CHAPTER 17
SEALS

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17.1 ELASTOMERIC SEAL RINGS

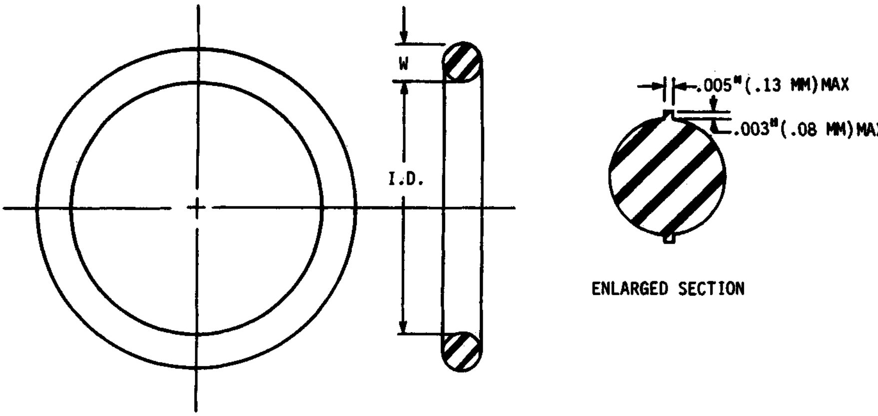
Seal rings of the O-ring type are used as both static and dynamic seals. Static seals serve the same purpose as gaskets; that is, they provide a seal between two members that are not intended to undergo relative motion. Dynamic seals, however, are used where rotating or reciprocating motion is intended to occur.

O-rings are molded to the size of the elastomeric material with a circular cross section, as shown in Fig. 17.1*a*. The *size* is designated by the cross-sectional diameter w and the nominal inside diameter (ID). The standard sizes specified in SAE J120a are summarized in Table 17.1. The first size number in a group is associated with the minimum inside diameter, and the last size number is associated with the maximum inside diameter. Some manufacturers provide additional sizes that extend the range of inside diameters for a particular cross-section size. The nominal inside diameters were selected to provide dynamic seals in cylinder bores dimensioned in inches and common fractions of inches. Either SAE J120a or manufacturers' recommendations should be consulted to obtain the recommended compression of the cross section. The compression is different for static and dynamic applications.

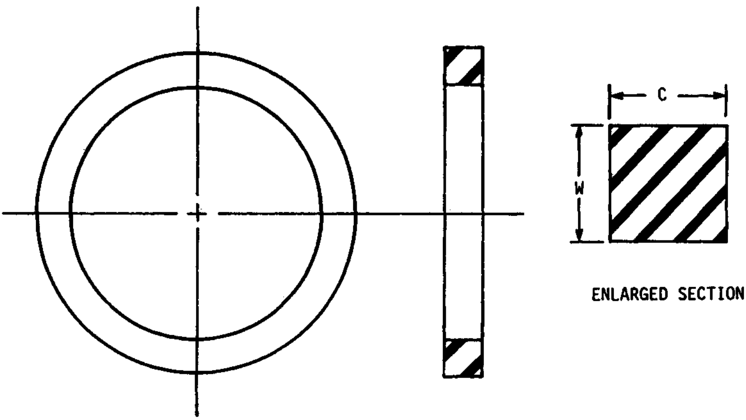
The rectangular-section ring in Fig. 17.1*b* is manufactured by cutting lengths from a tube of molded material. The standard sizes listed in SAE J120a are summarized in Table 17.2. The first size number in a group is associated with the minimum inside diameter, and the last size number is associated with the maximum inside diameter. Rectangular-section rings are suitable for static applications with pressures up to 1500 psi [10.3 newtons per square millimeter (N/mm^2)].

The standard shape of groove for sealing rings is shown in Fig. 17.2. The actual groove dimensions depend on the type and size of the seal ring cross section and the nature of the application. Recommended groove dimensions are provided in SAE J120a and in the manufacturers' literature. Because elastomeric materials are almost incompressible, it is necessary to provide sufficient volume for the seal ring in the groove. The recommended groove dimensions do so.

For static seals, a finish on surfaces contacted by the seal ring that is rougher than 32 μin (0.8 μm) may lead to leakage. Because rough finishes accelerate seal wear in dynamic seals, a surface finish of 5 to 16 μin (0.13 to 0.4 μm) is preferred. Friction is



(a)



(b)

FIGURE 17.1 Seal rings. (*a*) O-ring; (*b*) rectangular-section ring.

TABLE 17.1 Standard Sizes of O-Rings

Size no.	*w*, in	Actual ID, in	
		Minimum	Maximum
006 to 045	0.070 ± 0.003	0.114 ± 0.005	3.989 ± 0.015
110 to 163	0.103 ± 0.003	0.362 ± 0.005	5.987 ± 0.023
210 to 281	0.139 ± 0.004	0.734 ± 0.006	14.984 ± 0.060
325 to 349	0.210 ± 0.005	1.475 ± 0.010	4.475 ± 0.010
425 to 460	0.275 ± 0.006	4.475 ± 0.015	15.475 ± 0.015

TABLE 17.2 Standard Sizes of Rectangular-Section Rings

Size no.	w, in	c, in	Actual ID, in	
			Minimum	Maximum
R006 to R045	0.066 ± 0.004	0.066 ± 0.003	0.114 ± 0.005	3.989 ± 0.015
R110 to R163	0.099 ± 0.004	0.099 ± 0.003	0.362 ± 0.005	5.987 ± 0.023
R210 to R281	0.134 ± 0.004	0.134 ± 0.004	0.734 ± 0.006	14.984 ± 0.060
R325 to R349	0.203 ± 0.005	0.203 ± 0.005	1.475 ± 0.010	4.475 ± 0.015
R425 to R460	0.265 ± 0.005	0.265 ± 0.005	4.475 ± 0.015	15.475 ± 0.030

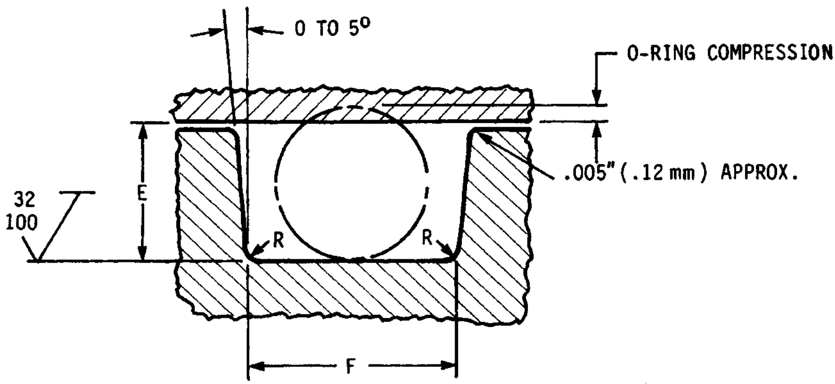


FIGURE 17.2 Shape of groove for seal rings.

reduced with the smoother finish, but surfaces smoother than 5 μin (0.13 μm) may not be satisfactory for reciprocating motion.

A static seal ring application in which the joint is subject to internal pressure only is shown in Fig. 17.3*a*. The groove design in Fig. 17.3*b* is for a joint subject to external pressure or internal vacuum only. It is generally advisable in these applications to use as large a seal ring cross section as possible because the tolerance on the groove depth is greater with larger cross sections. This requires less precise machining and tends to reduce manufacturing costs.

O-rings are also used as static seals for hydraulic tube fittings that are screwed into tapped holes. Recommended machining dimensions are provided in SAE J514 (June 1993).

Elastomeric sealing rings are most commonly made of nitrile (Buna N) compounds. These compounds are low in cost and are compatible with alcohol, gasoline, hydraulic fluids, lubricating oils, and water. They also are suitable for temperatures ranging from −67 to 257°F (−55 to 125°C). For resistance to higher temperatures or compatibility with other fluids, other compounds are employed. Among these compounds are butyl, ethylene propylene, neoprene, fluorocarbon, silicone, and polyurethane.

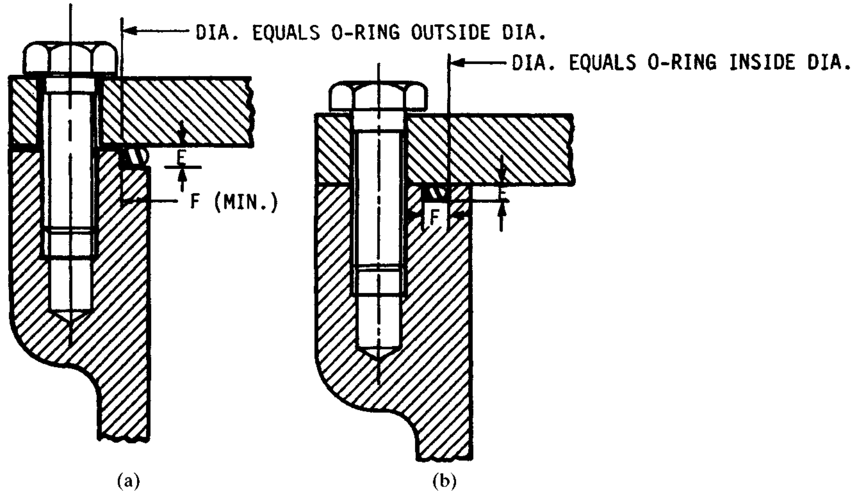


FIGURE 17.3 Static O-ring seals. (*a*) Joint subject to internal pressure only; (*b*) joint subject to external pressure.

17.2 SEALS FOR ROTARY MOTION

Seals are required on rotating shafts to retain working fluids, to retain lubricants, and to exclude dirt. The selection of a seal type depends on fluid pressure, shaft speed, and whether any leakage can be permitted. There are many variations of the basic seal types that are available from various manufacturers.

17.2.1 O-Rings

Attempts to use O-rings as seals for rotating shafts have not always been successful because the elastomers shrink when heated. If an O-ring is under tension, friction between the ring and the shaft generates heat that makes the ring shrink. Contraction of the ring creates additional heat, and failure occurs rapidly.

O-rings have been used successfully on rotating shafts when they are installed under compression by using a smaller-than-normal groove diameter in the housing. Satisfactory life can then be obtained at shaft speeds up to 750 feet per minute (ft/min) [3.8 meters per second (m/s)] and sealed pressures up to 200 psi (1.38 N/mm^2). Recommended O-ring cross sections are 0.139 in (3.53 mm) for speeds up to 400 ft/min (2.0 m/s), 0.103 in (2.62 mm) for speeds from 400 to 600 ft/min (2.0 to 3.0 m/s), and 0.070 in (1.78 mm) for speeds exceeding 600 ft/min (3.0 m/s) [17.1].

17.2.2 Radial Lip Seals

A section through a radial lip seal is shown in Fig. 17.4. This type is used primarily for retention of lubricants and exclusion of dirt. It is suited for conditions of low lubricant pressure, moderate shaft speeds, less-than-severe environmental conditions, and situations where slight leakage may be permitted. Radial lip seals are compact, effective, inexpensive, and easily installed.

The outer case is held in the bearing housing by an interference fit. The garter spring provides a uniform radial force to maintain contact between the elastomeric sealing lips and the shaft. Lubricant leakage is reduced when hydrodynamic sealing lips are used. Such lips have very shallow grooves molded into the primary sealing lip to pump lubricant out of the contact area. Hydrodynamic sealing lips are manufactured for rotation in one direction only or for rotation in either direction.

Sealing lips are most commonly made of nitrile (Buna N) rubber compounds because of their compatibility with greases, lubricating oils, and hydraulic fluids. The nitrile compounds have poor to fair compatibility with extreme-pressure (EP) additives used in some gear lubricants. A polyacrylate or fluoroelastomer compound is a better choice with EP lubricants.

Radial lip seal terminology is presented in SAE J111 (Jun. 88), and recommendations for applications are made in SAE J946 (Oct. 91). One of the purposes of the secondary sealing lip shown in Fig. 17.4 is to exclude dust. That lip, however, leads to higher seal temperatures because of the additional friction, and the higher temperatures lead to earlier seal failure. Dual lip seals are not recommended for shaft speeds exceeding 150 ft/min (0.76 m/s) [17.2].

A minimum hardness of Rockwell C 30 is recommended for the portions of shafts that contact the sealing lips in order to prevent scoring of the shaft. If the shaft may be damaged in handling, a minimum hardness of Rockwell C 45 will provide protection against damage. A hard surface can be provided for soft shafts by use of a hardened wear sleeve of thin steel that is held in place by an interference fit.

Radial lip seals function best with carbon-, alloy-, or stainless-steel shafts or nickel-plated surfaces. Use with aluminum alloys, brass, bronze, magnesium, zinc, or similar metals is not recommended. Shaft surface texture should be in the range of 10 to 20 μin (0.25 to 0.50 μm). This condition can best be met by plunge grinding.

This type of seal is limited to sealing pressures of 3 psig (0.02 N/mm^2 gauge) at shaft speeds exceeding 2000 ft/min (10.2 m/s) and 7 psig (0.05 N/mm^2 gauge) at speeds up to 1000 ft/min (5.1 m/s). When pressures exceed these limits, a mechanical face seal is preferable.

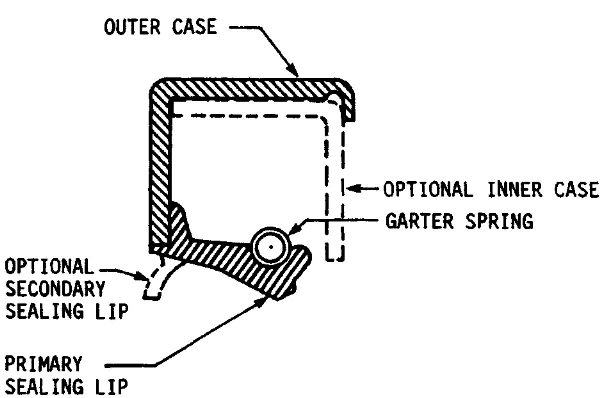


FIGURE 17.4 Cross section of radial lip seal.

17.2.3 Face Seals

Whereas a radial lip seal contacts the shaft circumference, a face seal acts against a surface perpendicular to the shaft axis. The seal may be mounted in a housing and seal against a shoulder or collar on the shaft. The seal also may be mounted on the shaft and seal against a surface on the housing. Some elastomeric face-sealing elements are loaded by mechanical springs, whereas others are not.

Elastomeric face seals (Fig. 17.5*a*) are used to retain lubricants. For high-speed applications, the lubricant should be on the side where centrifugal force throws the lubricant into the sealing area. These seals have the disadvantage of requiring rather precise location with respect to the sealing surface in order to provide the proper force on the seal. The sealing surface must be flat and smooth, with a surface finish of 10 to 20 μin (0.25 to 0.50 μm). Less rigid control of surface flatness is required for low speeds.

Mechanical face seals are used in situations where a radial lip seal or elastomeric face seal will not be satisfactory. Abrasive conditions, such as those encountered by earth-moving machinery and mining machinery, may dictate the use of a mechanical face seal. A mechanical seal is frequently used in the automotive coolant pump (Fig. 17.5*b*), in which the pressure is relatively low. Here, the stationary spring-loaded seal ring contacts a flat surface on the rotor.

If the seal in Fig. 17.5*b* were used with high-pressure fluid, a high axial sealing force would result. Friction between the rotor and stator could possibly cause overheating of the seal elements. Consequently, a balanced mechanical face seal, such as in Fig. 17.5*c*, is used for higher pressures. These seals are proportioned so that much of the force due to pressure is balanced. This leaves a small net force to provide contact between rotor and stator.

Another type of face seal, Fig. 17.6, is used to seal lubricants in rotating elements of machines that operate in environments where water, mud, or dust must be excluded. The seal consists of two hardened steel rings with lapped sealing surfaces that are forced together by surrounding elastomeric rings. The seal surfaces are tapered so that the point of contact moves inward as wear occurs. The seal rings do not contact the shaft that they surround; instead, one seal ring is driven by the rotating component through the elastomeric ring.

17.2.4 Metal Sealing Rings

Cast-iron sealing rings are used in hydraulic applications where oil must be introduced through a rotating shaft (Fig. 17.7). A typical application is to operate a clutch in an automatic transmission for a motor vehicle. Ring cross-sectional dimensions are similar to those for engine piston rings of the same outside diameter. Information on designing to accommodate these rings is provided in SAE J281 (Sept. 1980) and SAE J1236 (Apr. 1980).

17.2.5 Compression Packings

The stuffing box (Fig. 17.8) is used to seal fluids under pressure with either rotating or reciprocating shafts. Sealing between the packing and the shaft occurs as a result of axial movement of the gland when the nuts are tightened. Friction between the packing and the shaft causes wear, and so periodic tightening of the nuts is required.

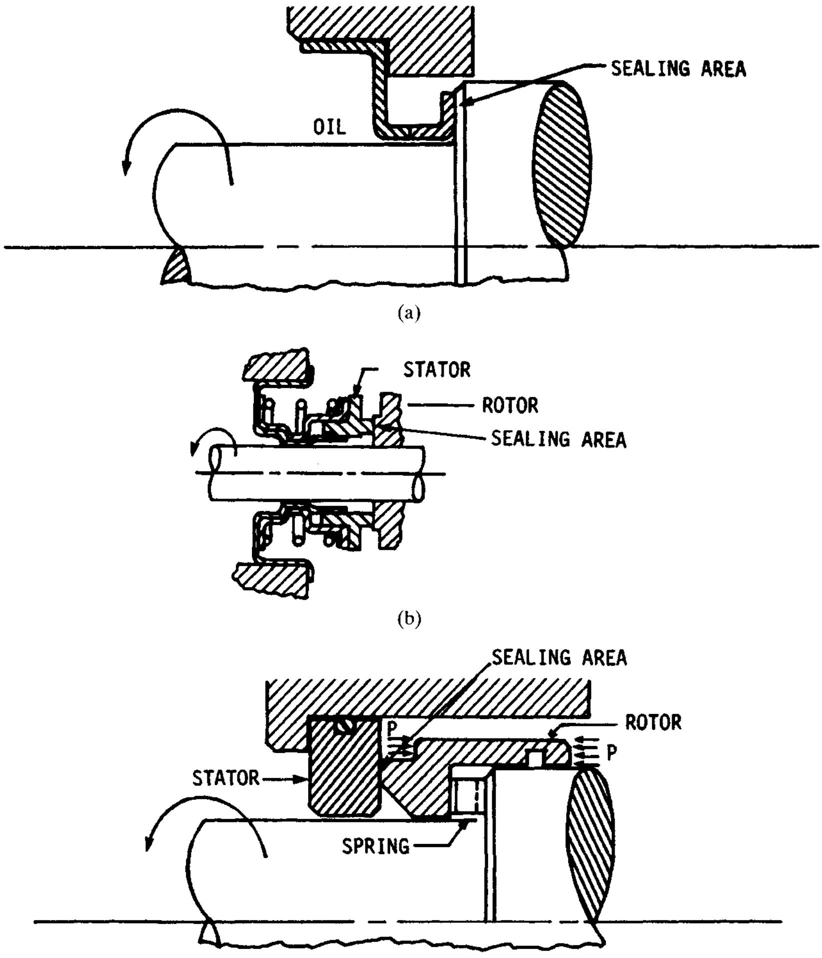


FIGURE 17.5 Face seals. (*a*) Housing-mounted elastomeric seal; (*b*) mechanical seal for engine coolant pump; (*c*) balanced mechanical seal.

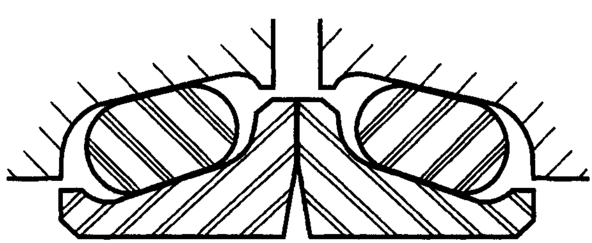


FIGURE 17.6 Cross section of a face seal for severe operating environments.

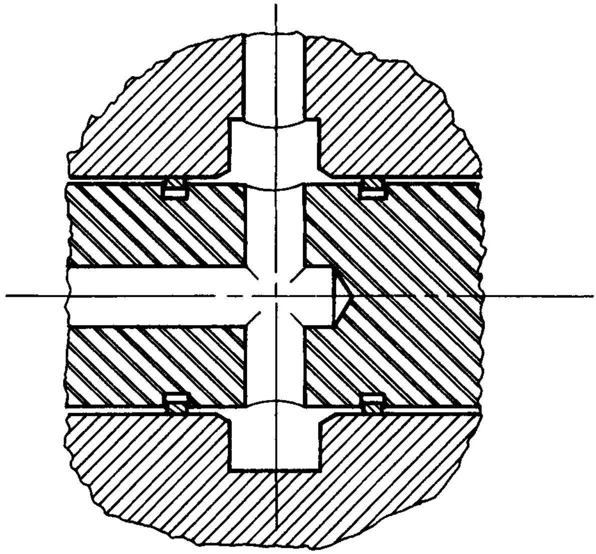


FIGURE 17.7 Metal seal ring application on a rotating shaft.

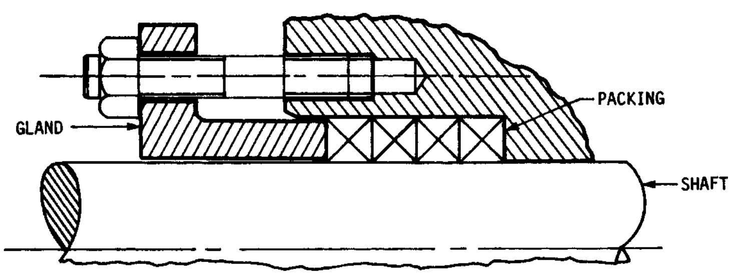


FIGURE 17.8 Stuffing box for a rotating shaft.

Packing material is usually obtained in straight lengths of square or rectangular cross section. Pieces are cut off and formed into rings that fit the stuffing box. The choice of packing material depends on the fluid to be sealed. Available packing materials include artificial fibers, asbestos, cotton, graphite, jute, leather, and metals. The metal packings are used for temperature conditions where the other materials are inadequate. The metal packings are formed from foil which is compressed into the proper packing shape.

In the design of stuffing boxes, small clearances are provided between the shaft and surrounding parts. The small clearances minimize extrusion of the packing into the clearance spaces.

Valve stems undergo a helical motion rather than a rotary motion when the valve is opened or closed. Investigations into the prevention of valve leakage and wear of valve stems resulted in a procedure for establishing packing dimensions for valve stems [17.3].

17.2.6 Noncontacting Seals

Frictional losses occur with sealing methods that utilize physical contact between a rotating and a stationary part. With high rubbing velocities, friction losses may be a significant factor. Those losses can be eliminated by using a seal that does not require physical contact. A noncontacting seal, however, cannot prevent leakage completely, although it does reduce it to a tolerable level.

One method of achieving a low leakage rate is to provide a very small clearance between the shaft and the surrounding housing or bushing. The longer the low-clearance passage, the greater the reduction in leakage.

A type of noncontacting seal called the *labyrinth seal* (Fig. 17.9) is used on such machines as large blowers and steam turbines. It can be used to retain lubricant in the bearings or to seal the working fluid in the machine. Effectiveness of the seal depends on small clearances between the seal and the shaft. In sealing the working fluid, the small clearances create a series of pressure drops between the working fluid and the atmosphere.

Labyrinth seals are usually made from a relatively soft metal such as aluminum or bronze so that the shaft is not damaged if contact between shaft and seal occurs. The simplest type is shown in Fig. 17.9, but other types are also used.

17.3 SEALS FOR RECIPROCATING MOTION

Some of the sealing methods used for rotary motion are also satisfactory for reciprocating motion. O-rings, compression packings, metal sealing rings, and some additional types are used to seal reciprocating rods, shafts, and pistons.

17.3.1 O-Rings

The O-ring is used extensively because of the low installed cost and effectiveness as a seal. It is well adapted to sealing reciprocating motion as well as for use as a static seal. Figure 17.10 shows applications on a piston and piston rod as well as a static seal application. Many such applications are for hydraulic cylinders in which the hydraulic oil acts as the lubricant for the O-rings.

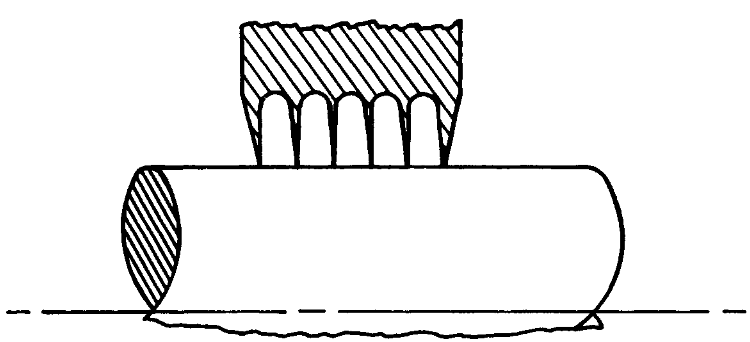


FIGURE 17.9 Labyrinth type of noncontacting seal.

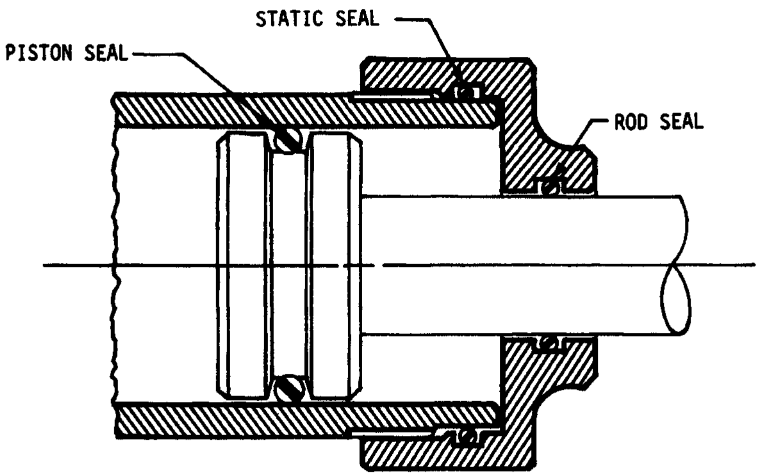
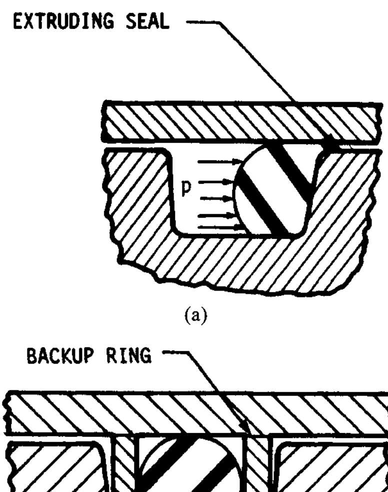
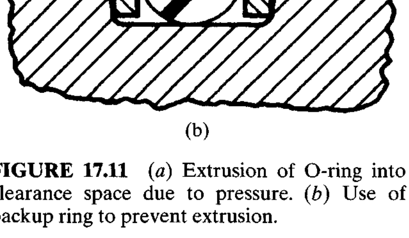


FIGURE 17.10 Applications of O-rings as a static seal and as seals for reciprocating motion.



(a)



(b)

FIGURE 17.11 (*a*) Extrusion of O-ring into clearance space due to pressure. (*b*) Use of backup ring to prevent extrusion.

Rectangular-section rings are not suited for reciprocating motion and are used only in static applications. The shape of the groove for circular-section O-rings for sealing reciprocating motion is the same as that for static applications (Fig. 17.2). The recommended groove depth *E*, however, is slightly different for reciprocating motion. Recommended dimensions are available in SAE J120a and in the manufacturers' literature.

An O-ring must seal the clearance space between the reciprocating and stationary parts, for example, between the piston and the cylinder in Fig. 17.10. The amount of clearance that can be permitted depends on the pressure differential across the O-ring and the ring hardness. If the clearance is too great, the O-ring is extruded into the clearance space (Fig. 17.11*a*). The reciprocating motion then tears away small pieces of the O-ring, which results in a leaking seal and contamination of the working fluid by the O-ring particles.

The pressure limitations of O-rings can be overcome by the use of backup rings (Fig. 17.11*b*) or other devices that prevent O-ring extrusion. Backup rings are made from leather, plastics, or metal. The metal rings are split like a piston ring for radial compression during assembly into the cylinder.

Recommendations on the combination of clearance and pressure for which backup rings are required vary to some extent among the various O-ring suppliers. Figure 17.12 provides one such recommendation [17.4]. In this figure, if the combination of fluid pressure and maximum gap falls to the right of a hardness curve, backup rings are required. If a piston or rod can be forced to one side of the bore, the maximum gap is the difference between the two diameters. If, however, the radial position of the piston or rod is restrained, as by bearings, the radial clearance is the maximum gap.

Both the compression of the O-ring cross section to effect a seal and fluid pressure acting on the seal cause friction forces that oppose reciprocating motion. Information for estimating friction factors is provided in Fig. 17.13 [17.4]. The friction factor due to compression of the cross section can be obtained from Fig. 17.13*a*. The seal compression is expressed as a percentage of the O-ring cross section. The friction factor is multiplied by the circumference of the surface where relative motion occurs to obtain the friction force. For a piston, the circumference of the cylinder bore is used; for a piston rod, the rod circumference applies.

The friction factor for pressure differential across the O-ring is obtained from Fig. 17.13*b*. That factor is multiplied by the projected area of the O-ring to obtain the friction force. For an O-ring in a piston, the projected area is the product of the diameter of the ring cross section and the circumference of the cylinder bore. The total estimated friction force is the sum of the friction forces due to compression and fluid pressure.

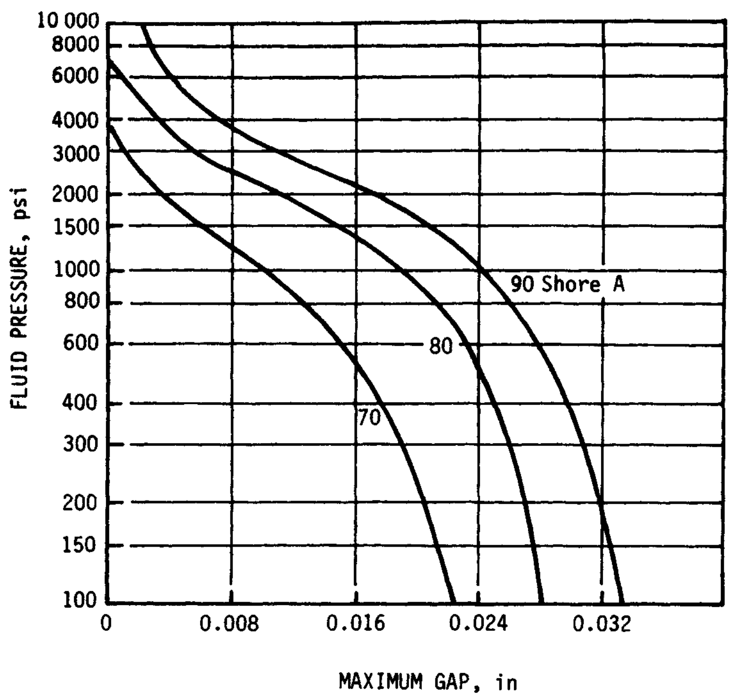


FIGURE 17.12 Extrusion limits for O-rings. *(From Ref. [17.4].)*

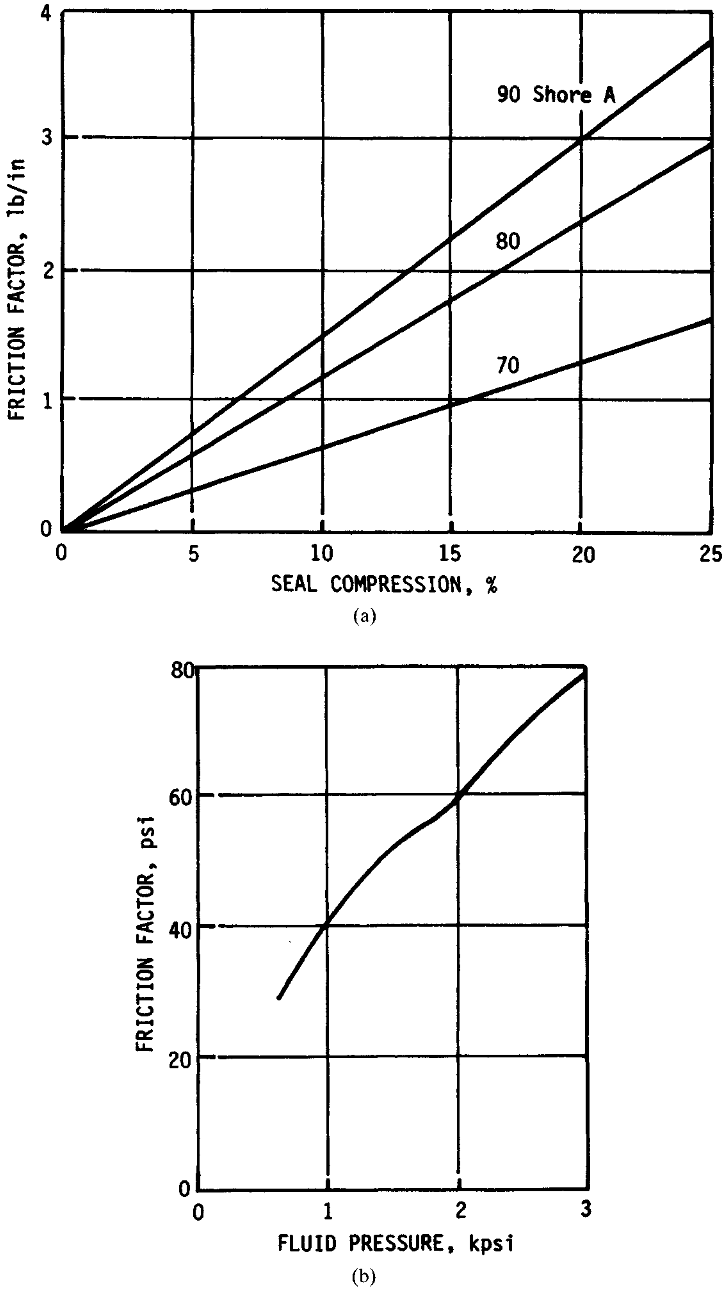


FIGURE 17.13 O-ring friction factors due to (*a*) compression of the cross section and (*b*) fluid pressure. *(From Ref. [17.4].)*

The finish of rubbing surfaces should be 8 to 16 μin (0.2 to 0.4 μm) for O-rings with a hardness of 70 Shore A. For rougher surfaces, a higher O-ring hardness should be used to ensure reasonable life.

O-rings may be damaged if they are forced over sharp corners during assembly. The addition of chamfers to corners is an inexpensive method of reducing damage. Serious damage to O-rings occurs if they are forced to pass over a hole in a cylinder wall while under pressure. If this occurs, the O-ring expands into the hole and must later be forced back into the groove. This action tends to shear pieces off the ring and thus destroy its ability to seal.

17.3.2 Lip Packings

Cup packings, U-seals, V-ring packings, and other forms of lip packings are used primarily to seal reciprocating motion. The packing material is usually leather, solid rubber, or fabric-reinforced rubber, although other compounds are available for difficult applications. An advantage of leather packings is a low coefficient of friction, on the order of 0.006 to 0.008 depending on the tanning process. Low friction increases the life of a packing because less heat is generated.

Cup packings (Fig. 17.14) were one of the first types of piston seals for hydraulic and pneumatic applications. The fluid pressure expands the cup outward against the cylinder wall and thus seals the piston in the cylinder. This action requires that a double-acting cylinder have two packings in order to seal the pressure in both directions of operation. The inner portion of the piston in Fig. 17.14 is a boss to prevent excessive tightening of the washer against the cup. If the cup is crushed against the piston, good sealing will not be obtained.

Figure 17.15 shows elastomeric U-seals on a double-acting piston. This type of seal is also used on piston rods. They have approximately the same pressure limitations as O-rings, and backup rings are required for higher pressures. When a U-seal is made of leather, a filler is required between the lips to prevent collapse of the seal.

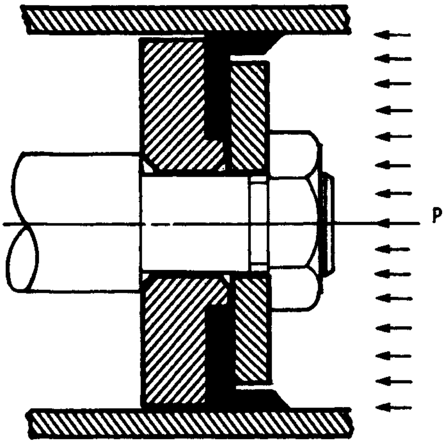


FIGURE 17.14 Cup packing for single-acting cylinder.

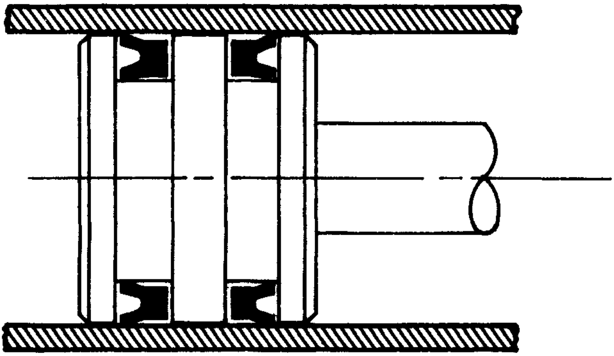


FIGURE 17.15 U-seals for a double-acting piston.

Rod scrapers are used on piston rods of hydraulic cylinders that are exposed to harsh environments. The purpose is to exclude mud, dust, and ice from the cylinder. A typical rod scraper is composed of a polyurethane element bonded to a metal shell which is pressed into the end cap of the cylinder. Molybdenum disulfide is sometimes added to the polyurethane to reduce friction. A rod scraper added to the end cap of a cylinder is shown in Fig. 17.16. The sealing lip is pointed outward to remove foreign material when the piston rod is retracted.

The chief use of the V-ring packing (Fig. 17.17) is for sealing piston rods or reciprocating shafts, although it can also be used to seal pistons. The ability to seal fluids under pressure depends on the type of packing material and number of packings used. The V-ring packing is considered superior to other lip types for sealing high pressures, especially above 50 000 psi (345 N/mm^2).

The V shape of the packing is obtained through the adapters that support the rings. Fluid pressure then expands the packing against the shaft and housing to seal

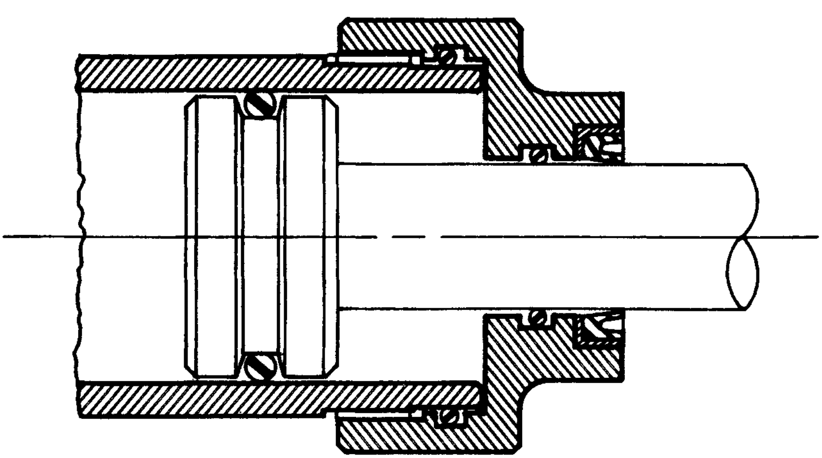


FIGURE 17.16 Rod scraper on piston rod.

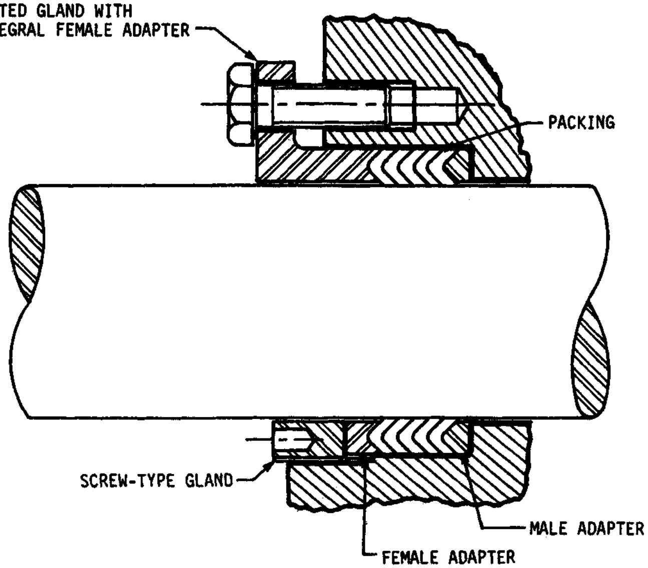


FIGURE 17.17 V-ring packing for a reciprocating shaft.

the fluid. A continuous packing provides better sealing than a series of split rings, although the latter are easier to install and remove.

17.3.3 Piston Rings

Piston rings for automotive engine applications are made of gray cast iron. The rings are used to seal gases in the cylinder and to restrict oil to the crankcase.

Piston rings must be split for assembly over the piston. This requires shaping the ring so that it will provide a uniform radial force against the cylinder wall. Piston ring manufacturers have developed methods of attaining this objective.

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