
Fluid Power (Part 2) – Hydraulic Power Units

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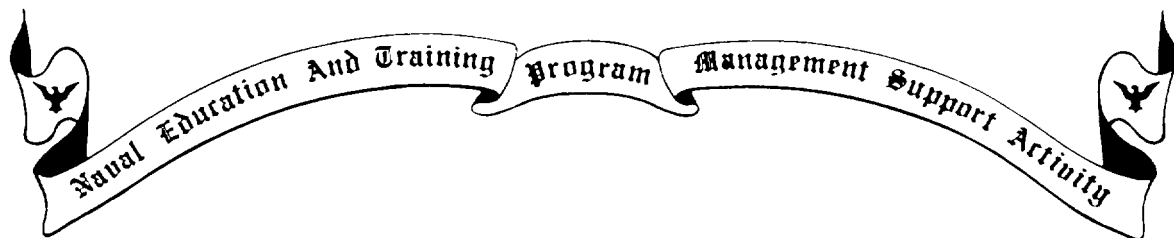
Fluid Power

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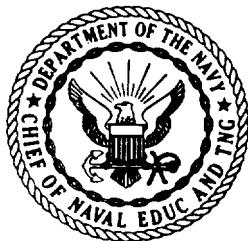


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FLUID POWER

NAVEDTRA 12964



*1990 Edition Prepared by
MWC Albert Beasley, Jr.*



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

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

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.

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

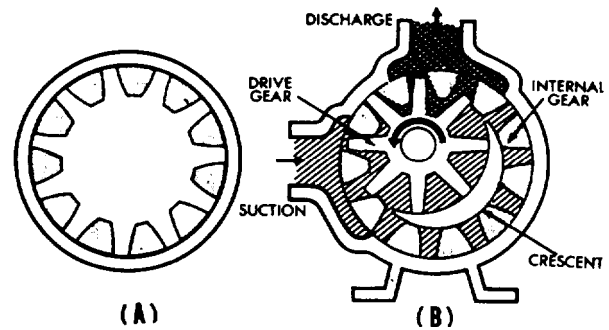


Figure 4-2.—Off-centered internal gear pump.

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

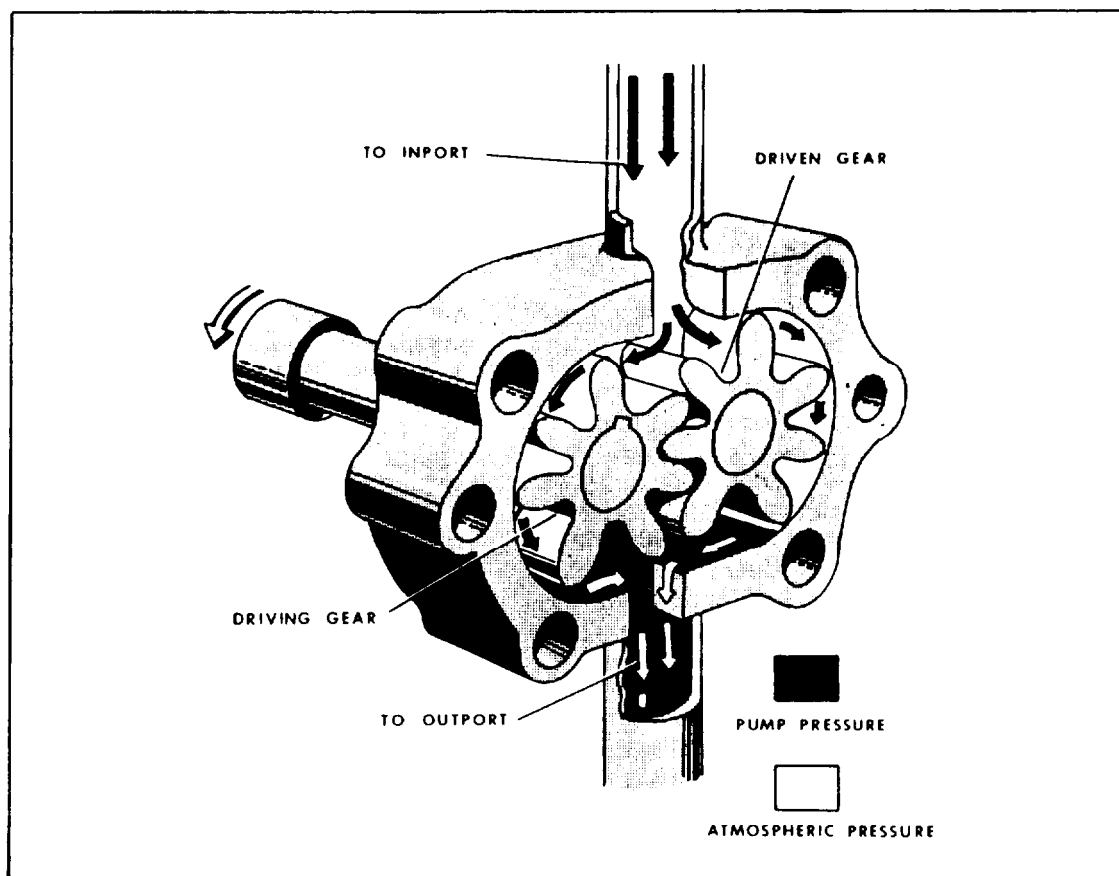


Figure 4-1.—Gear-type rotary pump.

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.

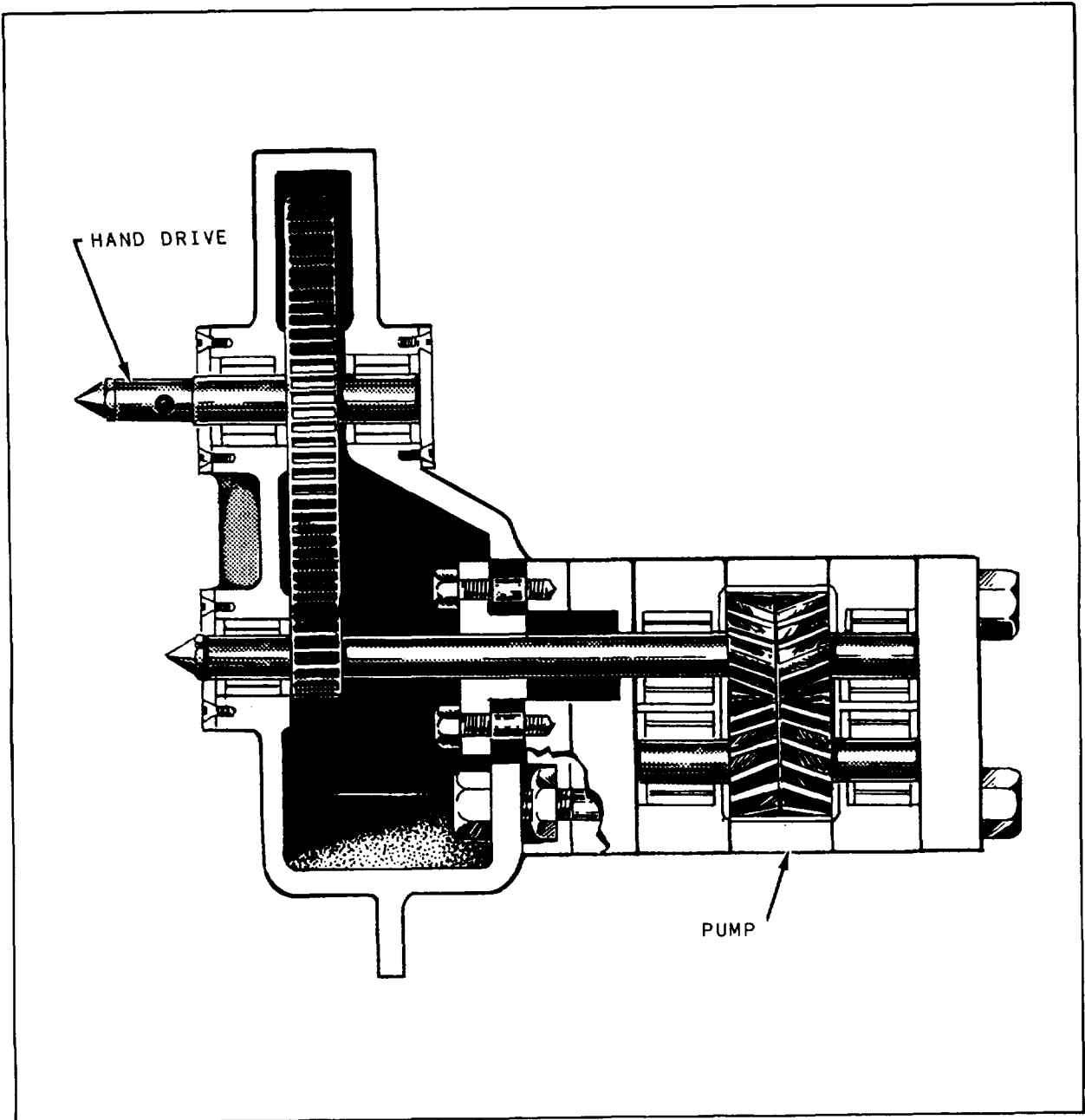


Figure 4-3.—Herringbone gear pump.

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.

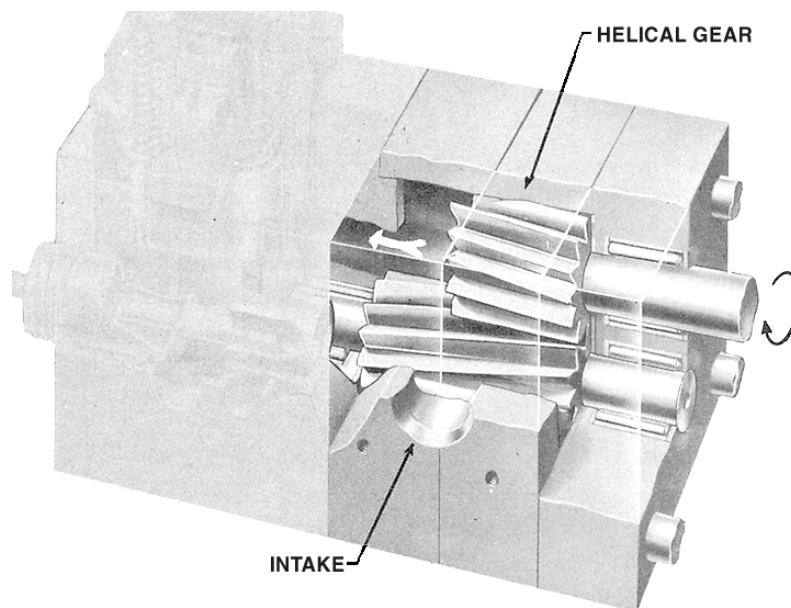


Figure 4-4.—Helical gear pump.

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

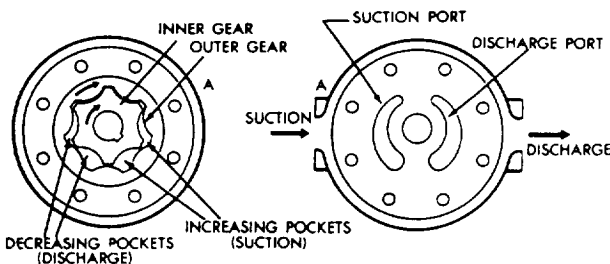


Figure 4-5.—Centered internal gear pump.

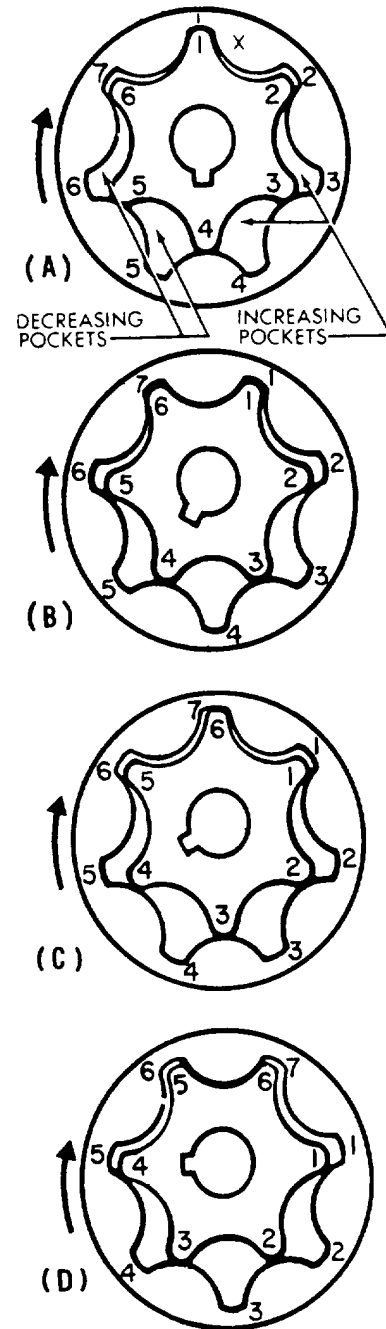


Figure 4-6.—Principles of operation of the internal gear pump.

reversed. Actually, A in one drawing covers A in the other.

LOBE PUMP

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

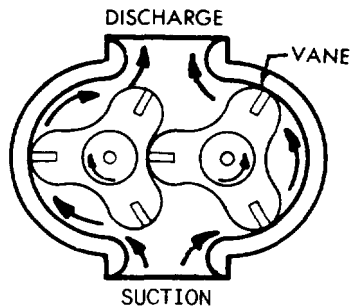


Figure 4-7.—Lobe pump.

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

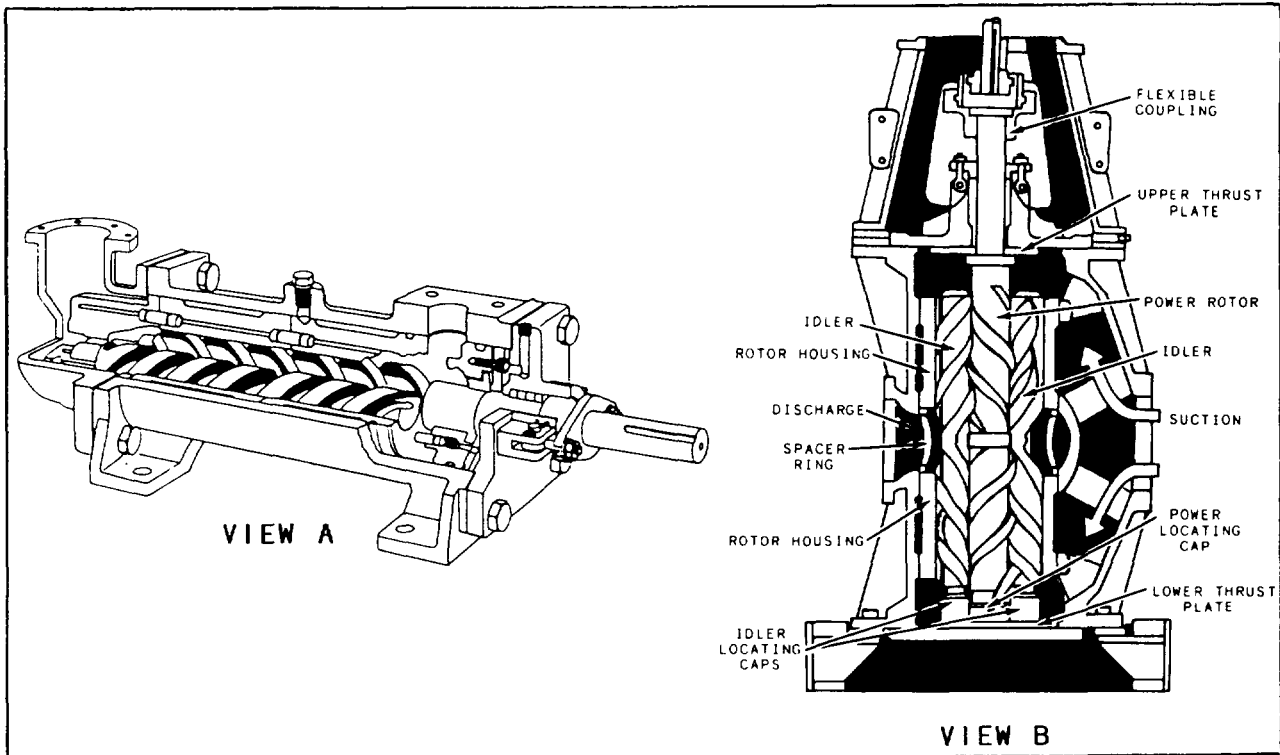


Figure 4-8.—Screw pumps.

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

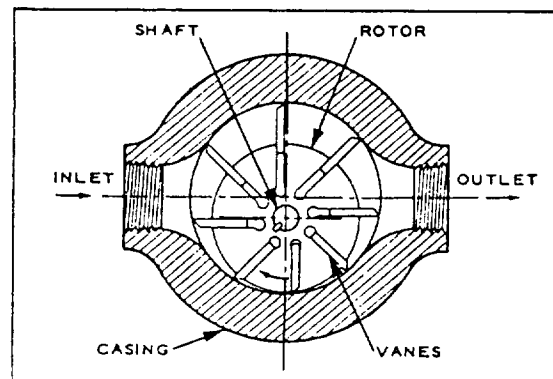


Figure 4-9.—Vane pump.

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

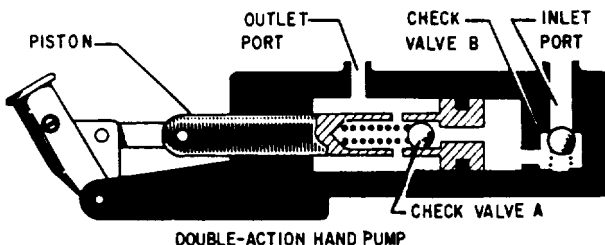


Figure 4-10.—Hydraulic hand pump.

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 reservoir acting on check valve B causes its spring to compress; thus, opening the check valve. This allows liquid to enter the inlet chamber.

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 reservoir, 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.

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

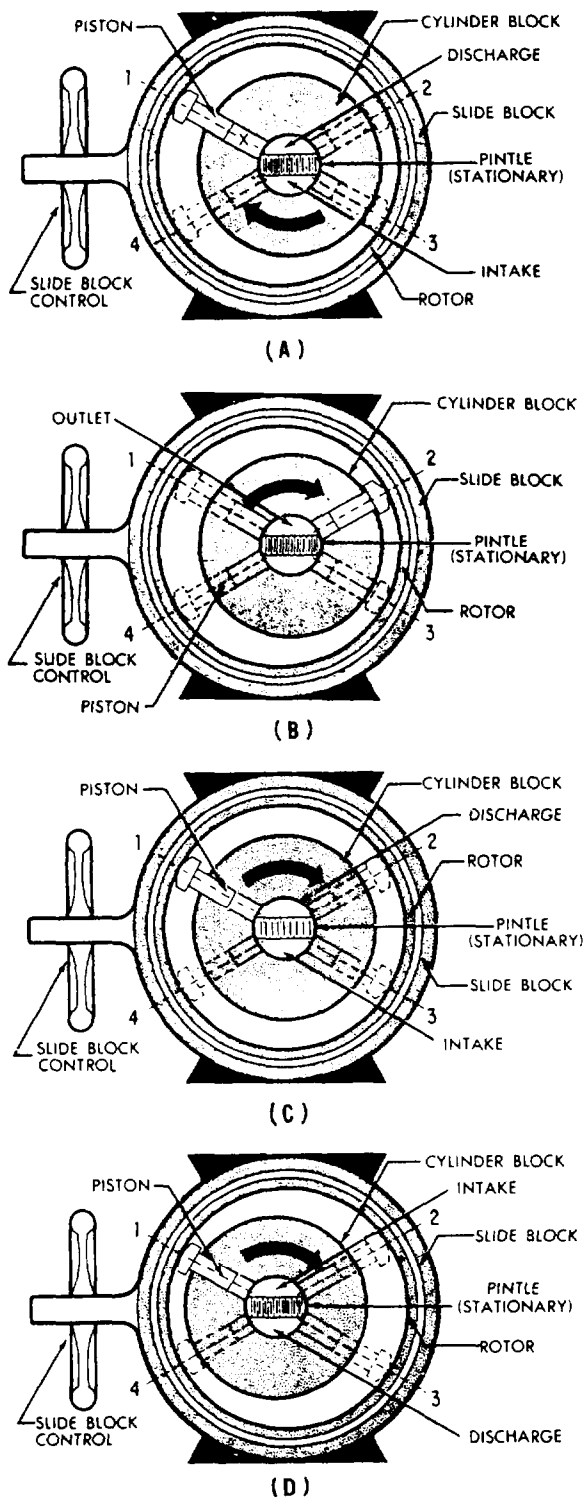


Figure 4-11.—Principles of operation of the radial piston pump.

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.

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.

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.

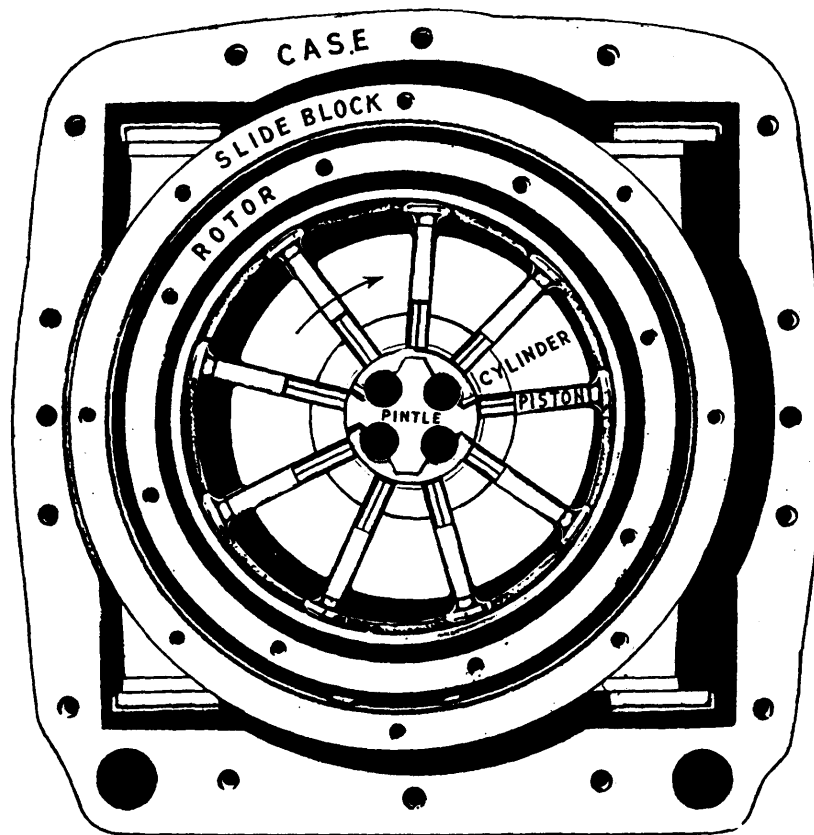


Figure 4-12.—Nine-piston radial piston pump.

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

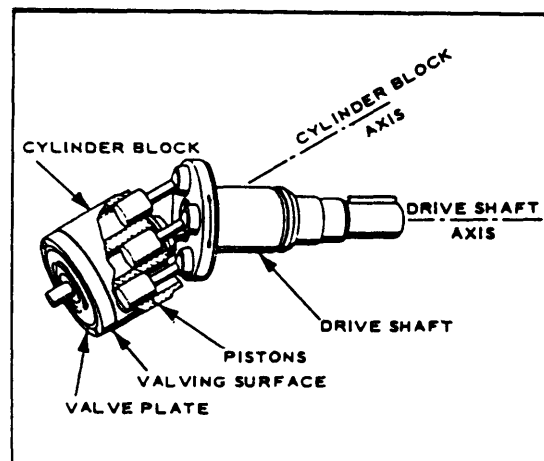


Figure 4-14.—Bent-axis axial piston pump.

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 arc 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.

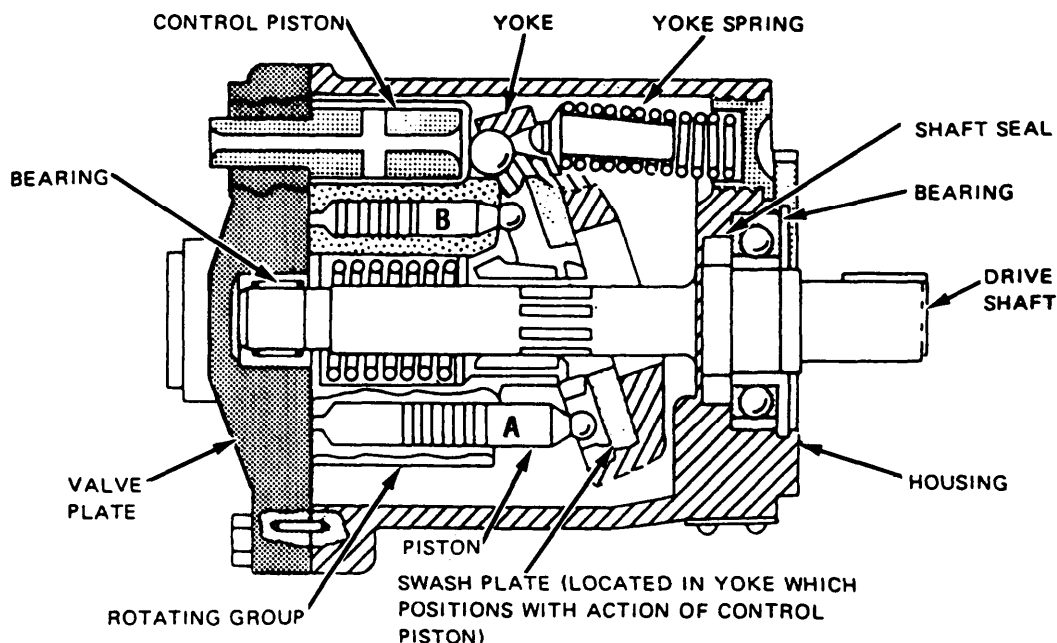


Figure 4-13.—In-line axial piston pump.

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.

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

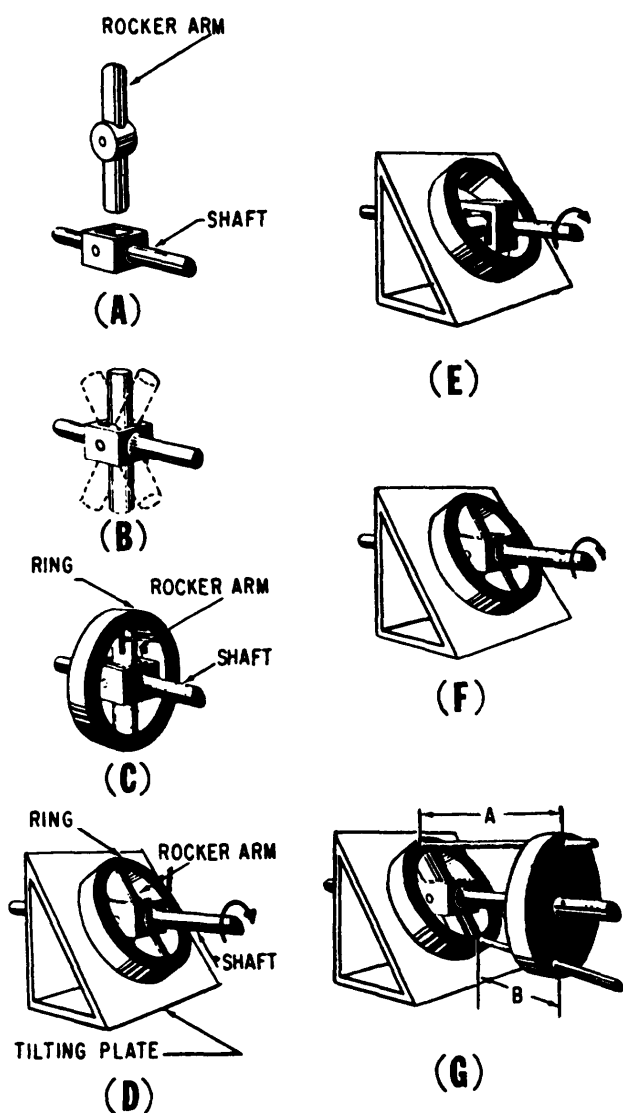


Figure 4-15.—Relationship of the universal joint in operation of the axial piston pump.

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 its 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 fixed center pivot and are held in constant 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

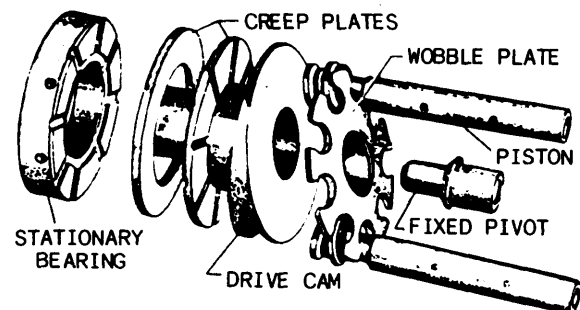


Figure 4-16.—Mechanical drive—Stratopower pump.

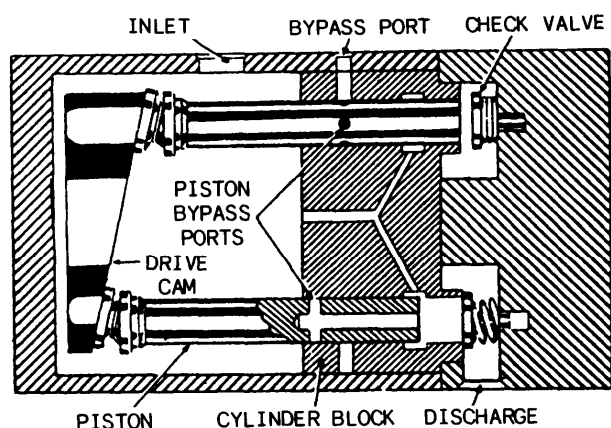


Figure 4-17.—Fluid displacement—Stratopower pump.

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.

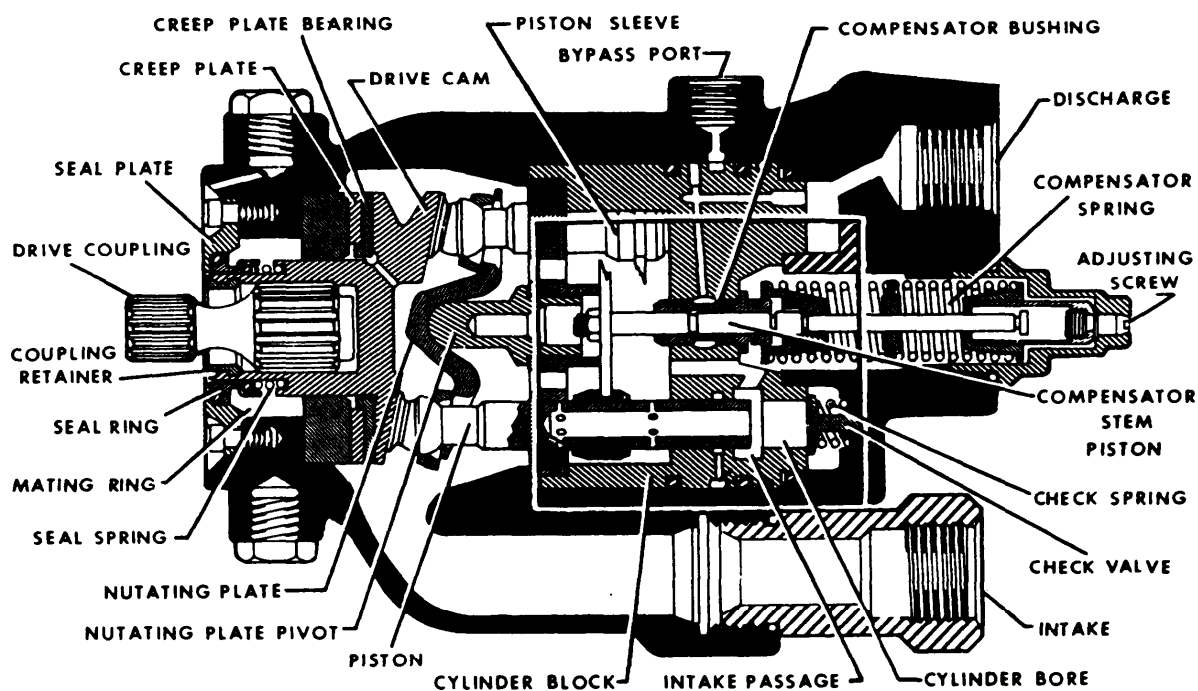


Figure 4-18.—Internal features of Stratopower variable-displacement pump.

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

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 ($\frac{1.315 - 1.049}{2}$), 0.179 ($\frac{1.315 - 0.957}{2}$), and 0.250 ($\frac{1.315 - 0.815}{2}$) 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

Table 5-2.—Tubing Size Designation

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

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

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.

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

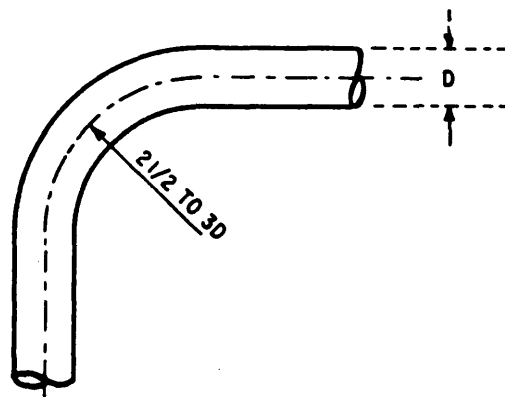


Figure 5-2.—Ideal bend radius.

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.

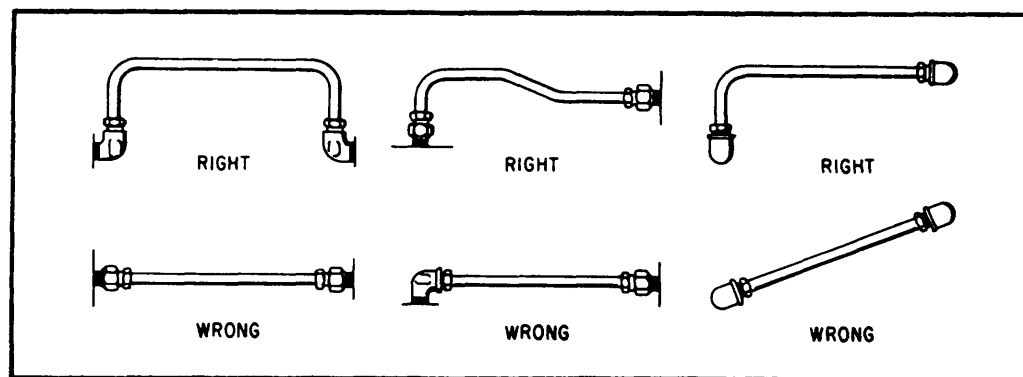


Figure 5-1.—Correct and incorrect methods of installing tubing.

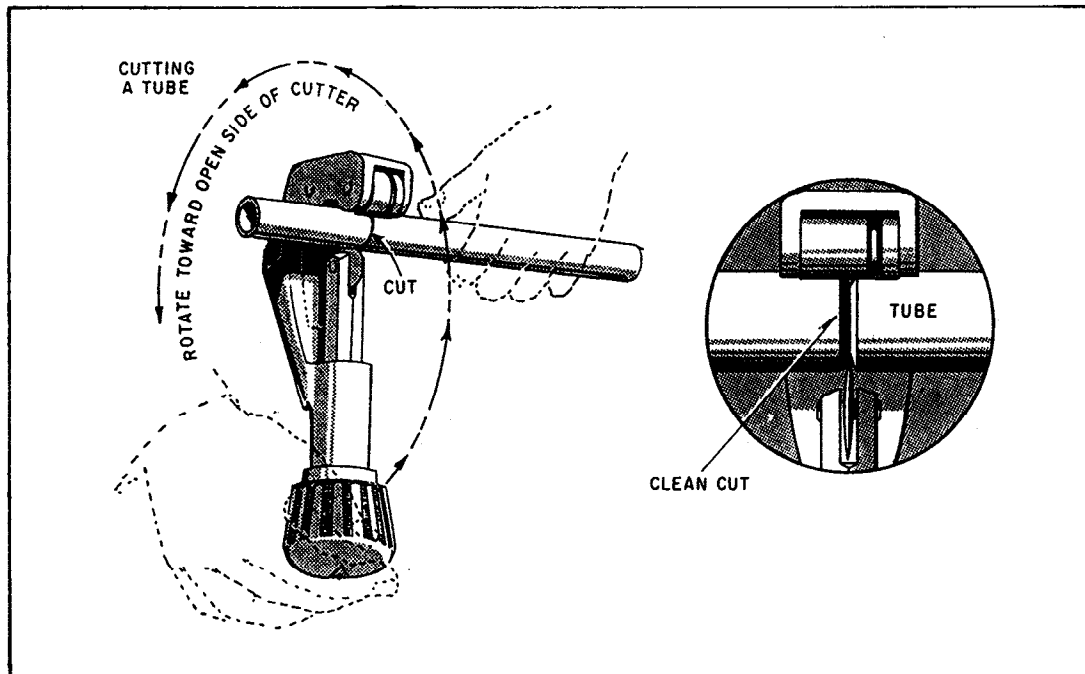


Figure 5-3.—Tube cutting.

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.

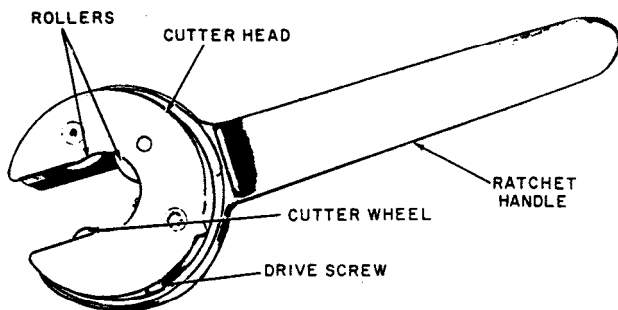


Figure 5-4.—Chipless cutter.

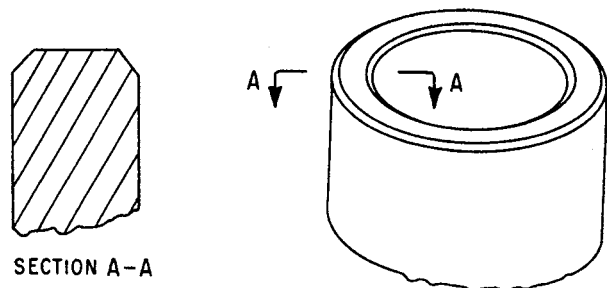


Figure 5-5.—Properly burred tubing.

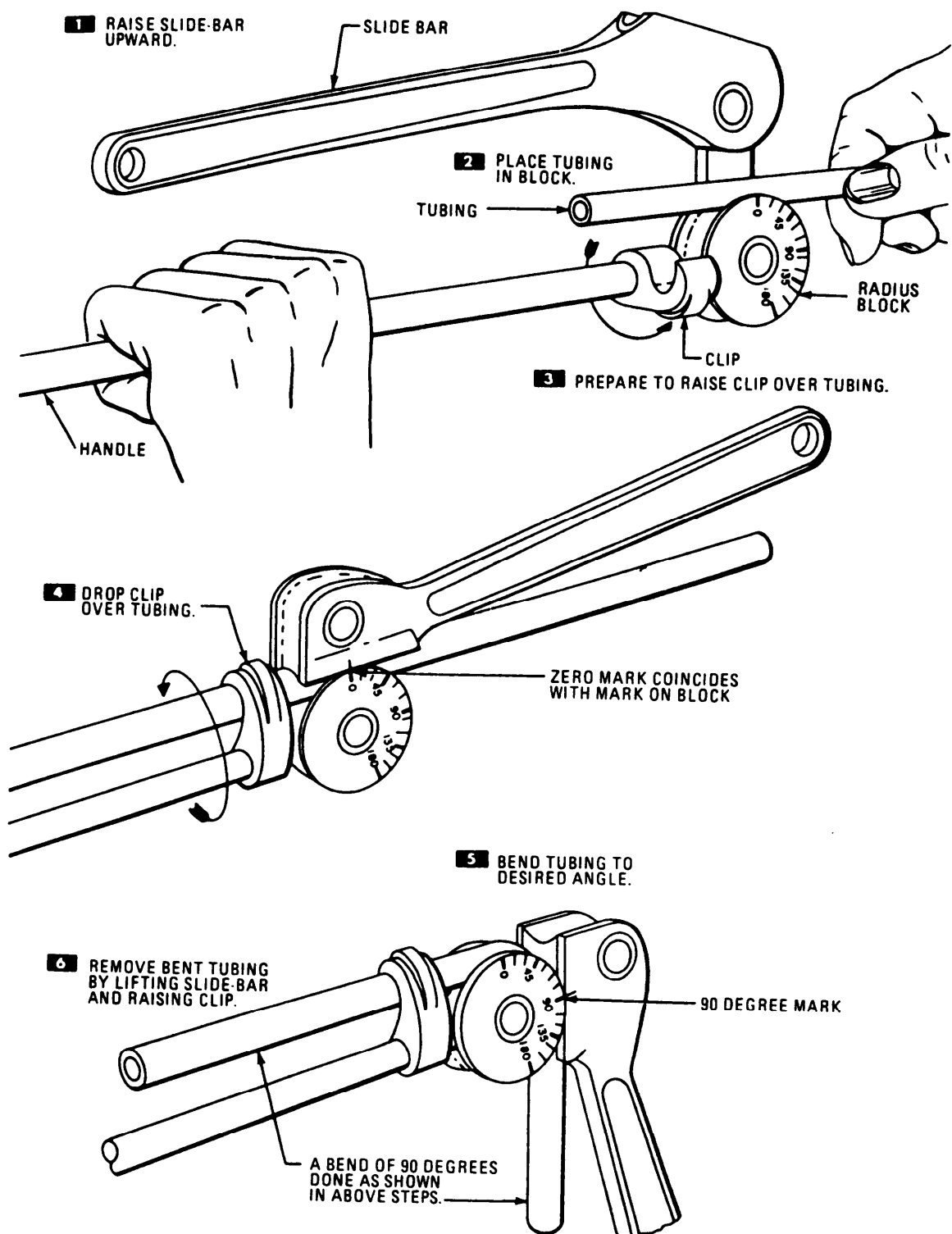


Figure 5-6.—Bending tubing with hand-operated tube bender.

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

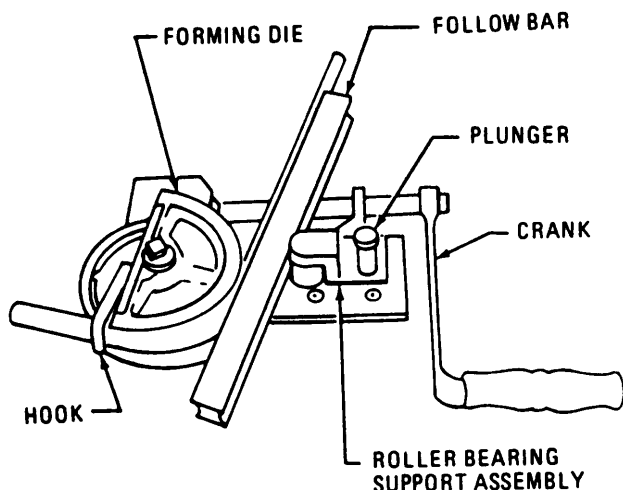


Figure 5-7.—Mechanically operated tube bender.

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.

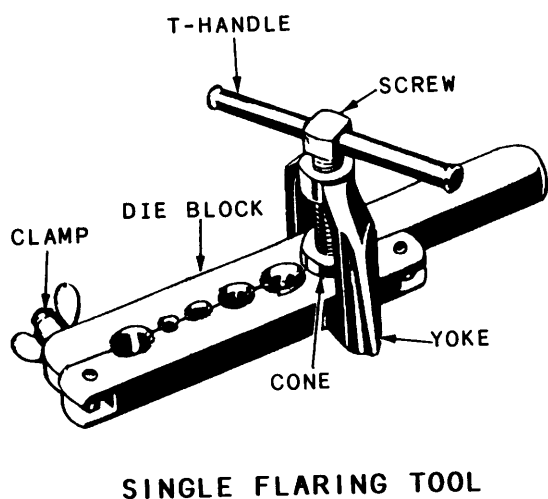
This type of tube bender uses a hand crank and gears. The forming die is keyed to the drive gear and is secured by a screw.

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.

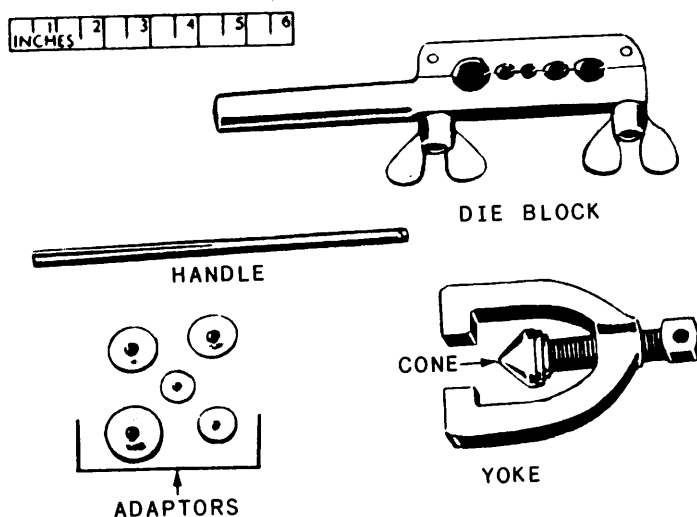
Tube Flaring

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,



SINGLE FLARING TOOL



DOUBLE FLARING TOOL

Figure 5-8.—Flaring tools.

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

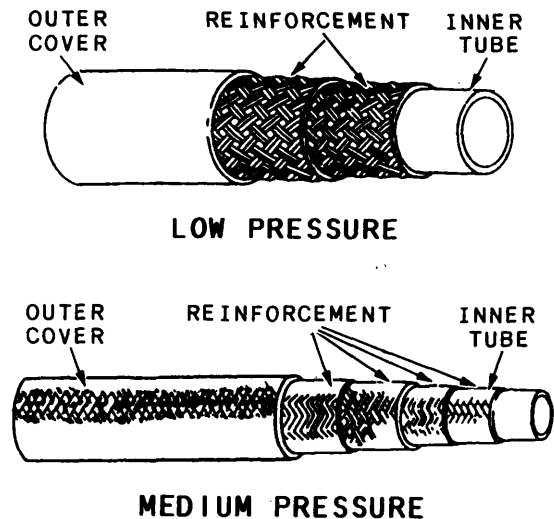


Figure 5-9.—Synthetic rubber hoses.

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.

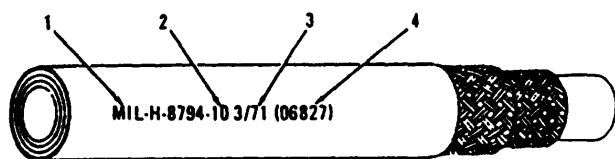
Synthetic rubber hoses with wire braid cover are identified by bands (fig. 5-10, view B) wrapped around the hose ends and at intervals along the length of the hose.

Sizing

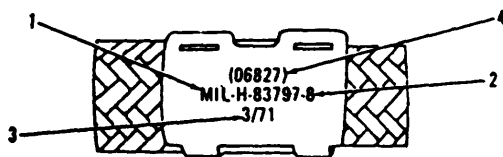
The size of a flexible hose is identified by the dash (-) number, which is the ID of the hose expressed in 16ths of an inch. For example, the ID of a -64 hose is 4 inches. For a few hose styles this is the nominal and not the true ID.

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,



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.

PTFE

PTFE 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 PTFE 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.

PTFE 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,

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.

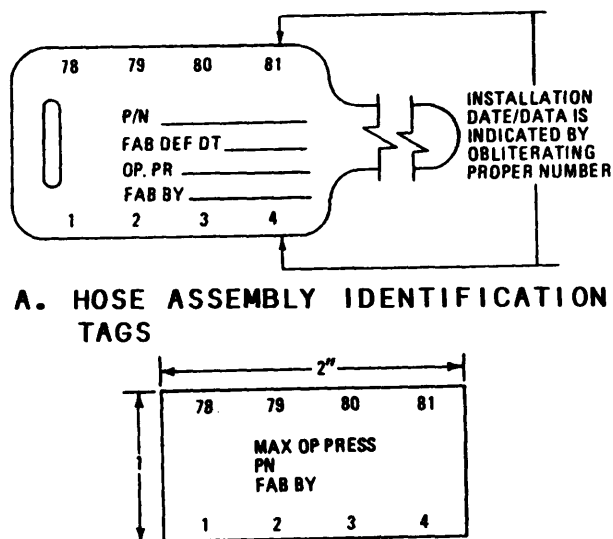
HOSE ASSEMBLY IDENTIFICATION TAG (SHIP _____)	
SRD DWG NO. _____	SYST. PRESSURE _____ PSI
SRP ITEM NO. _____	START SERVICE
HOSE TYPE/SIZE _____	DATE _____
SERVICE _____	

ID TAG WHEN SELECTED RECORD DRAWING IS AVAILABLE

HOSE ASSEMBLY IDENTIFICATION TAG (SHIP _____)	
PIPING ARR. DWG. NO. _____	SYST. PRESSURE _____ PSI
ASSY. PC. NO. _____	START SERVICE
HOSE TYPE/SIZE _____	DATE _____
SERVICE _____	

ID TAG WHEN SELECTED RECORD DRAWING DOES NOT EXIST

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



(MATERIAL: WHITE POLYESTER FILM PER MIL-P-38477.)

B. HOSE ASSEMBLY LABELS

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

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

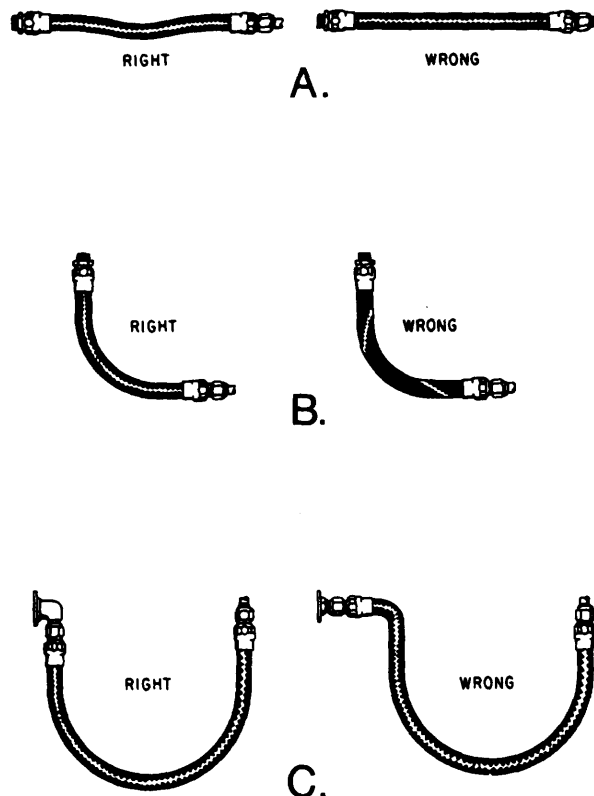


Figure 5-13.—Correct and incorrect installation of flexible hose.

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 pre-formed 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.

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

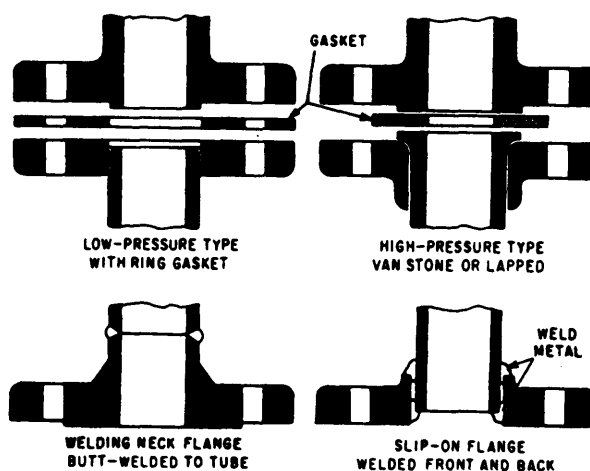


Figure 5-14.—Four types of bolted flange connectors.

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

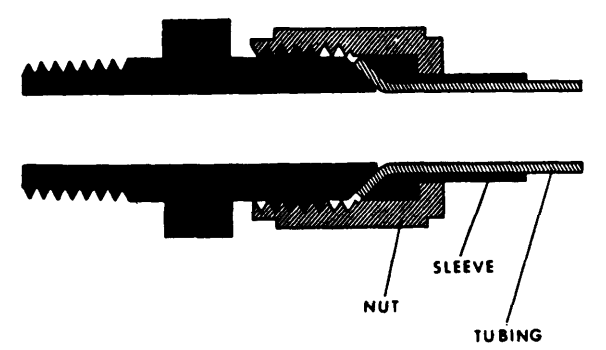


Figure 5-15.—Flared-tube fitting.

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

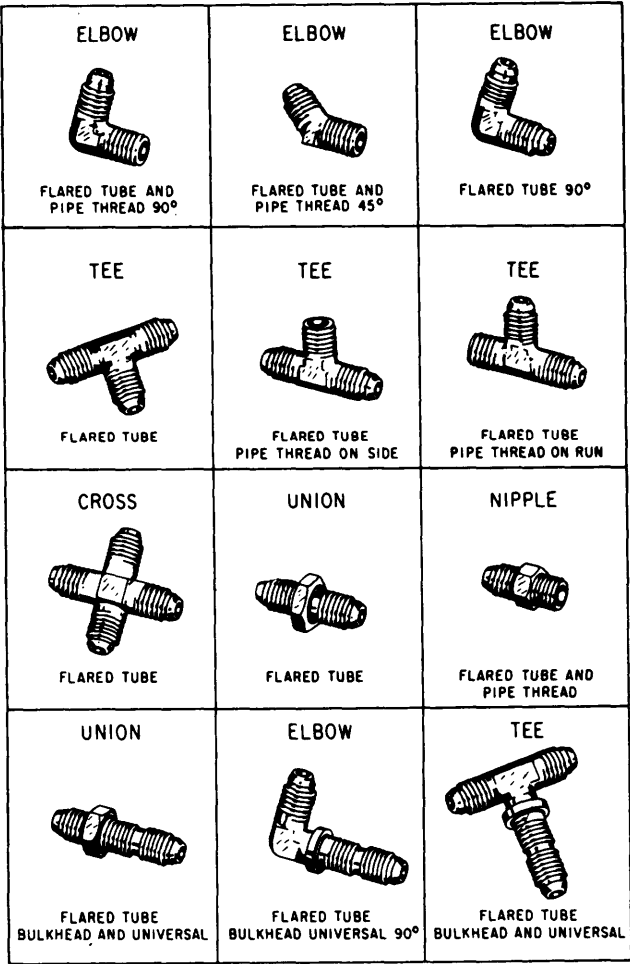


Figure 5-16.—Flared-tube fittings.

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 because 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.

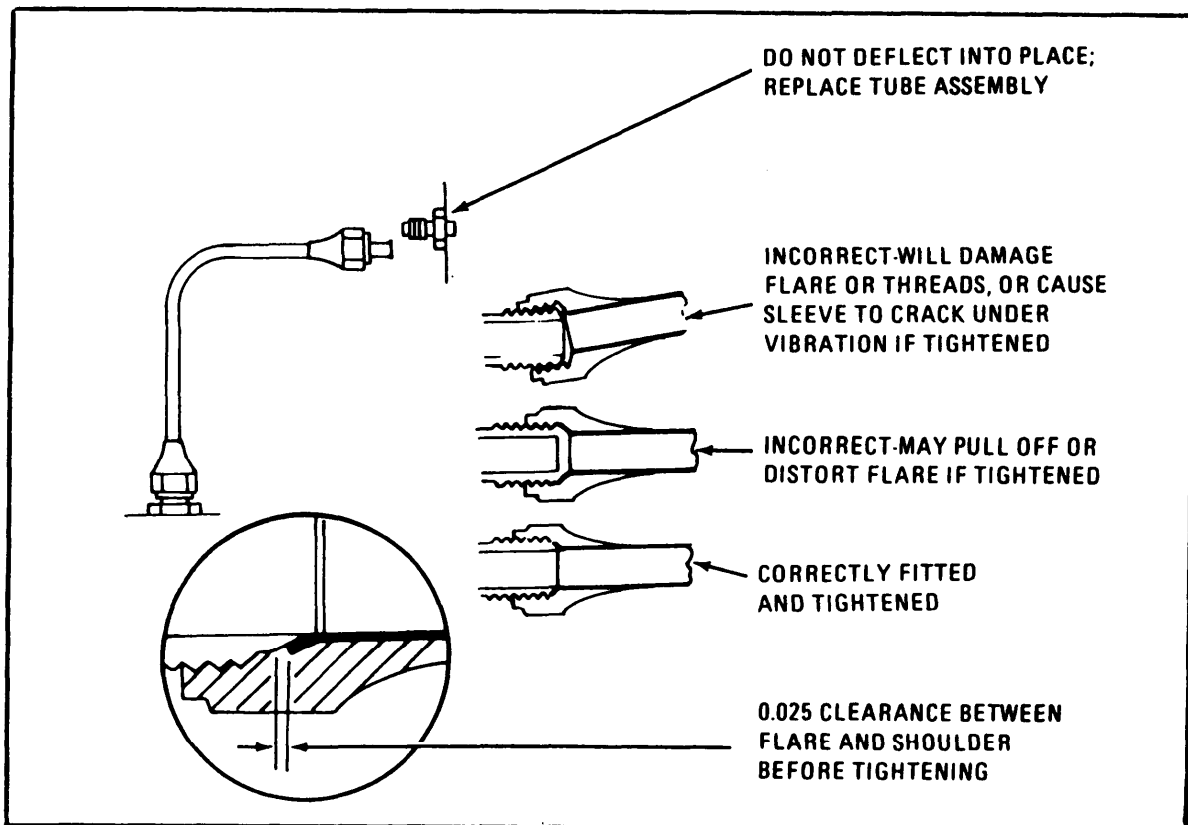


Figure 5-17.—Correct and incorrect methods of installing flared fittings.

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,

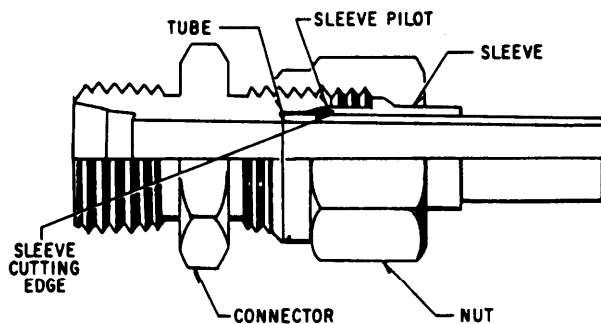


Figure 5-18.—Flareless-tube connector.

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

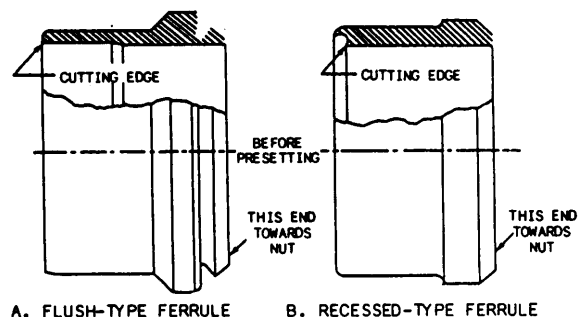


Figure 5-19.—Unused ferrules.

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.

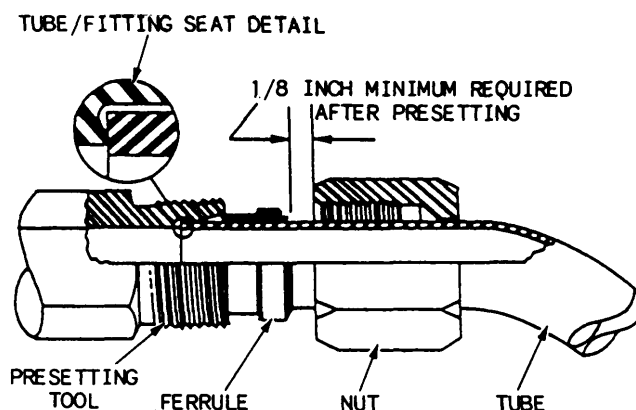


Figure 5-20.—Tube and ferrule assembled for presetting, showing nut position required for inspecting ferrule.

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)

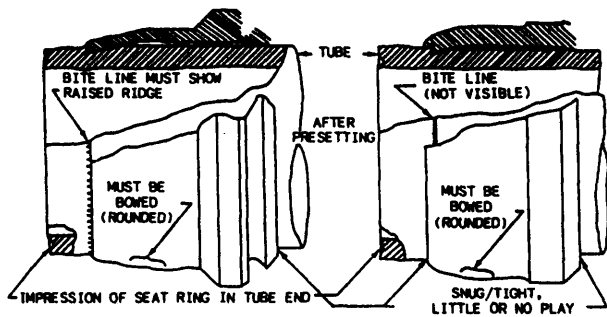


Figure 5-21.—Ferrules installed on tube, preset and removed for inspection.

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

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.

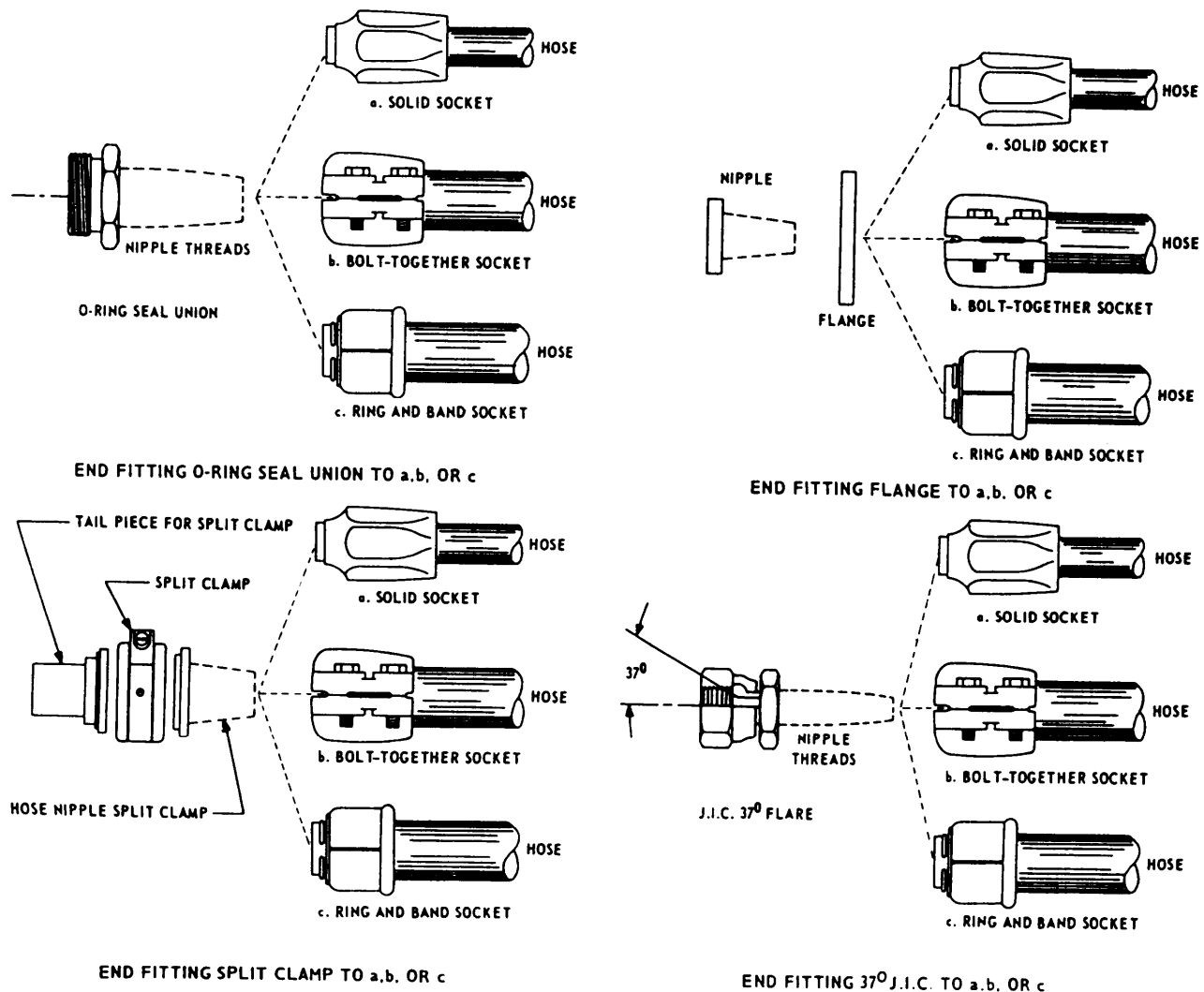


Figure 5-22.—End fittings and hose fittings.

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

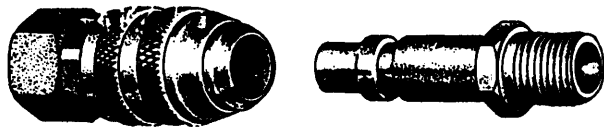


Figure 5-23.—Quick-disconnect coupling for air lines.

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.

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.

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.

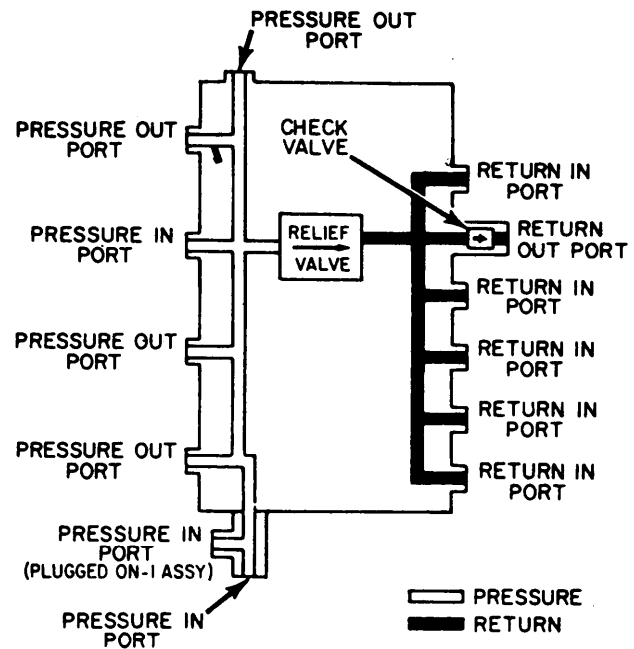


Figure 5-25.—Fluid manifold—flow diagram.

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

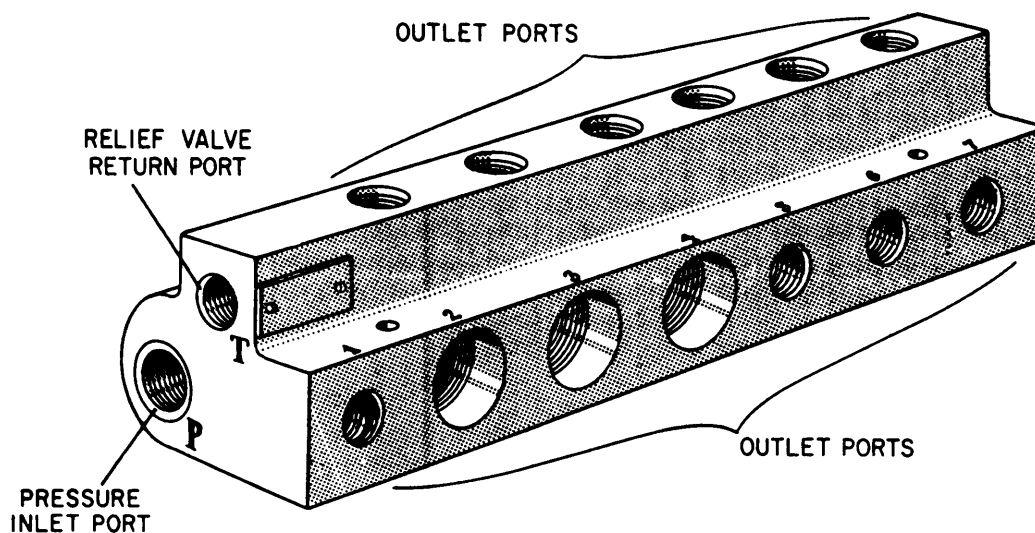


Figure 5-24.—Fluid manifold.

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.

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

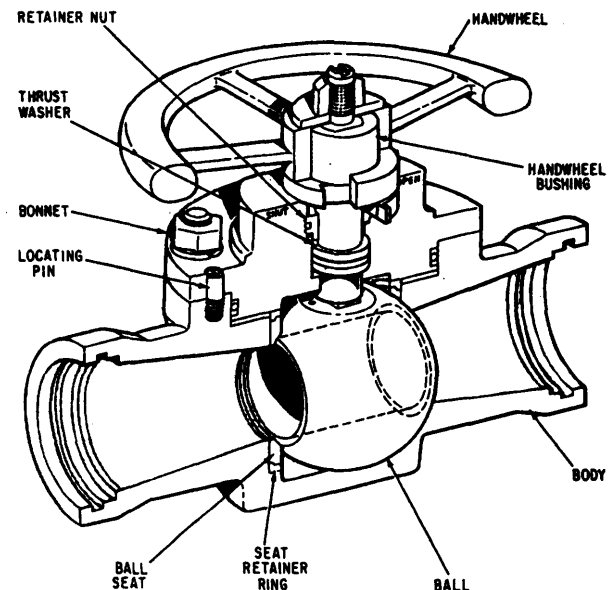


Figure 6-1.—Typical ball valve.

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).

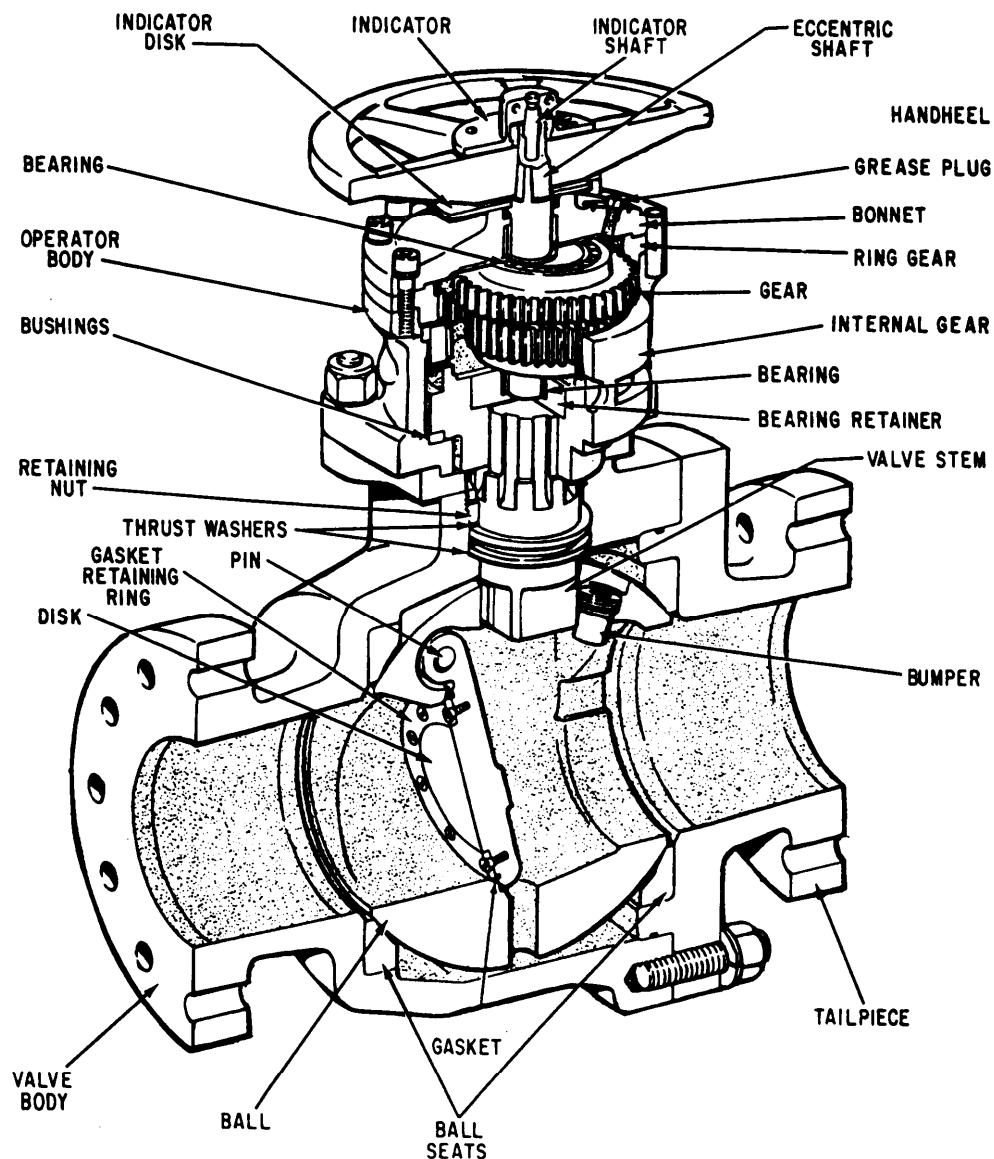


Figure 6-2.—Typical ball-stop, swing-check valve.

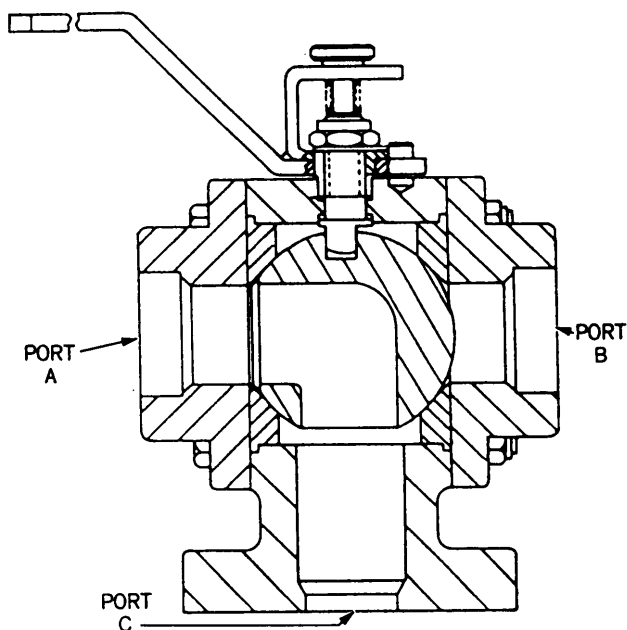


Figure 6-3.—Three-way ball valve.

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

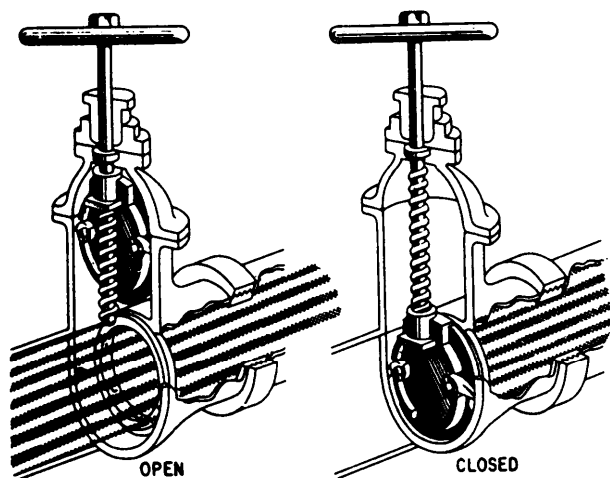


Figure 6-4.—Operation of a gate valve.

(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.

GLOBE VALVES

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

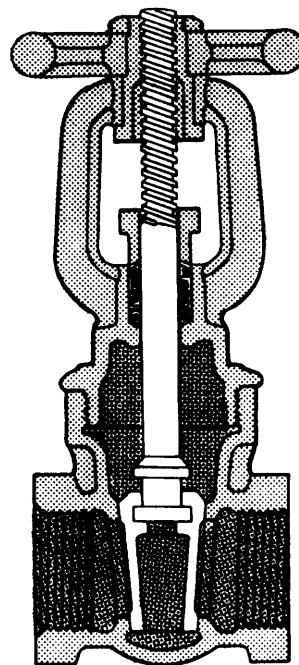


Figure 6-5.—Rising stem gate valve.

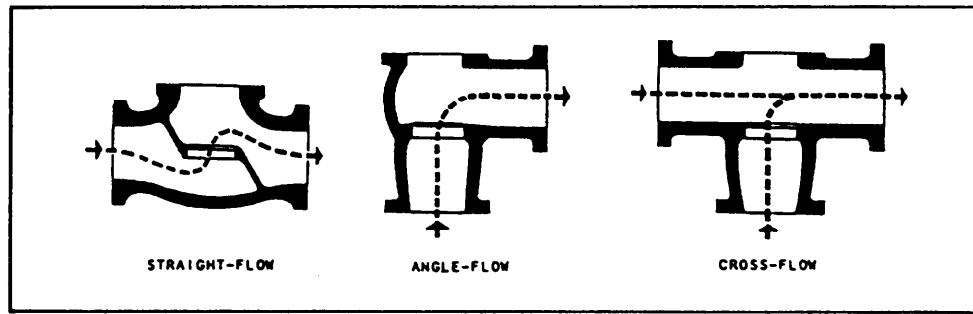


Figure 6-6.—Types of globe valve bodies.

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 form a 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

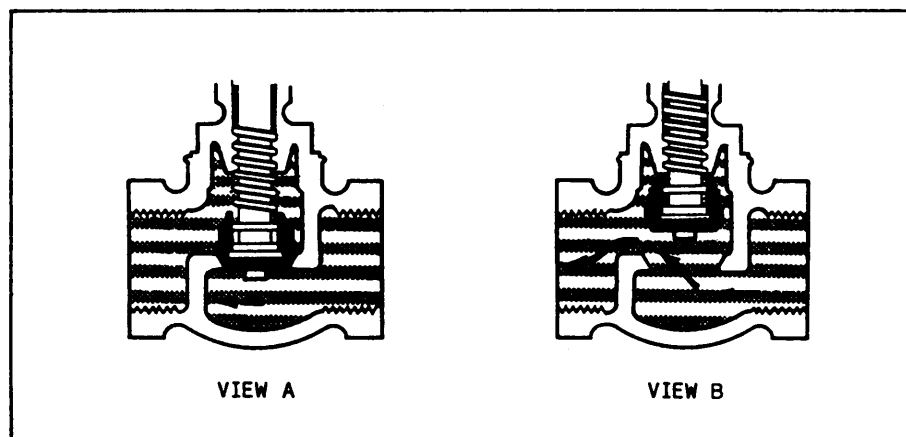


Figure 6-7.—Operation of a globe valve.

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

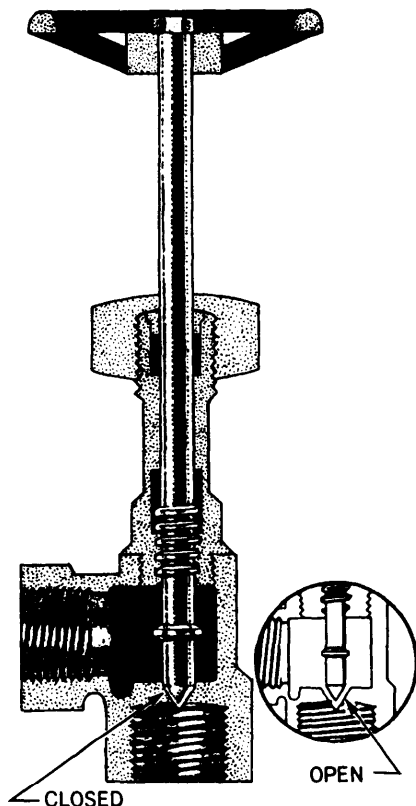


Figure 6-8.—Cross-sectional view of a needle valve.

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).

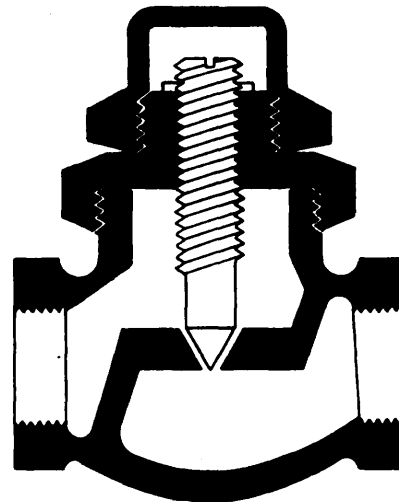


Figure 6-9.—Variable restrictor.

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 over-pressurization. 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.

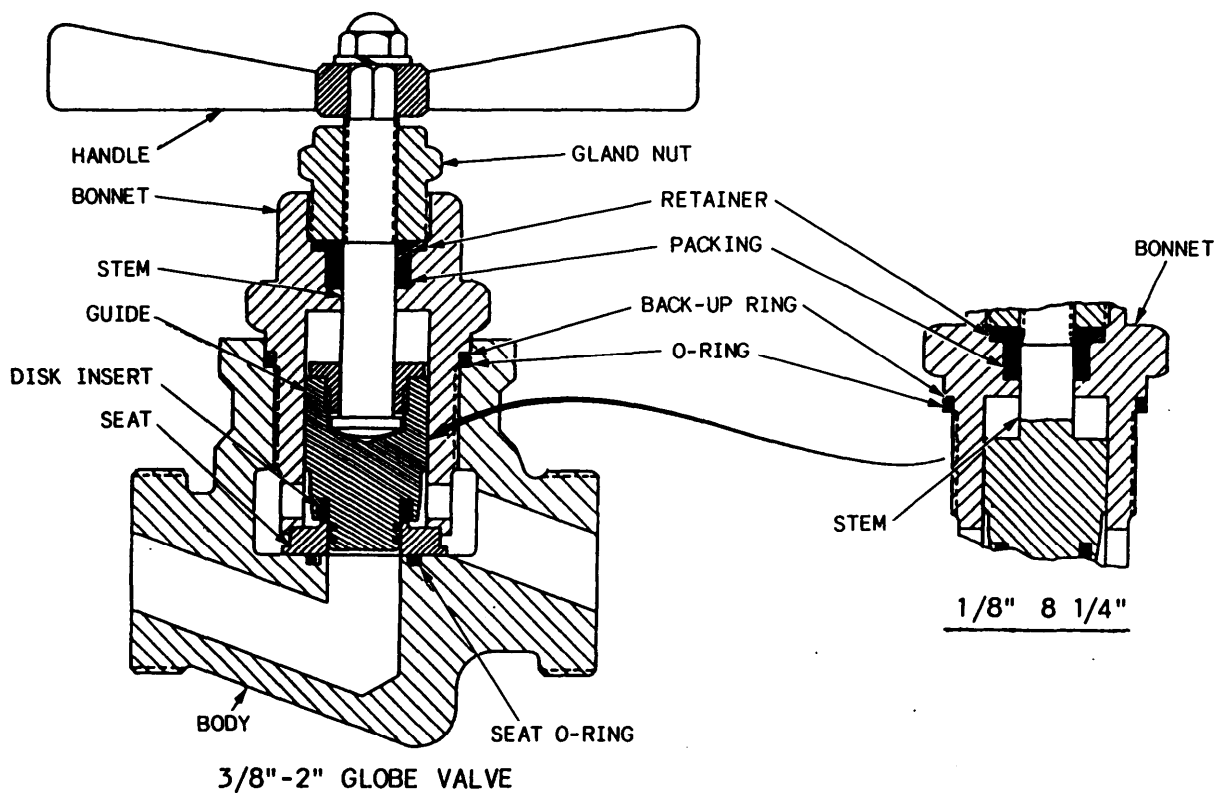


Figure 6-10.—Hydraulic and pneumatic globe valve (rising stem).

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

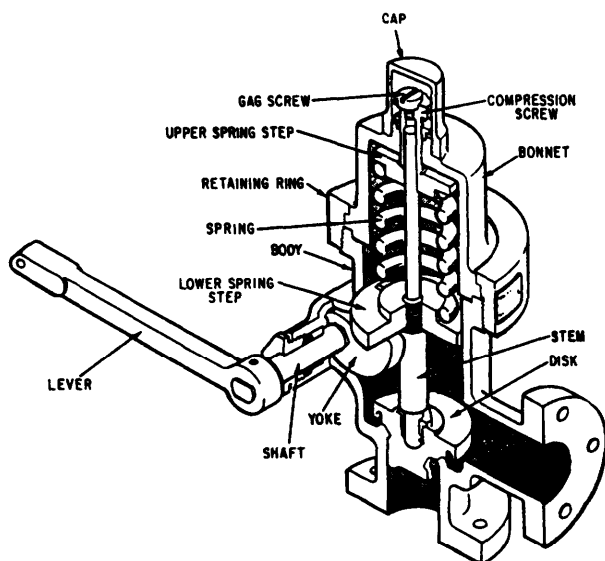


Figure 6-11.—Relief valve.

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

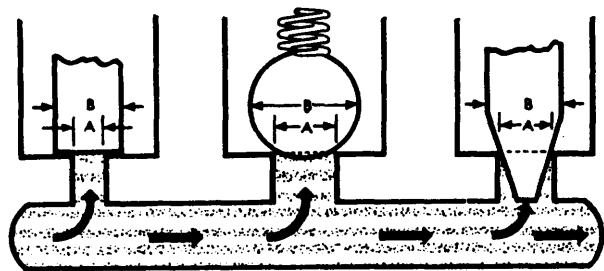


Figure 6-12.—Pressure acting on different areas.

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

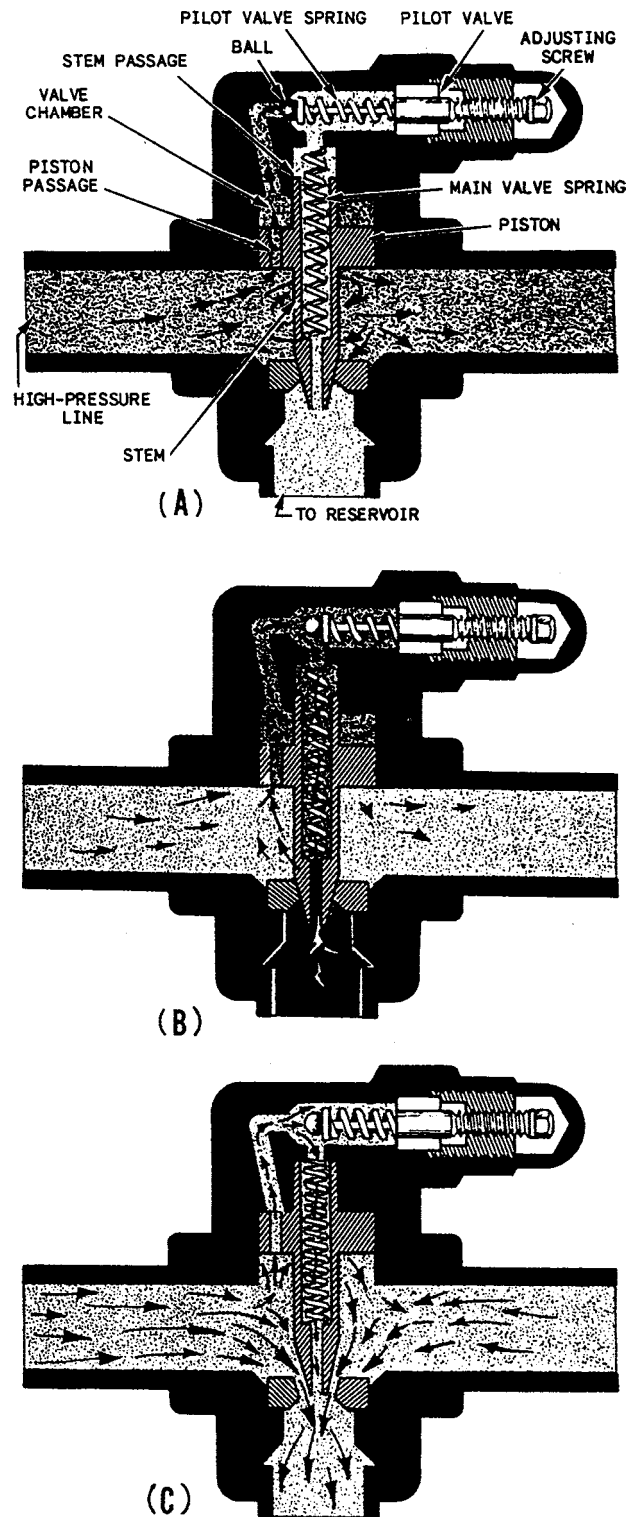


Figure 6-13.—Operation of compound relief valve,

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

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.

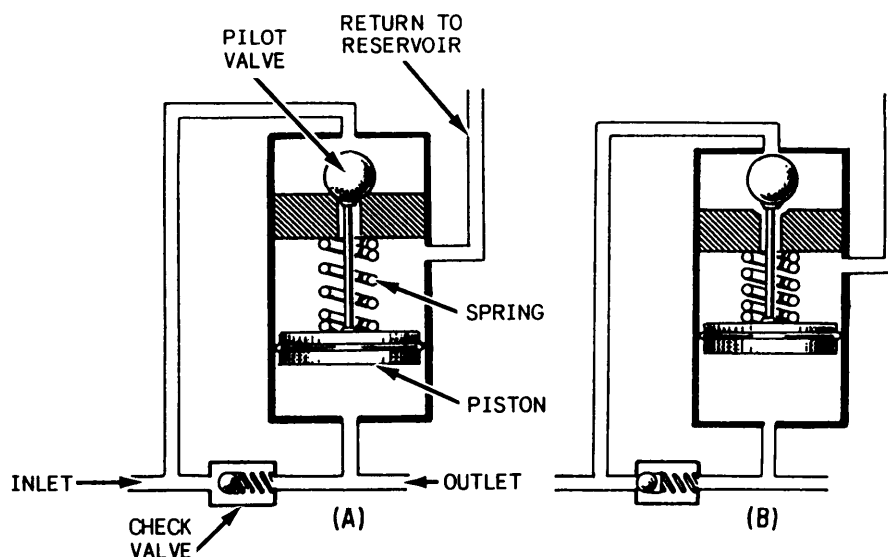


Figure 6-14.—Hydraulic pressure regulator.

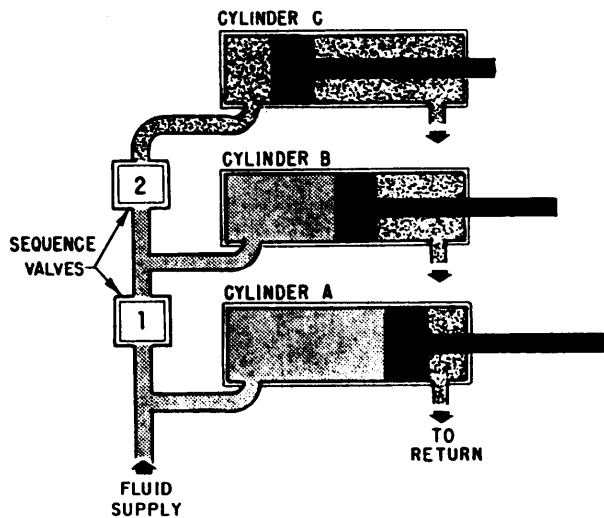


Figure 6-15.—Installation of sequence valves.

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.

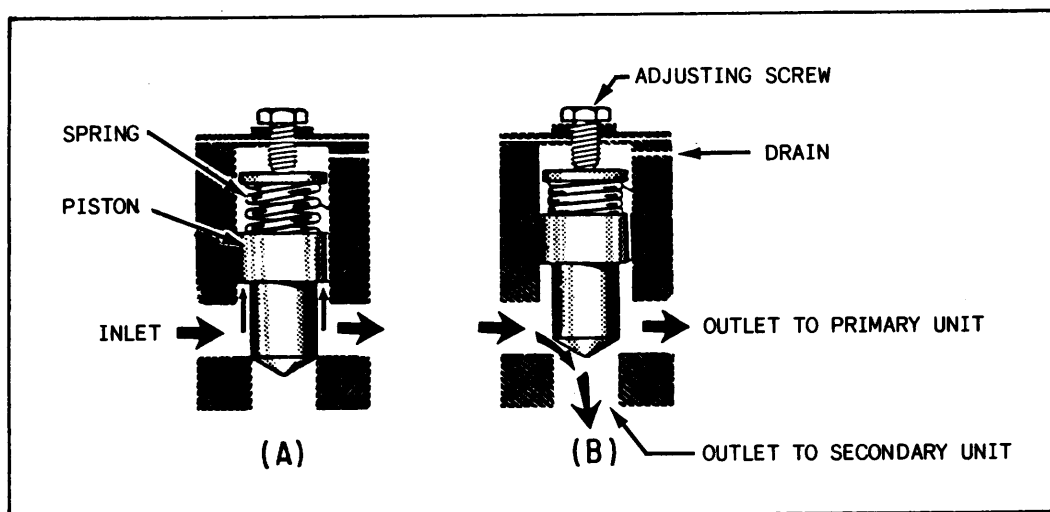


Figure 6-16.—Operation of a pressure-controlled sequence valve.

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

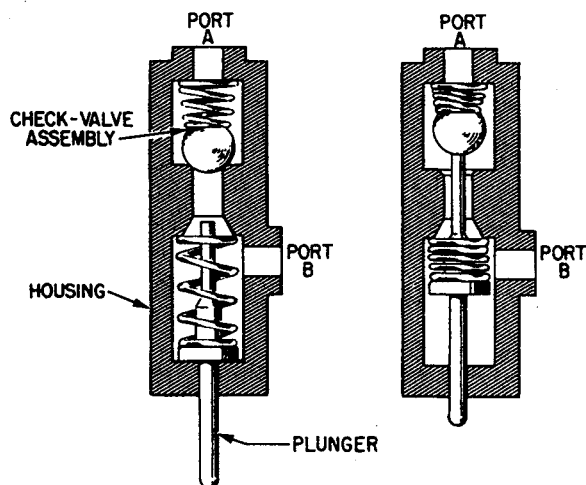


Figure 6-17.—Mechanically operated sequence valve.

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.

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.

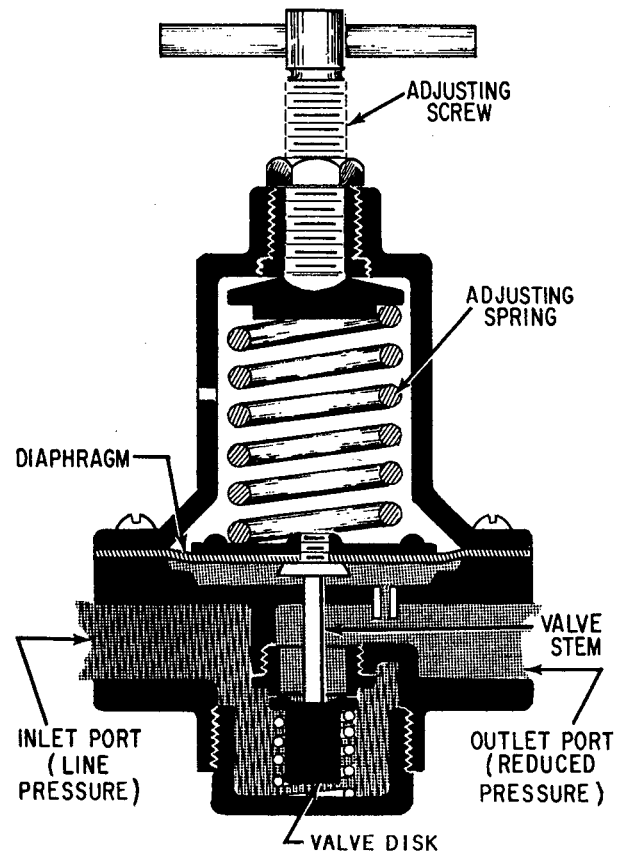


Figure 6-18.—Spring-loaded pressure-reducing valve.

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

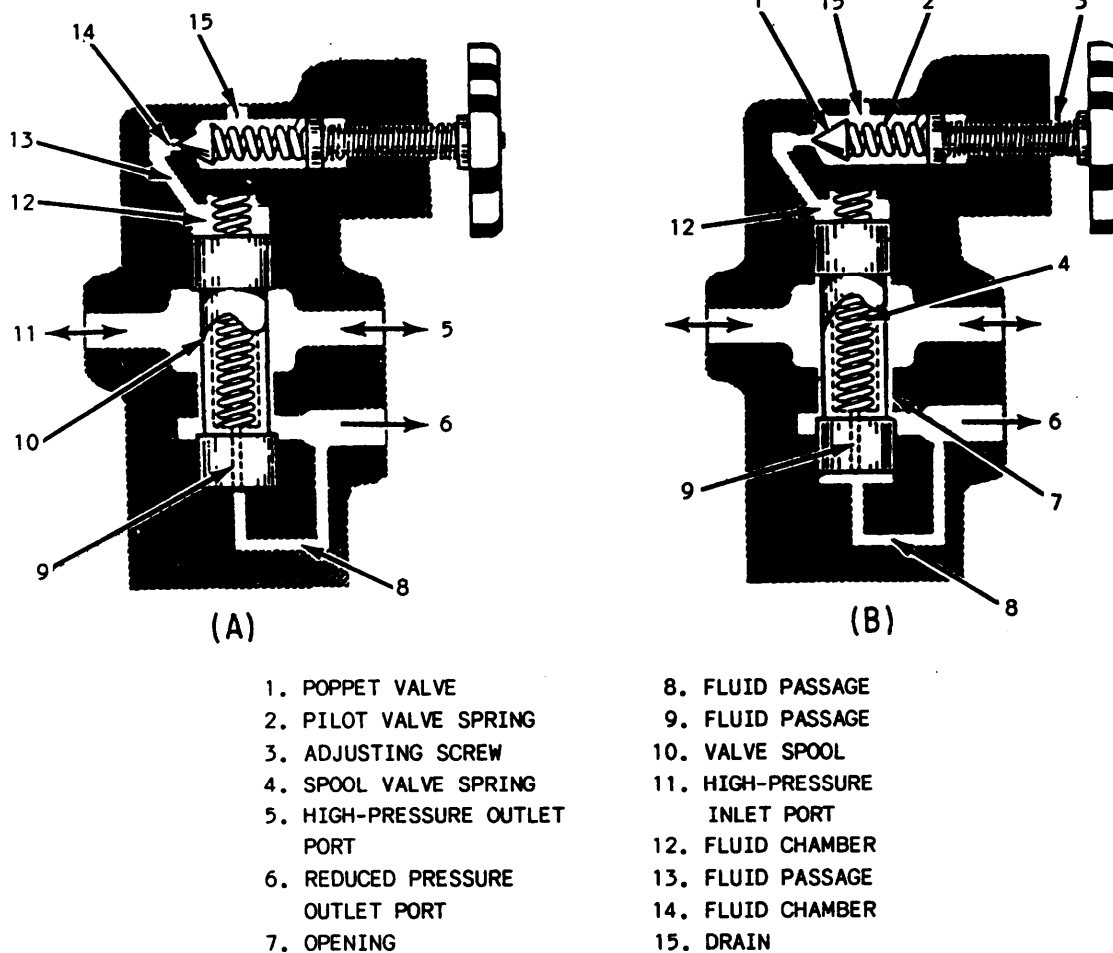


Figure 6-19.—Pilot-controlled pressure-reducing valve.

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

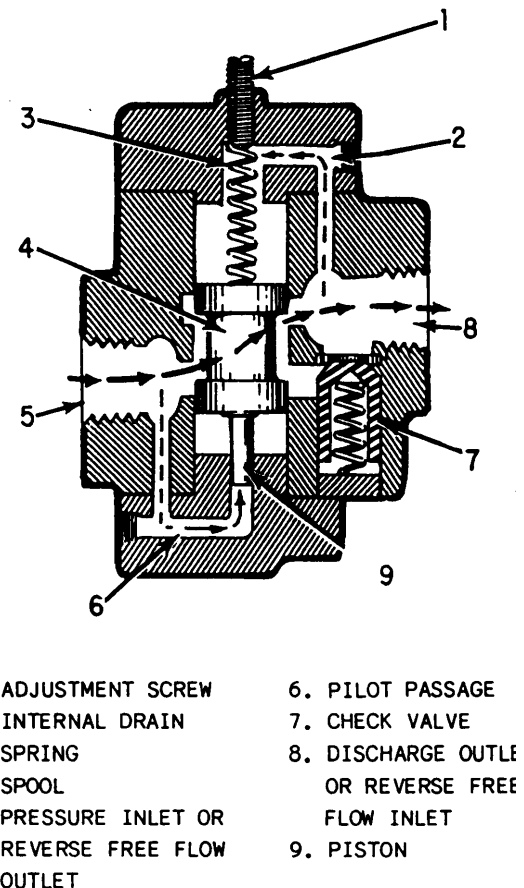


Figure 6-20.—Counterbalance valve.

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 may be 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

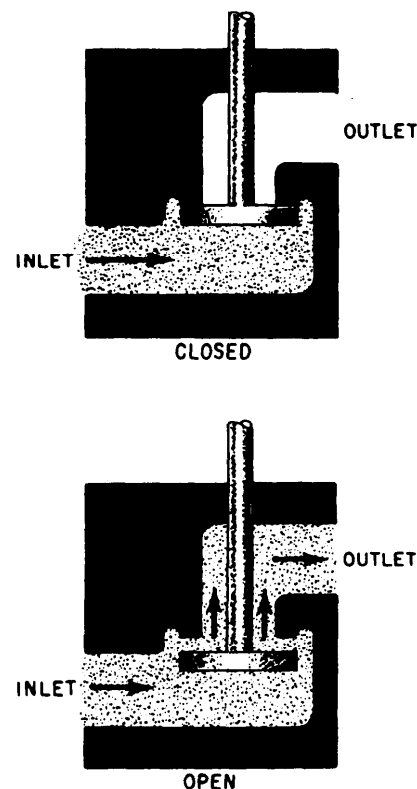


Figure 6-21.—Operation of a simple poppet valve.

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

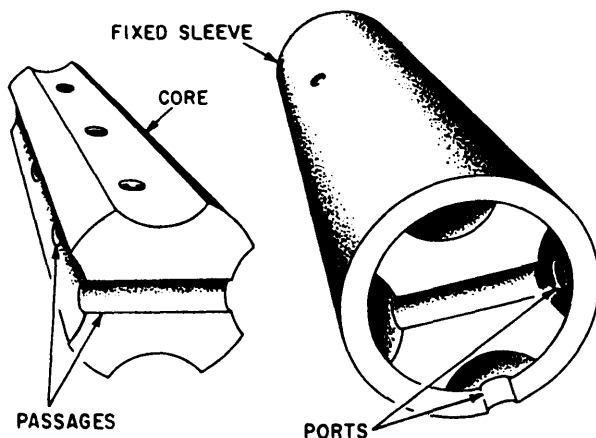


Figure 6-22.—Parts of a rotary spool directional control valve.

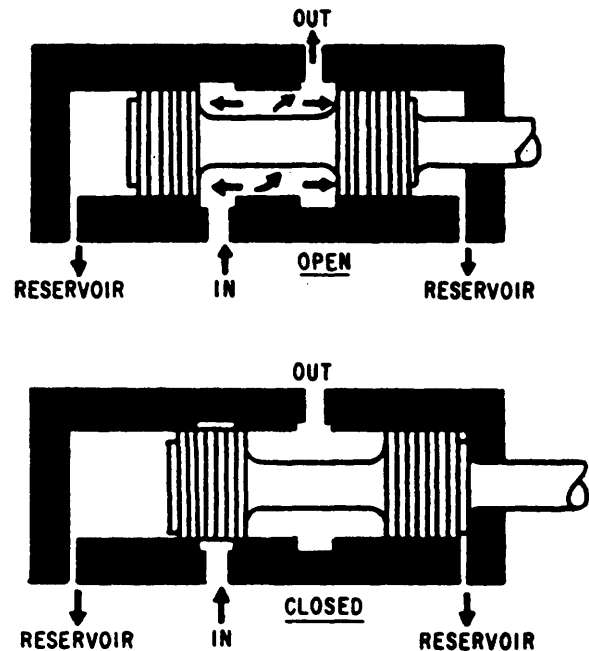


Figure 6-23.—Two-way, sliding spool directional control valve.

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.

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.

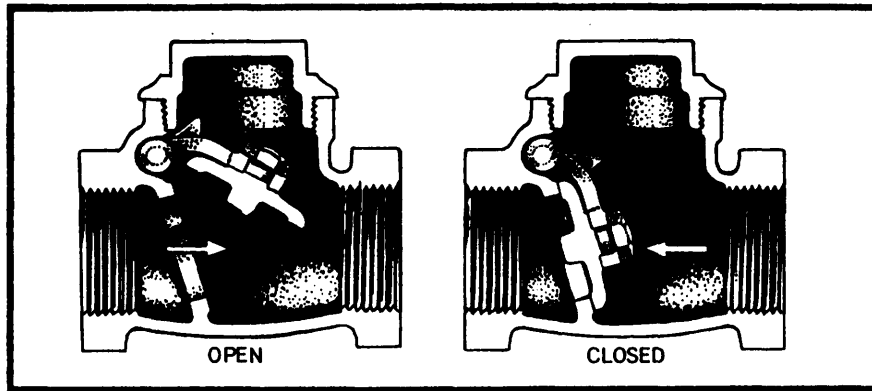


Figure 6-24.—Swing check valve.

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

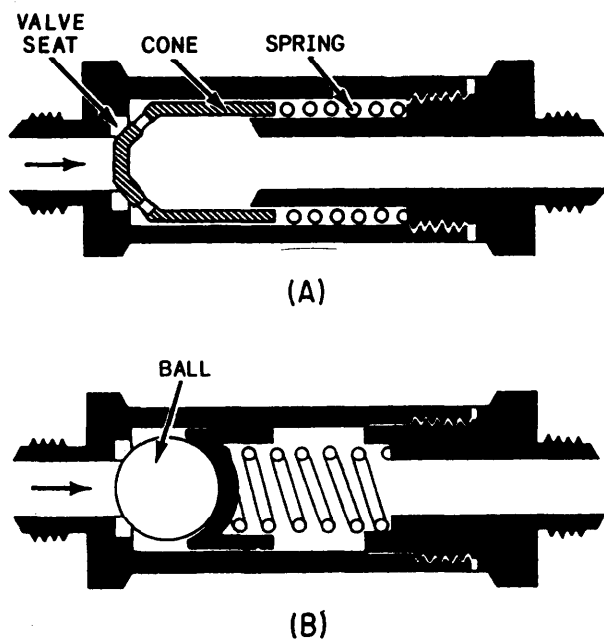


Figure 6-25.—Spring-loaded check valves.

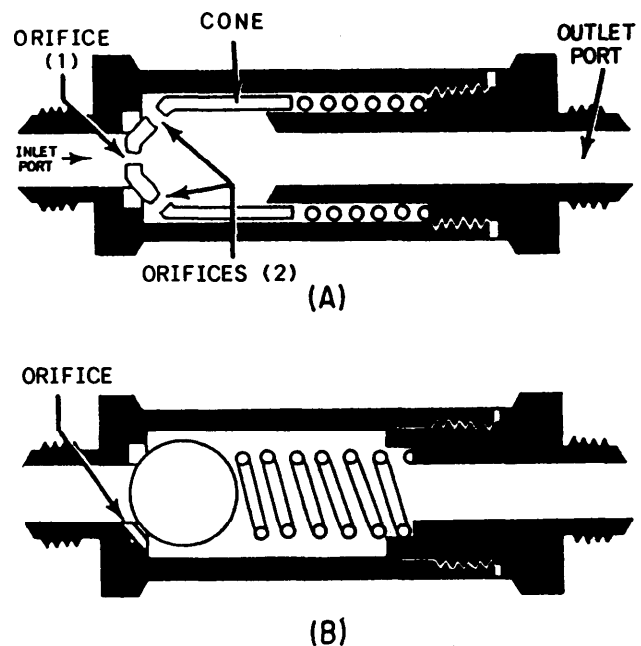


Figure 6-26.—Typical orifice check valves.

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.

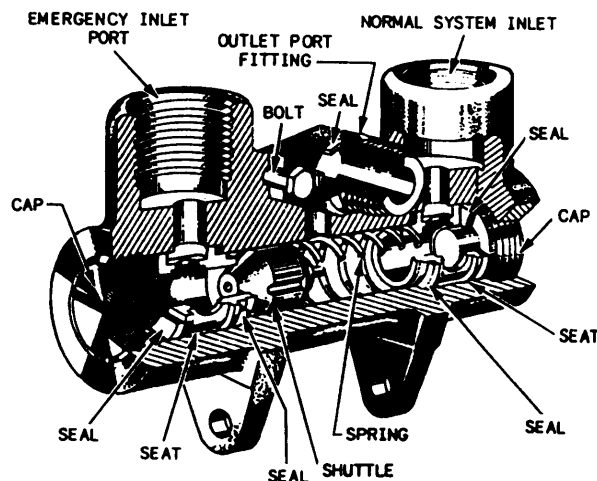


Figure 6-27.—Shuttle valve.

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

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.

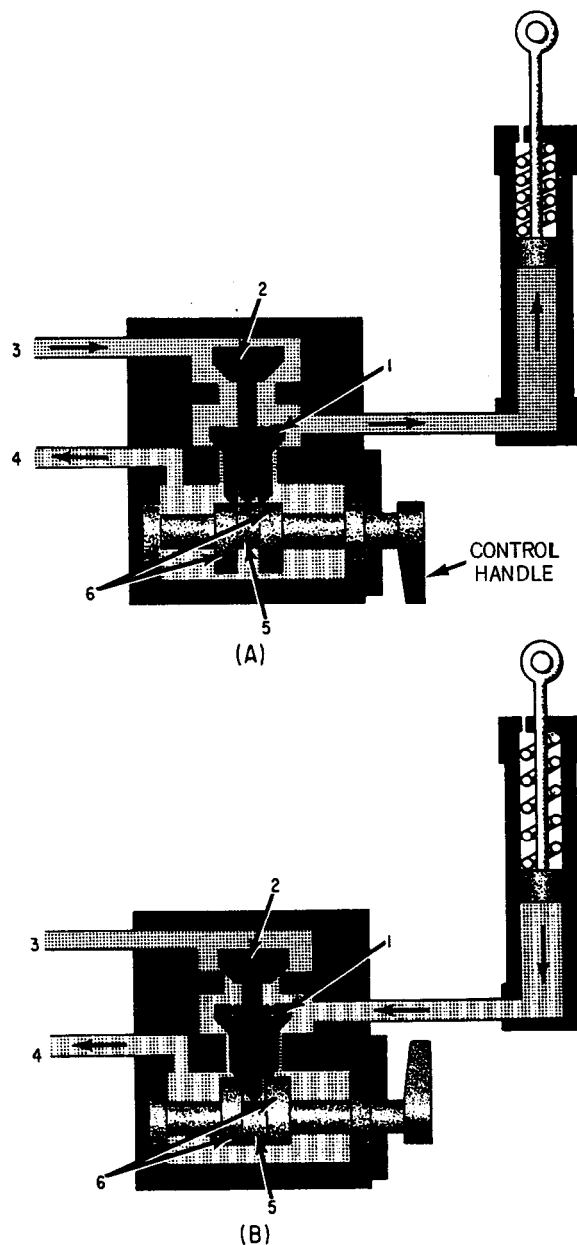
THREE-WAY VALVES

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).

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.

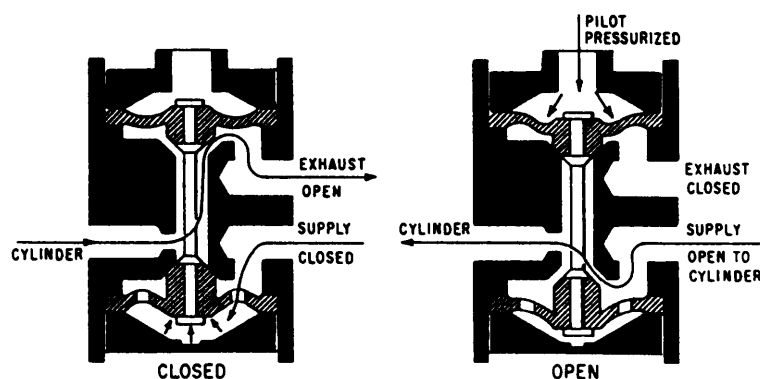


Figure 6-29.—Three-way, poppet-type, normally closed directional control valve (pilot-operated).

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

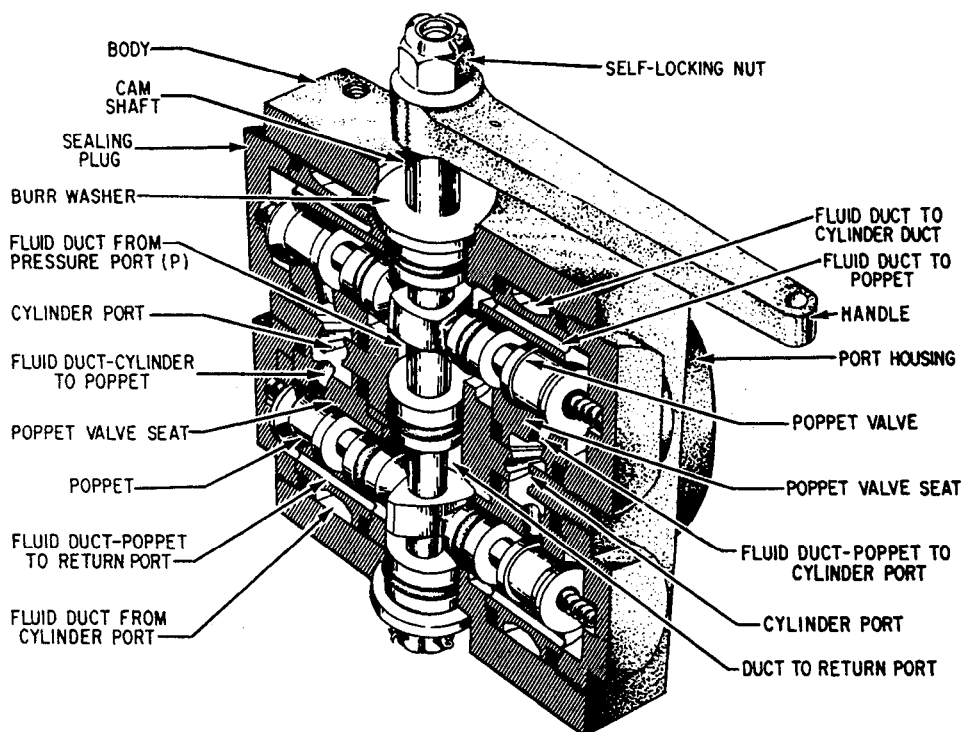


Figure 6-30.—Cutaway view of poppet-type, four-way directional control valve.

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.

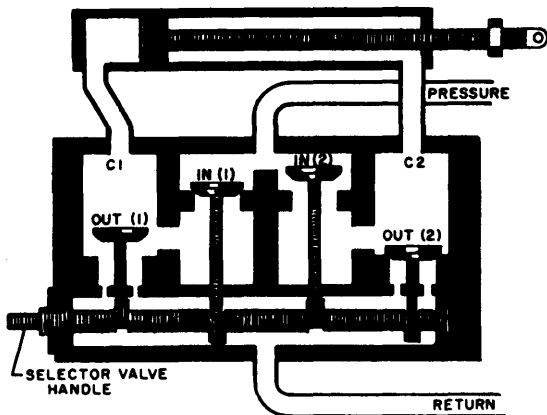


Figure 6-31.—Working view of a poppet-type, four-way directional control valve.

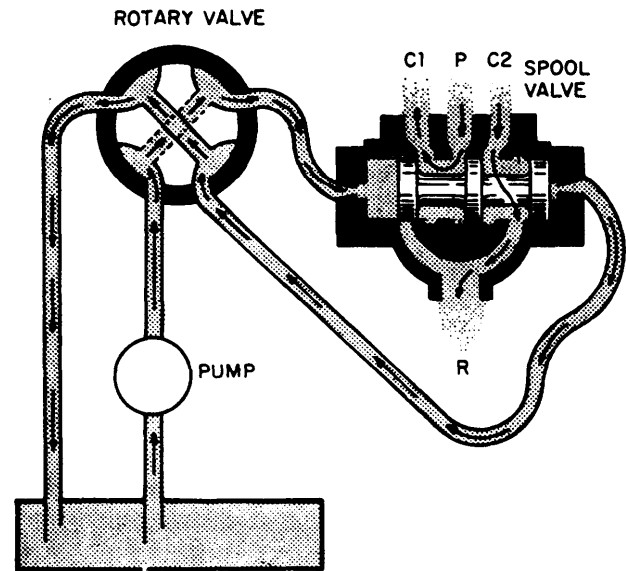


Figure 6-32.—Sliding spool valve controlled by a rotary spool valve.

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),

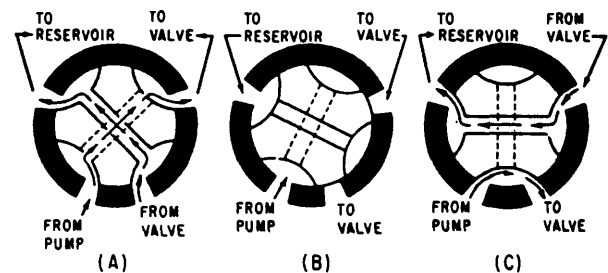


Figure 6-33.—Operation of a rotary spool, four-way directional control valve.

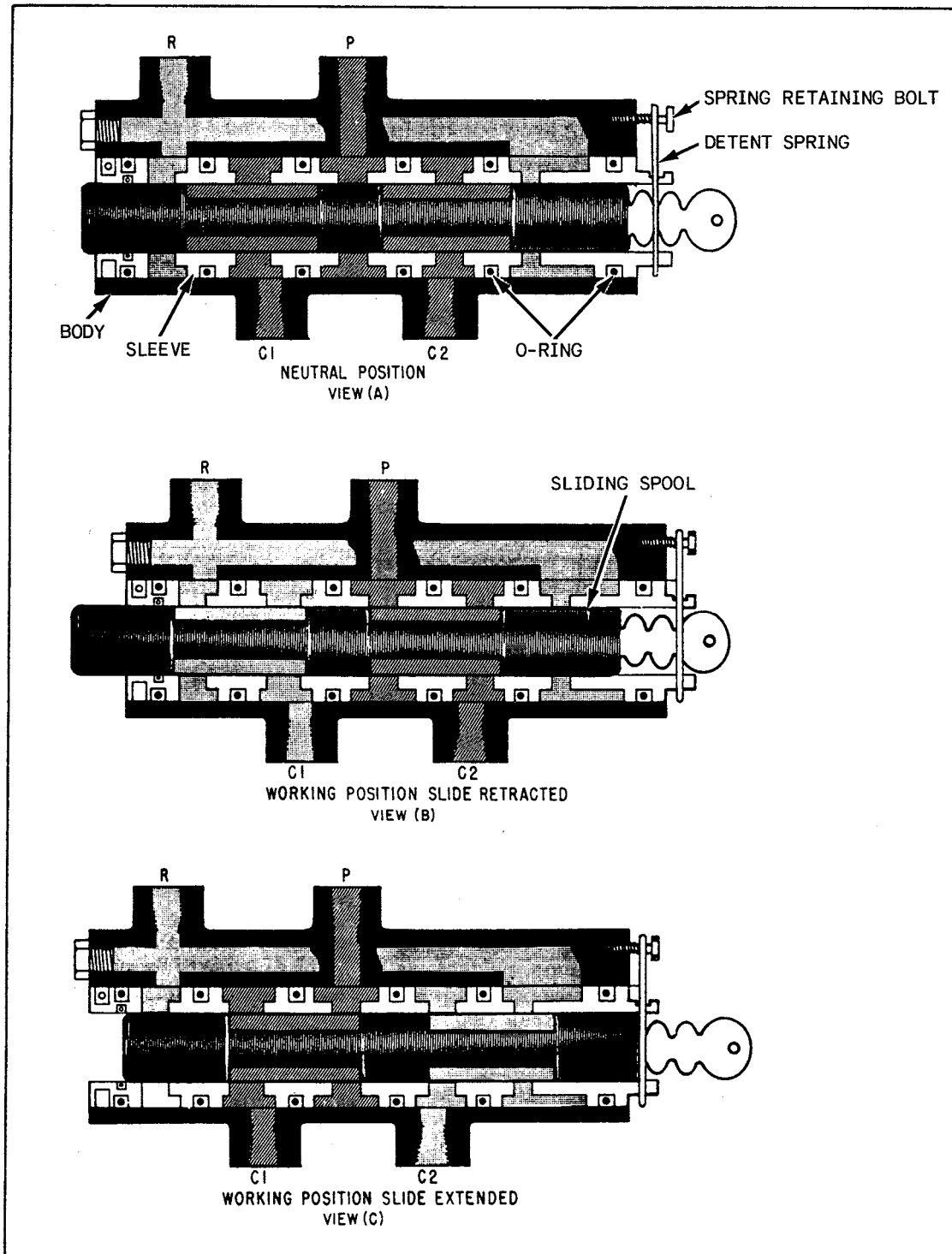


Figure 6-34.—Operation of a sliding spool, four-way directional control valve.

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 with 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

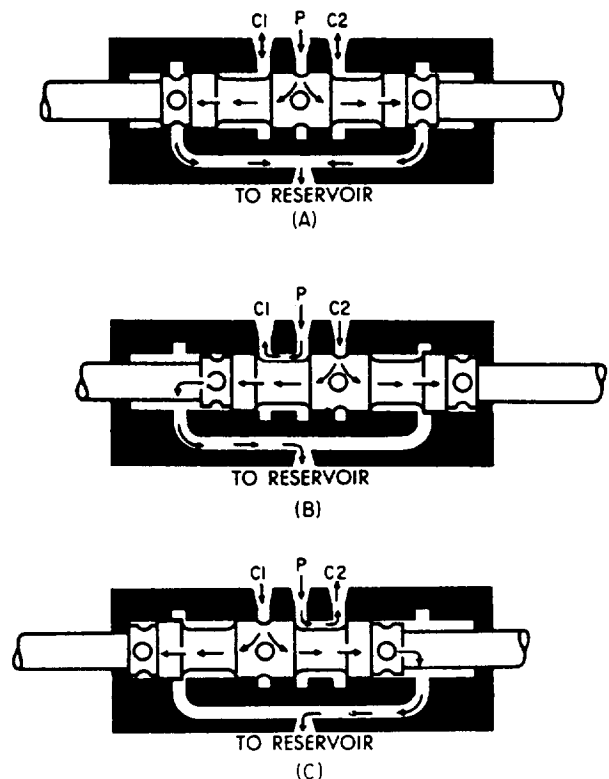


Figure 6-35.—Open center, sliding spool directional control valve.

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.

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.

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 degraded 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

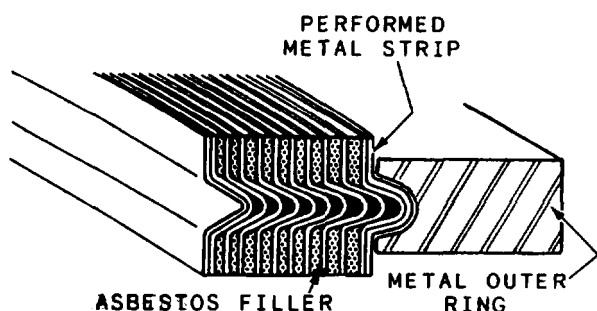


Figure 7-1.—Spiral-wound metallic-asbestos gasket.

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 (Nonmetallic)*, 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

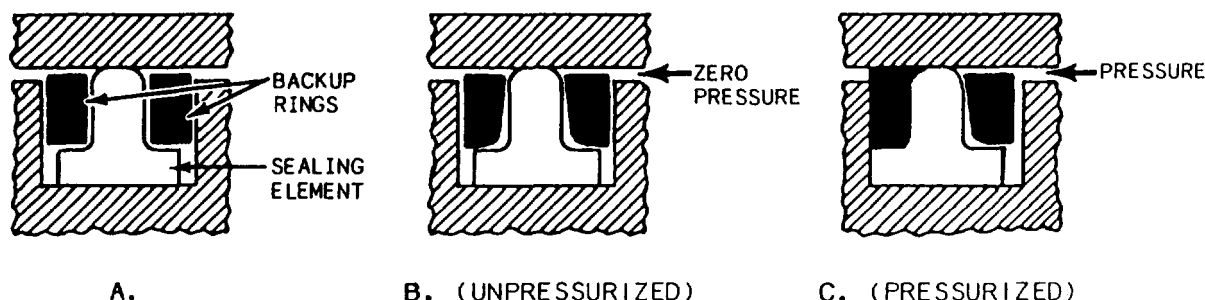


Figure 7-2.—T-seals.

Table 7-1.—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).									

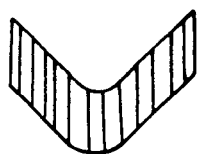
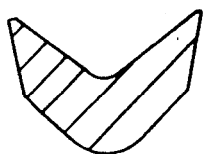


Figure 7-3.—V-rings.

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.

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

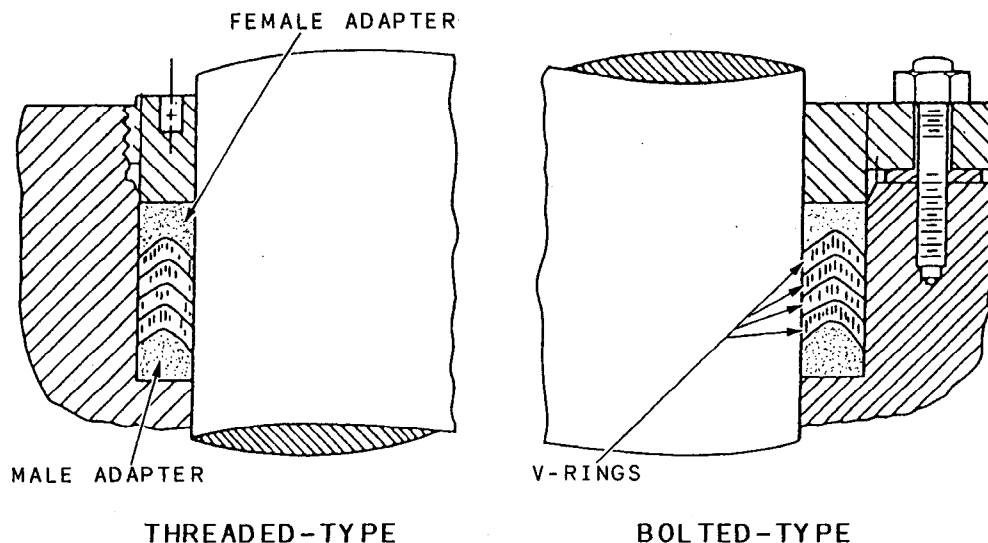


Figure 7-4.—Outside packed V-ring installations.

¹G-Tring[®] is a Greene, Tweed Trademark,

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

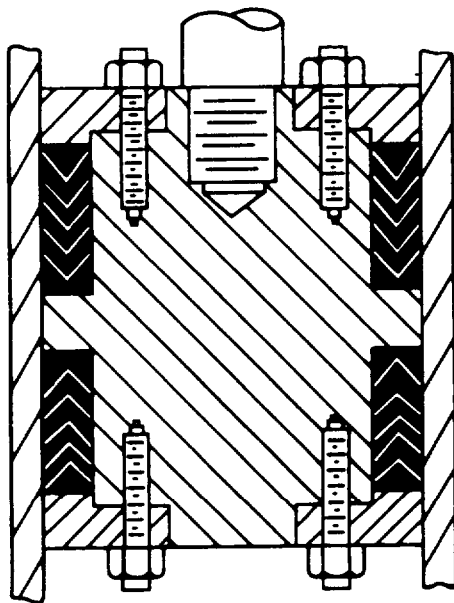


Figure 7-5.—Inside packed V-ring installation.

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

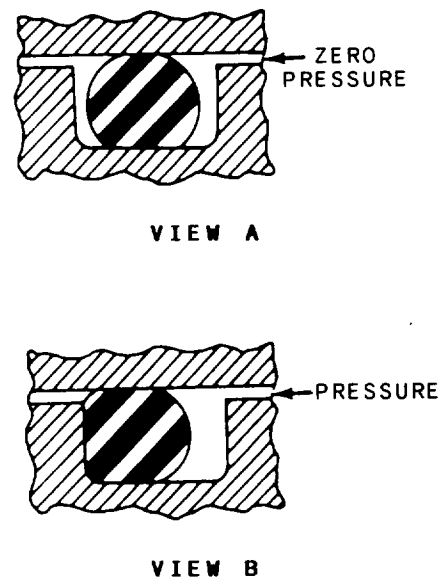


Figure 7-6.—O-ring installed in a gland.

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.

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.

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.

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

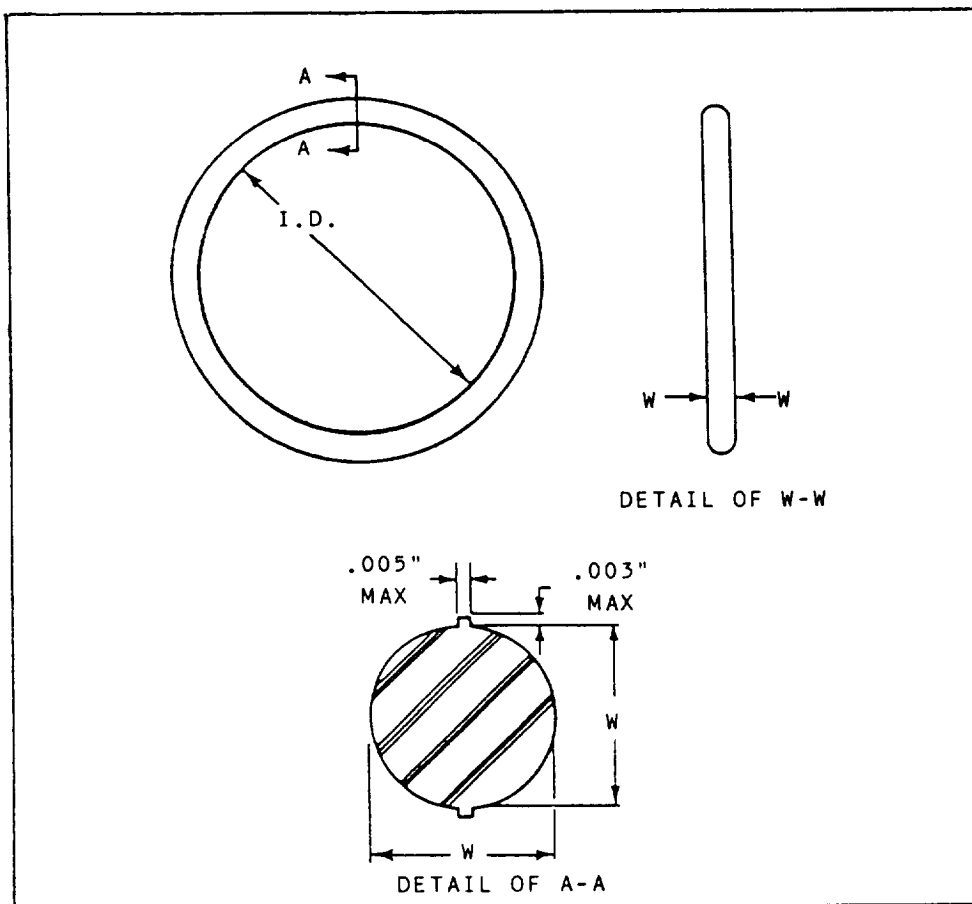


Figure 7-7.—Critical dimensions of an O-ring.

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

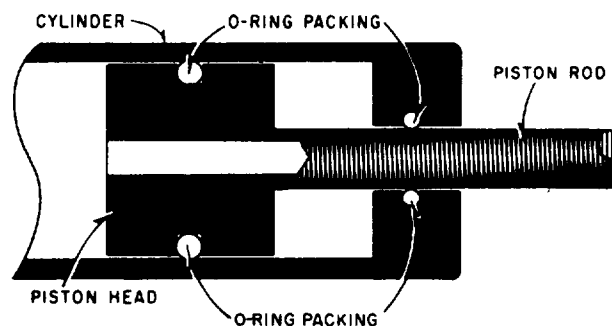


Figure 7-8.—Typical O-ring installation.

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

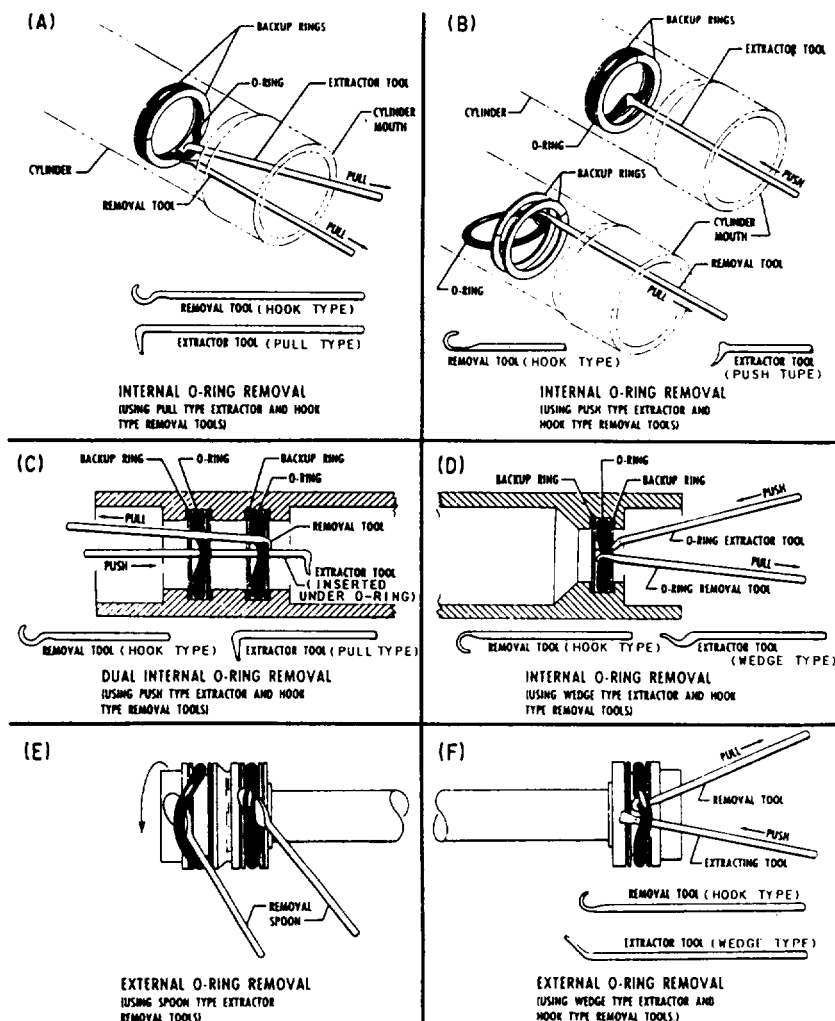


Figure 7-9.—O-ring tools and O-ring removal.

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

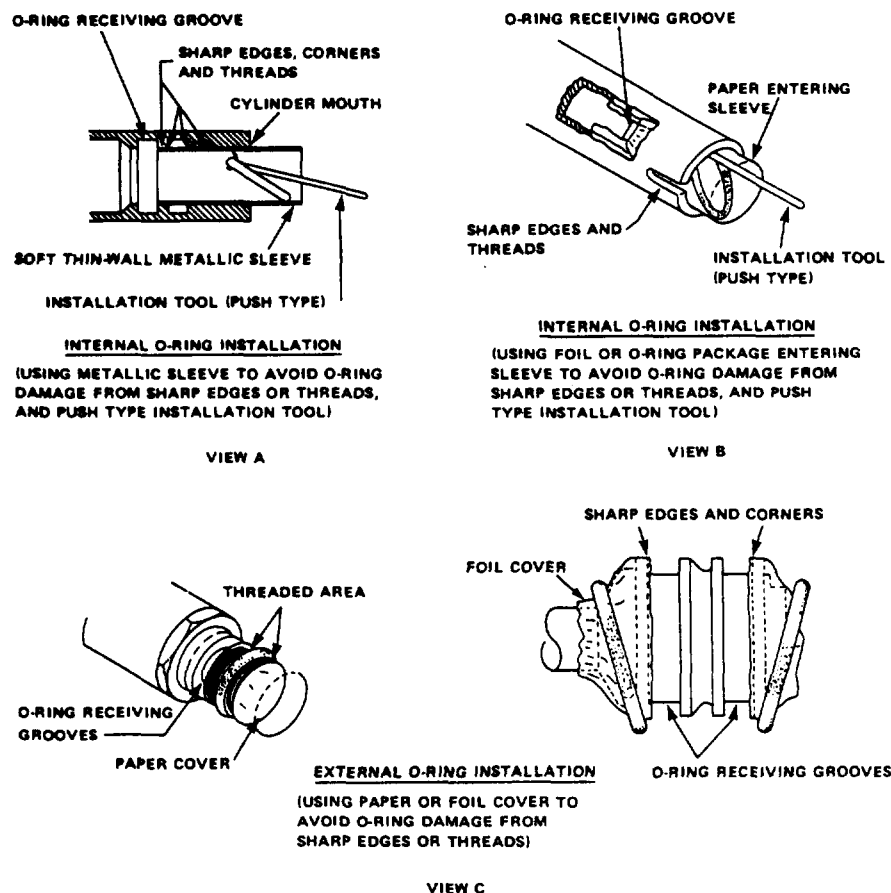


Figure 7-10.—O-ring installation.

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.

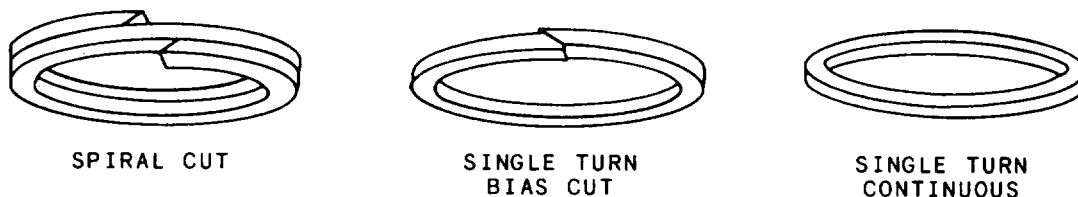


Figure 7-11.—Types of backup rings.

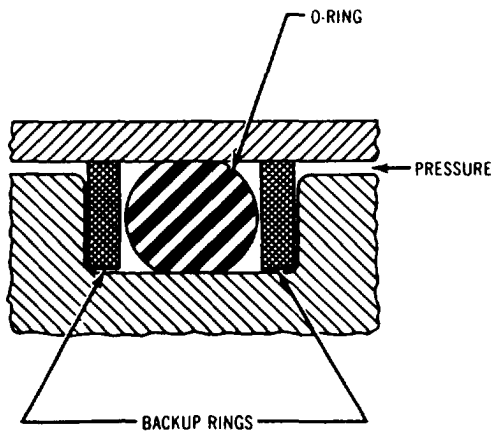
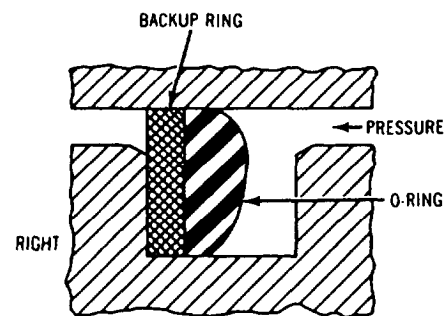


Figure 7-12.—Backup ring configuration.

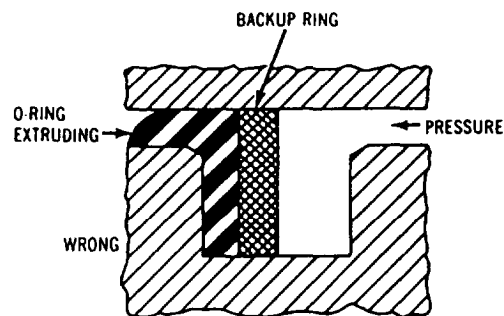
If only one backup ring is used, place the backup ring on the low-pressure side of the packing (fig. 7-13, view A). When a backup ring is placed on the high-pressure side of the packing, the pressure against the relatively hard surface of the backup ring forces the softer packing against the low-pressure 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



VIEW A—PROPER LOCATION OF A SINGLE BACKUP RING



VIEW B—IMPROPER LOCATION OF A SINGLE BACKUP RING

Figure 7-13.—Location of a single backup ring.

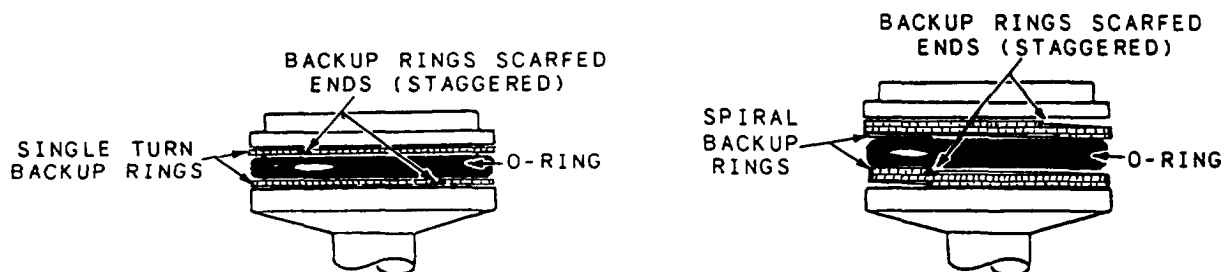
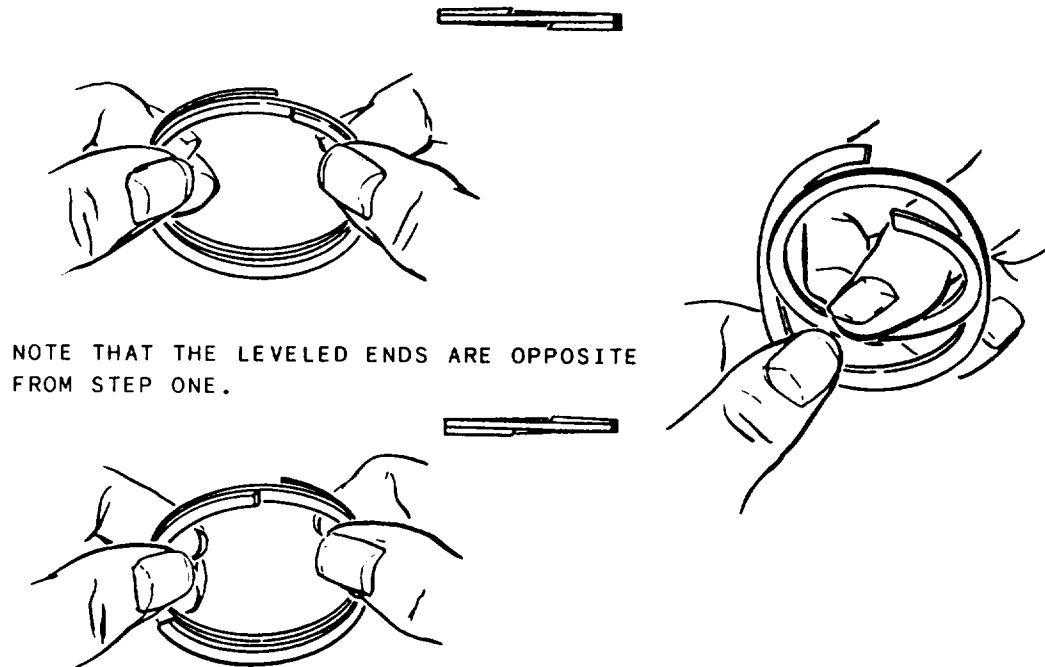


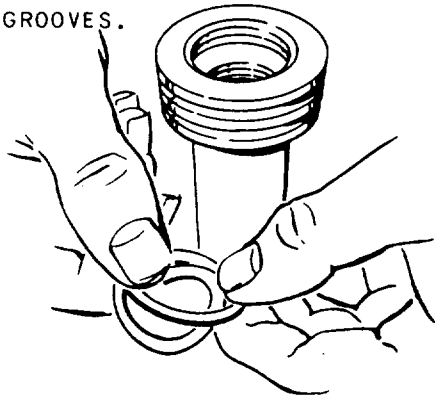
Figure 7-14.—Installation of cut dual backup rings.

CHANGING DIRECTION OF ROTATION OF SPIRAL BACKUP RINGS

(A) REVERSE THE SPIRAL OF A 5R-14 RING
(NORMALLY RH) TO A LEFT HAND SPIRAL.



(B) TO PREVENT OVERLAP, SLIGHTLY
STRETCH THE TEFLON RING BEFORE
INSTALLING IT INTO INTERNAL
GROOVES.



WORK THE RING INTO INTERNAL
GROOVES BY ROTATING THE ROD.

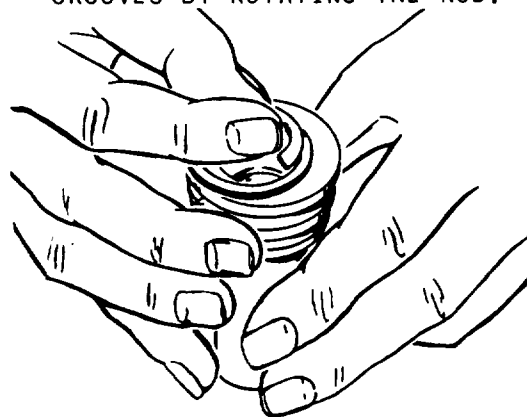


Figure 7-15.-Installation of TFE back up rings (internal).

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.

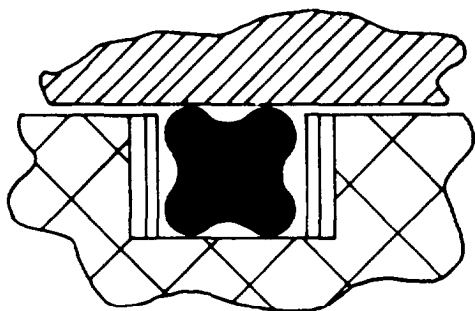


Figure 7-16.—Quad-Ring.

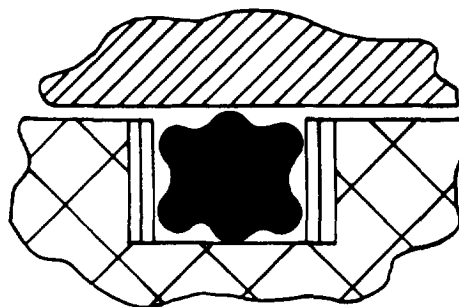


Figure 7-17.—Quad-O-Dyn® seal.

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

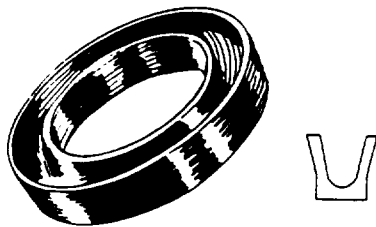


Figure 7-18.—Typical U-cup seal.

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.

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

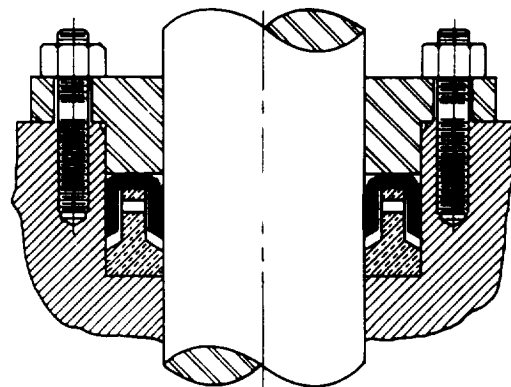


Figure 7-19.—U-packing.

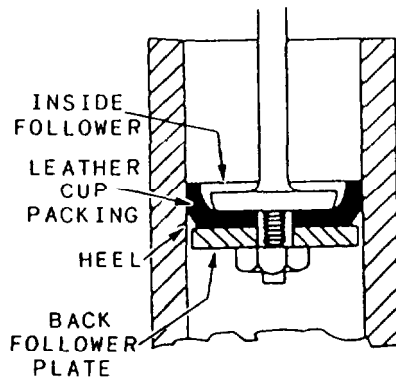


Figure 7-20.—Cup packing.

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.

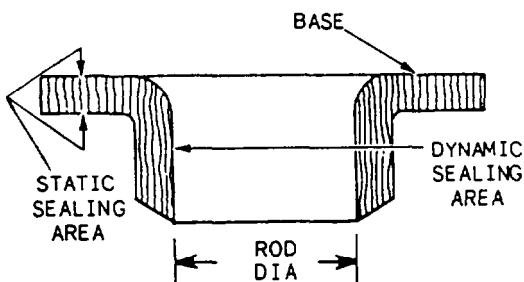


Figure 7-21.—Typical flange packing cross section.

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

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.