrators) are installed in the compressed air lines to minimize or eliminate moisture in systems where moisture would deteriorate system performance.

The supply of compressed air at the required volume and pressure is provided by an air compressor. (For information on air compressors, refer to *Naval Ships' Technical Manual,* chapter 551.) In most systems the compressor is part of the system with distribution lines leading from the compressor to the devices to be operated. In these systems a receiver is installed in-line between the compressor and the device to be operated to help eliminate pulsations in the compressor discharge line, to act as a storage tank during intervals when the demand for air exceeds the compressor's capacity, and to enable the compressor to shut down during periods of light load. Other systems receive their supply from cylinders which must be filled at a centrally located air compressor and then connected to the system.

Compressed air systems are categorized by their operating pressures as follows: high-pressure (HP) air, medium-pressure (MP) air, and low-pressure (LP) air.

High-Pressure Air Systems

HP air systems provide compressed air at a nominal operating pressure of 3000 psi or 5000 psi and are installed whenever pressure in excess

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of 1000 psi is required. HP compressed air plants support functions which require high pressures and high flow rates of compressed air by the addition of HP storage flasks to the system. An example of such a system is one that provides air for starting diesel and gas turbine engines. Reduction in pressure, if required, is done by using specially designed pressure-reducing stations.

Medium-Pressure Air

MP air systems provide compressed air at a nominal operating pressure of 151 psi to 1000 psi. These pressures are provided either by an MP air compressor or by the HP air system supplying air through an air bank and pressure-reducing stations.

Low-Pressure Air

LP air systems provide compressed air at a nominal operating pressure of 150 psi and below. The LP air system is supplied with LP air by LP air compressors or by the HP air system supplying air through an air bank and pressure-reducing stations. LP air is the most extensive and varied air system used in the Navy,

In addition to being used for various pneumatic applications, LP and HP compressed air are used in the production of nitrogen.

NITROGEN

For all practical purposes, nitrogen is considered to be an inert gas. It is nonflammable, does not form explosive mixtures with air or oxygen, and does not cause rust or decay. Due to these qualities, its use is preferred over compressed air in many pneumatic systems, especially aircraft and missile systems, and wherever an inert gas blanket is required.

Nitrogen is obtained by the fractional distillation of air. Oxygen/nitrogen-producing plants expand compressed air until its temperature decreases to –196°C (–320°F), the boiling point of nitrogen at atmospheric pressure. The liquid nitrogen is then directed to a storage tank. A liquid nitrogen pump pumps the low-pressure liquid nitrogen from the storage tank and discharges it as a high-pressure (5000 psi) liquid to the vaporizer where it is converted to a gas at 5000 psi. Oxygen/nitrogen-producing plants are located at many naval installations and on submarine tenders and aircraft carriers.

CONTAMINATION CONTROL

As in hydraulic systems, fluid contamination is also a leading cause of malfunctions in pneumatic systems. In addition to the solid particles of foreign matter which find a way to enter the system, there is also the problem of moisture. Most systems are equipped with one or more devices to remove this contamination. These include filters, water separators, air dehydrators, and chemical driers, which are discussed in chapter 9 of this manual. In addition, most systems contain drain valves at critical low points in the system. These valves are opened periodically to allow the escaping gas to purge a large percentage of the contaminants, both solids and moisture, from the system. In some systems these valves are opened and closed automatically, while in others they must be operated manually.

Complete purging is done by removing lines from various components throughout the system and then attempting to pressurize the system, causing a high rate of airflow through the system. The airflow will cause the foreign matter to be dislodged and blown from the system.

NOTE: If an excessive amount of foreign matter, particularly oil, is blown from any one system, the lines and components should be removed and cleaned or replaced.

In addition to monitoring the devices installed to remove contamination, it is your responsibility as a maintenance person or supervisor to control the contamination. You can do this by using the following maintenance practices:

1. Keep all tools and the work area in a clean, dirt-free condition.
2. Cap or plug all lines and fittings immediately after disconnecting them.
3. Replace all packing and gaskets during assembly procedures.
4. Connect all parts with care to avoid stripping metal slivers from threaded areas. Install and torque all fittings and lines according to applicable technical instructions.
5. Complete preventive maintenance as specified by MRCs.

Also, you must take care to ensure that the proper cylinders are connected to systems being supplied from cylinders.

Cylinders for compressed air are painted black. Cylinders containing oil-pumped air have

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two green stripes painted around the top of the cylinder, while cylinders containing water-pumped air have one green stripe. Oil-pumped air indicates that the air or nitrogen is compressed by an oil-lubricated compressor. Air or nitrogen compressed by a water-lubricated (or nonlubricated) compressor is referred to as water pumped. Oil-pumped nitrogen can be very dangerous in certain situations. For example, nitrogen is commonly used to purge oxygen systems. Oxygen will not burn, but it supports and accelerates combustion and will cause oil to burn easily and with great intensity. Therefore, oil-pumped nitrogen must never be used to purge oxygen systems. When the small amount of oil remaining in the nitrogen comes in contact with the oxygen, an explosion may result. In all situations, use only the gas specified by the manufacturer or recommended by the Navy. Nitrogen cylinders are painted gray. One black stripe identifies cylinders for oil-pumped nitrogen, and two black stripes identify cylinders for water-pumped nitrogen. In addition to these color codes, the exact identification of the contents is printed in two locations diametrically opposite one another along the longitudinal axis of the cylinder. For compressed air and nitrogen cylinders, the lettering is white.

POTENTIAL HAZARDS

All compressed gases are hazardous. Compressed air and nitrogen are neither poisonous nor flammable, but should not be handled carelessly. Some pneumatic systems operate at pressures exceeding 3000 psi. Lines and fittings have exploded, injuring personnel and property. Literally thousands of careless workers have blown dust or harmful particles into their eyes by the careless handling of compressed air outlets.

Nitrogen gas will not support life, and when it is released in a confined space, it will cause asphyxia (the loss of consciousness as a result of too little oxygen and too much carbon dioxide in the blood). Although compressed air and nitrogen seem so safe in comparison with other gases, do not let overconfidence lead to personal injury.

SAFETY PRECAUTIONS

To minimize personal injury and equipment damage when using compressed gases, observe all practical operating safety precautions, including the following:

1. Do not use compressed air to clean parts of your body or clothing, or to perform general space cleanup in lieu of vacuuming or sweeping.
2. Never attempt to stop or repair a leak while the leaking portion is still under pressure. Always isolate, repressurize and danger tag out the portion of the system to be repaired. For pressures of 1000 psi or greater, double valve protection is required to prevent injury if one of the valves should fail.
3. Avoid the application of heat to the air piping system or components, and avoid striking a sharp or heavy blow on any pressurized part of the piping system.
4. Avoid rapid operation of manual valves. The heat of compression caused by a sudden high-pressure flow into an empty line or vessel can cause an explosion if oil is present. Valves should be slowly cracked open until airflow is noted and should be kept in this position until pressures on both sides of the valve have equalized. The rate of pressure rise should be kept under 200 psi per second, if possible. Valves may then be opened fully.
5. Do not discharge large quantities of nitrogen into closed compartments unless adequate ventilation is provided.
6. Do not subject compressed gas cylinders to temperatures greater than 130°F.

Remember, any pressurized system can be hazardous to your health if it is not maintained and operated carefully and safely.

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CHAPTER 12

BASIC DIAGRAMS AND SYSTEMS

In the preceding chapters, you learned about hydraulic and pneumatic fluids and components of fluid power systems. While having a knowledge of system components is essential, it is difficult to understand the interrelationship of these components by simply watching the system operate. The knowledge of system interrelation is required to effectively troubleshoot and maintain a fluid power system. Diagrams provided in applicable technical publications or drawings are a valuable aid in understanding the operation of the system and in diagnosing the causes of malfunctions.

This chapter explains the different types of diagrams used to illustrate fluid power circuits, including some of the symbols that depict fluid power components. Included in this chapter are descriptions and illustrations denoting the differences between open-center and closed-center fluid power systems. The last part of the chapter describes and illustrates some applications of basic fluid power systems.

DIAGRAMS

As mentioned earlier in this chapter, to troubleshoot fluid power systems intelligently, a mechanic or technician must be familiar with the system on which he or she is working. The mechanic must know the function of each component in the system and have a mental picture of its location in relation to other components. This can best be done by studying the diagrams of the system.

A diagram may be defined as a graphic representation of an assembly or system that indicates the various parts and expresses the methods or principles of operations. The ability to read diagrams is a basic requirement for understanding the operation of fluid power systems. Understanding the diagrams of a system requires having a knowledge of the symbols used in the schematic diagrams.

SYMBOLS

The Navy uses two military standards that list mechanical symbols that must be used in preparing drawings that will contain symbolic representation. These standards are as follows:

1. *Military Standard, Mechanical Symbols (Other than Aeronautical, Aerospacecraft, and Spacecraft Use), Part 1,* MIL-STD-17B-1.
2. *Military Standard, Mechanical Symbols for Aeronautical, Aerospacecraft, and Spacecraft Use, Part 2,* MIL-STD-17B-2.

Some of the symbols frequently used in fluid power systems have been selected from these two standards and are shown in Appendixes II and III. Appendix II contains symbols from MIL-STD-17B-1. Appendix III contains symbols from MIL-STD-17B-2.

While the symbols shown in the appendixes are not all encompassing, they do provide a basis for an individual working with fluid power systems to build upon. Some rules applicable to graphical symbols for fluid diagrams are as follows:

1. Symbols show connections, flow paths, and the function of the component represented only. They do not indicate conditions occurring during transition from one flow path to another; nor do they indicate component construction or values, such as pressure or flow rate.
2. Symbols do not indicate the location of ports, direction of shifting of spools, or position of control elements on actual components.
3. Symbols may be rotated or reversed without altering their meaning except in cases of lines to reservoirs and vented manifolds.
4. Symbols may be drawn in any size.
5. Each symbol is drawn to show the normal or neutral condition of each component unless multiple circuit diagrams are furnished showing various phases of circuit operation.

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For more detailed information concerning the symbols used in fluid power diagrams, consult the above-mentioned military standards. Additional information concerning symbols and the reading of diagrams is contained in *BIueprint Reading and Sketching,* NAVEDTRA 10077-F1.

TYPES OF DIAGRAMS

There are many types of diagrams. Those that are most pertinent to fluid power systems are discussed in this text.

Pictorial Diagrams

Pictorial diagrams (fig. 12-1) show the general location and actual appearance of each component, all interconnecting piping, and the general piping arrangement. This type of diagram is sometimes referred to as an installation diagram. Diagrams of this type are invaluable to maintenance personnel in identifying and locating components of a system.

Cutaway Diagrams

Cutaway diagrams (fig. 12-2) show the internal working parts of all fluid power components in a system. This includes controls and actuating mechanisms and all interconnecting piping. Cutaway diagrams do not normally use symbols.

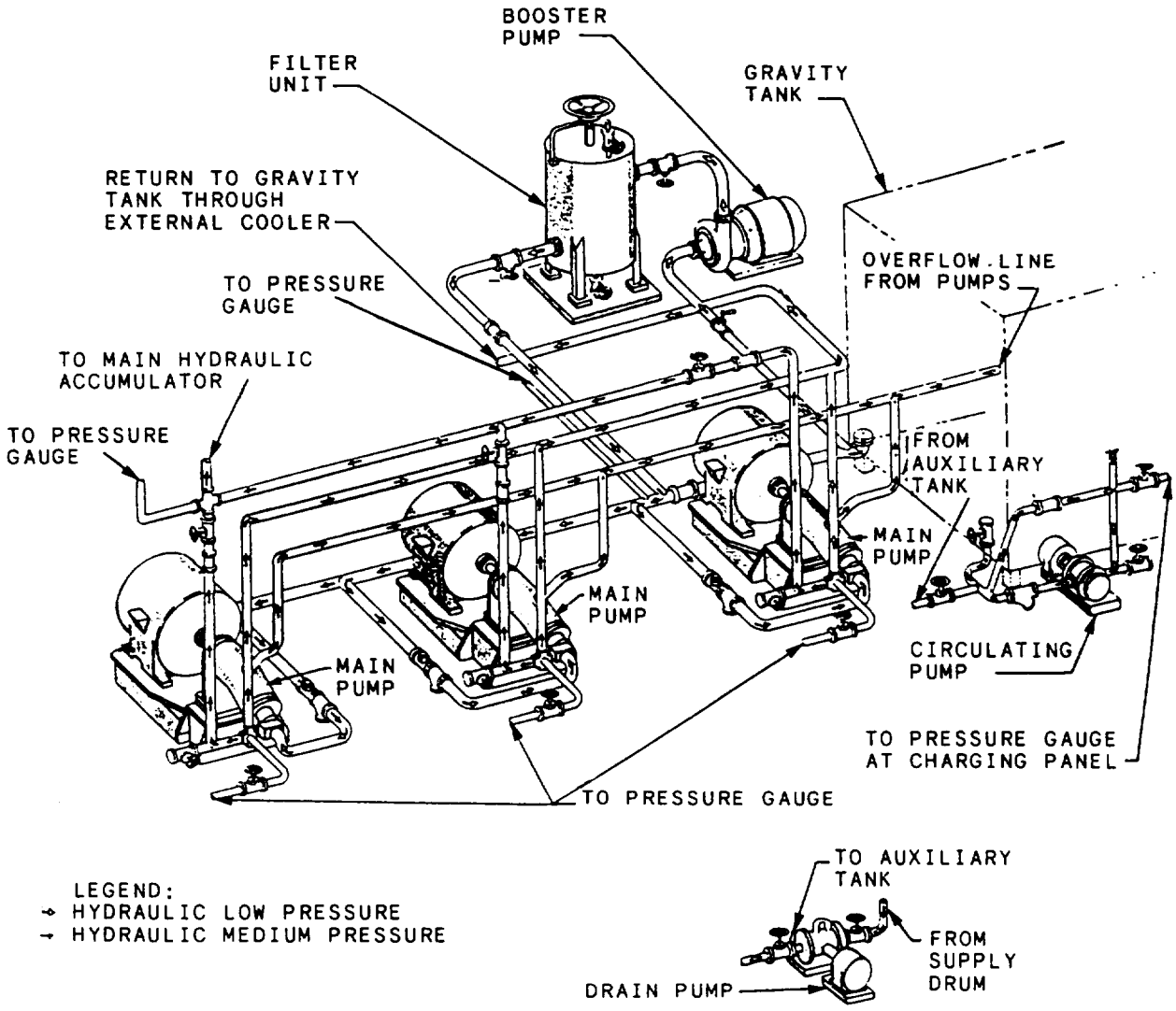

Figure 12-1.—Hydraulic system pictorial diagram.

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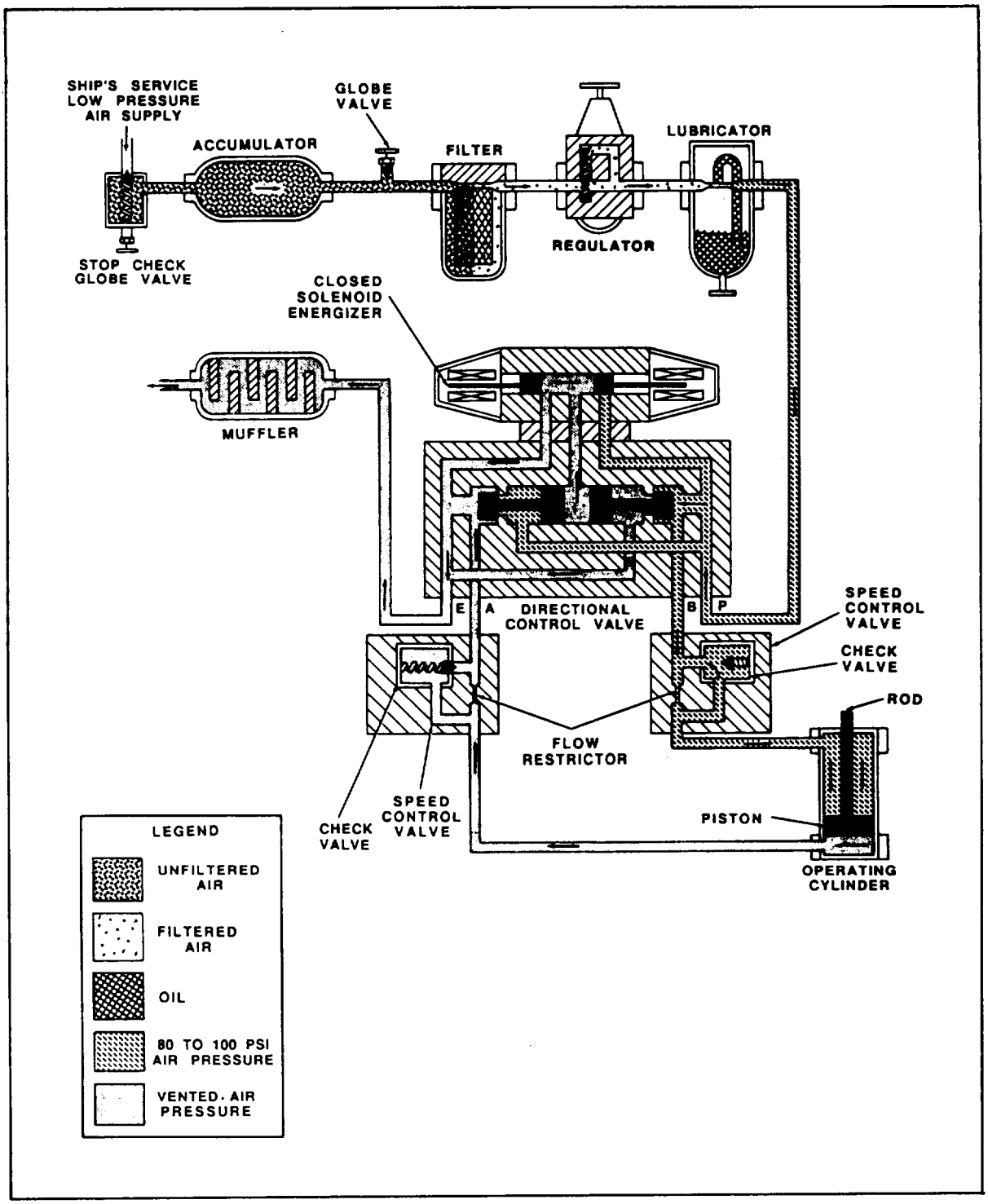

Figure 12-2.—Cutaway diagram—pneumatic.

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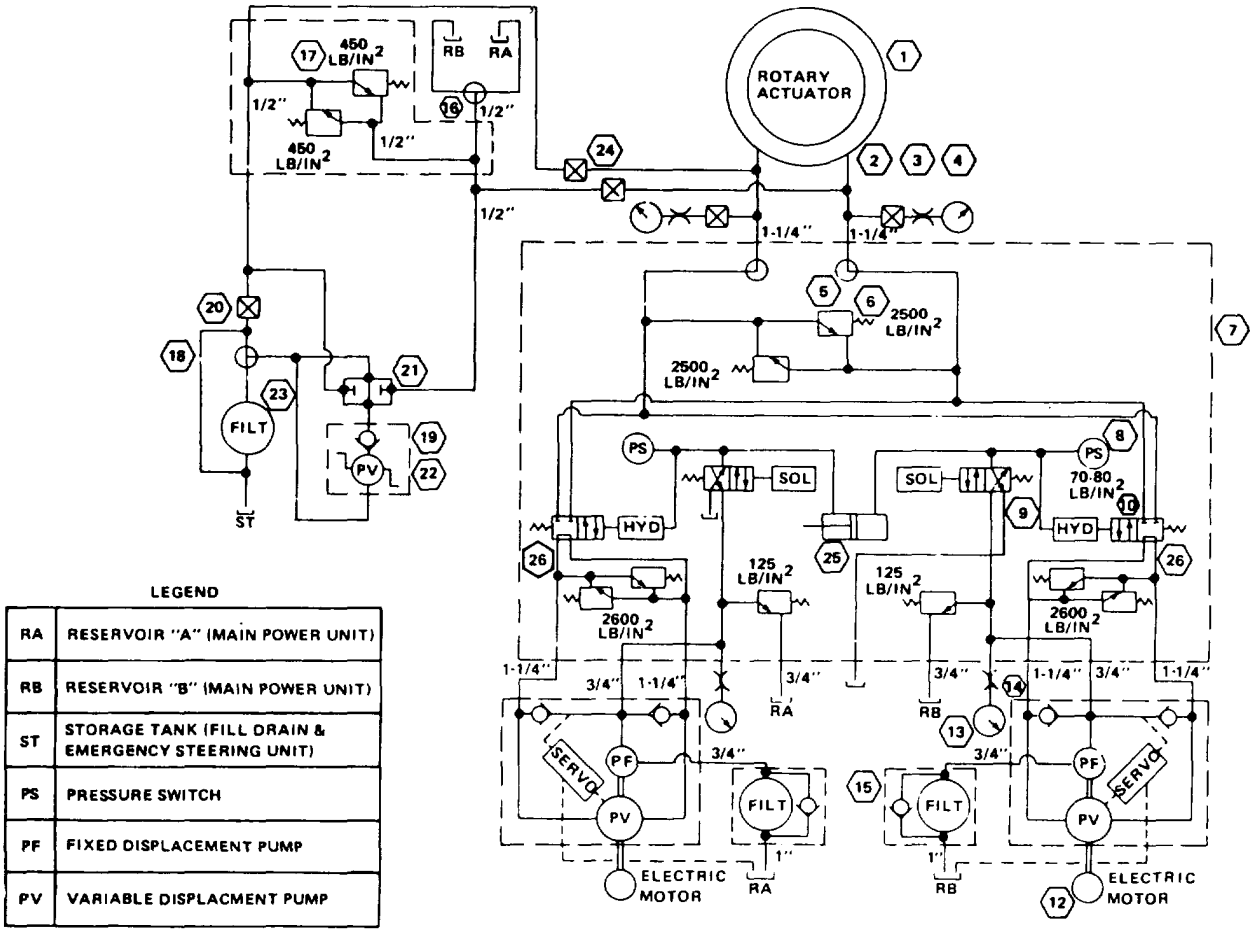
Graphic Diagrams

The primary purpose of a graphic (schematic) diagram is to enable the maintenance person to trace the flow of fluid from component to component within the system. This type of diagram uses standard symbols to show each component and includes all interconnecting piping. Additionally, the diagram contains a component list, pipe size, data on the sequence of operation, and other pertinent information. The graphic diagram (fig. 12-3) does not indicate the physical location of the various components, but it does show the relation of each component to the other components within the system.



LEGEND	
RA	RESERVOIR "A" (MAIN POWER UNIT)
RB	RESERVOIR "B" (MAIN POWER UNIT)
ST	STORAGE TANK (FILL DRAIN & EMERGENCY STEERING UNIT)
PS	PRESSURE SWITCH
PF	FIXED DISPLACEMENT PUMP
PV	VARIABLE DISPLACMENT PUMP

1. Rotary actuator
2. 1 1/4-inch shutoff valve
3. 0-5000 psi pressure snubber
4. 0-3000 psi pressure gauge
5. Manual 1-inch rotary selector valve
6. Pressure control valve
7. Manifold block
8. 7-250 psi pressure switch
9. 1/4-inch solenoid-operated 4-way valve
10. 3/4-inch pilot-operated 4-way valve
11. None
12. Electric motor (see NOTE)
13. 0-300 psi pressure gauge
14. Servo-operated variable-displacement pump
15. 10-micron filter
16. 1/2-inch manual rotary selector valve
17. Pressure control valve
18. 1/2-inch manual rotary selector valve
19. Common check valve with hand pump
20. 1/2-inch shutoff valve
21. 3/8-inch manual rotary selector valve
22. Hand pump
23. 10-micron filter
24. Shutoff valve
25. LVR/LVP selector cylinder
26. Relief valves

NOTE: Electric Motor is 40 horsepower, 1800 rpm, 440 volts ac, 3 phase, 60 hertz.

Figure 12-3.—Graphic diagram of LST 1182 class hydraulic steering gear.

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Notice that figure 12-3 does not indicate the physical location of the individual components with respect to each other in the system. For example, the 3/4-inch, solenoid-operated, 4-way valve (10) is not necessarily located directly above the relief valve (26). The diagram does indicate, however, that the 4-way valve is located in the working line, between the variable-displacement pump and the 1-inch rotary selector valve, and that the valve directs fluid to and from the rotary actuator.

Combination Diagrams

A combination drawing uses a combination of graphic, cutaway, and pictorial symbols. This drawing also includes all interconnecting piping.

FLUID POWER SYSTEMS

A fluid power system in which the fluid in the system remains pressurized from the pump (or regulator) to the directional control valve while the pump is operating is referred to as a closed-center system. In this type of system, any number of subsystems may be incorporated, with a separate directional control valve for each subsystem. The directional control valves are arranged in parallel so that system pressure acts equally on all control valves.

Another type of system that is sometimes used in hydraulically operated equipment is the open-center system. An open-center system has fluid flow but no internal pressure when the actuating mechanisms are idle. The pump circulates the fluid from the reservoir, through the directional control valves, and back to the reservoir. (See fig. 12-4, view A.) Like the closed-center system, the open-center system may have any number of subsystems, with a directional control valve for each subsystem. Unlike the closed-center system, the directional control valves of an open-center system are always connected in series with each other, an arrangement in which the system pressure line goes through each directional control valve. Fluid is always allowed free passage through each control valve and back to the reservoir until one of the control valves is positioned to operate a mechanism.

When one of the directional control valves is positioned to operate an actuating device, as shown in view B of figure 12-4, fluid is directed from the pump through one of the working lines to the actuator. With the control valve in this position, the flow of fluid through the valve to the reservoir is blocked. Thus, the pressure builds up in the system and moves the piston of the

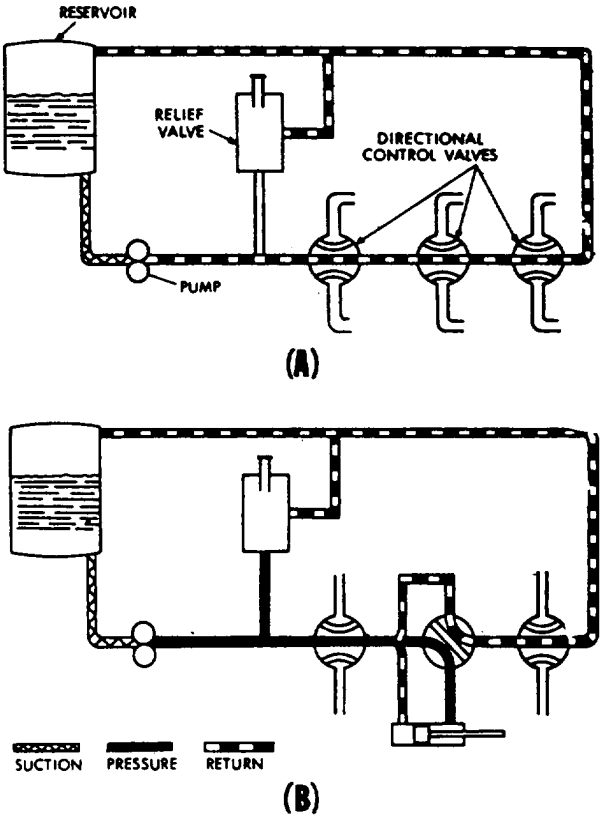

Figure 12-4.—Open-center hydraulic system.

actuating cylinder. The fluid from the other end of the actuator returns to the control valve through the opposite working line and flows back to the reservoir.

Several different types of directional control valves are used in the open-center system. One type is the manually engaged and manually disengaged. After this type of valve is manually moved to the operating position and the actuating mechanism reaches the end of its operating cycle, pump output continues until the system relief valve setting is reached. The relief valve then unseats and allows the fluid to flow back to the reservoir. The system pressure remains at the pressure setting of the relief valve until the directional control valve is manually returned to the neutral position. This action reopens the open-center flow and allows the system pressure to drop to line resistance pressure.

Another type of open-center directional control valve is manually engaged and pressure disengaged. This type of valve is similar to the valve discussed in the preceding paragraph; however, when the actuating mechanism reaches the end of its cycle and the pressure continues to

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rise to a predetermined pressure, the valve automatically returns to the neutral position and, consequently, to open-center flow.

One of the advantages of the open-center system is that the continuous pressurization of the system is eliminated. Since the pressure is gradually built up after the directional control valve is moved to an operating position, there is very little shock from pressure surges. This provides a smooth operation of the actuating mechanisms; however, the operation is slower than the closed-center system in which the pressure is available the moment the directional control valve is positioned. Since most applications require instantaneous operation, closed-center systems are the most widely used.

HYDRAULIC POWER DRIVE SYSTEM

The hydraulic power drive has been used in the Navy for many years. Proof of its effectiveness is that it has been used to train and elevate nearly all caliber guns, from the 40-mm gun mount to the 16-inch turret. In addition to gun mounts and turrets, hydraulic power drives are used to position rocket launchers and missile launchers, and to drive and control such equipment as windlasses, capstans, and winches.

In its simplest form, the hydraulic power drive consists of the following:

1. The prime mover, which is the outside source of power used to drive the hydraulic pump
2. A variable-displacement hydraulic pump
3. A hydraulic motor
4. A means of introducing a signal to the hydraulic pump to control its output
5. Mechanical shafting and gearing that transmits the output of the hydraulic motor to the equipment being operated

Hydraulic power drives differ in some respects, such as size, method of control, and so forth. However, the fundamental operating principles are similar. The unit used in the following discussion of fundamental operating principles is representative of the hydraulic power drives used to operate the 5"/38 twin mounts.

Figure 12-5 shows the basic components of the train power drive. The electric motor is constructed with drive shafts at both ends. The forward shaft drives the A-end pump through reduction gears, and the after shaft drives the auxiliary pumps through the auxiliary reduction gears. The reduction gears are installed because

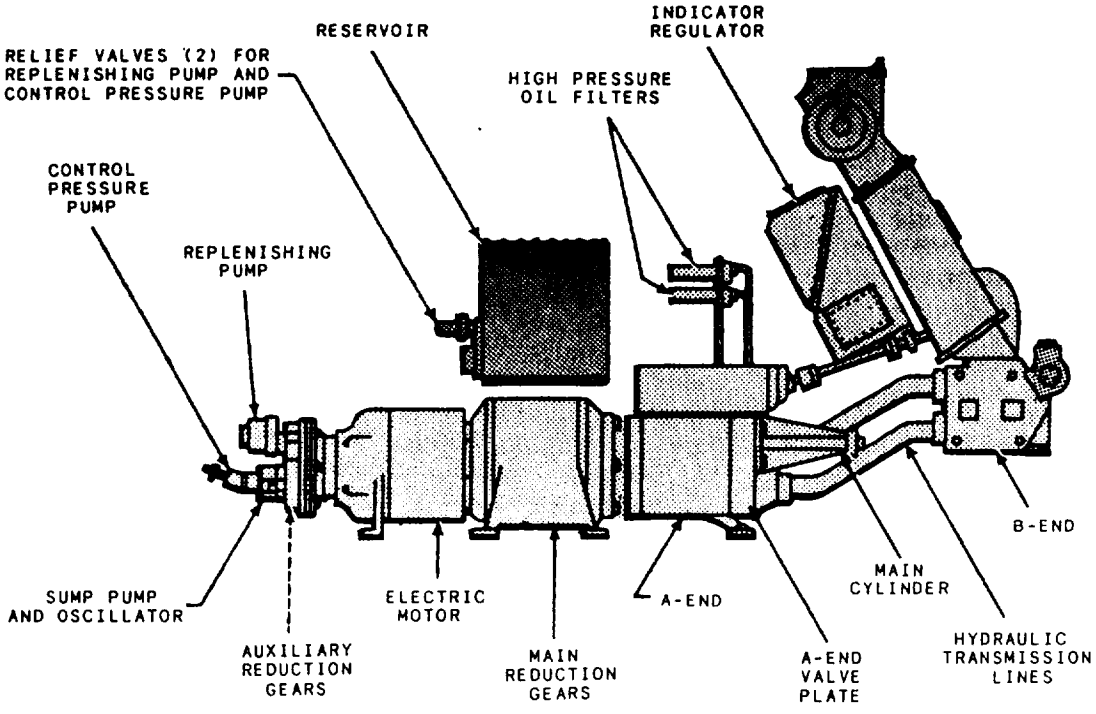

Figure 12-5.-Train power drive—components.

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the pumps are designed to operate at a speed much slower than that of the motor.

The replenishing pump is a spur gear pump. Its purpose is to replenish fluid to the active system of the power drive. It receives its supply of fluid from the reservoir and discharges it to the B-end valve plate. This discharge of fluid from the pump is held at a constant pressure by the action of a pressure relief valve. (Because the capacity of the pump exceeds replenishing demands, the relief valve is continuously allowing some of the fluid to flow back to the reservoir.)

The sump pump and oscillator has a twofold purpose. It pumps leakage, which collects in the sump of the indicator regulator, to the expansion tank. Additionally, it transmits a pulsating effect to the fluid in the response pressure system. Oscillations in the hydraulic response system help eliminate static friction of valves, allowing hydraulic control to respond faster.

The control pressure pump supplies high-pressure fluid for the hydraulic control system, brake pistons, lock piston, and the hand-controlled clutch operating piston. The control pressure pump is a fixed-displacement, axial-piston type. An adjustable relief valve is used to limit the operating pressure at the outlet of the pump.

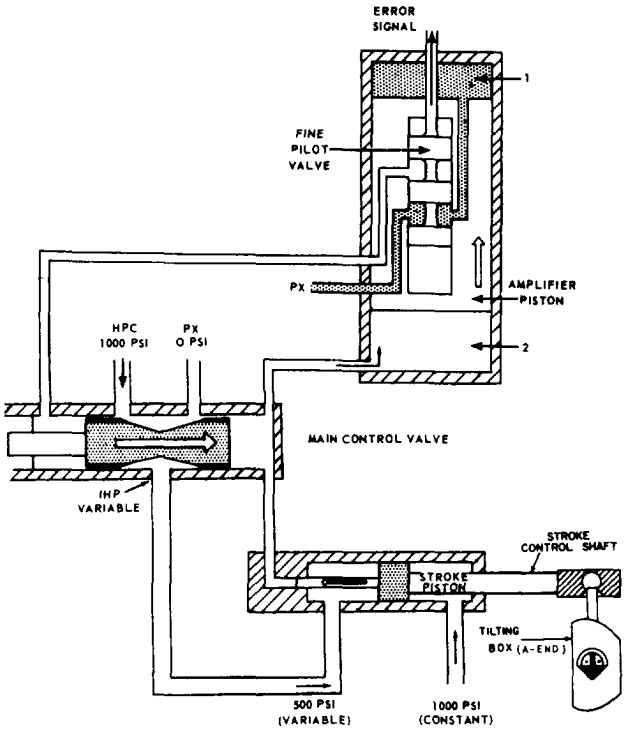

Figure 12–6.–Main cylinder assembly.

Control

For the purpose of this text, control constitutes the relationship between the stroke control shaft and the tilting box. The stroke control shaft is one of the piston rods of a double-acting piston-type actuating cylinder. This actuating cylinder and its direct means of control are referred to as the main cylinder assembly (fig. 12-6). It is the link between the hydraulic followup system and the power drive itself.

In hand control, the tilting box is mechanically positioned by gearing from the handwheel through the A-end control unit. In local and automatic control, the tilting box is positioned by the stroke control shaft. As shown in figure 12-6, the extended end of the control shaft is connected to the tilting box. Movement of the shaft will pivot the tilting box one way or the other; which, in turn, controls the output of the A-end of the transmission. The other end of the shaft is attached to the main piston. A shorter shaft is attached to the opposite side of the piston. This shaft is also smaller in diameter. Thus the working area of the left side of the piston is twice that of the area of the right side, as it appears in figure 12-6.

Intermediate high-pressure fluid (IHP) is transmitted to the left side of the piston, while high-pressure hydraulic fluid (HPC) is transmitted to the right side. The HPC is held constant at 1000 psi. Since the area of the piston upon which HPC acts is exactly one-half the area upon which IHP acts, the main piston is maintained in a fixed position when IHP is one-half HPC (500 psi). Whenever IHP varies from its normal value of 500 psi, the main piston will move, thus moving the tilting box.

Operation

Assume that a right train order signal is received. This will cause the pilot valve to be pulled upward. The fluid in the upper chamber of the amplifier piston can now flow through the lower land chamber of the fine pilot to exhaust. This will cause the amplifier piston to move upward, and the fluid in the right-hand chamber of the main control valve can flow into the lower chamber of the amplifier valve.

The main control valve will now move to the right, IHP will drop below 500 psi, and the stroke piston will move to the left. Movement of the

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stroke piston will cause tilt to be put on the tilt plate, and the A-end will cause the mount to train right.

Figure 12-7 is a simplified block diagram showing the main element of the hydraulic power drive system under automatic control for clockwise and counterclockwise rotation.

There are two principal problems in positioning a gun to fire. One is to get an accurate gun-order signal. This problem is solved by the director-computer combination. The other problem is to transmit the director signal promptly to the gun so that the position and movements of the gun will be synchronized with the signals from the director.

The problem of transforming gun-order signals to mount movements is solved by the power drive and its control—the indicator regulator. The indicator regulator controls the power drive, and this, in turn, controls the movement of the gun.

The indicator regulator receives an initial electrical gun-order from the director-computer, compares it to the existing mount position, and sends an error signal to the hydraulic control mechanism in the regulator. The hydraulic control mechanism controls the flow to the stroke control shaft, which positions the tilting box in the A-end of the transmission. Its tilt controls the volume and direction of fluid pumped to the B-end and, therefore, the speed and direction of the drive shaft of the B-end. Through mechanical linkage, the B-end output shaft moves the gun in the direction determined by the signal. At the same time, B-end response is transmitted to the indicator regulator and continuously combines with incoming gun-order signals to give the error between the two. This error is modified hydraulically, according to the system of mechanical linkages and valves in the regulator. When the gun is lagging behind the signal, its movement is accelerated; and when it begins to catch up, its movement is slowed down so that it will not overrun excessively.

LANDING GEAR EMERGENCY SYSTEM

If the landing gear in a naval aircraft fails to extend to the down and locked position, the aircraft has an emergency method to extend the landing gear. This text will cover the nitrogen system.

The nitrogen storage bottle system is a one-shot system powered by nitrogen pressure stored in four compressed nitrogen bottles (fig. 12-8). When the landing gear control handle is used to actuate the emergency landing gear system, a cable between the control and the manually operated nitrogen bottle opens the emergency gear down release valve on the bottle. Nitrogen from this bottle actuates the release valves on the other three bottles so that they discharge. Nitrogen flows from the manually operated bottle, actuates the dump valves, and causes the shuttles within the shuttle valves on the

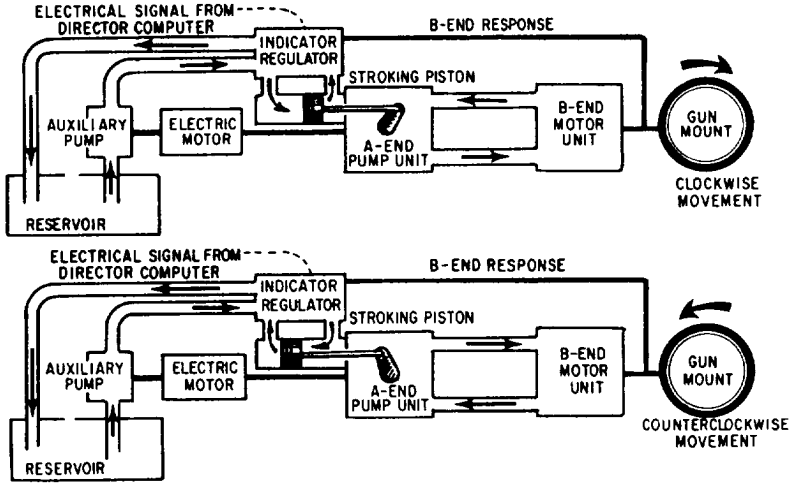

Figure 12-7.—Operation of the hydraulic power drive.

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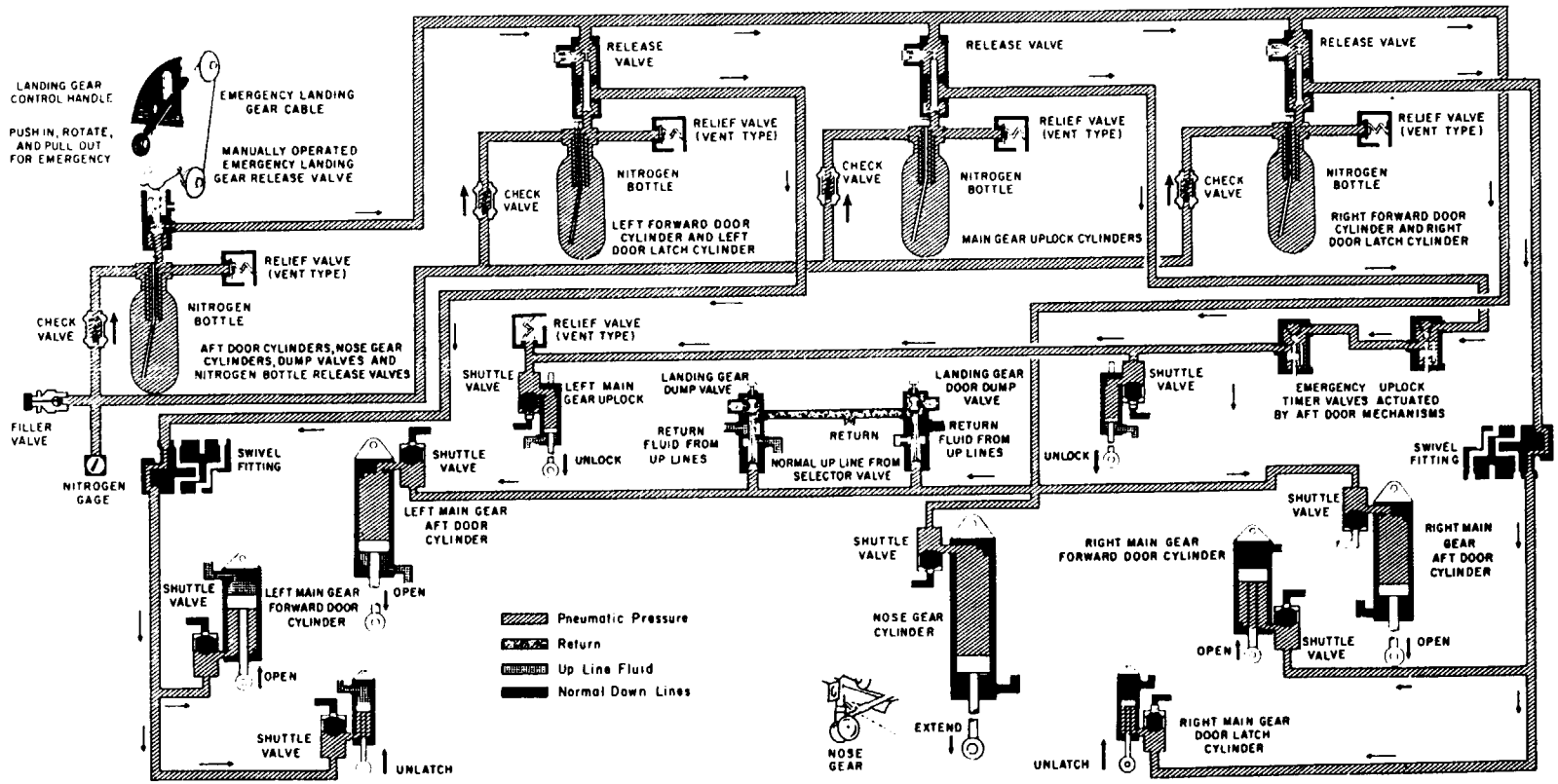

Figure 12-8.—Landing gear emergency extension system.

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aft doors' cylinders and the shuttle valve on the nose gear cylinder to close off the normal port and operate these cylinders. The nose gear cylinder extends; this unlocks the uplock and extends the nose gear. The nitrogen flowing into the aft door cylinders opens the aft doors. Fluid on the close side of the door cylinder is vented to return through the actuated dump valves. Nitrogen from another bottle actuates the shuttle valves on the uplock cylinders. Nitrogen flows into the uplock cylinders and causes them to disengage the uplocks. As soon as the uplocks are disengaged, the main gear extends by the force of gravity. Fluid on the up side of the main gear cylinders is vented to return through the actuated dump valves, preventing a fluid lock.

JET BLAST DEFLECTORS

Jet blast deflectors (JBD) onboard aircraft carriers are raised and lowered by hydraulic cylinders through mechanical linkage. Two hydraulic cylinders are attached to each JBD panel shaft by crank assemblies. (See fig. 12-9.) The shaft is rotated by the push and pull operation of the hydraulic cylinders. Shaft rotation extends or retracts the linkage to raise or lower the JBD panels. This operation is designed so that in the event of a failure of one of the hydraulic cylinders, the other one will raise or lower the panels.

Figure 12-10 is a diagram of the hydraulic control system of a JBD during the raise cycle. Hydraulic fluid from the catapult hydraulic supply system is supplied to the JBD hydraulic system through an isolation valve and a filter to the 4-way control valve assembly. (The 4-way control valve assembly consists of a pilot-operated control valve, a direct- or solenoid-operated control valve, and a sequence valve, which is not shown.)

To raise the JBD, solenoid B of the 4-way control valve assembly is energized. The spools of the 4-way valve assembly shift, allowing medium-pressure hydraulic fluid to flow into port A of the hydraulic cylinder. The cylinders extend,

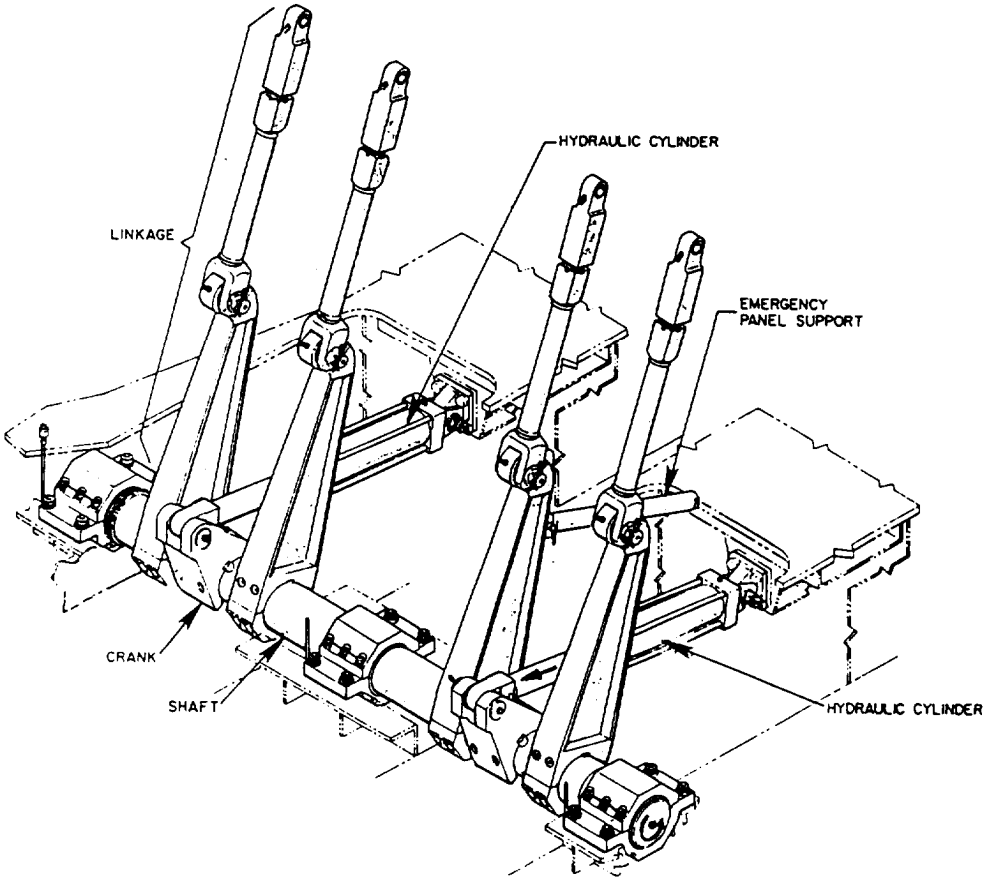

Figure 12-9.—Operating gear assembly (panels raised).

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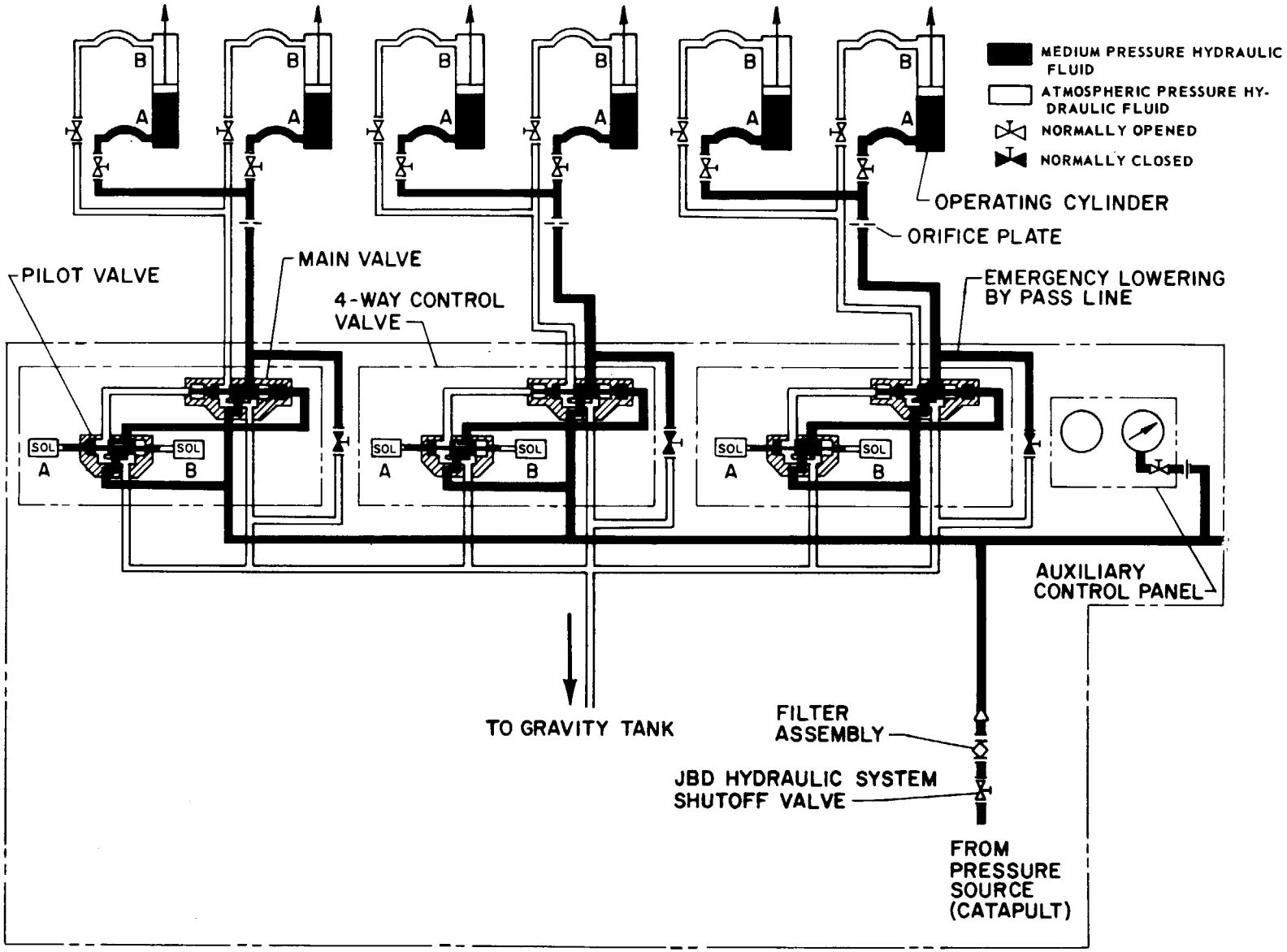

Figure 12-10.—Hydraulic system flow diagram, raise cycle.

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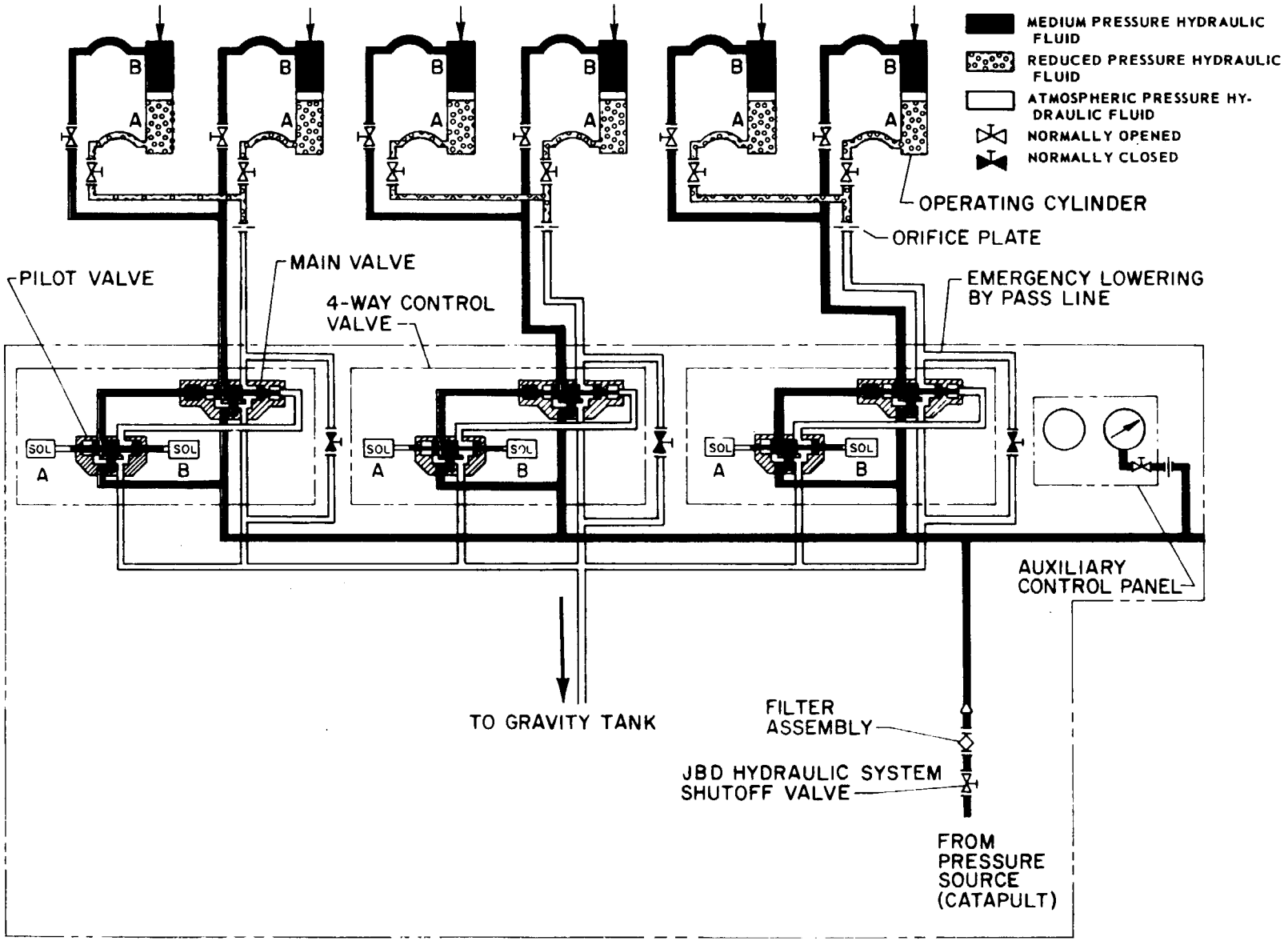

Figure 12-11.—Hydraulic system flow diagram, lower cycle.

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pushing the crank assembly aft and rotating the shaft. The rotation of the shaft extends the operating gear linkage and raises the panel assemblies. Fluid from port B of the piston is directed through the 4-way valve assembly and back to the gravity tank.

To lower the JBD (fig. 12-11), solenoid A of the 4-way control valve assembly is energized. The spools of the 4-way valve assembly shift, allow medium-pressure hydraulic fluid to flow into port B of the hydraulic cylinder. The cylinders retract, pulling the crank assembly forward and rotating the shaft. The rotation of the shaft retracts the operating gear linkage and lowers the panel assemblies. Fluid from port A of the piston is directed through the 4-way valve assembly and back to the gravity tank.

To lower the JBD in the event of hydraulic control failure, each JBD panel is equipped with a manual bypass valve, which allows bypassing the 4-way control valve. This allows venting the hydraulic pressure from the “raise” side of the cylinder back to the gravity tank.

The three lines to port A of the hydraulic cylinders have orifice assemblies in them. These orifice assemblies control the flow of hydraulic fluid in both the raise and lower operations.

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APPENDIX I

GLOSSARY

A part of this glossary has been extracted from the *American Standard Glossary of Terms for Fluid Power* (ASA B93.2-1965) with permission of the publisher, The National Fluid Power Association.

ABSOLUTE TEMPERATURE—The temperature measured using absolute zero as a reference. Absolute zero is –273.16°C or –459.69°F.

ACCELERATION—Time rate of change of velocity.

ACCUMULATOR—A device for storing liquid under pressure. It usually consists of a chamber separated into a gas compartment and a liquid compartment by a piston or diaphragm. An accumulator also serves to smooth out pressure surges in a hydraulic system.

ACTUATOR—A device that converts fluid power into mechanical force and motion.

ADDITIVE—A chemical compound or compounds added to a fluid to change its properties.

AIR, COMPRESSED—Air at any pressure greater than atmospheric pressure.

AMBIENT—Surrounding, such as ambient air, meaning surrounding air.

BAROMETER—An instrument that measures atmospheric pressure.

BERNOULLI'S PRINCIPLE—If a fluid flowing through a tube reaches a constriction, or narrowing of the tube, the velocity of the fluid flowing through the constriction increases and the pressure decreases.

BLEEDER, AIR—A bleeder for the removal of air.

BOYLE'S LAW—The absolute pressure of a fixed mass of gas varies inversely as the volume, provided the temperature remains constant.

CAVITATION—A localized gaseous condition within a liquid stream that occurs where the pressure is reduced to the vapor pressure.

CELSIUS—The temperature scale using the freezing point of water as zero and the boiling point as 100, with 100 equal divisions between, called degrees. This scale was formerly known as the centigrade scale.

CENTIGRADE—(See Celsius.)

CENTRIFUGAL FORCE—A force exerted on a rotating object in a direction outward from the center of rotation.

CHARLES'S LAW—If the pressure is constant, the volume of dry gas varies directly with the absolute temperature.

CHEMICAL CHANGE—A change that alters the composition of the molecules of a substance.

CIRCUIT—An arrangement of interconnected component parts.

COMPRESSIBILITY—The change in volume of a unit volume of a fluid when it is subjected to a unit change of pressure.

COMPRESSOR—A device that converts mechanical force and motion into pneumatic fluid power.

COMPUTER—A device capable of accepting information, applying prescribed processes to the information, and supplying the results of these processes.

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CONDENSATION—The change from a gaseous (or vapor) state to a liquid state.

CONTAMINANT—Detrimental matter in a fluid.

CONTINUITY EQUATION—The mass rate of fluid flow into any fixed space is equal to the mass flow rate out. Hence, the mass flow rate of fluid past all cross sections of a conduit is equal.

CONTROL—A device used to regulate the function of a component or system.

CONTROL, CYLINDER—A control in which a fluid cylinder is the actuating device.

CONTROL, ELECTRIC—A control actuated electrically.

CONTROL, HYDRAULIC—A control actuated by a liquid.

CONTROL, MANUAL—A control actuated by the operator.

CONTROL, MECHANICAL—A control actuated by linkages, gears, screws, cams, or other mechanical elements.

CONTROL, PNEUMATIC—A control actuated by air or other gas pressure.

CONTROL, SERVO—A control actuated by a feedback system that compares the output with the reference signal and makes corrections to reduce the difference.

CONTROLS, PUMP—Controls applied to positive-displacement variable delivery pumps to adjust their volumetric output or direction of flow.

CONVERGENT—That which inclines and approaches nearer together, as the inner walls of a tube that is constricted.

COOLER—A heat exchanger, which removes heat from a fluid.

COOLER, AFTERCOOLER—A device that cools a gas after it has been compressed.

COOLER, INTERCOOLER—A device that cools a gas between the compressive steps of a multiple stage compressor.

COOLER, PRECOOLER—A device that cools a gas before it is compressed.

CORROSION—The slow destruction of materials by chemical agents and electromechanical reactions.

CYCLE—A single complete operation consisting of progressive phases starting and ending at the neutral position.

CYLINDER—A device that converts fluid power into linear mechanical force and motion. It usually consists of a movable element, such as a piston and piston rod, plunger, or ram, operating within a cylindrical bore.

CYLINDER, CUSHIONED—A cylinder with a piston-assembly deceleration device at one of both ends of the stroke.

CYLINDER, DOUBLE-ACTING—A cylinder in which fluid force can be applied to the movable element in either direction.

CYLINDER, DOUBLE-ROD—A cylinder with a single piston and a piston rod extending from each end.

CYLINDER, DUAL-STROKE—A cylinder combination that provides two working strokes.

CYLINDER, PISTON—A cylinder in which the movable element has a greater cross-sectional area than the piston rod.

CYLINDER, PLUNGER—A cylinder in which the movable element has the same cross-sectional area as the piston rod.

CYLINDER, SINGLE-ACTING—A cylinder in which the fluid force can be applied to the movable element in only one direction.

CYLINDER, SINGLE-ROD—A cylinder with a piston rod extending from one end.

CYLINDER, SPRING-RETURN—A cylinder in which a spring returns the piston assembly.

CYLINDER, TANDEM—Two or more cylinders with interconnected piston assemblies.

CYLINDER, TELESCOPING—A cylinder with nested multiple tubular rod segments which provide a long working stroke in a short retracted envelope.

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DENSITY—The weight per unit volume of a substance.

DIAGRAM, COMBINATION—A drawing using a combination of graphical, cutaway, and pictorial symbols.

DIAGRAM, CUTAWAY—A drawing showing principal internal parts of all components, controls, and actuating mechanisms, all interconnecting lines and functions of individual components.

DIAGRAM, GRAPHICAL—A drawing or drawings showing each piece of apparatus including all interconnecting lines by approved standard symbols.

DIAGRAM, PICTORIAL—A drawing showing each component in its actual shape according to the manufacturer's installation.

DIAGRAM, SCHEMATIC—(See Diagram, graphical.)

DIAPHRAGM—A dividing membrane or thin partition.

DIFFUSER—A duct of varying cross section designed to convert a high-speed gas flow into low-speed at an increased pressure.

DISPLACEMENT—The volume of fluid that can pass through a pump, motor, or cylinder in a single revolution or stroke.

DIVERGENT—Moving away from each other, as the inner wall of a tube that flares outward.

EFFICIENCY—The ratio of the output power to the input power, generally expressed as a percentage.

ENERGY—The ability or capacity to do work.

EQUILIBRIUM—A state of balance between opposing forces or actions.

FAHRENHEIT—The temperature scale using the freezing point of water as 32 and the boiling point as 212, with 180 equal divisions between, called degrees.

FEEDBACK—A transfer of energy from the output of a device to its input.

FILTER—A device whose primary function is the retention by a porous media of insoluble contaminants from a fluid.

FILTER ELEMENT—The porous device that performs the actual process of filtration.

FILTER MEDIA—The porous materials that perform the actual process of filtration.

FILTER MEDIA, SURFACE—Porous materials that primarily retain contaminants on the influent face.

FLASH POINT—The temperature to which a liquid must be heated under specified conditions of the test method to give off sufficient vapor to form a mixture with air that can be ignited momentarily by a specified flame.

FLOW, LAMINAR—A flow situation in which fluid moves in parallel layers (also referred to as streamline flow).

FLOW, METERED—Flow at a controlled rate.

FLOW, TURBULENT—A flow situation in which the fluid particles move in a random manner.

FLOW RATE—The volume, mass, or weight of a fluid passing through any conductor per unit of time.

FLOWMETER—An instrument used to measure quantity or the flow rate of a fluid motion.

FLUID—A liquid or a gas.

FLUID FLOW—The stream or movement of a fluid, or the rate of its movement.

FLUID FRICTION—Friction due to the viscosity of fluids.

FLUID, FIRE-RESISTANT—A fluid difficult to ignite, which shows little tendency to propagate flame.

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FLUID, HYDRAULIC—A fluid suitable for use in a hydraulic system.

FLUID, PETROLEUM—A fluid composed of petroleum oil. It may contain additives.

FLUID, PHOSPHATE ESTER BASE—A fluid that contains a phosphate ester as one of the major components.

FLUID, SILICONE—A fluid composed of silicones. It may contain additives.

FLUID, WATER-GLYCOL—A fluid whose major constituents are water and one or more glycols or polyglycols.

FLUID STABILITY—Resistance of a fluid to permanent change in properties.

FLUID POWER—Energy transmitted and controlled through the use of fluids under pressure.

FLUID POWER SYSTEM—A system that transmits and controls power through use of a pressurized fluid within an enclosed circuit.

FOOT-POUND—The amount of work accomplished when a force of 1 pound produces a displacement of 1 foot.

FORCE—The action of one body on another tending to change the state of motion of the body acted upon.

FREE FLOW—Flow that encounters negligible resistance.

FRICTION—The action of one body or substance rubbing against another, such as fluid flowing against the walls of pipe; the resistance to motion caused by this rubbing.

FRICTION PRESSURE DROP—The decrease in the pressure of a fluid flowing through a passage attributable to the friction between the fluid and the passage walls.

GAS—The form of matter that has neither a definite shape nor a definite volume.

GASKET—A class of seals that provides a seal between two stationary parts.

GAUGE—An instrument or device for measuring, indicating, or comparing a physical characteristic.

GAUGE PRESSURE—Pressure above atmospheric pressure.

GAUGE SNUBBER—A device installed in the line to the pressure gauge used to dampen pressure surges and thus provide a steady reading and a protection for the gauge.

GAUGE, BELLOWS—A gauge in which the sensing element is a convoluted closed cylinder. A pressure differential between the outside and the inside causes the cylinder to expand or contract axially.

GAUGE, BOURDON TUBE—A pressure gauge in which the sensing element is a curved tube that tends to straighten out when subjected to internal fluid pressure.

GAUGE, DIAPHRAGM—A gauge in which the sensing element is relatively thin and its inner portion is free to deflect with respect to its periphery.

GAUGE, PRESSURE—A gauge that indicates the pressure in the system to which it is connected.

GAUGE, VACUUM—A pressure gauge for pressures less than atmospheric.

GRAVITY—The force that tends to draw all bodies toward the center of the earth. The weight of a body is the resultant of gravitational force acting on the body.

HEAD—The height of a column or body of fluid above a given point expressed in linear units. Head is often used to indicate gauge pressure. Pressure is equal to the height times the density of the fluid.

HEAD, FRICTION—The head required to overcome the friction at the interior surface of a conductor and between fluid particles in motion. It varies with flow, size, type, and condition of conductors and fittings, and fluid characteristics,

HEAD, STATIC—The height of a column or body of fluid above a given point.

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HEAD, VELOCITY—The equivalent head through which the liquid would have to fall to attain a given velocity. Mathematically it is equal to the square of the velocity (in feet) divided by 64.4 feet per second square.

HEAT EXCHANGER—A device that transfers heat through a conducting wall from one fluid to another.

HYDRAULICS—Engineering science pertaining to liquid pressure and flow.

HYDROMETER—An instrument for determining the specific gravities of liquids.

HYDROPNEUMATICS—Pertaining to the combination of hydraulic and pneumatic fluid power.

HYDROSTATICS—Engineering science pertaining to the energy of liquids at rest.

IMPACT PRESSURE—The pressure of a moving fluid brought to rest that is in excess of the pressure the fluid has when it does not flow; that is, total pressure less static pressure. Impact pressure is equal to dynamic pressure in incompressible flow; but in compressible flow, impact pressure includes the pressure change owing to the compressibility effect.

IMPINGEMENT—The striking or dashing upon with a clash or sharp collision, as air impinging upon the rotor of a turbine or motor.

IMPULSE TURBINE—A turbine driven by a fluid at high velocity under relatively low pressure.

INERTIA—The tendency of a body at rest to remain at rest, and a body in motion to continue to move at a constant speed along a straight line, unless the body is acted upon in either case by an unbalanced force.

INHIBITOR—Any substance which slows or prevents chemical reactions such as corrosion or oxidation.

INVERSE PROPORTION—The relation that exists between two quantities when an increase in one of them produces a corresponding decrease in the other.

KELVIN SCALE—The temperature scale using absolute zero as the zero point and divisions that are the same size as centigrade degrees.

KINETIC ENERGY—The energy that a substance has while it is in motion.

KINETIC THEORY—A theory of matter that assumes that the molecules of matter are in constant motion.

LINE—A tube, pipe, or hose that is used as a conductor of fluid.

LIQUID—A form of matter that has a definite volume but takes the shape of its container.

LOAD—The power that is being delivered by any power-producing device. The equipment that uses the power from the power-producing device.

LUBRICATOR—A device that adds controlled or metered amounts of lubricant into a fluid power system.

MANIFOLD—A type of fluid conductor that provides multiple connections ports.

MANOMETER—A differential pressure gauge in which pressure is indicated by the height of a liquid column of known density. Pressure is equal to the difference in vertical height between two connected columns multiplied by the density of the manometer liquid. Some forms of manometers are U tube, inclined tube, well, and bell types.

MATTER—Any substance that occupies space and has weight.

MECHANICAL ADVANTAGE—The ratio of the resisting weight to the acting force. The ratio of the distance through which the force is exerted divided by the distance the weight is raised.

METER-IN—To regulate the amount of fluid into a system or an actuator.

METER-OUT—To regulate the flow of fluid from a system or actuator.

MICRON—A millionth of a meter or about 0.00004 inch.

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MOLECULE—A small natural particle of matter composed of two or more atoms.

MOTOR—A device that converts fluid power into mechanical force and motion. It usually provides rotary mechanical motion.

MOTOR, FIXED-DISPLACEMENT—A motor in which the displacement per unit of output motion cannot be varied.

MOTOR, LINEAR—(See Cylinder.)

MOTOR, ROTARY—A motor capable of continuous rotary motion.

MOTOR, ROTARY LIMITED—A rotary motor having limited motion.

MOTOR, VARIABLE-DISPLACEMENT—A motor in which the displacement per unit of output motion can be varied.

NEOPRENE—A synthetic rubber highly resistant to oil, light, heat, and oxidation.

NEUTRALIZATION NUMBER—A measure of the total acidity or basicity of an oil; this includes organic or inorganic acids or bases or a combination of them.

OXIDATION—The process by which oxygen unites with some other substance, causing rust or corrosion.

PACKING—A class of seal that is used to provide a seal between two parts of a unit which move in relation to each other.

PASCAL'S LAW—A pressure applied to a confined fluid at rest is transmitted with equal intensity throughout the fluid.

PERIPHERY—The outside surface, especially that of a rounded object or body.

PIPE—A type of fluid line whose dimensions are designated by nominal (approximate) inside diameter and wall thickness.

PNEUMATICS—Engineering science pertaining to gaseous pressure and flow.

PORT—An internal or external terminus of a passage in a component.

POTENTIAL ENERGY—The energy a substance has because of its position, its condition, or its chemical composition.

POUR POINT—The lowest temperature at which a liquid will flow under specified conditions.

POWER UNIT—A combination of pump, pump drive, reservoir, controls, and conditioning components which may be required for its application.

POWER—The rate of doing work or the rate of expanding energy.

PRESSURE—The amount of force distributed over each unit of area, usually expressed in pounds per square inch.

PRESSURE, ABSOLUTE—The sum of atmospheric and gauge pressures.

PRESSURE, ATMOSPHERIC—Pressure exerted by the atmosphere at any specific location.

PRESSURE, BACK—The pressure encountered on the return side of a system.

PRESSURE, DIFFERENTIAL—The difference in pressure between any two points of a system or a component.

PRESSURE, HEAD—The pressure due to the height of a column or body of fluid. It is usually expressed in feet.

PRESSURE, OPERATING—The pressure at which a system operates.

PRESSURE, PRECHARGE—The pressure of compressed gas in an accumulator prior to the admission of a liquid.

PRESSURE, PROOF—The nondestructive test pressure in excess of the maximum rated operating pressure.

PRESSURE, STATIC—The pressure in a fluid at rest.

PRESSURE SWITCH—An electrical switch operated by the increase or decrease of fluid pressure.

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PRIME MOVER—The source of mechanical power used to drive the pump or compressor.

PUMP—A device that converts mechanical force and motion into hydraulic fluid power.

PUMP, AXIAL PISTON—A pump having multiple pistons disposed with their axes parallel.

PUMP, CENTRIFUGAL—A pump that produces fluid velocity and converts it to pressure head.

PUMP, FIXED-DISPLACEMENT—A pump in which the displacement per cycle cannot be varied.

PUMP, RADIAL PISTON—A pump having multiple pistons disposed radially actuated by an eccentric element.

PUMP, VARIABLE-DISPLACEMENT—A pump in which the volume of fluid per cycle can be varied.

RANKINE SCALE—A thermometer scale based on absolute zero of the Fahrenheit scale, in which the freezing point of water is approximately 492°R.

RATIO—The value obtained by dividing one number by another, indicating their relative proportions.

RECEIVER—A container in which gas is stored under pressure as a supply source for pneumatic power.

RECIPROCATING—Moving back and forth, as a piston reciprocating in a cylinde.,

RESERVOIR—A container for storage of liquid in a fluid power system.

RESPONSE TIME—The time lag between a signal input and the resulting change of output.

RESTRICTOR—A device that reduces the cross-sectional flow area.

RESTRICTOR, ORIFICE—A restrictor, the length of which is relatively small with respect to its cross-sectional area. The orifice may be fixed or variable. Variable types are noncompensated, pressure compensated, or pressure and temperature compensated.

RETURN LINE—A line used for returning fluid back into the reservoir or atmosphere.

SEPARATOR—A device whose primary function is to isolate undesirable fluids and or contaminants by physical properties other than size.

SERVO—A device used to convert a small movement into a greater movement of force.

SOLID—The form of matter that has a definite shape and a definite volume.

SPECIFIC GRAVITY—The ratio of the weight of a given volume of a substance to the weight of an equal volume of some standard substance.

STEADY FLOW—A flow in which the velocity, pressure, and temperature at any point in the fluid do not vary with time.

STRAINER—A coarse filter.

STOKE—The standard unit of kinematic viscosity in the cgs system. It is expressed in square centimeters per second; 1 centistoke equals 0.01 stoke.

STUFFING BOX—A cavity and closure with manual adjustment for a sealing device.

SUPPLY LINE—A line that conveys fluid from the reservoir to the pump.

SURGE—A momentary rise of pressure in a circuit.

SYNCHRONIZE—To make two or more events or operations occur at the proper time with respect to each other.

SYNTHETIC MATERIAL—A complex chemical compound that is artificially formed by the combining of two or more simpler compounds or elements.

TANK—A container for the storage of fluid in a fluid power system.

THEORY—A scientific explanation, tested by observations and experiments.

THERMAL EXPANSION—The increase in volume of a substance due to temperature change.

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TORQUE—A force or combination of forces that produces or tends to produce a twisting or rotary motion.

TUBING—A type of fluid line whose dimensions are designated by actual measured outside diameter and by actual measured wall thickness.

TURBINE—A rotary motor actuated by the reaction, impulse, or both, of a flow of pressurized fluid.

VALVE—A device that controls fluid flow direction, pressure, or flow rate.

VALVE, CHECK—A directional control valve that permits flow of fluid in only one direction.

VALVE, COUNTERBALANCE—A pressure control valve that maintains back pressure to prevent a load from falling.

VALVE, DIRECTIONAL CONTROL—A valve whose primary function is to direct or prevent flow through selected passages.

VALVE, FLOW CONTROL—A valve whose primary function is to control flow rate.

VALVE, HYDRAULIC—A valve for controlling liquid.

VALVE, PILOT—A valve used to operate another valve or control.

VALVE, PNEUMATIC—A valve for controlling gas.

VALVE, PRESSURE REDUCING—A pressure control valve whose primary function is to limit outlet pressure.

VALVE, PRIORITY—A valve that directs flow to one operating circuit at a fixed rate and directs excess flow to another operating circuit.

VALVE, RELIEF—A pressure control valve whose primary function is to limit system pressure.

VALVE, SELECTOR—A directional control valve whose primary function is to selectively interconnect two or more ports.

VALVE, SEQUENCE—A valve whose primary function is to direct flow in a predetermined sequence.

VALVE, SERVO—A directional control valve that modulates flow or pressure as a function of its input signal.

VALVE, SHUTOFF—A valve that operates fully open or fully closed.

VALVE, UNLOADING—A pressure control valve whose primary function is to permit a pump or compressor to operate at minimum load.

VELOCITY—The rate of motion in a particular direction. The velocity of fluids is usually expressed in feet per second.

VENTURI—A tube having a narrowing throat or constriction to increase the velocity of fluid flowing through it. The flow through the venturi causes a pressure drop in the smallest section, the amount being a function of the velocity of flow.

VISCOSITY—A measure of the internal friction or resistance of a fluid to flow.

VISCOSITY INDEX—A measure of the viscosity-temperature characteristics of a fluid as referred to that of two arbitrary reference fluids.

VISCOSITY, SAYBOLT UNIVERSAL SECONDS (SUS)—The time in seconds for 60 milliliters of oil to flow through a standard orifice at a given temperature.

VISCOSITY, KINEMATIC—The absolute viscosity divided by the density of the fluid. It is usually expressed in centistokes.

VOLUME OF FLOW—The quantity of fluid that passes a certain point in a unit of time. The volume of flow is usually expressed in gallons per minute for liquids and cubic feet per minute for gases.

WORK—The transference of energy from one body or system to another. That which is accomplished by a force acting through a distance.

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APPENDIX II

MECHANICAL SYMBOLS OTHER THAN AERONAUTICAL FOR FLUID POWER DIAGRAMS

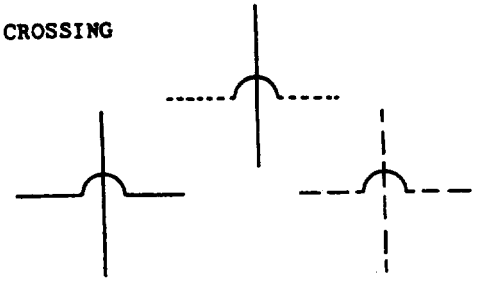
LINES

LINES, WORKING

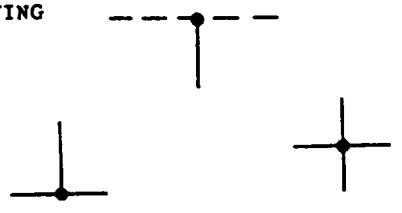
LINES, PILOT

LINES, LIQUID DRAIN OR AIR EXHAUST

LINES, CROSSING

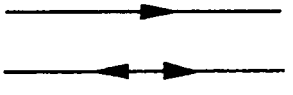


LINES, JOINING



LINES, FLEXIBLE

FLOW, DIRECTION OF

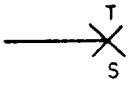


LINES TO RESERVOIR BELOW FLUID LEVEL

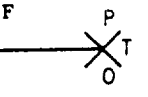
ABOVE FLUID LEVEL

PLUG OR PLUGGED CONNECTION

TESTING STATION

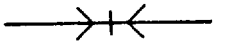


FLUID POWER TAKE-OFF STATION

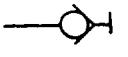


RESTRICTION, FIXED

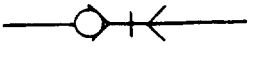
QUICK DISCONNECT WITHOUT CHECKS



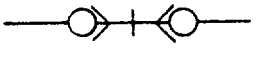
WITH CHECKS DISCONNECTED



WITH ONE CHECK



WITH TWO CHECKS



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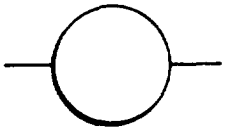
PUMPS, COMPRESSORS & ROTARY MOTORS

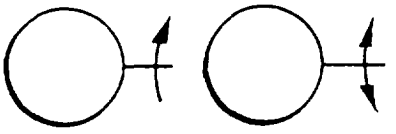
BASIC SYMBOL ENVELOPE

PORTS



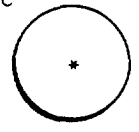
LINES OUTSIDE ENVELOPE ARE NOT PART OF SYMBOL, BUT REPRESENT FLOW LINES CONNECTED THERETO.

SHAFTS, ROTATING



ARROW INDICATES DIRECTION OF ROTATION BY ASSUMING IT IS ON NEAR SIDE OF SHAFT.

PUMPS, HYDRAULIC

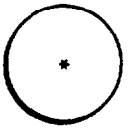


APPROPRIATE SYMBOLS SHALL BE ADDED TO INDICATE SHAFTS, CONNECTING LINES, AND METHOD OF CONTROL.

* TYPE OF PUMP SHALL BE INDICATED WITHIN BASIC SYMBOL BY APPROPRIATE LETTERS LISTED BELOW.

PF FIXED DISPLACEMENT
PK KINETIC - CENTRIFUGAL
PV VARIABLE DISPLACEMENT

COMPRESSORS, AIR

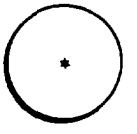


APPROPRIATE SYMBOLS SHALL BE ADDED TO INDICATE SHAFTS, CONNECTING LINES, AND METHOD OF CONTROL.

* TYPE OF COMPRESSOR SHALL BE INDICATED WITHIN BASIC SYMBOL BY APPROPRIATE LETTERS LISTED BELOW.

CF FIXED DISPLACEMENT
CK KINETIC

FLUID MOTORS, ROTARY



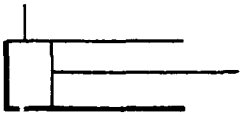
APPROPRIATE SYMBOLS SHALL BE ADDED TO INDICATE SHAFTS, CONNECTING LINES, AND METHOD OF CONTROL.

* TYPE OF MOTOR SHALL BE INDICATED WITHIN BASIC SYMBOL BY APPROPRIATE LETTERS LISTED BELOW.

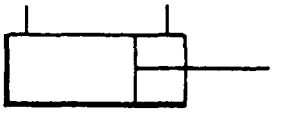
MF FIXED DISPLACEMENT
MO OSCILLATING
MV VARIABLE DISPLACEMENT

CYLINDERS

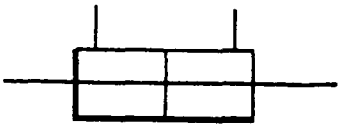
SINGLE ACTING



DOUBLE ACTING SINGLE END ROD



DOUBLE END ROD



RESERVOIRS

VENTED

PRESSURIZED

RECEIVER FOR AIR OR OTHER GASES

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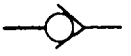
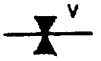
VALVE

VALVE, CHECK 

VALVE, RESTRICTION, CHOKE VARIABLE VISCOUS

VALVE, RESTRICTION, ORIFICE VARIABLE NON-VISCOUS 

VALVE, BASIC SYMBOL (INSERT MODEL NO. FOR SPECIAL VALVES)

METHOD OF INDICATING INTERNAL FLOW

VALVE EXAMPLES

VALVE, MANUAL SHUT OFF

VALVE, RELIEF MAXIMUM PRESSURE

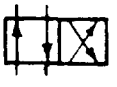
VALVE, RELIEF REMOTELY OPERATED

VALVE, SEQUENCE DIRECTLY OPERATED

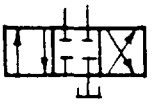
VALVE, PRESSURE REDUCING

VALVE, SHUT OFF 2 POSITION-2 CONNECTION

VALVE, DIRECTIONAL 2 POSITION-3 CONNECTION

VALVE, DIRECTIONAL 2 POSITION-4 CONNECTION 

VALVE, DIRECTIONAL 3 POSITION-4 CONNECTION OPEN CENTER

VALVE, DIRECTIONAL 3 POSITION-4 CONNECTION CLOSED CENTER 

TYPE OF CONTROL

CONTROL, BASIC SYMBOL

CONTROL, CENTRIFUGAL

CONTROL, COMPENSATOR

CONTROL, COMPENSATOR PRESSURE

CONTROL, COMPENSATOR TEMPERATURE

CONTROL, CYLINDER

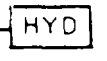
CONTROL, DETENT

CONTROL, MANUAL

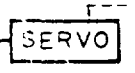
CONTROL, MECHANICAL

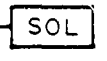
CONTROL, MOTOR ELECTRIC

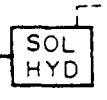
CONTROL, MOTOR HYDRAULIC

CONTROL, PILOT HYDRAULIC 

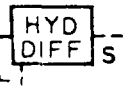
CONTROL, PILOT AIR

CONTROL, SERVO 

CONTROL, SOLENOID 

CONTROL, SOLENOID HYD. PILOT OPERATED 

CONTROL, THERMAL

CONTROL, PILOT HYD. DIFFERENTIAL AREA 

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MISCELLANEOUS UNITS

MOTOR, ELECTRIC

HEAT EXCHANGER

INTENSIFIER

ACCUMULATOR

FILTER

STRAINER

PRESSURE SWITCH

PRESSURE GAGE

SPRING

COMPONENT ENCLOSURE

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APPENDIX III

AERONAUTICAL MECHANICAL SYMBOLS FOR FLUID POWER DIAGRAMS

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TUBE AND HOSE LINES

BRAKE

DOWN (OR CLOSE)

EMERGENCY PRESSURE

HOSE CONNECTION (RIGID TUBING)

HOSE, FLEXIBLE

RETURN

SUPPLY FLUID (PUMP SUCTION)

SUCTION GRAVITY

SUPPLY PRESSURE

UP (OR OPEN)

VENT

EQUIPMENT

ACCUMULATOR

AIR BOTTLE, EMERGENCY

BRAKE CONTROL

BUNGEE, AIR-OIL

COUPLING, SELF-SEALING

CYLINDER, ACTUATING

DEBOOSTER, BRAKE

FILTER OR STRAINER

FITTING, SWIVEL

GAUGE, PRESSURE

GAUGE AND SNUBBER, PRESSURE

PUMP, HAND

PUMP, POWER DRIVEN

RESERVOIR

VALVE, CHECK, AUTOMATIC

VALVE, CHECK, MANUAL

VALVE, BRAKE CONTROL

VALVE, GUN CHARGER CONTROL

VALVE, PRESSURE REGULATING (UNLOADING) AUTOMATIC

VALVE, PRESSURE REGULATING (UNLOADING) MANUAL

VALVE, RELIEF

VALVE, RESTRICTOR, BOTH WAYS

VALVE, RESTRICTOR, PARTIAL ONE-WAY

VALVE, SHUTTLE

VALVE OR SELECTOR, DIRECTIONAL CONTROL

ALPHABETIC CODE

A - AIR
B - BRAKE
BC - BRAKE CONTROL
D - DOWN (OR CLOSE)
F - FLUID (LIQUID)
HP - HANDPUMP
P - PRESSURE
R - RETURN
S - SUCTION (OR SUPPLY)
U - UP (OR OPEN)

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