Section V – Dynamic O-Ring Sealing

5.1 Introduction 5-2
5.2 Hydraulic Reciprocating O-Ring Seals 5-2
5.3 Surface Finishes 5-3
5.4 Temperature Effects on Dynamic Seals 5-5
5.5 Side Loads 5-5
5.6 Direction of Pressure 5-5
5.7 Shock Loads and Pressures 5-5
5.8 High Frequency Motion or Vibration 5-6
5.9 Squeeze 5-6
5.10 Stretch 5-6
5.11 Friction 5-7
5.11.1 Break-Out Friction 5-7
5.11.2 Running Friction 5-7
5.12 Calculate Rubbing Surface 5-8
5.13 Methods to Reduce Friction 5-8
5.14 Friction and Wear 5-8
5.14.1 Friction 5-8
5.14.2 Pneumatic Seals 5-11
5.14.3 Wear 5-12
5.14.4 Interdependence of Friction Wear and an Effective Seal 5-12
5.15 Spiral Failure 5-13
5.15.1 Speed of Stroke 5-14
5.15.2 Lack of Lubrication 5-14
5.15.3 Pressure Differential and Direction 5-14
5.15.4 Squeeze 5-14
5.15.5 Shape of Groove and Split Groove 5-14
5.15.6 Temperature of Operation 5-14
5.15.7 Length of Stroke 5-14
5.15.8 Surface Finish 5-14
5.15.9 Back-Up Rings 5-14
5.16 Modifications for Special Applications 5-14
5.16.1 Small Amount of Leakage 5-15
5.16.2 Early Stress-Aging 5-15
5.16.3 Low Temperature Leakage 5-15
5.16.4 Excessive Swell (above 20%) 5-15
5.17 Gland Dimensions for Reciprocating Hydraulic O-Ring Seals 5-15
5.18 Floating Glands 5-15
5.19 Pneumatic Reciprocating O-Ring Seals 5-16
5.20 Temperature 5-16
5.21 Silicone Compounds 5-16
5.22 High-Pressure 5-16
5.23 Lubrication 5-16
5.24 Gland Dimensions 5-16
5.25 Floating Seal 5-16
5.26 Uni-Directional Gland 5-17
5.27 Rotary Seal 5-17
5.28 Oscillating Seal 5-19
5.29 Seat Seal 5-19
5.30 Drive Belts 5-20
5.30.1 Calculation of Drive Belt Open Design 5-20
5.30.2 Calculation of Drive Belt Crossed Design . . 5-21
5.31 O-Ring Glands 5-21
5.31.1 O-Ring Glands for Aerospace Hydraulic Packings and Gaskets 5-21
5.31.2 O-Ring Glands for Industrial Reciprocating Seals 5-30
5.31.3 O-Ring Glands for Pneumatic Floating Piston Ring Seals 5-35
5.31.4 O-Ring Glands for Rotary Seals 5-39
5.32 Dynamic Vacuum Sealing 5-44

Dynamic O-Ring Sealing

5.1 Introduction

Dynamic O-ring sealing applications are considerably more involved than static applications due to the implied motion against the O-ring seal interface. Resistance to fluids must be more carefully scrutinized than in conventional static seal designs since a volumetric increase in the O-ring in excess of approximately 20% may lead to friction and wear difficulties, and only a minimum of shrinkage (at most 4%), can be tolerated.

The metal or other surface over which the O-ring will move also becomes critical. It must be hard and wear resistant. It also must be sufficiently smooth so that it will not abrade the rubber, and yet there must be small microfine "pockets" on the moving surfaces to hold lubricant.

The greatest dynamic use of O-rings is in reciprocating hydraulic rod and piston seals. These are discussed first, but many of the ideas expressed are also applicable to other dynamic applications. Considerations applying only to other types of dynamic seals are discussed in greater detail later in the section.

5.2 Hydraulic Reciprocating O-ring Seals

O-rings are best when used on short-stroke, relatively small-diameter applications. Millions of O-rings however, are used very successfully in reciprocating hydraulic, pneumatic, and other fluid systems which employ long stroke, large diameter seals. If designed properly, an O-ring seal will give long, trouble-free service. The following discussion is presented so that common troubles and misuses can be avoided.

If the engineer or designer is to become his own seal expert, he must learn the basic types and causes of seal failure. In this section we present a discussion of failures and causes of various seal failure modes even though it may overemphasize the problems.

Reciprocating seals are affected by extrusion, breathing, surface finish of the metal, and hardness of the seal as discussed in O-Ring Applications, Section III. These factors should therefore be considered in any reciprocating gland design. There are also additional factors discussed in this chapter that must be considered in order to avoid future difficulty.

Materials for the surface(s) over which moving O-rings slide should be chosen carefully. Those that give the maximum life to moving O-ring seals are: Cast iron or steel for bores, hardened steel for rods, or hard chrome plated surfaces.

Soft metals such as aluminum, brass, bronze, monel and some stainless steels should be avoided in most dynamic applications, although they may be used in low-pressure pneumatics. If the cylinder bore surface can be hardened, as by carburizing, cylinder life will be increased. Hardness of the piston should always be lower than the cylinder walls to minimize the possibility of damage to the cylinder bore surface.

Preferably, metallic moving surfaces sealed by an O-ring should never touch, but if they must, then the one containing the O-ring groove should be a soft bearing material. It is impossible to run a highly polished piston rod through a hard bearing without inflicting scratches on the rod. It is likewise impossible to slide a hard piston in a highly polished cylinder and not inflict scratches on the cylinder wall. The scratches are usually caused by small hard particles that are loosened and picked up by the oil which sooner or later become jammed between the moving surfaces and score them. Though they may be hairlines, they are longitudinal scratches and will therefore reduce sealing efficiency and life of the O-ring.

The most satisfactory bearing material tried for this purpose is babbitt metal. Babbitt makes an excellent bearing and the hard particles become imbedded and captured in it without damage to the hardened rod. In fact after millions of cycles, the babbitt imparts a glass-like finish to the rod. Nylon may also be used as a bearing material, but the bearing may need to be split in some fashion to allow for nylon's relatively high coefficient of thermal expansion.

In a suggested design, Figure 5-1, the piston is surfaced with babbitt. The gland is also lined with babbitt. The O-ring may be located in the babbitt lining or in the supporting metal which should be relieved 0.051 or 0.076 mm (0.002 or 0.003 inches) so there will be no chance of the hard metals running together.

Lubrication, as explained in O-Ring Application, Section III, is useful in all O-ring seals. It is doubly important in dynamic applications where a lubricating film between the O-ring, and the surface it slides over, will protect the ring from abrasion, frictional heating and rapid wear.

In pneumatic applications, a back-up ring will trap some lubricant, and extend the useful life of seals that are lubricated infrequently. It will also help retain oil in applications powered with lubricated air.

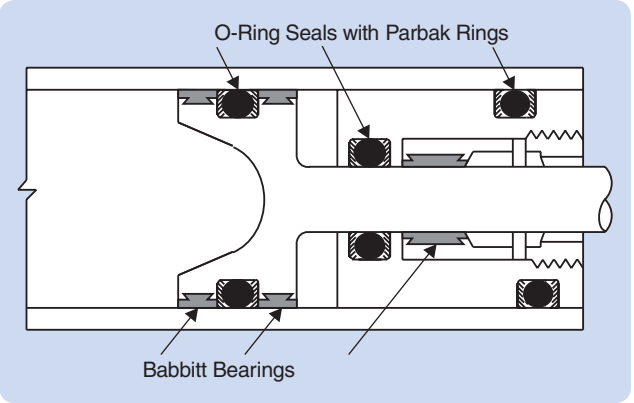

Figure 5-1: O-ring Seals with Bearings

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When a cylinder rod extends out into a dirty environment where it can pick up dirt, lint, metal chips, etc., this foreign material can nullify the effect of the best lubricant and cause rapid abrasive wear of both the O-ring and the rod. Equipment exposed to such conditions should be fitted with a wiper/scraper ring to prevent the dirt from reaching the O-ring seal. It is also good practice to install a felt ring between the scraper and the seal to insure proper lubrication of the rod on its return stroke. Figure 5-2 illustrates this concept.

A felt ring may cause corrosion in some installations, as felt also tends to collect moisture. A second O-ring may be used for the wiper, but it must not actually seal because a pressure trap condition is likely to develop between two reciprocating O-ring seals. This can be prevented by cutting the outer O-ring so it cannot seal. Since this can easily be forgotten, it is preferable to provide a vent hole between the two O-rings. It should vent downward so it will not become clogged with dirt. The sample problem provided in Table 5-1 explains how to design the gland for such an O-ring wiper.

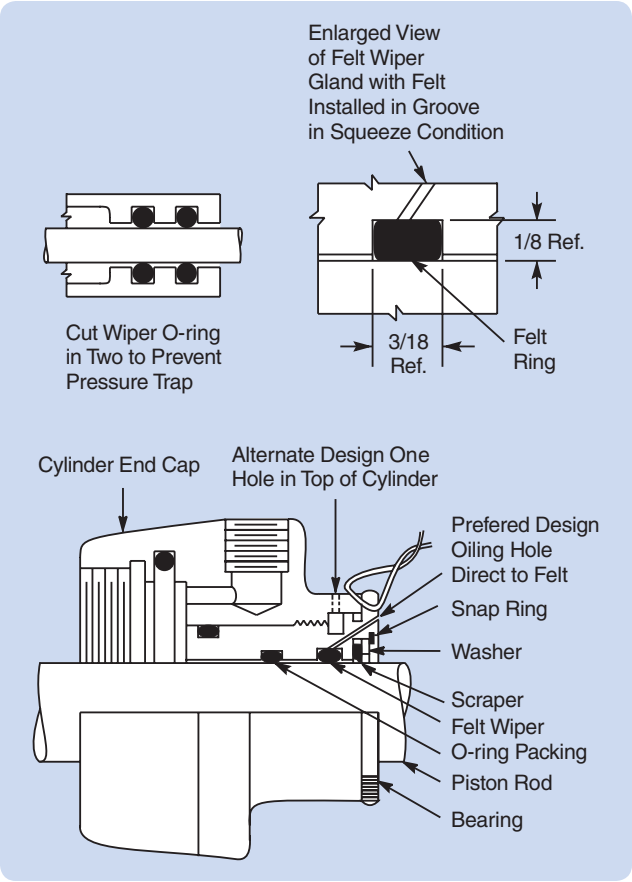

Figure 5-2: Lubrication Between Scraper and Seal Rings

5.3 Surface Finishes

Finishes of contact surfaces have much to do with the life of dynamic O-ring seals. Limits of maximum roughness for glands are given on the drawings accompanying the design charts in this section and represent accepted practice for military and industrial use. Surface roughness values less than 5 micro-inches are not recommended for dynamic seals, however, as an extending rod will be wiped completely dry and will not be lubricated when it retracts. The surface must be rough enough to hold small amounts of oil. Ideally, a microscopic "orange peel" type of surface is best, presenting smooth rounded surfaces for the O-ring to slide on, with small crevices between to act as oil reservoirs. This kind of surface may be approximated by peening the rod with metal shot or glass beads. An even better surface can be obtained by electropolishing. The most desirable surface roughness value is from 10 to 20 micro-inches.

The roughness of a surface as measured comprises several elements which can be handled separately according to DIN 4760:

Level 1 — dimensional deviations within tolerance band

Level 2 — surface undulations (waves)

Levels 3 to 5 — range of roughness

All these deviations from the ideal finish are superimposed as measurements are carried out and represent the surface roughness (see Figure 5-3).

Surface finish is often quantified in terms of R_t and R_a (see Figure 5-4). R_t is the vertical distance between the highest and the lowest peaks in a roughness profile over a test length l_m. R_t is increasingly being replaced by the maximum depth of roughness, R_{max}. R_{max} is the greatest single roughness found in five consecutive single trace lengths lm.

Problem: To design a wiper gland for a 25.4 mm (1.000 in.) OD piston rod.

Procedural Steps:	Example:
(A) Select O-ring with actual ID slightly smaller than Rod OD, B.	(A) Parker No. 2-214 (ID = 0.984)
(B) Divide the actual minimum squeeze given in Design Chart A6-5 for this O-ring size by two (the same squeeze is permissible in most cases).	(B) Squeeze 0.012/2 = 0.006
(C) Add this amount to both max. and min. gland depth, L, given in Design Chart A6-5 to get proper gland depth for wiper, L_W.	(C) L_W min = 0.121 + .006 = 0.127 L_W max = 0.123 + .006 = 0.129
(D) Calculate balance of gland dimensions same as for piston rod seal.	

Table 5-1: Wiper Gland Design Example

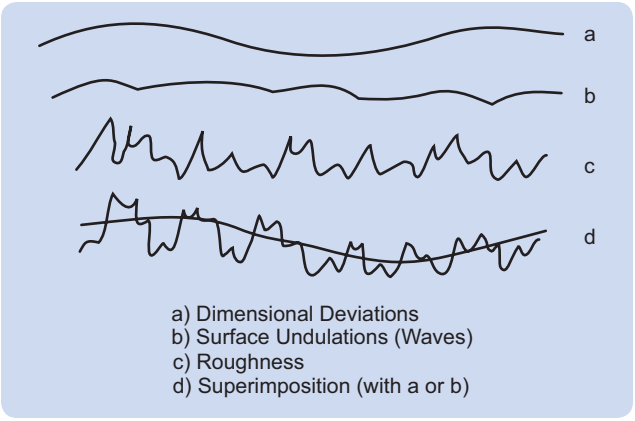

Figure 5-3: Surface Finish Structure

This is given in Figure 5-4 by the roughest profile Z_4. In this case $Z_4 = R_{max}$ does not include extreme roughness peaks as is the case of R_t.

The medium roughness value R_a is an arithmetic mean of all components of the roughness trace within the trace length l_m. The average roughness value R_z of five consecutive trace lengths often is preferred to R_e.

If R_a is known, R_z can be taken from Figure 5-5 and vice versa. Figure 5-5 is taken from DIN 4768, part 1, attachment 1. Should R_z reach the upper portion of the graph, it can be assumed that the specified R_a values will not be exceeded.

The lower limits would be taken if an R_z value should be specified.

Finally, the depth of roughness R_p also is of interest and is the vertical distance between the highest point on the roughness trace and the center line of that trace.

Values for R_t are of very little assistance in reaching a conclusion regarding the suitability of a surface roughness from the sealing point of view. Table 5-1 shows that for a similar R_t all levels of roughness can be produced. R_a values are unsuitable for comparison because profiles 6 and 7 have the same R_a value. R_p values without reference to the load area t_p also gives a false impression of roughness.

A static sealing surface $R_t \leq 6.3$ µm ($\overline{VVV}$ roughness DIN 3141) is rougher than the dynamic surface requirements. Seal manufacturers recommend a roughness $R_t \leq 2.5$ µm for a dynamic sealing surface ($R_a = 0.25$ to 0.5 mm) ($\overline{VVV}$ roughness DIN 3141) when the load area is over 50%, or when the surface finish roughness R_p is under 50%. These limitations often are overlooked, nevertheless the connection between surface finish and load area is very important because an "open" profile can have sharp edges (e.g., profiles 2 through 6 in Table 5-2). These open profiles are a product of cutting processes such as turning or grinding. A much larger load area is produced by cold forming processes such as rolling, drawing or sinking.

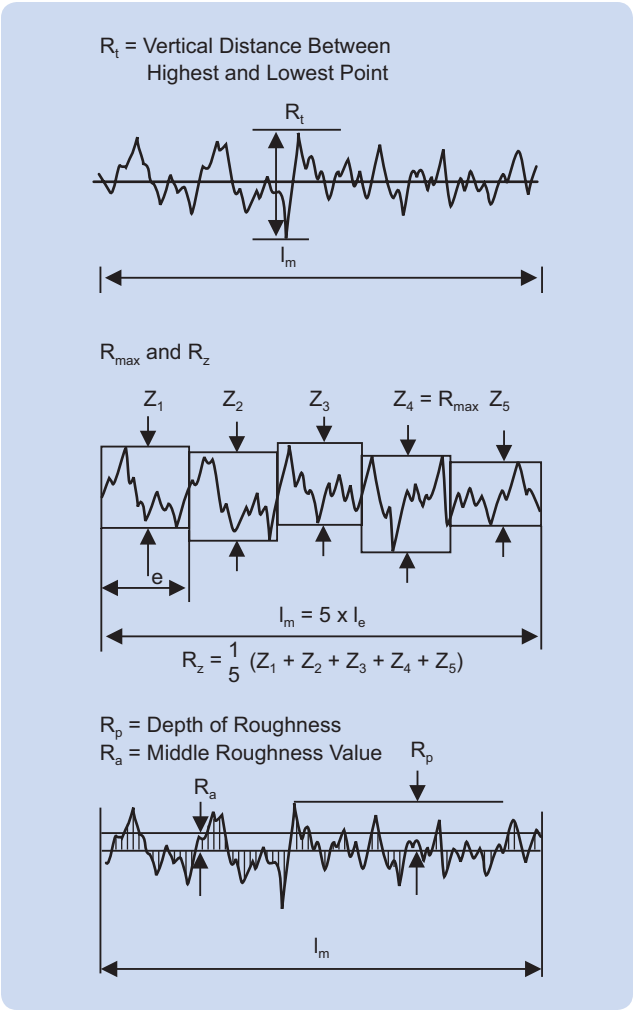

Figure 5-4: Roughness Terminology

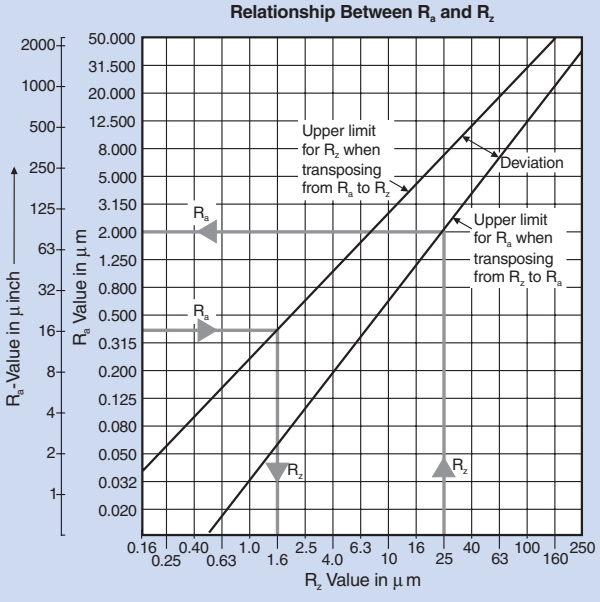

Figure 5-5: Relationship Between R_a and R_z

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It can be clearly seen from Figure 5-6 that surfaces produced by roller burnishing have no sharp peaks which can cause damage to a seal. Further, the valleys form potential lubrication reservoirs which improve the dynamic behavior of a seal.

Surface finish values obtained in a single test are possibly not typical. For this reason several readings should be taken. When several results are to be compared, the length of the test surface must be stated — for different trace lengths, results are not comparable because they result from other profile heights.

5.4 Temperature Effects On Dynamic Seals

High Temperatures — It should be remembered that the higher the temperature (above 38°C) (100°F) in and around a reciprocating gland, the more critical the application becomes. The higher the interface temperature, the greater the tendency of the lighter fractions of the oil to evaporate from an exposed surface. Lack of lubrication will cause greatly accelerated seal wear. If the temperature is high enough, the tacky residue (resins) which remains after oil evaporation will char and create a hard, abrasive surface which, if not removed, will quickly abrade away the seal until leakage or complete seal failure occurs.

Low temperatures — Low temperature environments are most troublesome, especially if the seal has been operating at a high temperature for some time. This is because the elastomer in the seal will take a compression set at high temperature. When the seal is then subjected to low temperature, there may be insufficient elastic memory to overcome the relatively high coefficient of shrinkage (10 times that of steel) at low temperatures.

	R_t	R_p	R_a	t_p (%)		
	µm	µm	µm	0.25	0.50	0.75 R_t
1. R_t	1	0.5	0.5	50	50	50
2. R_t	1	0.5	0.5	50	50	75
3. R_t	1	0.5	0.5	50	50	75
4. R_t	1	0.75	0.28	12.5	25	37.5
5. R_t	1	0.25	0.28	62.5	75	87.5
6. R_t	1	0.785	0.188	3.5	14	35
7. R_t	1	0.215	0.188	65	86	96.5
8. R_t	1	0.5	0.39	43	50	57

Table 5-2: Diagramatic Representation of Surface Profiles

Once unseated from a spot on a given metal surface, the seal must be reseated by internal seal resilience or system pressure. Therefore, it is much easier to seal a hydraulic system that goes from zero-pressure to high-pressure almost instantaneously. Low-pressure fuel, pneumatic, oil, and similar fluid systems are prone to leak if an O-ring is used as a dynamic seal at -54°C (-65°F) because there is insufficient pressure to keep the O-ring tightly seated during and immediately after motion of the gland. Remember that the -54°C (-65°F) compound is flexible and capable of acceptable seal performance at -54°C (-65°F) but may not be resilient below -43°C (-45°F).

5.5 Side Loads

Side loads on a piston or rod can cause the clearance in the gland to be on one side only. If excess clearance is created by side-loading, extrusion will result. If adequate squeeze has not been applied, leakage will result. The higher unit load on the opposite side causes uneven friction on the seal, and if high enough, the rod or barrel will be galled or scored.

5.6 Direction of Pressure

The placement of a groove can be determined from the direction of the system pressure in relation to the direction of the moving friction force. If the friction of the moving metal surface across the O-ring is in the same direction as the direction of pressure, the O-ring will tend to be dragged into the gap more readily and thus extrude at only 30 to 40% of the pressure normally necessary to cause extrusion. By placing the groove in the opposite metal part, any friction will work against pressure. Snubbing cylinders, in which the motion and force create the pressure, are the usual culprits.

5.7 Shock Loads and Pressures

Shock pressures, such as those created by the sudden stopping of a rapidly descending hydraulic hoist cylinder on which there is a heavy load, are often far in excess of the pressure for which the seal and the system were designed. The same could be said about the whip of a gun barrel, of a tank on rough roads, or a truck tailgate and others if they

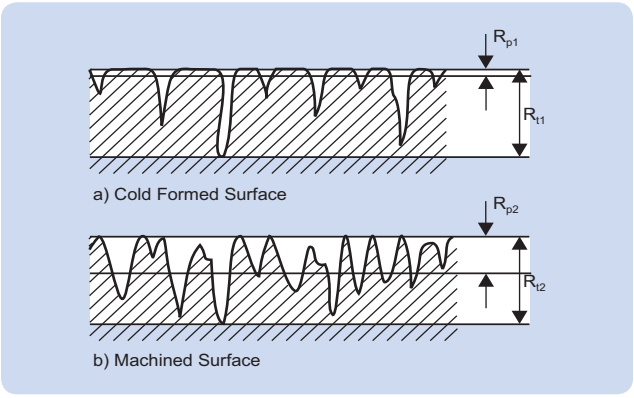

Figure 5-6: Surfaces Produced by Roller Burnishing (a) and by Normal Machining (b)

are designed to ride on the hydraulic system during transit. Transient pressures of 690 Bar (10,000 psi) are not uncommon in these cases. A mechanical lock or brake should be provided to hold a position once it is attained. The hydraulic cylinder should be used only to raise and lower the load if it does not have a relief valve within it to prevent excessive pressure build-up by shock loads.

5.8 High Frequency Motion or Vibration

O-rings or other seals can be worn excessively by small frequent motions which are usually encountered when equipment is in transit. For example: the tilt cylinder of a lift truck, a hydraulic tailgate lift, and a road scraper blade. Normally, the hydraulic cylinder is intended as an actuator and not as a locking device or a snubber. It will be noted that brick pavements and dirt roads cause the most trouble when this type of effect is encountered. A mechanical lock is also recommended as a cure in this case.

5.9 Squeeze

The best squeeze for a reciprocating O-ring seal must be a compromise of all the factors involved. The design tables in this chapter are generally satisfactory. The greater the temperature range to be sealed, the greater the squeeze that is needed. The same is true if low pressure or vacuums are encountered. On the other hand, too much squeeze will cause excessive friction, wear, and occasionally spiral failure. Some rubber compounds require more squeeze than others in order to seal. The nitrile (buna-N) base compounds are recommended whenever possible because they are more extrusion-resistant, more wear-resistant, and require less squeeze to seal, than any other oil-resistant rubber developed to date.

The military services have found that more than 0.432 mm (0.017") squeeze (per side) on a 5.334 mm (0.210") cross section makes an O-ring prone to spiral failure. Yet much less than this amount of squeeze will allow leakage at low temperature.

As discussed before, the amount of squeeze is a vital factor in friction. Therefore, one should carefully consider the squeeze applied to the O-ring in any gland design.

Squeeze is actually necessary only during periods of very low or no pressure sealing because at high pressures the O-ring seeks the path of least resistance, the clearance gap, and tends to seal tighter and tighter as the pressure is increased.

Enough squeeze must always be provided to offset the great difference in coefficient of shrinkage of the rubber and the metal, take up the tolerances of the metal and rubber parts, and compensate for the shrinkage (if any) of the rubber in the fluid. The following example illustrates how the squeeze can vary in a typical piston installation:

Consider Parker size 2-012 and Design Table 5-2:

1. With perfect concentricity

$$\text{Gland Depth, } L_{max} = \frac{0.501 - 0.387 = 0.057}{2}$$

$$\text{Radial clearance, max} = \frac{0.501 - 0.496 = 0.0025}{2}$$

Cross section, W_{min} = .067

Reduction of W, due to installation stretch = 0.003 (see Figure 3-3)

W_{min}, installed = 0.064

less L_{max} = 0.057 (from 1. above)

squeeze, min = 0.007

2. With maximum radial displacement (piston tangent with bore)

squeeze, min = 0.007 (from 1. above)

radial piston shift, max = 0.0025

squeeze = 0.0045 min possible

3. With maximum eccentricity of 0.002 T.I.R. between piston and groove OD

squeeze, min = 0.0045 (from 2. above)

radial piston shift, max = 0.0010

squeeze, min. = 0.0035 with adverse tolerance build-up.

If the O-ring is made in a compound that will shrink in the fluid, the minimum possible squeeze under adverse conditions then must be at least .076 mm (.003").

5.10 Stretch

When an O-ring must be stretched more than two or three percent as installed in a piston groove, the reduction in the squeeze diameter that results should be allowed for in determining the gland depth so that the desired percent squeeze will be applied to the reduced section. The percent of stretch should therefore be checked whenever the catalog gland dimensions are not used.

Large diameter O-rings may fit the piston so loosely that they must be carefully stuffed into the groove as the piston enters the cylinder to prevent damage. For these, the danger of damage is reduced if the next smaller size O-ring is used. Since this will likely cause a stretch close to five percent, it will usually be necessary to adjust the gland depth as mentioned above. See Figure 3-3 for the reduction in squeeze diameter with stretch.

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5.11 Friction

Friction, either break-out, running, or both, can become troublesome in some applications. At any given time, there are anomalies and difficulties in the prediction of developed friction. These are accentuated if one of the surfaces involved is deformable as in O-ring piston or shaft seals. An understanding of the principles may prove helpful in the solution of specific problems.

5.11.1 Break-Out Friction

In addition to the usual causes of running friction: hardness of the rubber, type of surface, surface finish, squeeze on the O-ring, amount and type of lubrication, fluid pressure/temperature, the amount of break-out friction that a system will generate depends on the length of time the surfaces of the metal and the seal element have been in physical contact at rest. See Figures 5-7 and 5-8.

The theory has been proposed and generally accepted that the increase of friction on standing is caused by the rubber O-ring flowing into the microfine grooves or surface irregularities of the mating part. As a general rule for a 70 durometer rubber against an 8 micro-inch surface, the maximum break-out friction that will develop in a system is 3 times the running friction. This ratio can be reduced by the use of a softer rubber. Table 5-3 shows some of the factors which may be used to adjust friction.

Coefficient of friction has little bearing on lubricated rubber's break-out and running friction. The other variables listed are much more important in the practical solution to problems.

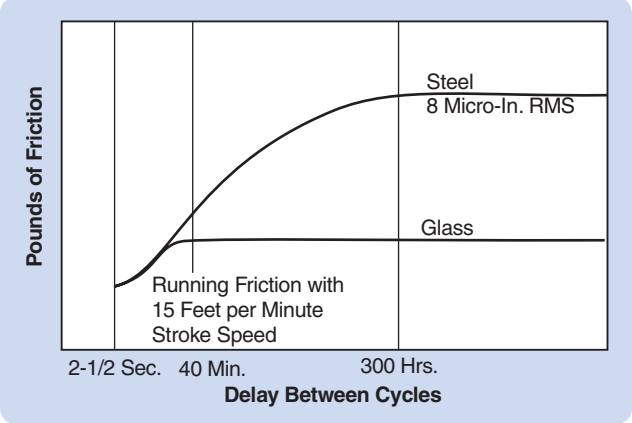

Figure 5-7: Change of O-ring Friction with Time at Rest

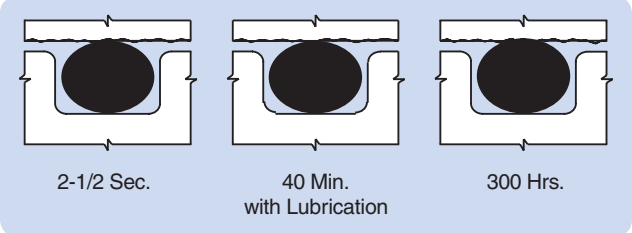

Figure 5-8: Flow of O-Ring into Metallic Surfaces

5.11.2 Running Friction

High running friction may cause difficulty by wearing soft metal parts. Metals such as copper, brass or aluminum can be rapidly worn away by a moving O-ring. This is especially true if high pressures are involved. If unexplained leakage occurs with these or other soft metals, it is good practice to check the metal dimensions for signs of wear.

The following formulas may be used for estimating the running friction of O-rings.

Piston Groove	Rod Groove
$F_C = f_c \times L_p$	$F_C = f_c \times L_r$
$F_H = f_h \times A_p$	$F_h = f_h \times A_r$
$F = F_C + F_H$	$F = F_c + F_H$

A_p = Projected area of seal for piston groove applications.

A_r = Projected area of seal for rod groove applications.

F = Total seal friction in pounds.

F_C = Total friction due to seal compression.

F_H = Total friction due to hydraulic pressure on the seal.

fc = Friction due to O-ring compression obtained from Figure 5-9.

f_h = Friction due to fluid pressure obtained from Figure 5-10.

L_p = Length of seal rubbing surface in inches for piston groove applications.

L_r = Length of seal rubbing surface in inches for rod groove applications.

Example:
Parker 2-214 rubbing against OD of O-ring at 103.5 Bar (1500 psi), 10% compression, 70 durometer:

$F_C = 0.7 \times 3.93 = 2.75$
$F_H = 48 \times 0.44 = 20.90$
$F = F_C + F_H = 23.65$ pounds

Data for the coefficients (fc and fh) are given in Figures 5-9 and 5-10. Projected areas and lengths of rubbing surface are given in Table 5-4.

Friction Factors (In Order of Importance)

To Increase Friction	Factor	To Decrease Friction
Increase	Unit Load (squeeze)	Decrease
Increase RMS	Surface Finish (metal)	Decrease RMS
Increase	Durometer	Decrease
Decrease	Speed of Motion	Increase
Increase	Cross Section of O-Ring	Decrease
Increase	Pressure	Decrease
Omit Lubrication	Lubrication	Use Lubrication
Decrease	Temperature	Increase
Decrease	Groove Width	Increase
Increase	Diameter of Bore or Rod	Decrease
Decrease	Surface Finish (O-Ring)	Increase
Stretch O-ring	Joule Effect*	Compress O-Ring
Lower Durometer of O-ring	Coefficient of Friction#	Increase Durometer

* Refer to rotary seals.
\# A minor factor and should be ignored in design work other than for ultra high speeds.

Table 5-3: Friction Factors

5.12 Calculate Rubbing Surface

The areas and lengths given in Table 5-4 are based on the dimensions given in Design Table 5-2 at the end of this section. If the application differs, use dimensions from the applicable table, i.e. Table Design 5-1 for aerospace, and calculate the area and length.

The following example illustrates the procedure:

Projected Area: $A_p = (\pi/4)\,[A^2_{max} - (B\text{-}1)^2_{min}]$

$A_r = (\pi/4)\,[(A\text{-}1)^2_{max} - B^2_{min}]$

Rubbing Surface Length: $L_p = \pi A_{max}$

$L_r = \pi B_{max}$

For Parker Size No. 2-113: $A_{max} = 0.751$ $B_{min} = 0.559$

$A\text{-}1_{max} = 0.739$ $B\text{-}1_{min} = 0.571$

$B_{max} = 0.561$

Projected Area:

$A_p = (\pi/4)\,[(0.751)^2 - (0.571)^2] = 0.187$ sq. in.

$A_r = (\pi/4)\,[(0.739)^2 - (0.559)^2] = 0.184$ sq. in.

Rubbing Surface Length:

$L_p = 0.751\pi = 2.36$ in.

$L_r = 0.561\pi = 1.76$ in.

5.13 Methods To Reduce Friction

The foregoing formulas for estimating O-ring friction are intended for applications in which standard O-ring compound types are to be used in systems lubricated with hydraulic oil. In pneumatic or other dynamic applications, Parker Seal can help reduce friction in several ways. O-Lube and Super-O-Lube greases are available from Parker distributors, and O-rings may be ordered that have received special friction reducing treatments. These include internally lubricated rings and Lube Treated rings.

5.14 Friction and Wear

O-rings load a sealing surface due to their own resilience compounded with any system pressure. When the surface to be sealed moves relative to the O-ring, frictional forces are set up producing two effects: one leads to wear and the other reduces the useful load which a cylinder can transmit.

5.14.1 Friction

In dynamic applications difference must be made between break-out and running friction. Break-out friction must be overcome at the beginning of movement and also is known as start-up friction. Once movement is established the frictional forces drop to a lower level and gliding begins. This can be clearly seen in reciprocating cylinders.

The running friction of seals depends on countless factors making a mathematical analysis practically impossible. For this reason it is difficult to make exact statements regarding the level of friction which can be expected. The most important factors are:

Related to the seal:

- Geometrical form including production tolerances and resulting deformation;
- Hardness and surface finish;
- Friction values for dry and lubricated compound;
- Swell and temperature characteristics.

Related to the hydraulic fluid:

- Tendency to build up a lubricating film and its distribution;
- Viscosity and temperature/viscosity relationship.

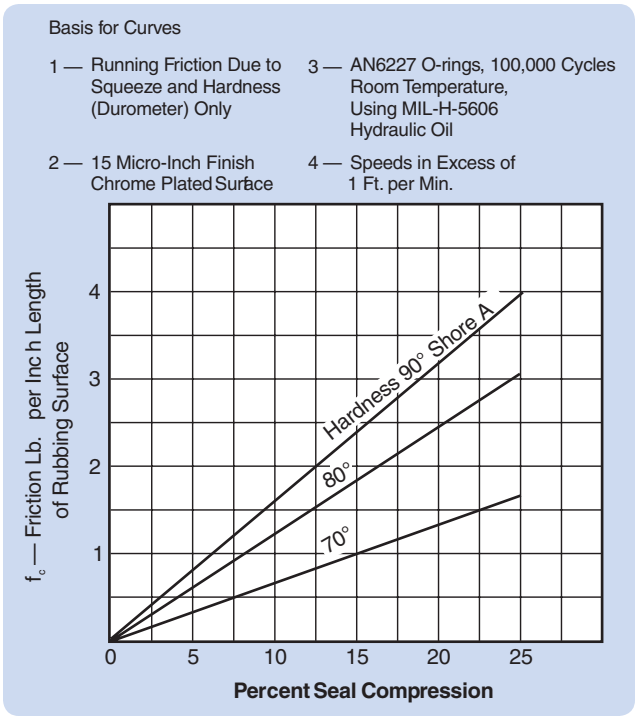

Figure 5-9: Friction Due to O-ring Compression

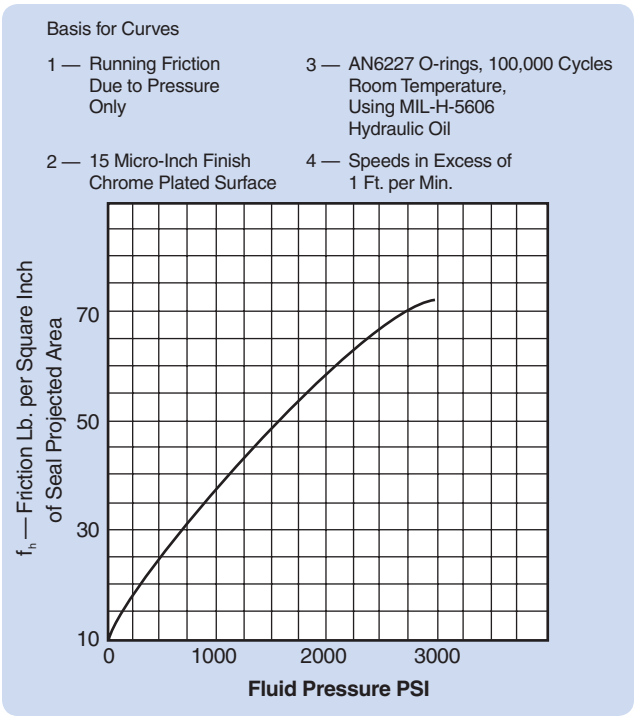

Figure 5-10: Friction Due to Fluid Pressure

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Projected Areas and Lengths of Rubbing Surface for O-Rings

O-Ring Size Parker No. 2-	Piston Groove A_p Sq. In.	Piston Groove L_p In.	Rod Groove A_r Sq. In.	Rod Groove L_r In.
006	.03	.79	.03	.39
007	.04	.89	.04	.49
008	.05	.98	.04	.58
009	.05	1.08	.05	.68
010	.06	1.18	.05	.78
011	.07	1.38	.07	.98
012	.08	1.57	.08	1.17
013	.09	1.77	.09	1.37
014	.10	1.97	.10	1.57
015	.11	2.16	.11	1.76
016	.12	2.36	.12	1.96
017	.14	2.56	.13	2.16
018	.15	2.75	.14	2.35
019	.16	2.95	.16	2.55
020	.17	3.14	.17	2.75
021	.18	3.34	.18	2.94
022	.19	3.54	.19	3.14
023	.20	3.73	.20	3.33
024	.21	3.93	.21	3.53
025	.22	4.13	.22	3.73
026	.24	4.32	.23	3.92
027	.25	4.52	.24	4.12
028	.26	4.72	.26	4.32
110	.13	1.77	.13	1.17
111	.15	1.97	.15	1.37
112	.17	2.16	.17	1.57
113	.19	2.36	.18	1.76
114	.20	2.56	.20	1.96
115	.22	2.75	.22	2.16
116	.24	2.95	.24	2.35
117	.26	3.14	.25	2.55
118	.28	3.34	.27	2.75
119	.29	3.54	.29	2.94
120	.31	3.73	.31	3.14
121	.33	3.93	.32	3.33
122	.35	4.13	.34	3.53
123	.36	4.32	.36	3.73
124	.38	4.52	.38	3.92
125	.40	4.72	.40	4.12
126	.42	4.91	.41	4.32
127	.43	5.11	.43	4.51
128	.45	5.30	.45	4.71
129	.47	5.50	.47	4.90
130	.49	5.70	.48	5.10
131	.50	5.89	.50	5.30
132	.52	6.09	.52	5.49
133	.54	6.29	.54	5.69
134	.56	6.48	.55	5.89
135	.58	6.68	.57	6.08
136	.59	6.88	.59	6.28
137	.61	7.07	.61	6.47
138	.63	7.27	.63	6.67
139	.65	7.46	.64	6.87
140	.66	7.66	.66	7.07
141	.68	7.86	.68	7.26
142	.70	8.05	.70	7.46
143	.72	8.25	.71	7.65
144	.73	8.45	.73	7.85
145	.75	8.64	.75	8.05
146	.77	8.84	.77	8.24
147	.79	9.04	.78	8.44
148	.80	9.23	.80	8.64
149	.82	9.43	.82	8.83
210	.34	3.15	.34	2.35
211	.36	3.34	.36	2.54
212	.39	3.54	.38	2.74
213	.41	3.74	.41	2.94
214	.44	3.93	.43	3.14
215	.46	4.13	.46	3.33
216	.48	4.33	.48	3.53
217	.51	4.52	.50	3.72
218	.53	4.72	.53	3.92
219	.56	4.91	.55	4.12
220	.58	5.11	.58	4.31
221	.61	5.31	.60	4.51
222	.63	5.50	.63	4.71
325	1.00	5.90	.99	4.71
326	1.07	6.29	1.07	5.10
327	1.14	6.68	1.14	5.49
328	1.22	7.07	1.21	5.88
329	1.29	7.47	1.29	6.28
330	1.36	7.86	1.36	6.67
331	1.44	8.25	1.43	7.06
332	1.51	8.65	1.51	7.45
333	1.58	9.04	1.58	7.85
334	1.66	9.43	1.65	8.24
335	1.73	9.82	1.73	8.63
336	1.81	10.22	1.80	9.03
337	1.88	10.61	1.87	9.42
338	1.95	11.00	1.95	9.81
339	2.03	11.40	2.02	10.20
340	2.10	11.79	2.10	10.59
341	2.17	12.18	2.17	10.99
342	2.25	12.58	2.24	11.38
343	2.32	12.97	2.31	11.77
344	2.39	13.36	2.39	12.16
345	2.47	13.75	2.46	12.56
346	2.54	14.15	2.54	12.95
347	2.62	14.54	2.61	13.34
348	2.69	14.93	2.68	13.73
349	2.76	15.32	2.76	14.13
425	3.59	15.72	3.57	14.13
426	3.69	16.11	3.66	14.52
427	3.78	16.51	3.76	14.91
428	3.87	16.90	3.85	15.31
429	3.97	17.29	3.95	15.70
430	4.06	17.68	4.04	16.09
431	4.16	18.08	4.14	16.48
432	4.25	18.47	4.23	16.88
433	4.35	18.86	4.32	17.27
434	4.44	19.25	4.42	17.66
435	4.53	19.65	4.51	18.05
436	4.63	20.04	4.61	18.45
437	4.72	20.43	4.70	18.84
438	4.91	21.22	4.89	19.63
439	5.10	22.00	5.08	20.41
440	5.29	22.79	5.27	21.20
441	5.48	23.57	5.46	21.98
442	5.67	24.36	5.64	22.77
443	5.85	25.15	5.83	23.55
444	6.04	25.93	6.02	24.34
445	6.23	26.72	6.21	25.12
446	6.61	28.29	6.59	26.69
447	6.98	29.86	6.96	28.26
448	7.36	31.43	7.34	29.84
449	7.74	33.00	7.72	31.41
450	8.12	34.57	8.09	32.98
451	8.49	36.14	8.47	34.55
452	8.87	37.71	8.85	36.12
453	9.25	39.28	9.22	37.69
454	9.62	40.85	9.60	39.26
455	10.00	42.42	9.98	40.83
456	10.38	43.99	10.36	42.40
457	10.75	45.57	10.73	43.97
458	11.13	47.14	11.11	45.54
459	11.51	48.71	11.49	47.11
460	11.89	50.28	11.86	48.69

Table 5-4: Projected Areas and Lengths of Rubbing Surface for O-rings

Related to the working conditions:

- Working pressure;
- Velocity of movement;
- Type of material and surface finish of surfaces;
- Working tolerances;
- Axial loads and wear bands on pistons.

These factors cannot be quantified because they overlap and act cumulatively.

At the beginning of a stroke the seal goes through three friction phases. Initially the seal is in direct contact with the sealing face with few lubricated fields, e.g., $\mu = 0.3$. Then follows a wider area of mixed friction where the coefficient of friction can drop as low as 0.06 to 0.08 according to the proportion of lubrication/non-lubricated areas (Figure 5-11). Finally, pure hydrodynamic friction which does not allow direct contact between the seal and the running surfaces is rarely reached.

As complete lubrication (= flooding) occurs, loss of fluid from a system increases.

Friction depends on a compound's sliding properties. Hardness and deformation of the seal influence the seal pressure. Specific seal pressure is in general related to, but not strictly proportional, to the system pressure.

The working pressure controls the width of clearance gaps and thereby the thickness of the lubricating film. The result depends on the geometry of the seal. Friction caused by O-rings increases with increasing pressure. Lip seals are more sensitive to pressure, friction increases quicker than with seals without a lip. This shows that the geometry of a seal directly affects the amount of friction.

Friction is proportional to the working pressure and therefore it is necessary to keep seal friction low, especially at low pressures.

Unfortunately, reduction of the sealing force also results in an increased tendency to leakage. This relationship can be modified within certain limits by selection of the seal geometry.

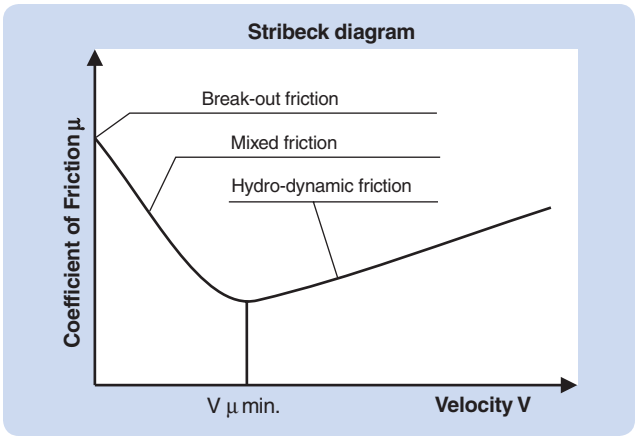

Figure 5-11: Stribeck Diagram

Normally the decision must be made between lower friction and high leakage.

Additionally, an unstable seal geometry due to swelling in the medium plays a role. Swelling means increase sealing force and increased friction.

When the medium is mineral oil it would seem that sufficient lubrication is assured. However, the seal geometry once again plays a role when, for example, a wiper seal scrapes a shaft dry. Leakage at a wiper seal will not occur until the seal wears. On the other hand lubrication can cause leakage amounting to the thick lubricating film with every stroke.

The optimum condition is a relatively thin lubricating film with sufficient adhesive properties.

The dynamic piston actually causes less friction with increasing velocity. In absolute terms there are very large discrepancies according to the thickness of the lubricating film. The reduction of friction with increasing velocity stems from the hydrodynamic properties of the lubricating fluid. This is also true for harder compounds. At low pressures the friction varies to the piston speed. At high pressures friction is seen to be more or less constant.

Friction is directly influenced by the seal diameter because the wear-area is greater. The greater the metal surface roughness, the more the contact surface consists of metallic “islands” and therefore again mixed friction occurs.

As in many other areas break-out friction of elastomers is significantly higher than running friction. Apart from compound type and seal geometry, tendency to adhesion, deformation, the down-time and the surface finish play a role in increasing break-out friction. The longer the down-time, the more lubrication is squeezed from between the seal and the running surface resulting in a non-lubricated vacuum. In this condition the level of starting friction approaches that for dry friction and is up to 10 times that found in running friction (Figures 5-12 and 5-11).

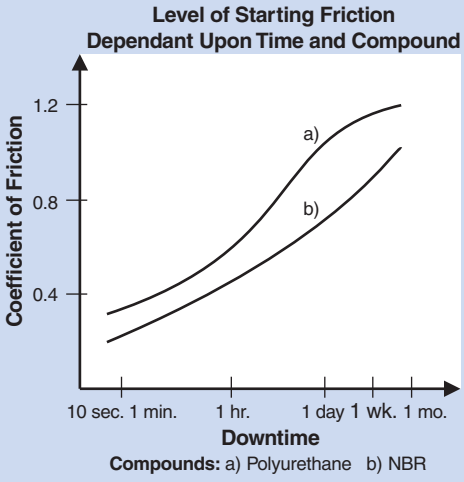

Figure 5-12: Level of Starting Friction Dependant Upon Time and Compound

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For the same conditions, friction at high temperature (= low viscosity) is high because the lubricating film is often interrupted.

The most important factors can be seen in Figure 5-13. Here friction is shown as a function of pressure and velocity. Figure 5-13 is valid only for a specific seal in a particular application. For other seals and applications the interdependence varies.

The stick-slip effect also is related to the friction at the sealing face. The friction, or better expressed the difference between break-out and running friction, plays an important role in evaluation and selection of a suitable elastomer.

Break-out friction occurs when the three following conditions are present:

- When the break-out friction is higher than the running friction a running velocity Vμ min (see Figure 5-11);
- The running velocity is Vμ min;
- The power is transmitted through the elastic body of the "compressible" oil.

To assist in the explanation of the term stick-slip, please refer to Figure 5-14. To accelerate a mass m from zero to maximum velocity, the break-out friction μH must be overcome by F1. The spring element is loaded with F1 and with increasing velocity the friction value μH reduces to μG and the force to F2. The potential energy stored in the spring accelerates the mass even further. When the stored energy is used, the mass is decelerated by the increasing friction in direction μH. This requires once again an increase in force level of F1, and the procedure repeats again.

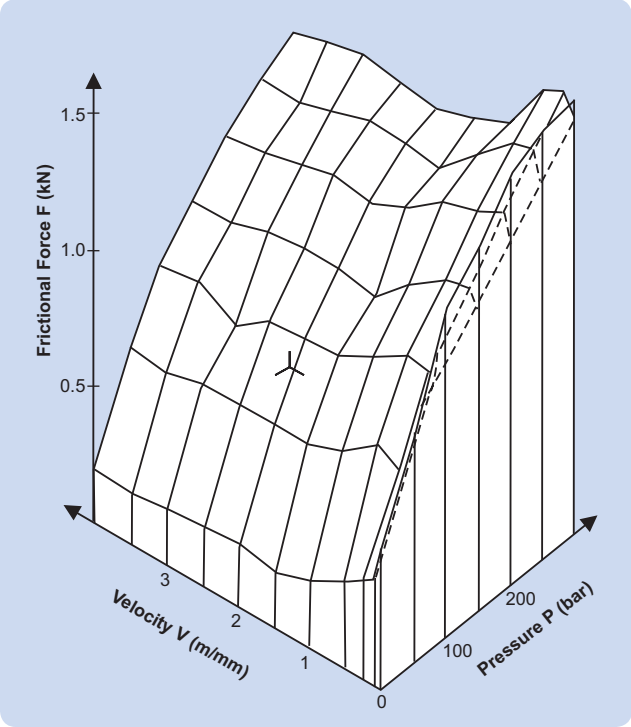

Figure 5-13: Frictional Force is Dependent Upon Pressure and Velocity – Compact Rod Seal 90° Shore A

Running velocity is a product of seal friction, the piston mass and the load. Of all these factors, only friction can be influenced and makes for a better relationship between sealing surface finish, lubricating film and surface finish. Certain improvements can be made making the system stiffer, this means the smallest possible oil volume under pressure on the hydraulic side.

Radial oscillation of the piston will occur when the lubricating film breaks down. Conversely oils with strong film building properties do not break down under the same working conditions using the same seals.

5.14.2 Pneumatic Seals

In principle the same conditions apply here as for the hydraulic seal, except that the effects of certain extreme conditions are more serious. This is particularly the case when lubrication is poor, as found when lubricated air is not available. Lubricated air gives more or less the same results as in a hydraulic application.

When lubricating grease is not continually replaced, it can eventually be removed by a seal lip. The effectiveness of lubrication with grease depends on the thickness of the original film and the running velocity of the seal (Figure 5-15).

The lower the velocity the thinner will become the lubricating film. With an O-ring seal the loss of grease can lead to total breakdown of the hydrodynamic lubricating film after only a few slow strokes.

Breakdown of the lubricating film after long operation also results in contact between the seal and the metal surfaces. This makes the seal move in the mixed friction range, the increase in friction causes high wear. The lubricating film therefore must be protected by rounding of the seal wiper edges and complete wiping of grease from the running surface must be prevented.

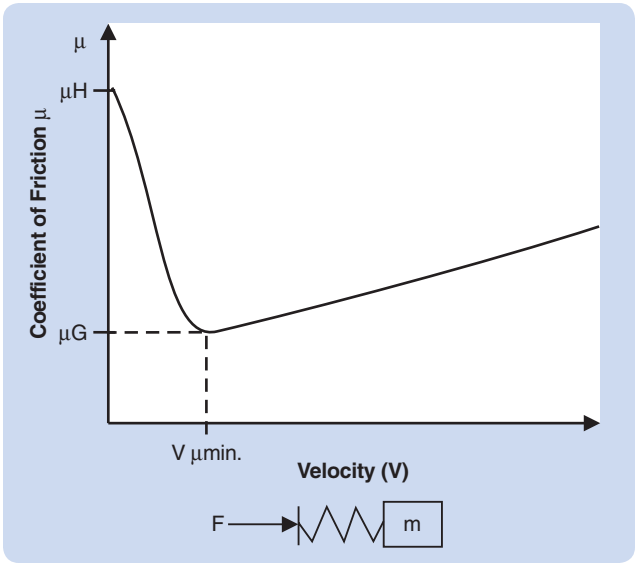

Figure 5-14: System Diagram for Stick-Slip Effect

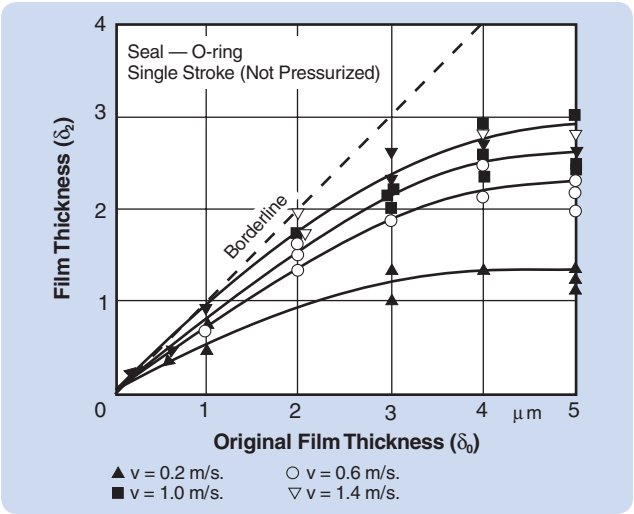

Figure 5-15: Change in Original Film Thickness as a Function of Running Speed During a Single Stroke (Border Line Δ Theoretical maximum Values)

This action has little effect upon the starting friction but brings a noticeable improvement in running friction levels.

If slow pneumatic piston velocities are achieved by throttling the pressurizing air, the risk of high stick-slip increases. Stick-slip is effected directly and negatively by long seal lips and sharp seal edges. An extremely rough or fine polished metallic running surface both cause equally higher stick-slip.

5.14.3 Wear

Friction causes wear. However, friction can be anticipated and taken into consideration in the design stage. The wear rate however is difficult to predict but directly governs the lifetime of an O-ring and the frequency of maintenance.

Today's high precision machinery tends in most cases to eliminate hydrodynamic lubrication because of the increased wipe-off effect. This means the seal always functions in semidry condition and for this reason wear resistance depends on:

- properties of the compound;
- lubricating properties of the medium;
- running surface roughness;
- working conditions.

Wear in fluid solutions can be divided into four groups:

- Scuff wear develops with metal-to-metal contact in the semidry condition where both materials tend to form mixed crystals. High Performance Lubricating (HPL) oils help to prevent this contact because of their additives. These additives have no influence in rubber/steel or rubber/metal combinations.
- Fatigue wear becomes evident when particles are released from the metal structure and is usually the result of pulsating loads.
- Corrosion wear manifests itself in the form of rust and can normally be reduced by suitable oil additives. Seals are not directly affected by the above types of wear. However, in dynamic applications particularly these wear conditions can cause the seal to fail through abrasion.
- Abrasive wear can affect both metallic and seal areas. Metals are abraded by hard compounds or by hard foreign matter in the medium. A rough metal surface normally is the cause of elastomer abrasion.

The seal user normally has no profound knowledge of seal wear characteristics. It is therefore recommended to consult the manufacturer about details of all extreme application conditions so that the correct seal can be offered.

5.14.4 Interdependence of Friction Wear and an Effective Seal

In order to obtain a problem-free seal it is necessary to have stability with regard to the clearance gap to avoid possible extrusion. However, stability is difficult to achieve because the relevant parameters often work conversely.

The first consideration is the lubricating film in the clearance gap. To estimate friction, lifetime and leakage it is necessary to know the width of the gap and how it varies under working conditions. To keep friction as low as possible the lubricating film should be fairly substantial. This, however, can result in leakage because the "thick" film is wiped off the rod surface during the return stroke. In the other extreme a lack of lubricating film causes problems due to high friction. The effectiveness of a seal and friction therefore are inversely proportional.

Hardness, together with the width and length of a clearance gap is very important. The hardness determines the elasticity of the seal and assures that the seal gives way to the lubricating film under pressure. The instantaneous viscosity of the fluid also plays an important role in resisting the wiping effect of the seal.

It is still not known which factors influence the lubricating film and which mechanisms act in the clearance gap. A soft compound favors a thicker film. Hard and soft compounds behave differently at high velocities, harder compounds help form a lubricating film whereas a soft compound will hinder this by strong adhesion to the running surface.

The lubricating film is very important but only one of the factors affecting seal friction. Other factors are, for example, the seal compound, seal shape, pressure, velocity, and changes in direction. Often many of these factors are difficult to measure or reproduce.

It is therefore quite understandable that seal manufacturers cannot give customers fixed figures regarding friction and wear for an individual seal. Information about seal lifetimes only can be made when all parameters affecting the seal are known and reproducible. General assumptions from a few tests are not acceptable because laboratory tests never can reproduce real working situations.

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5.15 Spiral Failure

A unique type of failure sometimes occurs on reciprocating O-rings which is called spiral failure. This name was given to this type of failure because when it occurs the seal looks as if it had been cut about halfway through the O-ring cross section in a spiral or corkscrew pattern. Oddly enough, the O-ring usually seals satisfactorily until a complete break or separation occurs at one place. Sometimes the seal is twisted in two without evidence of the spiral pattern, but in general, the same factors cause the break.

A properly used O-ring slides during all but a small fraction of any reciprocating stroke. This type of seal does not normally tend to roll or twist because:

1. The hydraulic pressure, acting through the O-ring, produces a greater holding force within the groove (friction on a larger area) than that produced by the sliding surface (rod or cylinder wall) opposite the groove (see Figure 5-16).
2. The smoother finish of the sliding surface, in relation to the groove surface-finish, produces less friction.
3. Running friction is lower than break-out friction.
4. The torsional resistance of the O-ring tends to resist twisting.

The conditions which cause spiral failure are those that simultaneously cause segments of the ring to slide and others to roll. A small amount of twisting is not detrimental but, when excessive, torsional failure or spiral failure will occur. True spiral failure occurs after the seal has been excessively twisted, but not broken, and then subjected to relatively high pressure. The twisted seal is forced into the sharp corner at the clearance gap by the pressure which puts an additional stress on this portion of the seal. Rapid stress-aging, or stress above the elastic limit of the rubber, causes a rupture of the O-ring to start adjacent to the clearance gap. Slight flexing, motion, or working of the O-ring apparently causes the rupture to penetrate about half way through the cross section. When the O-ring is removed from the gland, it returns to its original shape and the rupture appears as a tight spiral around the cross section.

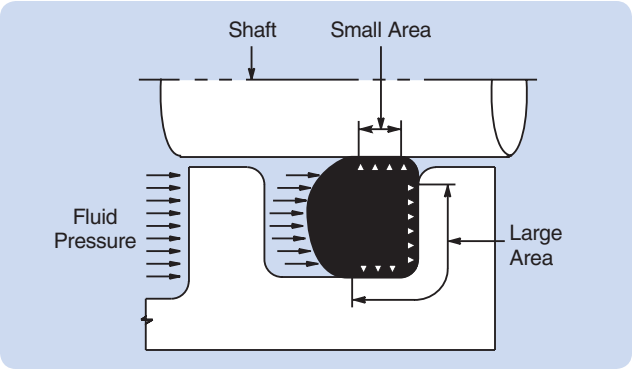

Figure 5-16: Action of Fluid Pressure to Prevent Rolling of O-ring

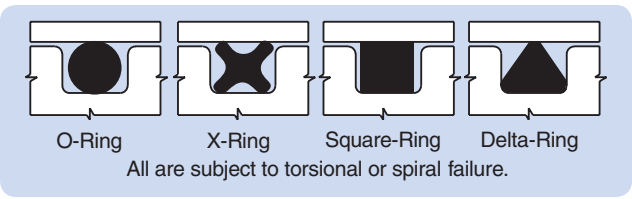

Figure 5-17: Ring Cross-Sections for Reciprocating Seals

Torsional or spiral failure is not limited to the O-ring or torus type of seal. Square, delta, four-leaf clover, and other cross sectional shapes (see Figure 5-17) are also prone to fail by twisting if the proper conditions exist.

The design and operational factors which contribute to spiral failure of a seal are listed below in the order of their relative importance:

1. Speed of stroke
2. Lack of lubrication
3. Pressure differential and direction
4. Squeeze
5. Shape of groove or split grooves
6. Temperature of operation
7. Length of stroke
8. Surface finish of gland
9. Type of metal surface
10. Side loads
11. ID to W ratio of O-ring
12. Contamination or gummy deposits on metal surface
13. Type of metal rubbing surface
14. Breathing
15. Concentricity of mating metal parts
16. Stretch of O-ring (see rotary shaft seals)
17. Lack of back-up rings
18. Poor installation of O-rings

Only the very important or less obvious factors which contribute to spiral failure will be discussed. Some of those which have been discussed elsewhere will also be omitted here. It should be remembered that before spiral failure can occur, an O-ring must be twisted by one or more of the above interrelated factors. Usually, several factors combine to produce any failure that develops. Some of the other seal designs will leak excessively when twisted. The O-ring usually seals until complete failure occurs.

5.15.1 Speed of Stroke

Investigations have disclosed that one of the primary causes of spiral failure is by reciprocating speeds of less than one foot per minute. It appears that at this slow speed, the sliding or running seal friction created is very high and comparable to break-out friction. Extreme twisting will occur on low or balanced pressure components, such as hydraulic accumulators, in a relatively few (about 200) cycles if the temperature is above 39°C (100°F). O-ring seals are not recommended, therefore, for speeds less than one foot per minute when the pressure differential is less than 27.6 Bar (400 psi). If the system pressure is slowly lost, as through slow valve leaks, and a sealed piston moves slowly through a cylinder a number of times, spiral failure of the O-ring very probably will result. The obvious remedy here is to provide good maintenance of the system so that slow leaks are prevented, or make it an operational practice to quickly exhaust the system after the day's work.

5.15.2 Lack of Lubrication

The lack of lubrication on a surface exposed to the atmosphere is one of the prime contributors to spiral failure. Excessive wear will normally occur. However, twisting of the seal and spiral failure can result if the unlubricated surface is actuated through the seal with little or no pressure on the seal to hold it and prevent it from rolling. This applies primarily to long stroke (greater than 152.4 mm (6")) applications.

The remedy for this situation is to:

a. Use lubricating (or lubricated) wiper rings.
b. Apply a suitable grease, that will not evaporate, to the exposed surface.
c. Use a fluid that will not tend to evaporate or become tacky at the operating temperature.
d. Lubricate metal surface prior to assembly.
e. Use a metal or surface plating that will produce less friction.

5.15.3 Pressure Differential and Direction

As explained earlier, the direction of pressure and seal friction should oppose each other. Spiral failure is more likely to occur if the pressure and seal friction are both in the same direction. In other words, seals in a pump are more likely to spiral than are those in an actuator.

Normally an O-ring will not twist when the pressure differential across the seal is greater than 27.6 Bar (400 psi) during operation.

5.15.4 Squeeze

The aerospace industry has generally found that more than 0.043 mm (0.017 in.) of squeeze on the side of a 5.3 mm (0.210") cross section (W) O-ring will make some long stroke applications prone to spiral failure. It can be easily seen that more rolling force is created on the cross section with an increase in squeeze. Other factors are normally involved when failure occurs with the standard squeezes recommended for reciprocating seals.

5.15.5 Shape of Groove and Split Groove

If a V-shaped groove is used, it is evident that the hydraulic holding force is reduced because the area on the side of the V-groove is less than at the bottom and side of a square groove. V-grooves are much more prone to produce spiral failures. This is especially true if any of the other factors are out of balance. Split grooves give trouble if the hydraulic holding force on the O-ring against both the side and the bottom of the groove is not maintained. Great care should be used when designing glands which have an opening in the bottom in order to make sure the normal holding force will be maintained (see Figure 5-16).

5.15.6 Temperature of Operation

When the temperature in and around a system is substantially increased, the seals are more prone to fail. This is because lubricants are more likely to evaporate, or lose their, "light ends", and/or lose some of their lubricity, the seal becomes softer, the squeeze is increased due to the rubber expansion, and the metal clearances may become greater.

5.15.7 Length of Stroke

As a general rule, the longer the stroke of a cylinder or rod, the greater the eccentricity, bending, side load, and other factors that contribute to wear and/or spiral failure. We do not recommend an O-ring for service when the stroke is greater than 304.8 mm (12") unless extra precautions are taken to avoid trouble.

5.15.8 Surface Finish

When a cylinder or rod is actuated, side loads, bending, chips or other foreign material, and non perfect machining, drilling and finishing all in some way tend to contribute to scoring, galling, marring, or scratching of the surface over which the seal must slide (refer to metals and floating glands). When this occurs, the roughness is unevenly distributed around the circumference or periphery. Even though it may be very slight, it creates an uneven friction condition and thus can contribute to spiral failure and/or uneven, excessive wear.

5.15.9 Back-Up Rings

Back-up rings sometimes provide enough extra lubrication on the return stroke to assist in the prevention of spiral failure. For further information see the discussion on back-up rings in Section VI.

5.16 Modifications for Special Applications

Normally, the gland dimensions given in Design Tables 5-1 and 5-2 are adequate and give trouble-free service. If not applicable, the following modifications will help solve specific problems:

- Small Amount of Leakage
- Early Stress-Aging
- Low Temperature Leakage
- Excessive Swells (above 20%)

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5.16.1 Small Amount of Leakage

1. Examine the O-ring for signs of cutting during installation.
2. Increase the squeeze on the cross-section of O-ring.
3. Reduce the groove length. A wide groove may cause leakage because of pumping action of the O-ring. This is especially possible when the piston is cycled rapidly.
4. Improve the surface finish of metal rubbing surface.
5. Check for eccentric machining of gland.

5.16.2 Early Stress-Aging

1. Redesign groove to reduce stretch of the O-ring.
2. Redesign groove to reduce squeeze of the O-ring.
3. Use a more heat-resistant rubber compound.
4. Make certain O-ring is not being twisted during dry assembly.
5. Use larger O-ring to reduce stretch.
6. Make sure O-rings are not closer than six feet from an electric motor (operating) during shelf storage. Ozone causes rapid deterioration of most elastomers.

5.16.3 Low Temperature Leakage

1. Make certain that O-ring compound was designed for operation at low temperatures.
2. Increase squeeze of the O-ring. Coefficient of contraction of rubber is about 10 times that of steel and several times greater than aluminum.
3. Spring load the O-ring (see Figure 3-1).
4. Make sure all gland surfaces are smooth enough (see paragraph 5.3).

Note: Minute leakage is to be expected and is in fact, desirable, when an O-ring is used as a reciprocating seal. An O-ring that does not by-pass a little fluid at each stroke is running dry and high friction and rapid seal wear will result.

5.16.4 Excessive Swell (above 20%)

1. Replace O-ring with one made from a compound more resistant to the fluid being sealed.
2. Increase groove length. If the volume of the groove is too small, increased friction and excessive stress may cause premature failure of the O-ring (refer to discussions of friction and spiral failure).

5.17 Gland Dimensions for Reciprocating Hydraulic O-Ring Seals

For most reciprocating applications in which an O-ring is sealing a liquid of any kind (the design is not limited to hydraulic oils), the dimensions of either Design Table 5-1, the military design, or Design Table 5-2, the industrial design, would be suitable. Of the two, Parker Seal Group normally recommends the Table 5-2 dimensions because these industrial designs, in most cases, allow for the use of standard drill rod sizes and standard boring tools. The dimensions in this table are actually in good agreement with early versions of the aerospace table. The military dimensions cause less stretch on the O-rings. The percent reduction is so slight, except in the smallest sizes, that the effect cannot be significant, while the cost of the special machined rods and boring tools that are required could be high.

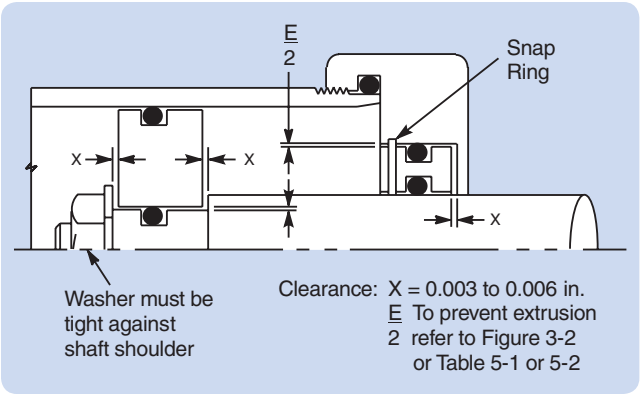

Figure 5-18: Floating Gland

In reciprocating applications for which neither table applies because of a predetermined dimension that does not agree, the following procedure may be used to find gland dimensions.

1. For piston seals, select an O-ring having an OD near to or preferably slightly larger than the cylinder bore diameter.
2. For rod seals, select on O-ring having an ID closest to the rod diameter. It may be slightly larger or smaller, but ID stretch should not exceed 5% as installed for optimum design.
3. In all reciprocating seals, make sure minimum squeeze recommendations are considered.

5.18 Floating Glands

Since it is impossible to bore, drill or tap perfect, true holes, and to machine perfect parts providing perfect alignment, the engineer should consider the floating gland. Eccentricity (lack of concentricity) is allowable, but it does cause high unit loads on small portions of bearing surfaces. In turn, this causes minute scratches on the metal surface on which the O-ring must rub (with the possible exception of very soft bearing materials, such as babbitt).

In order to reduce or eliminate the high bearing loads, the relatively inexpensive floating gland should be used whenever possible. The object of this gland is to allow the piston or rod bearing (containing the O-ring groove), to pivot, adjust, or float a small amount, offsetting misalignment. (See Figure 5-18.)

This gland design increases the life of the O-ring and eliminates many of the spasmodic or unscheduled failures, as well as reducing the maintenance cost.

5.19 Pneumatic Reciprocating O-Ring Seals

The past few years have shown a rapid increase of interest in pneumatic systems, not only for new equipment, but as a replacement for some existing hydraulic components. Some of the more general reasons are:

1. Increased non-flammability.
2. Light weight.
3. Leakage is less critical and does not contaminate the surrounding area.
4. The atmosphere acts as a giant reservoir.
5. System fluid is not decomposed by high temperature.

5.20 Temperature

Nitrile rubber is generally the first compound considered for a seal. It should be remembered, however, that it is less resistant to dry heat than it is to hot oils or other liquids. Nitrile compounds are used for pneumatic applications more than any other polymer, but in this kind of use, temperatures are usually low. In pneumatic applications above 104°C (220°F) for extended periods of time, consider ethylene propylene, fluorocarbon, or even silicone or fluorosilicone. The choice depends on temperature extremes, internal lubricant, severity of service, and overall cost.

5.21 Silicone Compounds

If silicone compounds are used, extra attention is necessary to make sure that all foreign material and sharp edges or corners are removed from the gland. This is necessary because of the relatively poor resistance to cutting and abrasion which is characteristic of silicone compounds. Recent developments have improved the abrasion resistance and oil resistance of the silicones, but they are still far short of many other synthetic rubbers.

5.22 High-Pressure

The most difficult gland to seal for any type of packing is that in a high-pressure pneumatic system because, in addition to the problems encountered with liquids, the following must be considered:

1. It is the hardest type seal to keep lubricated.
2. Oxygen in the air comes in direct contact with the seal and causes rapid aging and/or deterioration. This problem is amplified as system pressures and temperatures increase. (More oxygen is present due to the compression of the air.)
3. Gaseous molecules are very small, and will pass into and through (permeate) materials and openings which liquids will not. (See Table 3-19.)
4. The large change in volume which a gas undergoes with fluctuations in pressure often make necessary very special rubber seal materials so gases that have entered the seal can be rapidly expelled. Otherwise, blisters, ruptures, and chunks will be blown out of the seal when decompression of the system occurs. All organic materials are permeable, so the gas cannot be kept out of the seal.
5. Compounds other than those used in hydraulic systems are often necessary because the requirements are entirely different. This is especially true at high temperatures above 71°C (160°F) and high pressure (69 Bar to 207 Bar) (1000 psi to 3000 psi).

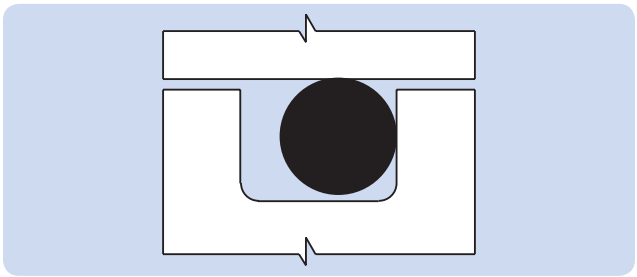

Figure 5-19: Floating O-Ring Seal

5.23 Lubrication

Most conventional pneumatic applications that fail prematurely do so because of inadequate lubrication. Rubber has an inherently high coefficient of friction with all metals and most non-metallic surfaces. Disregarding the necessity for lubrication will result in high friction, excessive abrasion or rapid wear of the rubber O-ring and heat build-up. For pneumatic seal applications it is especially important that adequate lubrication be provided. Of course, a lubricant must be selected that will not cause deterioration of the O-ring.

5.24 Gland Dimensions

Normally, the static and reciprocating gland dimensions given at the end of the Static Seals section and this section are adequate and give trouble-free pneumatic service. Much lower squeeze designs are permissible and used frequently in low pressure pneumatic applications (i.e. using shop air pressure for machine tools, holding devices, and similar applications.)

Since the temperature range is very moderate and a little leakage is not critical, some liberties can be taken with soft metals, surface finish and other design criteria without seriously reducing the life expectancy of this type of seal (low pressure cases). In fact, successful designs are in service which vary between the relatively high-squeeze hydraulic gland recommendations and the no-squeeze floating seal design discussed below. Each application seems to have an optimum design depending on what is desired.

5.25 Floating Seal

It has been found possible to modify the standard gland design for moving seals and reduce breakout friction as much as 60%. By allowing the O-ring to float, the frictional forces are greatly reduced and longer life can be expected from the seal. (See Figure 5-19.) There is a slight increase in leakage at the beginning of a stroke which for most pneumatic applications is undetectable. Because of this leakage and other considerations, the design is recommended for a temperature range from -23°C to 82°C (-10°F to 180°F) and for low pressure (up to 13.8 Bar (200 psi)) air service only.

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Recommended dimensions for floating pneumatic piston seal glands are tabulated in Design Chart 5-3 and Design Table 5-3 at the end of this section. The "floating" feature of this design is the virtual lack of squeeze on the O-ring cross-section. Sealing is accomplished by the peripheral squeeze applied to the outside diameter of the O-ring as it is assembled into the bore, and air pressure moving the ring into facial contact with the wall of the groove.

When this principle is understood, it will be seen that when the direction of pressurized air is reversed, a puff of air escapes between the inside diameter of the O-ring and the bottom of the groove during the small fraction of a second it takes the O-ring to move to the other side of the gland. This is the primary reason for the slight increase in leakage mentioned for this design.

The floating seal will not trap pressure between two O-rings in separate grooves unless considerable rubber swell is encountered.

Five or six O-rings are used in adjoining floating seal glands. This design has been used for some hot water and steam applications as a method of increasing O-ring life. The full effect of the hot steam is brought to bear on the inner rings and a lesser amount on the outer rings. Consequently, the seal is effective long after a single O-ring would have failed.

For the design of pneumatic reciprocating rod seals, use Design Chart 5-2 and Design Table 5-2. This is the cross section squeeze design used for hydraulic piston and rod seals. Floating seals are not recommended for pneumatic rods, as they would require stretching the O-ring, causing early aging. Furthermore, since pneumatically actuated shafts often move rapidly, a stretched O-ring in this situation would be subject to the Gow-Joule effect described in the rotary seal discussion. For static pneumatic seal designs, use Design Chart 4-1 and Design Table 4-1.

5.26 Uni-Directional Gland

This design modification utilizes a uni-directional floating seal groove and more than one O-ring (see Figure 5-20). The addition of drilled holes in the grooves causes each O-ring to seal in one direction only, preventing a pressure trap of non-compressible liquid between the O-rings. When using this design, the gland dimensions given in Design Table 5-3 are suggested and the holes should be drilled into the pressure side of the outside grooves on the piston and the inside of the rod glands. As many individual seals as required may be used as long as each groove is vented. The O-rings on either end, alternately seal and release trapped pressure. The three piston O-rings and the two rod O-rings are never all sealing at the same time. The vents are not necessary in pneumatic designs.

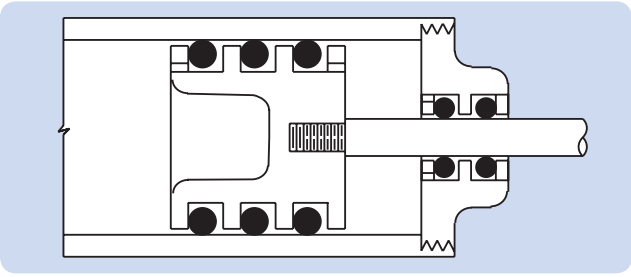

Figure 5-20: Use of Multiple O-Rings in a Floating O-Ring Design

5.27 Rotary Seal

An O-ring has proved to be a practical rotary shaft seal in many applications. With the correct design, Parker O-Ring rotary seal compound N1090-85, will provide satisfactory service at surface speeds up to 1500 feet per minute.

The design conditions are most critical for rotary seals, as would be expected. Relatively high durometer compounds, close control of tolerances, and minimum cross section are required.

Rotary seals usually should not be used at temperatures below -40°C (-40°F) even though flexibility to -54°C (-65°F) is claimed, since thermal shrinkage and loss of resilience tend to cause loss of contact with the shaft. In some cases, initial leakage of frozen seals may be tolerable until heat build-up occurs in higher speed shafts. Spring loading may be helpful in some situations.

High-speed shafts of soft metal should be avoided since they will normally wear more rapidly than the rubber, opening the clearance and allowing leakage. Hardened steel shafts in the range of 55 Rockwell are desirable, but not mandatory. Attention to clearances, side thrust, and end-play are critical in designing effective rotary O-ring seals.

Whenever it can be avoided, an O-ring should not be installed in a gland that holds it in more than a minimum of tensional stress. This principle is especially important to consider when designing for an O-ring rotary shaft seal. Most elastomers when heated in the stressed, or stretched condition will contract. This is of practical importance in a rotary seal because it results in a tendency for the O-ring to seize the high-speed rotating shaft. This phenomenon, known as the Gow-Joule effect, occurs only if the rubber is under tensile stress.

The friction between the O-ring and the rotating shaft creates heat. When it is installed in more than a minimum of tensional stress, the O-ring tends to contract when heated and seize the high speed rotating shaft. This contraction causes more friction which in turn causes more heat and the process becomes self-perpetuating, until the O-ring is destroyed.

Even at low surface speeds, where heating is not a problem, a stretched O-ring tends to rotate with the shaft and leak. For speeds below 200 feet per minute, the squeeze recommended in Design Chart 5-2 may be used. However, the shaft diameter should be no larger than the free state ID of the O-ring.

Shaft seal applications where the O-ring is installed in a groove in the shaft are not recommended if the shaft rotates. This is due to the centrifugal action which causes the O-ring to rotate and rub on all surfaces which generally causes early seal leakage or failure.

O-Ring Sections for Rotary Seals	
Speed (fpm*)	**Maximum Recommended "W" Dimension**
0 to 200	Usually not critical (Use chart 5-2)
200 to 400	0.139
200 to 600	0.103
200 to 1500	0.070

*Feet per minute = 0.26 X Shaft Diameter (inches) X rpm.

Table 5-5: O-Ring Sections for Rotary Seals

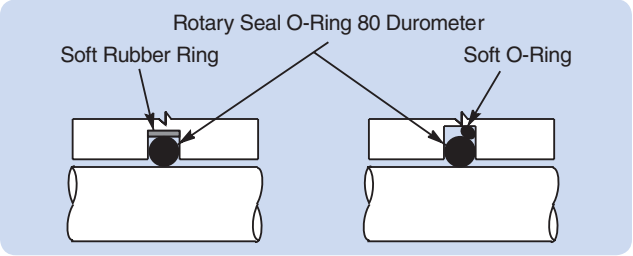

Figure 5-21: Spring-Loading for Rotary Seal

The use of O-rings as high speed rotary shaft seals is usually not recommended for applications requiring lower than -40°C (-40°F) or higher than 121°C (250°F) operating temperatures.

The O-ring gland in a rotary shaft application should not be used as a bearing surface. The shaft should be contained by bearings that will permit the O-ring to operate under the lowest possible heat and load. Because of the limited interference that must be used to avoid frictional heat, the O-ring will not compensate for shafts that are out of round or rotate eccentrically. Shafts should remain concentric within .013 mm (0.0005") T.I.R.

Bearings of all types cause considerable local heat and seals placed too close to them will fail prematurely.

Provision should be made for the dissipation of any heat that may be generated because of friction. The nearer to room temperature the seal interface, the longer the O-ring will seal. There are two methods commonly used to prevent high bearing heat build-up:

Preferred: Provide a clearance of 0.203 mm (0.008") on a side between the rotating shaft and the O-ring housing. Make sure that the shaft does not rub the housing. For pressures exceeding 55.2 Bar (800 psi), decrease the diametrical clearance per Figure 3-2.

Alternate: The bearing length should be at least 10 times the "W" dimension of the O-ring used. This provides for a greater area for heat transfer. If the clearance must be kept to a minimum to prevent high pressure extrusion, the 10 times "W" rule also applies. A floating gland (see Figure 5-18) is preferred to avoid high unit load at a local point or area.

Experience has proven that it is desirable to use the O-ring with the smallest "W", or cross-section diameter, available for the ID required. It is recommended that a "W" dimension of 0.103 be considered maximum for all speeds over 600 feet per minute. (See Table 5-5.)

All metals and plastics suitable for the housing or gland construction of seal assemblies requiring rotary shaft seals can be used with O-rings. However, since most rotary seal compounds contain graphite as a compound ingredient, any metal, such as stainless steel, or surface treatment that may be adversely affected by this material should be avoided.

To ensure maximum O-ring life, use an O-ring compound that has been specially developed for rotary seal applications and provides the required characteristics that are necessary

Problem: To design a rotary seal gland for a 76.2 mm (3") (desired) shaft running at 1750 RPM with oil pressure at 6.9 Bar (100 psi).

Procedural Steps:	Example:
(A) Calculate surface speed.	(A) Speed = 0.26 X 3 X 1750 = 1365 fpm
(B) Determine O-ring cross section that may be used from Table 5-5.	(B) .070 (larger cross sections are eliminated due to speed)
(C) Select .070 cross section O-ring with actual ID closest to desired shaft OD from Design Table 5-4.	(C) Parker No. 2-041
(D) Add 0.002 to O-ring ID to determine max. actual shaft OD, B.	(D) B max. = 2.969 + 0.002 = 2.991 (TOL: + .000, - .001)
(E) Determine gland depth, L from Design Chart 5-4.	(E) 0.065 to 0.067
(F) Calculate Gland Groove ID, A-1 A-1 min = B max. + 2L min. A-1 max. = B min. + 2L max.	(F) A-1 min. = 2.991 + 2(0.065) = 3.121 A-1 max. = 2.990 + (0.067) = 3.124 A-1 = 3.121 (TOL: + .003, - .000)
(G) Determine diametral clearance, E from Design Chart 5-4.	(G) 0.012 + 0.016
(H) Calculate shaft bore D D min. = B max. + E min. D max. = B min. + E max.	(H) D min. = 2.991 + 0.012 = 3.003 D max. = 2.990 + 0.016 = 3.006 D = 3.003 (TOL: + .003, -.000)
(I) Determine groove width, G from Design Chart 5-4.	(I) 0.075 - 0.079
(J) Check Figure 3-2 to make sure design is extrusion safe.	

Table 5-6: Rotary Seal Design Example

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for this service. See Section II, Basic O-Ring Elastomers, for more information on rotary seal compounds.

Figure 5-21 shows two methods of "spring loading" the hard rotary seal. Either of these should only be used when absolutely necessary to obtain the desired seal.

See Table 5-6 for a rotary seal design example.

5.28 Oscillating Seal

In this guide, two types of oscillating seals are considered:

1. Faucet or valve stems are excellent examples of assemblies that can be simplified by the use of an O-ring seal. Compression type or multiple-lip packing can be eliminated, reducing space requirements and eliminating the need for adjusting or take-up devices. For applications of this type, if the speed is under 200 feet per minute, use Design Table 5-2 for selecting O-ring sizes and gland dimensions.
2. Constantly oscillating shafts, such as those used on timing and metering devices, can be sealed satisfactorily with O-rings. If the motion is continuous for long periods of time, use Design Table 5-4 for O-ring sizes and gland dimensions.

5.29 Seat Seals

A properly designed check or poppet type valve, with an O-ring on the seat, will give an exceptionally long, non-leaking service. Many designers and engineers make the costly mistake of trying to use a conventional groove (square or rectangular) design to hold the O-ring.

With this type of groove, "blow-out" will normally occur when the valve is unseated.

"Blow-out" is a type of seal failure caused by the action of the pressure in the system on the side of the O-ring, forcing it out of the groove into some other part of the valve or system. "Blow-out" usually occurs at differential pressures above 5.5 Bar (80 psi). The exact pressure will depend on the gas or fluid, valve design and the physical properties of the O-ring when a non-retaining or conventional type groove is used.

It should be kept in mind that blow-out is similar to extrusion, but that it occurs at considerably lower pressures.

Figure 5-22 shows an O-ring on the seat of a check valve in a conventional groove. The seal is satisfactory as long as the valve is not opened at or near the pressure necessary to cause blow-out.

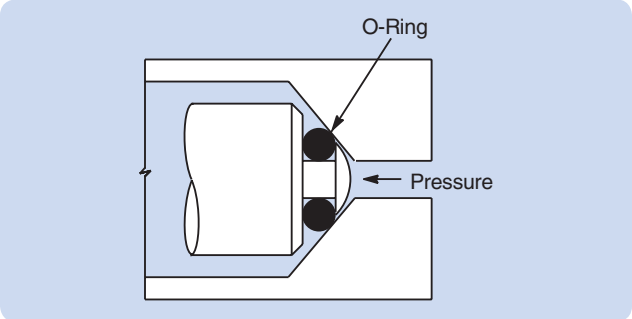


Figure 5-22: Valve Seat Seal, Standard Groove

Figure 5-23 illustrates a valve opening above "blow-out" pressure. As the valve opens, the space between the two faces becomes increasingly larger. The pressure opening the valve is also acting on the O-ring, causing it to continue to seal the opening until it is stretched completely out of the groove and is blown out or forced into another part of the system.

Gases such as air, LPG, CO2, etc. enter or permeate the O-ring. With release of pressure, the gas inside the O-ring can cause the seal to "balloon" or swell momentarily. (The amount depends on the pressure.) The ballooning effect that can occur at very low pressure usually pops the O-ring out of the groove the same as blow-out. "Ballooning" and "blow-out" often combine to cause valve seal failure. Another term often used to describe this phenomenon is "explosive decompression." O-ring blow-out may be prevented by using a groove design which encloses more than 180° of the O-ring cross section or by venting the groove. Typical methods used are shown in Figure 5-24. If a rectangular groove must be used, alter the dimensions as follows:

Groove depth — 0.015 to 0.025 less than O-ring cross section diameter.

Groove width — 1.00 to 1.10 times the O-ring cross section diameter.

Groove side angle — 0°, if possible.

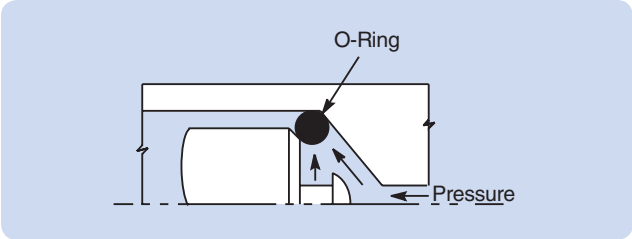


Figure 5-23: O-Ring Blow-Out, Standard Groove

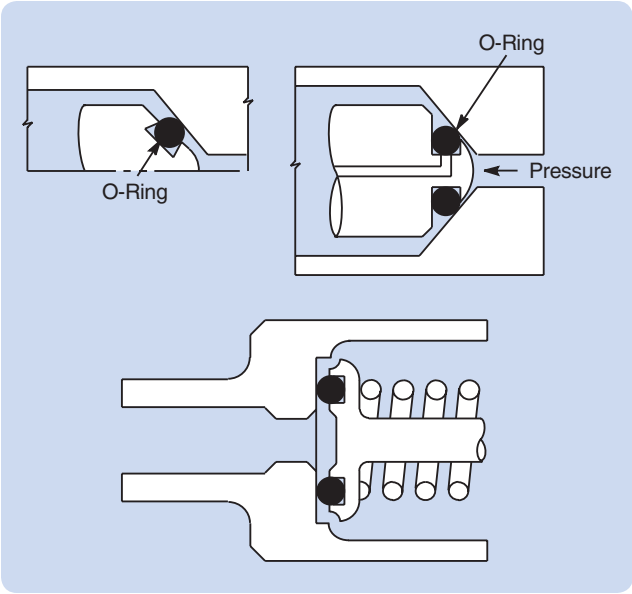


Figure 5-24: Groove Designs to Prevent Blow-Out

5.30 Drive Belts

O-rings can be used as low power transmission elements. They are not only an economic solution but also offer many advantages:

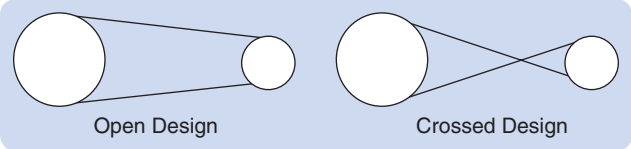
- Simple installation
- Constant tension
- Flexible fitting
- Because of their elastic properties, O-ring compounds require no adjustment,
- Freely available in standard compounds and sizes
- Greatest possible tolerances in positioning of pulleys.

An O-ring compound is selected for minimum stretch relaxation (tensile set) and maximum dynamic properties. The choice of elastomer is made to the environment:

- Contact medium, e.g. ozone, oil grease,
- Extreme temperatures

The general requirements are:

- Good aging resistance
- Wear resistance
- Relatively low tendency to return to original shape under tension and temperature caused by friction; this means a higher resistance to the Joule effect;
- Good bending flexibility



Compound Selection

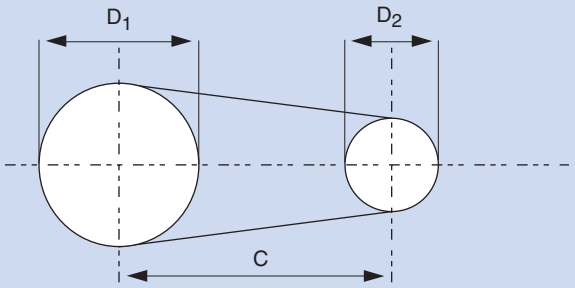
Please see Section III, paragraph 3.17 for information about drive belt compound selection

Design Information

- Direct contact with fluids should be avoided. For contact medium see medium Compatibility Table.
- The smaller pulley minimum diameter is D_2 mm = 6 x d_2 (cross section).
- The elongation of the O-ring inner diameter d_1 is a maximum of 15% (average elongation between 8% and 12%).
- Tension when fitted approximately 0.6 to 1.0 N/mm^2.
- Cross section d_2 should be greater or equal to 2,62mm.

Ordering Detail

All O-rings which are used as drive belts are subject to additional quality inspection procedures and inspection for surface defects under elongation. O-rings ordered for this application are to be coded as follows: “E0540 DBA2-250”.



Abbreviations:
C Center line distance of pulleys (mm)
D_1 Diameter of driven pulley (mm)
D_2 Diameter of drive pulley (mm)
S Elongation as a decimal (e.g. 10% = 0.1)
d_1 O-ring inner diameter (mm)
d_2 O-ring cross-section (mm)
L Length of drive belt (mm)
B Calculation factor

5.30.1 Calculation of Drive Belt Open Design

1. Calculation of O-Ring size d_1:

Known – D_1 and D_2, diameter of pulley
C center line distance of pulleys
S elongation as a decimal (e.g. 10% = 0.1)

a) Calculation of drive belt L:

$$L = 2 \times C + 1.57 \times (D_1 + D_2) + \frac{(D_1 + D_2)^2}{4 \times C}$$

b) Calculation of O-ring inside diameter d_1:

$$d_1 = \frac{L}{3.14 \times (1.0 + S)}$$

c) O-ringis selected according to the O-ring size list. If a size is required between the sizes then the smaller size should be taken.

2. Calculation of elongation S:

Known – d_1 inside diameter of O-ring
C center line distance of pulleys
D_1 and D_2, diameter of pulleys

a) Calculation of drive belt L: (see above, 1a)

b) Calculation of elongation S as a decimal:

$$S = \frac{L}{3.14 \times d_1} - 1$$

3. Calculation of center line distance C of pulley:

Known – d_1 inside diameter of O-ring
S elongation as a decimal (e.g. 10% = 0.10)
D_1 and D_2, diameters of pulleys

a) Calculation of factor B:

$$B = 3.14 \times d_1 \times (S + 1) - 1.57 \times (D_1 + D_2)$$

b) Thereafter calculation of center line distance C:

$$C = \frac{B + \sqrt{B2 - (D_1 - D_2)^2}}{4}$$

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5.30.2 Calculation of Drive Belt Crossed Design

1) Calculation of O-Ring size d_1:

Known – D_1 and D_2, diameter of pulley
C center line distance of pulleys
S elongation as a decimal (e.g. 10% = 0.1)

a) Calculation of drive belt L:

$$L = 2 \times C + 1.57 \times (D_1 + D_2) + \frac{(D_1 - D_2)^2}{4 \times C}$$

b.) Calculation of O-Ring inside diameter d_1:

$$d_1 = \frac{L}{3.14 \times (1.0 + S)}$$

c) O-ring is selected according to the O-Ring size list. If a size is required between the sizes then the smaller size should be taken.

2) Calculation of elongation S:

Known - d_1 inside diameter of O-Ring
C center line distance of pulleys
D_1 and D_2, diameter of pulleys

a) Calculation of drive belt L: (see above, 1a)
b) Calculation of elongation S as a decimal:

$$S = \frac{L}{3.14 \times d_1} - 1$$

3. Calculation of center line distance C of pulley:

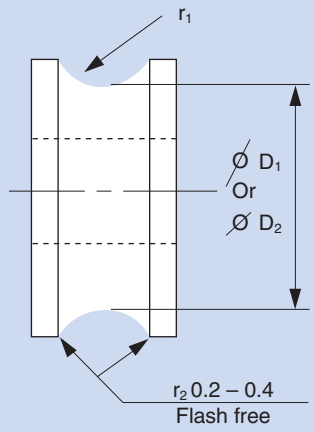
Known – d_1 inside diameter of O-ring
S elongation as a decimal (e.g. 10% = 0.10)
D_1 and D_2, diameters of pulleys

a) Calculation of factor B:

$$B = 3.14 \times d_1 \times (S + 1) - 1.57 \times (D_1 + D_2)$$

b) Thereafter calculation of center line distance C:

$$C = \frac{B + \sqrt{B2 - (D_1 - D_2)^2}}{4}$$



d_2 in mm	r_1 in mm
2.62	1.25 + 0.1
3.53	1.70 + 0.1
5.33	2.60 + 0.1
6.99	3.50 + 0.15

For other cross-sections
$r_1 = 0.49 \times d_2$
Surface roughness:
Rmax < 6.3 µm
Ra < 1.6 µm

Pulley gland radius

5.31 O-Ring Glands

5.31.1 O-Ring Glands (Per SAE AS4716) for Aerospace Hydraulic (Reciprocating) Packings and Gaskets

Design Chart 5-1 provides the basis for calculating gland dimensions for standard O-ring sizes. These dimensions have been calculated and are listed in Design Table 5-1. The procedures for the use of Design Table 5-1 are outlined in Design Guide 5-1.

After selecting gland dimension, read horizontally to determine proper O-ring size number per AS568A.

There are a number of various O-ring gland design specifications in use throughout industry. These include Aerospace Recommended Practice (ARP) 1232, 1233 and 1234. There also is the International Standards Organization (better known as ISO) Specification 3601/2. Each of these and other less accepted documents have slight dimensional variations from those found in this Handbook.

Guide For Design Table 5-1

If Desired Dimension is Known for	Select Closest Dimension in Column	Read Horizontally in Column	To Determine Dimension for
Cylinder Bore or Male Gland Cylinder Bore I.D	A	G C F	Groove Width* Piston or Cylinder O.D. Groove O.D.
Piston or Cylinder O.D.	C	G A F	Groove Width* Cylinder Bore or Male Gland Cylinder Bore I.D. Groove O.D.
Rod or Gland Sleeve O.D.	B	G J H	Groove Width* Groove I.D. Rod Bore or Female Gland Housing Bore I.D.
Rod Bore or Female Gland Housing Bore I.D.	H	G J B	Groove Width* Groove I.D. Rod or Gland Sleeve O.D.

*For information on groove width refer to Design Chart 5-1A

Design Guide 5-1: Guide For Design Chart 5-1

Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

Standard Gland Width for Zero, One, and Two Backup Rings

Gland and AS568 Dash Number	O-ring Cross Section W		Gland Width G No Backup Ring		Gland Width G One Backup Ring		Gland Width G Two Backup Rings	
	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
001	.037	.043	.070	.075	—	—	—	—
002	.047	.053	.077	.082	—	—	—	—
003	.057	.063	.088	.093	—	—	—	—
004 to 009	.067	.073	.098	.103	.154	.164	.210	.220
010 to 028	.067	.073	.094	.099	.150	.160	.207	.217
110 to 149	.100	.106	.141	.151	.183	.193	.245	.255
210 to 247	.135	.143	.188	.198	.235	.245	.304	.314
325 to 349	.205	.215	.281	.291	.334	.344	.424	.434
424 to 460	.269	.281	.375	.385	.475	.485	.579	.589

Design Chart 5-1A: Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

Standard Gland Diametral Clearance Dimensions

Gland and AS568 Dash Number	O-ring Cross Section W		Diametral Clearance E Max.	
	Min.	Max.	Exterior	Interior
001	.037	.043	.004	.004
002	.047	.053	.004	.004
003	.057	.063	.004	.004
004 to 012	.067	.073	.004	.004
013 to 028	.067	.073	.005	.005
110 to 126	.100	.106	.005	.005
127 to 129	.100	.106	.005	.006
130 to 132	.100	.106	.006	.006
133 to 140	.100	.106	.006	.007
141 to 149	.100	.106	.007	.007
210 to 222	.135	.143	.005	.005
223 and 224	.135	.143	.006	.006
225 to 227	.135	.143	.006	.007
228 to 243	.135	.143	.007	.007
244 and 245	.135	.143	.008	.007
246 and 247	.135	.143	.008	.008
325 to 327	.205	.215	.006	.006
328 and 329	.205	.215	.006	.007
330 to 345	.205	.215	.007	.007
346 to 349	.205	.215	.008	.007
425 to 438	.269	.281	.009	.009
439 to 445	.269	.281	.009	.010
446	.269	.281	.010	.010
447 to 460	.269	.281	.011	.010

Design Chart 5-1B: Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

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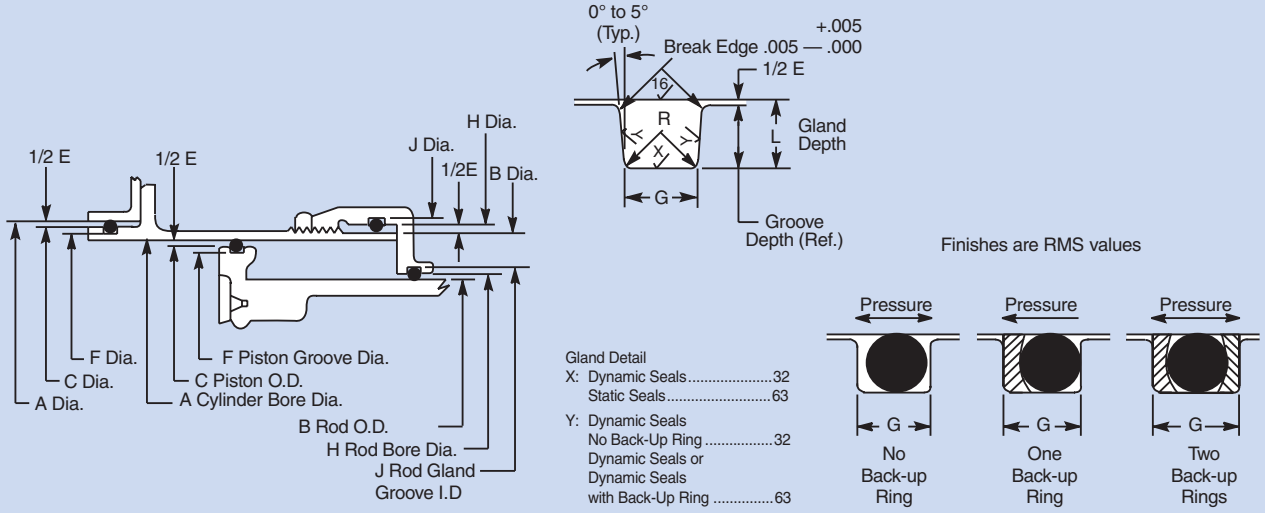

Groove Wall Angle (X)	
Pressure (PSIG)	**X (Degrees)**
≤ 3000	0.0 +5.0/-0.0
4000 to 6000	0.0 +2.0/-0.0
>6000 to 8000	0.0 ±0.5

Break Edge	
Pressure (PSIG)	**Groove Edge Break**
≤ 4000	0.005 +0.005/-0.000
> 4000	0.002 +0.008/-0.000

Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

Gland and AS568 Dash No.	Piston or Cylinder OD C	Cylinder Bore OD A	Gland OD F	Rod or Gland Sleeve OD B	Rod Bore ID H	Gland ID J	Actual Minimum Squeeze Piston	Actual Minimum Squeeze Rod	Actual Maximum Squeeze Piston	Actual Maximum Squeeze Rod
001	0.093	0.095	0.033	0.033	0.035	0.095	0.0000	–	0.0145	–
	0.092	0.096	0.032	0.032	0.036	0.096	–	0.0000	–	0.0145
002	0.126	0.128	0.048	0.048	0.050	0.128	0.0000	–	0.0139	–
	0.125	0.129	0.047	0.047	0.051	0.129	–	0.0000	–	0.0139
003	0.157	0.159	0.063	0.063	0.065	0.159	0.0000	–	0.0154	–
	0.156	0.160	0.062	0.062	0.066	0.160	–	0.0000	–	0.0154
004	0.188	0.190	0.076	0.076	0.078	0.190	0.0003	–	0.0176	–
	0.187	0.191	0.075	0.075	0.079	0.191	–	0.0003	–	0.0176
005	0.219	0.221	0.108	0.108	0.110	0.221	0.0020	–	0.0178	–
	0.218	0.222	0.107	0.107	0.111	0.222	–	0.0020	–	0.0178
006	0.233	0.235	0.123	0.123	0.125	0.235	0.0023	–	0.0177	–
	0.232	0.236	0.122	0.122	0.126	0.236	–	0.0023	–	0.0177
007	0.264	0.266	0.154	0.154	0.156	0.266	0.0032	–	0.0179	–
	0.263	0.267	0.153	0.153	0.157	0.267	–	0.0032	–	0.0179
008	0.295	0.297	0.189	0.185	0.187	0.294	0.0050	–	0.0193	–
	0.294	0.298	0.188	0.184	0.188	0.295	–	0.0053	–	0.0196
009	0.327	0.329	0.220	0.217	0.219	0.327	0.0052	–	0.0192	–
	0.326	0.330	0.219	0.216	0.220	0.328	–	0.0052	–	0.0193
010	0.358	0.360	0.250	0.248	0.250	0.359	0.0052	–	0.0190	–
	0.357	0.361	0.249	0.247	0.251	0.360	–	0.0050	–	0.0189
011	0.420	0.422	0.312	0.310	0.312	0.421	0.0057	–	0.0193	–
	0.419	0.423	0.311	0.309	0.313	0.422	–	0.0054	–	0.0191
012	0.483	0.485	0.375	0.373	0.375	0.484	0.0060	–	0.0194	–
	0.482	0.486	0.374	0.372	0.376	0.485	–	0.0057	–	0.0192
013	0.548	0.550	0.441	0.435	0.437	0.545	0.0050	–	0.0196	
	0.547	0.552	0.439	0.433	0.438	0.547	–	0.0050	–	0.0198
014	0.611	0.613	0.504	0.498	0.500	0.608	0.0052	–	0.0197	
	0.610	0.615	0.502	0.496	0.501	0.610	–	0.0051	–	0.0199

Design Table 5-1: Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716) (Continued)

Gland and AS568 Dash No.	Piston or Cylinder OD C	Cylinder Bore OD A	Gland OD F	Rod or Gland Sleeve OD B	Rod Bore ID H	Gland ID J	Actual Minimum Squeeze Piston	Actual Minimum Squeeze Rod	Actual Maximum Squeeze Piston	Actual Maximum Squeeze Rod
015	0.673	0.675	0.566	0.560	0.562	0.670	0.0052	–	0.0200	–
	0.672	0.677	0.564	0.558	0.563	0.672	–	0.0051	–	0.0202
016	0.736	0.738	0.629	0.623	0.625	0.733	0.0052	–	0.0203	–
	0.735	0.740	0.627	0.621	0.626	0.735	–	0.0051	–	0.0205
017	0.798	0.800	0.691	0.685	0.687	0.795	0.0054	–	0.0204	–
	0.797	0.802	0.689	0.683	0.688	0.797	–	0.0052	–	0.0205
018	0.861	0.863	0.753	0.748	0.750	0.858	0.0050	–	0.0200	–
	0.860	0.865	0.751	0.746	0.751	0.860	–	0.0053	–	0.0205
019	0.923	0.925	0.815	0.810	0.812	0.920	0.0051	–	0.0200	–
	0.922	0.927	0.813	0.808	0.813	0.922	–	0.0053	–	0.0205
020	0.989	0.991	0.881	0.873	0.875	0.983	0.0050	–	0.0198	–
	0.988	0.993	0.879	0.871	0.876	0.985	–	0.0054	–	0.0205
021	1.051	1.053	0.943	0.935	0.937	1.045	0.0051	–	0.0199	–
	1.050	1.055	0.941	0.933	0.938	1.047	–	0.0055	–	0.0205
022	1.114	1.116	1.006	0.998	1.000	1.108	0.0051	–	0.0200	–
	1.113	1.118	1.004	0.996	1.001	1.110	–	0.0055	–	0.0205
023	1.176	1.178	1.068	1.060	1.062	1.170	0.0052	–	0.0200	–
	1.175	1.180	1.066	1.058	1.063	1.172	–	0.0055	–	0.0205
024	1.239	1.241	1.131	1.123	1.125	1.233	0.0052	–	0.0200	–
	1.238	1.243	1.129	1.121	1.126	1.235	–	0.0056	–	0.0205
025	1.301	1.303	1.193	1.185	1.187	1.295	0.0052	–	0.0201	–
	1.300	1.305	1.191	1.183	1.188	1.297	–	0.0056	–	0.0205
026	1.384	1.366	1.256	1.248	1.250	1.358	0.0053	–	0.0201	–
	1.363	1.368	1.254	1.246	1.251	1.360	–	0.0056	–	0.0205
027	1.426	1.428	1.318	1.310	1.312	1.420	0.0053	–	0.0202	–
	1.425	1.430	1.316	1.308	1.313	1.422	–	0.0056	–	0.0205
028	1.489	1.491	1.381	1.373	1.375	1.483	0.0053	–	0.0203	–
	1.488	1.493	1.379	1.371	1.376	1.485	–	0.0056	–	0.0205
110	0.548	0.550	0.379	0.373	0.375	0.546	0.0053	–	0.0204	–
	0.547	0.552	0.377	0.371	0.376	0.548	–	0.0052	–	0.0204
111	0.611	0.613	0.441	0.435	0.437	0.609	0.0052	–	0.0202	–
	0.610	0.615	0.439	0.433	0.438	0.611	–	0.0050	–	0.0201
112	0.673	0.675	0.502	0.498	0.500	0.672	0.0053	–	0.0201	–
	0.672	0.677	0.500	0.496	0.501	0.674	–	0.0053	–	0.0203
113	0.736	0.738	0.565	0.560	0.562	0.734	0.0052	–	0.0204	–
	0.735	0.740	0.563	0.558	0.563	0.736	–	0.0052	–	0.0207
114	0.798	0.800	0.627	0.623	0.625	0.797	0.0053	–	0.0210	–
	0.797	0.802	0.625	0.621	0.626	0.799	–	0.0052	–	0.0211
115	0.861	0.863	0.689	0.685	0.687	0.859	0.0050	–	0.0206	–
	0.860	0.865	0.687	0.683	0.688	0.861	–	0.0054	–	0.0212
116	0.923	0.925	0.751	0.748	0.750	0.923	0.0053	–	0.0208	–
	0.922	0.927	0.749	0.746	0.751	0.925	–	0.0050	–	0.0207
117	0.989	0.991	0.817	0.810	0.812	0.985	0.0050	–	0.0205	–
	0.988	0.993	0.815	0.808	0.813	0.987	–	0.0050	–	0.0209
118	1.051	1.053	0.879	0.873	0.875	1.048	0.0052	–	0.0207	–
	1.050	1.055	0.877	0.871	0.876	1.050	–	0.0051	–	0.0209
119	1.114	1.116	0.942	0.935	0.937	1.110	0.0053	–	0.0206	–
	1.113	1.118	0.940	0.933	0.938	1.112	–	0.0052	–	0.0209
120	1.176	1.178	1.003	0.998	1.000	1.173	0.0050	–	0.0204	–
	1.175	1.180	1.001	0.996	1.001	1.175	–	0.0053	–	0.0209
121	1.239	1.241	1.066	1.060	1.062	1.235	0.0050	–	0.0203	–
	1.238	1.243	1.064	1.058	1.063	1.237	–	0.0054	–	0.0209
122	1.301	1.303	1.128	1.123	1.125	1.298	0.0052	–	0.0204	–
	1.300	1.305	1.126	1.121	1.126	1.300	–	0.0055	–	0.0209
123	1.364	1.366	1.191	1.185	1.187	1.360	0.0051	–	0.0205	–
	1.363	1.368	1.189	1.183	1.188	1.362	–	0.0054	–	0.0210

Design Table 5-1: Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

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Gland and AS568 Dash No.	Piston or Cylinder OD C	Cylinder Bore OD A	Gland OD F	Rod or Gland Sleeve OD B	Rod Bore ID H	Gland ID J	Actual Minimum Squeeze Piston	Actual Minimum Squeeze Rod	Actual Maximum Squeeze Piston	Actual Maximum Squeeze Rod
124	1.426	1.428	1.253	1.248	1.250	1.423	0.0052	–	0.0206	–
	1.425	1.430	1.251	1.246	1.251	1.425	–	0.0055	–	0.0210
125	1.489	1.491	1.316	1.310	1.312	1.485	0.0052	–	0.0206	–
	1.488	1.493	1.314	1.308	1.313	1.487	–	0.0055	–	0.0210
126	1.551	1.553	1.378	1.373	1.375	1.548	0.0053	–	0.0207	–
	1.550	1.555	1.376	1.371	1.376	1.550	–	0.0056	–	0.0210
127	1.614	1.616	1.441	1.435	1.437	1.610	0.0053	–	0.0206	–
	1.613	1.618	1.439	1.433	1.439	1.612	–	0.0051	–	0.0215
128	1.676	1.678	1.503	1.498	1.500	1.673	0.0054	–	0.0207	–
	1.675	1.680	1.501	1.496	1.502	1.675	–	0.0052	–	0.0215
129	1.739	1.741	1.566	1.560	1.562	1.735	0.0053	–	0.0208	–
	1.738	1.743	1.564	1.558	1.564	1.737	–	0.0051	–	0.0215
130	1.802	1.805	1.631	1.623	1.625	1.798	0.0053	–	0.0217	–
	1.801	1.807	1.629	1.621	1.627	1.800	–	0.0051	–	0.0215
131	1.864	1.867	1.693	1.685	1.687	1.860	0.0053	–	0.0217	–
	1.863	1.869	1.691	1.683	1.689	1.862	–	0.0052	–	0.0215
132	1.927	1.930	1.756	1.748	1.750	1.923	0.0054	–	0.0217	–
	1.926	1.932	1.754	1.746	1.752	1.925	–	0.0052	–	0.0215
133	1.989	1.992	1.818	1.810	1.813	1.984	0.0054	–	0.0218	–
	1.988	1.994	1.816	1.808	1.815	1.986	–	0.0052	–	0.0225
134	2.052	2.055	1.881	1.873	1.876	2.047	0.0055	–	0.0218	–
	2.051	2.057	1.879	1.871	1.878	2.049	–	0.0053	–	0.0225
135	2.115	2.118	1.944	1.936	1.939	2.110	0.0054	–	0.0219	–
	2.114	2.120	1.942	1.934	1.941	2.112	–	0.0052	–	0.0225
136	2.177	2.180	2.006	1.998	2.001	2.172	0.0055	–	0.0219	–
	2.176	2.182	2.004	1.996	2.003	2.174	–	0.0053	–	0.0225
137	2.240	2.243	2.069	2.061	2.064	2.235	0.0055	–	0.0219	–
	2.239	2.245	2.067	2.059	2.066	2.237	–	0.0053	–	0.0225
138	2.302	2.305	2.131	2.123	2.126	2.297	0.0055	–	0.0219	–
	2.301	2.307	2.129	2.121	2.128	2.299	–	0.0053	–	0.0225
139	2.365	2.368	2.194	2.186	2.189	2.360	0.0056	–	0.0219	–
	2.364	2.370	2.192	2.184	2.191	2.362	–	0.0054	–	0.0225
140	2.427	2.430	2.256	2.248	2.251	2.422	0.0056	–	0.0219	–
	2.426	2.432	2.254	2.246	2.253	2.424	–	0.0054	–	0.0225
141	2.490	2.493	2.319	2.311	2.314	2.485	0.0050	–	0.0225	–
	2.488	2.495	2.317	2.309	2.316	2.487	–	0.0053	–	0.0225
142	2.552	2.555	2.381	2.373	2.376	2.547	0.0051	–	0.0225	–
	2.550	2.557	2.379	2.371	2.378	2.549	–	0.0053	–	0.0225
143	2.615	2.618	2.444	2.436	2.439	2.610	0.0051	–	0.0225	–
	2.613	2.620	2.442	2.434	2.441	2.612	–	0.0054	–	0.0225
144	2.677	2.680	2.506	2.498	2.501	2.672	0.0051	–	0.0225	–
	2.675	2.682	2.504	2.496	2.503	2.674	–	0.0054	–	0.0225
145	2.740	2.743	2.569	2.561	2.564	2.735	0.0052	–	0.0225	–
	2.738	2.745	2.567	2.559	2.566	2.737	–	0.0054	–	0.0225
146	2.802	2.805	2.631	2.623	2.626	2.797	0.0052	–	0.0225	–
	2.800	2.807	2.629	2.621	2.628	2.799	–	0.0054	–	0.0225
147	2.865	2.868	2.694	2.686	2.689	2.860	0.0051	–	0.0225	–
	2.863	2.870	2.692	2.684	2.691	2.862	–	0.0054	–	0.0225
148	2.927	2.930	2.756	2.748	2.751	2.922	0.0052	–	0.0225	–
	2.925	2.932	2.754	2.746	2.753	2.924	–	0.0054	–	0.0225
149	2.990	2.993	2.819	2.811	2.814	2.985	0.0052	–	0.0225	–
	2.988	2.995	2.817	2.809	2.816	2.987	–	0.0054	–	0.0225
210	0.989	0.991	0.750	0.748	0.750	0.989	0.0052	–	0.0244	–
	0.988	0.993	0.748	0.746	0.751	0.991	–	0.0054	–	0.0247
211	1.051	1.053	0.812	0.810	0.812	1.051	0.0054	–	0.0244	–
	1.050	1.055	0.810	0.808	0.813	1.053	–	0.0055	–	0.0248

Design Table 5-1: Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716) (Continued)

Gland and AS568 Dash No.	Piston or Cylinder OD C	Cylinder Bore OD A	Gland OD F	Rod or Gland Sleeve OD B	Rod Bore ID H	Gland ID J	Actual Minimum Squeeze Piston	Actual Minimum Squeeze Rod	Actual Maximum Squeeze Piston	Actual Maximum Squeeze Rod
212	1.114	1.116	0.874	0.873	0.875	1.115	0.0051	–	0.0242	–
	1.113	1.118	0.872	0.871	0.876	1.117	–	0.0052	–	0.0243
213	1.176	1.178	0.936	0.935	0.937	1.177	0.0052	–	0.0242	–
	1.175	1.180	0.934	0.933	0.938	1.179	–	0.0053	–	0.0244
214	1.239	1.241	0.999	0.998	1.000	1.240	0.0054	–	0.0243	–
	1.238	1.243	0.997	0.996	1.001	1.242	–	0.0054	–	0.0244
215	1.301	1.303	1.064	1.060	1.062	1.302	0.0067	–	0.0254	–
	1.300	1.305	1.062	1.058	1.063	1.304	–	0.0056	–	0.0244
216	1.364	1.366	1.124	1.123	1.125	1.365	0.0054	–	0.0246	–
	1.363	1.368	1.122	1.121	1.126	1.367	–	0.0055	–	0.0247
217	1.426	1.428	1.186	1.185	1.187	1.427	0.0055	–	0.0246	–
	1.425	1.430	1.184	1.183	1.188	1.429	–	0.0056	–	0.0247
218	1.489	1.491	1.249	1.248	1.250	1.490	0.0056	–	0.0246	–
	1.488	1.493	1.247	1.246	1.251	1.492	–	0.0057	–	0.0247
219	1.551	1.553	1.311	1.310	1.312	1.552	0.0057	–	0.0246	–
	1.550	1.555	1.309	1.308	1.313	1.554	–	0.0058	–	0.0248
220	1.614	1.616	1.374	1.373	1.375	1.615	0.0058	–	0.0247	–
	1.613	1.618	1.372	1.371	1.376	1.617	–	0.0059	–	0.0248
221	1.676	1.678	1.436	1.435	1.437	1.677	0.0059	–	0.0247	–
	1.675	1.680	1.434	1.433	1.438	1.679	–	0.0059	–	0.0248
222	1.739	1.741	1.499	1.498	1.500	1.740	0.0057	–	0.0250	–
	1.738	1.743	1.497	1.496	1.501	1.742	–	0.0058	–	0.0250
223	1.864	1.867	1.625	1.623	1.625	1.865	0.0053	–	0.0254	–
	1.863	1.869	1.623	1.621	1.627	1.867	–	0.0054	–	0.0255
224	1.989	1.992	1.750	1.748	1.750	1.990	0.0055	–	0.0254	–
	1.988	1.994	1.748	1.746	1.752	1.992	–	0.0056	–	0.0255
225	2.115	2.118	1.876	1.873	1.876	2.115	0.0054	–	0.0255	–
	2.114	2.120	1.874	1.871	1.878	2.117	–	0.0050	–	0.0260
226	2.240	2.243	2.001	1.998	2.001	2.240	0.0055	–	0.0255	–
	2.239	2.245	1.999	1.996	2.003	2.242	–	0.0051	–	0.0260
227	2.365	2.368	2.126	2.123	2.126	2.365	0.0056	–	0.0255	–
	2.364	2.370	2.124	2.121	2.128	2.367	–	0.0052	–	0.0260
228	2.490	2.493	2.251	2.248	2.251	2.490	0.0051	–	0.0260	–
	2.488	2.495	2.249	2.246	2.253	2.492	–	0.0052	–	0.0260
229	2.615	2.618	2.376	2.373	2.376	2.615	0.0051	–	0.0260	–
	2.613	2.620	2.374	2.371	2.378	2.617	–	0.0053	–	0.2060
230	2.740	2.743	2.501	2.498	2.501	2.740	0.0052	–	0.0260	–
	2.738	2.745	2.499	2.496	2.503	2.742	–	0.0054	–	0.0260
231	2.865	2.868	2.626	2.623	2.626	2.865	0.0053	–	0.0260	–
	2.863	2.870	2.624	2.621	2.628	2.867	–	0.0054	–	0.0260
232	2.990	2.993	2.751	2.748	2.751	2.990	0.0052	–	0.0260	–
	2.988	2.995	2.749	2.746	2.753	2.992	–	0.0053	–	0.0260
233	3.115	3.118	2.876	2.873	2.876	3.115	0.0053	–	0.0260	–
	3.113	3.120	2.874	2.871	2.878	3.117	–	0.0054	–	0.0260
234	3.240	3.243	3.001	2.997	3.000	3.239	0.0053	–	0.0260	–
	3.238	3.245	2.999	2.995	3.002	3.241	–	0.0055	–	0.0260
235	3.365	3.368	3.126	3.122	3.125	3.364	0.0054	–	0.0260	–
	3.363	3.370	3.124	3.120	3.127	3.366	–	0.0055	–	0.0260
236	3.490	3.493	3.251	3.247	3.250	3.489	0.0054	–	0.0260	–
	3.488	3.495	3.249	3.245	2.252	3.491	–	0.0056	–	0.0260
237	3.615	3.618	3.376	3.372	3.375	3.614	0.0055	–	0.0260	–
	3.613	3.620	3.374	3.370	3.377	3.616	–	0.0056	–	0.0260
238	3.740	3.743	3.501	3.497	3.500	3.739	0.0055	–	0.0260	–
	3.738	3.745	3.499	3.495	3.502	3.741	–	0.0057	–	0.0260
239	3.865	3.868	3.626	3.622	3.625	3.864	0.0055	–	0.0260	–
	3.863	3.870	3.624	3.620	3.627	3.866	–	0.0056	–	0.0260

Design Table 5-1: Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

Parker Hannifin Corporation • O-Ring Division
2360 Palumbo Drive, Lexington, KY 40509
Phone: (859) 269-2351 • Fax: (859) 335-5128
www.parkerorings.com

Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716) (Continued)

Gland and AS568 Dash No.	Piston or Cylinder OD C	Cylinder Bore OD A	Gland OD F	Rod or Gland Sleeve OD B	Rod Bore ID H	Gland ID J	Actual Minimum Squeeze Piston	Actual Minimum Squeeze Rod	Actual Maximum Squeeze Piston	Actual Maximum Squeeze Rod
240	3.990	3.993	3.751	3.747	3.750	3.989	0.0055	–	0.0260	–
	3.988	3.995	2.749	3.745	3.752	3.991	–	0.0056	–	0.0260
241	4.115	4.118	3.876	3.872	3.875	4.114	0.0056	–	0.0260	–
	4.113	4.120	3.874	3.870	3.877	4.116	–	0.0057	–	0.0260
242	4.240	4.243	4.001	3.997	4.000	4.239	0.0056	–	0.0260	–
	4.238	4.245	3.999	3.995	4.002	4.241	–	0.0057	–	0.0260
243	4.365	4.368	4.126	4.122	4.125	4.364	0.0056	–	0.0260	–
	4.363	4.370	4.124	1.120	4.127	4.366	–	0.0057	–	0.0260
244	4.489	4.493	4.251	4.247	4.250	4.489	0.0051	–	0.0265	–
	4.487	4.495	4.249	4.245	4.252	4.491	–	0.0057	–	0.0260
245	4.614	4.618	4.376	4.372	4.375	4.614	0.0051	–	0.0265	–
	4.612	4.620	4.374	4.370	4.377	4.616	–	0.0058	–	0.0260
246	4.739	4.743	4.501	4.497	4.501	4.739	0.0052	–	0.0265	–
	4.737	4.745	4.499	4.495	4.503	4.741	–	0.0053	–	0.0265
247	4.864	4.868	4.626	4.622	4.626	4.864	0.0052	–	0.0265	–
	4.862	4.870	4.624	4.620	4.628	4.866	–	0.0053	–	0.0265
325	1.864	1.867	1.495	1.498	1.500	1.870	0.0081	–	0.0322	–
	1.863	1.869	1.493	1.496	1.502	1.872	–	0.0079	–	0.0318
326	1.989	1.992	1.620	1.623	1.625	1.995	0.0083	–	0.0323	–
	1.988	1.994	1.618	1.621	1.627	1.997	–	0.0081	–	0.0319
327	2.115	2.118	1.746	1.748	1.750	2.120	0.0085	–	0.0322	–
	2.114	2.120	1.744	1.746	1.752	2.122	–	0.0083	–	0.0320
328	2.240	2.243	1.871	1.873	1.876	2.245	0.0087	–	0.0323	–
	2.239	2.245	1.869	1.871	1.878	2.247	–	0.0080	–	0.0325
329	2.365	2.368	1.996	1.998	2.001	2.370	0.0086	–	0.0326	–
	2.364	2.370	1.994	1.996	2.003	2.372	–	0.0080	–	0.0329
330	2.490	2.493	2.121	2.123	2.126	2.495	0.0083	–	0.0332	–
	2.488	2.495	2.119	2.121	2.128	2.497	–	0.0081	–	0.0330
331	2.615	2.618	2.246	2.248	2.251	2.620	0.0084	–	0.0332	–
	2.613	2.620	2.244	2.246	2.253	2.622	–	0.0083	–	0.0330
332	2.740	2.743	2.371	2.373	2.376	2.745	0.0085	–	0.0332	–
	2.738	2.745	2.369	2.371	2.378	2.747	–	0.0084	–	0.0330
333	2.865	2.868	2.496	2.498	2.501	2.870	0.0085	–	0.0334	–
	2.863	2.870	2.494	2.496	2.503	2.872	–	0.0084	–	0.0332
334	2.990	2.993	2.621	2.623	2.626	2.995	0.0087	–	0.0334	–
	2.988	2.995	2.619	2.621	2.628	2.997	–	0.0085	–	0.0332
335	3.115	3.118	2.746	2.748	2.751	3.120	0.0088	–	0.0334	–
	3.113	3.120	2.744	2.746	2.753	3.122	–	0.0087	–	0.0332
336	3.240	3.243	2.871	2.873	2.876	3.245	0.0089	–	0.0334	–
	3.238	3.245	2.869	2.871	2.878	3.247	–	0.0088	–	0.0332
337	3.365	3.368	2.996	2.997	3.000	3.369	0.0087	–	0.0335	–
	3.363	3.370	2.994	2.995	3.002	3.371	–	0.0087	–	0.0335
338	3.490	3.493	3.121	3.122	3.125	3.494	0.0088	–	0.0335	–
	3.488	3.495	3.119	3.120	3.127	3.496	–	0.0088	–	0.0335
339	3.615	3.618	3.246	3.247	3.250	3.619	0.0089	–	0.0335	–
	3.613	3.620	3.244	3.245	3.252	3.621	–	0.0089	–	0.0335
340	3.740	3.743	3.371	3.372	3.375	3.744	0.0090	–	0.0335	–
	3.738	3.745	3.369	3.370	3.377	3.746	–	0.0090	–	0.0335
341	3.865	3.868	3.496	3.497	3.500	3.869	0.0091	–	0.0335	–
	3.863	3.870	3.494	3.495	3.502	3.871	–	0.0090	–	0.0335
342	3.990	3.993	3.621	3.622	3.625	3.994	0.0090	–	0.0335	–
	3.988	3.995	3.619	3.620	3.627	3.996	–	0.0089	–	0.0335
343	4.115	4.118	3.746	3.747	3.750	4.119	0.0090	–	0.0335	–
	4.113	4.120	3.744	3.745	3.752	4.121	–	0.0090	–	0.0335
344	4.240	4.243	3.871	3.872	3.875	4.244	0.0091	–	0.0335	–
	4.238	4.245	3.869	3.870	3.877	4.246	–	0.0091	–	0.0335

Design Table 5-1: Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716) (Continued)

Gland and AS568 Dash No.	Piston or Cylinder OD C	Cylinder Bore OD A	Gland OD F	Rod or Gland Sleeve OD B	Rod Bore ID H	Gland ID J	Actual Minimum Squeeze Piston	Actual Minimum Squeeze Rod	Actual Maximum Squeeze Piston	Actual Maximum Squeeze Rod
345	4.365	4.368	3.966	3.997	4.000	4.369	0.0092	–	0.0335	–
	4.363	4.370	3.994	3.995	4.002	4.371	–	0.0091	–	0.0335
346	4.489	4.493	4.121	4.122	4.125	4.494	0.0087	–	0.0340	–
	4.487	4.495	4.119	4.120	4.127	4.496	–	0.0092	–	0.0335
347	4.614	4.618	4.246	4.247	4.250	4.619	0.0087	–	0.0340	–
	4.612	4.620	4.244	4.245	4.252	4.621	–	0.0092	–	0.0335
348	4.739	4.743	4.371	4.372	4.375	4.744	0.0088	–	0.0340	–
	4.737	4.745	4.369	4.370	4.377	4.746	–	0.0092	–	0.0335
349	4.864	4.868	4.496	4.497	4.500	4.869	0.0088	–	0.0340	–
	4.862	4.870	4.494	4.495	4.502	4.871	–	0.0093	–	0.0335
425	4.970	4.974	4.497	4.497	4.501	4.974	0.0175	–	0.0480	–
	4.968	4.977	4.494	4.494	4.503	4.977	–	0.0175	–	0.0480
426	5.095	5.099	4.622	4.622	4.626	5.099	0.0176	–	0.0480	–
	5.093	5.102	4.619	4.619	4.628	5.102	–	0.0176	–	0.0480
427	5.220	5.224	4.747	4.747	4.751	5.224	0.0176	–	0.0480	–
	5.218	5.227	4.744	4.744	4.753	5.227	–	0.0176	–	0.0480
428	5.345	5.349	4.872	4.872	4.876	5.349	0.0177	–	0.0480	–
	5.343	5.352	4.869	4.869	4.878	5.352	–	0.0177	–	0.0480
429	5.470	5.474	4.997	4.997	5.001	5.474	0.0176	–	0.0480	–
	5.468	5.477	4.994	4.994	5.003	5.477	–	0.0176	–	0.0480
430	5.595	5.599	5.122	5.122	5.126	5.599	0.0176	–	0.0480	–
	5.593	5.602	5.119	5.119	5.128	5.602	–	0.0176	–	0.0480
431	5.720	5.724	5.247	5.247	5.251	5.724	0.0177	–	0.0480	–
	5.718	5.727	5.244	5.244	5.253	5.727	–	0.0177	–	0.0480
432	5.845	5.849	5.372	5.372	5.376	5.849	0.0178	–	0.0480	–
	5.843	5.852	5.369	5.369	5.378	5.852	–	0.0178	–	0.0480
433	5.970	5.974	5.497	5.497	5.501	5.974	0.0178	–	0.0480	–
	5.968	5.977	5.494	5.494	5.503	5.977	–	0.0178	–	0.0480
434	6.095	6.099	5.622	5.622	5.626	6.099	0.0179	–	0.0480	–
	6.093	6.102	5.619	5.619	5.628	6.102	–	0.0179	–	0.0480
435	6.220	6.224	5.747	5.747	5.751	6.224	0.0179	–	0.0480	–
	6.218	6.227	5.744	5.744	5.753	6.227	–	0.0179	–	0.0480
436	6.345	6.349	5.872	5.872	5.876	6.349	0.0180	–	0.0480	–
	6.343	6.352	5.869	5.869	5.878	6.352	–	0.0180	–	0.0480
437	6.470	6.474	5.997	5.997	6.001	6.474	0.0180	–	0.0480	–
	6.468	6.477	5.994	5.994	6.003	6.477	–	0.0180	–	0.0480
438	6.720	6.724	6.247	6.247	6.251	6.724	0.0180	–	0.0480	–
	6.718	6.727	6.244	6.244	6.253	6.727	–	0.0180	–	0.0480
439	6.970	6.974	6.497	6.497	6.501	6.974	0.0181	–	0.0480	–
	6.968	6.977	6.494	6.494	6.504	6.977	–	0.0176	–	0.0485
440	7.220	7.224	6.747	6.747	6.751	7.224	0.0182	–	0.0480	–
	7.218	7.227	6.744	6.744	6.754	7.227	–	0.0177	–	0.0485
441	7.470	7.474	6.997	6.997	7.001	7.474	0.0182	–	0.0480	–
	7.468	7.477	6.994	6.994	7.004	7.477	–	0.0177	–	0.0485
442	7.720	7.724	7.247	7.247	7.251	7.724	0.0181	–	0.0480	–
	7.718	7.727	7.244	7.244	7.254	7.727	–	0.0176	–	0.0485
443	7.970	7.974	7.497	7.497	7.501	7.974	0.0182	–	0.0480	–
	7.968	7.977	7.494	7.494	7.504	7.977	–	0.0177	–	0.0485
444	8.220	8.224	7.747	7.747	7.751	8.224	0.0183	–	0.0480	–
	8.218	8.227	7.744	7.744	7.754	8.227	–	0.0178	–	0.0485
445	8.470	8.474	7.997	7.997	8.001	8.474	0.0183	–	0.0480	–
	8.468	8.477	7.994	7.994	8.004	8.477	–	0.0178	–	0.0485
446	8.970	8.974	8.497	8.497	8.501	8.974	0.0177	–	0.0485	–
	8.967	8.977	8.494	8.494	8.504	8.977	–	0.0177	–	0.0485
447	9.470	9.474	8.997	8.997	9.001	9.474	0.0168	–	0.0485	–
	9.467	9.478	8.994	8.994	9.004	9.478	–	0.0173	–	0.0485

Design Table 5-1: Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

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Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716) (Continued)

Gland and AS568 Dash No.	Piston or Cylinder OD C	Cylinder Bore OD A	Gland OD F	Rod or Gland Sleeve OD B	Rod Bore ID H	Gland ID J	Actual Minimum Squeeze		Actual Maximum Squeeze	
							Piston	Rod	Piston	Rod
448	9.970	9.974	9.497	9.497	9.501	9.974	0.0169	–	0.0485	–
	9.967	9.978	9.494	9.494	9.504	9.978	–	0.0174	–	0.0485
449	10.470	10.474	9.997	9.997	10.001	10.474	0.0170	–	0.0485	–
	10.467	10.478	9.994	9.994	10.004	10.478	–	0.0175	–	0.0485
450	10.970	10.974	10.497	10.497	10.501	10.974	0.0170	–	0.0485	–
	10.967	10.978	10.494	10.494	10.504	10.978	–	0.0175	–	0.0485
451	11.470	11.474	10.997	10.997	11.001	11.474	0.0170	–	0.0485	–
	11.467	11.478	10.994	10.994	11.004	11.478	–	0.0176	–	0.0485
452	11.970	11.974	11.497	11.497	11.501	11.974	0.0171	–	0.0485	–
	11.967	11.978	11.494	11.494	11.504	11.978	–	0.0176	–	0.0485
453	12.470	12.474	11.997	11.997	12.001	12.474	0.0172	–	0.0485	–
	12.467	12.478	11.994	11.994	12.004	12.478	–	0.0177	–	0.0485
454	12.970	12.974	12.497	12.497	12.501	12.974	0.0173	–	0.0485	–
	12.967	12.978	12.494	12.494	12.504	12.978	–	0.0178	–	0.0485
455	13.470	13.474	12.997	12.997	13.001	13.474	0.0173	–	0.0485	–
	13.467	13.478	12.994	12.994	13.004	13.478	–	0.0178	–	0.0485
456	13.970	13.974	13.497	13.497	13.501	13.974	0.0172	–	0.0485	–
	13.967	13.978	13.494	13.494	13.504	13.978	–	0.0177	–	0.0485
457	14.470	14.474	13.997	13.997	14.001	14.474	0.0173	–	0.0485	–
	14.467	14.478	13.994	13.994	14.004	14.478	–	0.0178	–	0.0485
458	14.970	14.974	14.497	14.497	14.501	14.974	0.0173	–	0.0485	–
	14.967	14.978	14.494	14.494	14.504	14.978	–	0.0178	–	0.0485
459	15.470	15.474	14.997	14.997	15.001	15.474	0.0174	–	0.0485	–
	15.467	15.478	14.994	14.994	15.004	15.478	–	0.0179	–	0.0485
460	15.970	15.974	15.497	15.497	15.501	15.974	0.0174	–	0.0485	–
	15.967	15.978	15.494	15.494	15.504	15.978	–	0.0179	–	0.0485

Design Table 5-1: Gland Design, O-Ring and Other Elastomeric Seals (SAE AS4716)

5.31.2 O-Ring Glands for Industrial Reciprocating Seals

Design Chart 5-2 provides a reasonable basis for calculating reciprocating O-ring seal glands. Design Table 5-2, which follows it, contains recommended gland dimensions for the standard AS568A O-ring sizes. The major difference from the military gland dimensions (Design Table 5-1) is the use of standard cylinder bore and standard rod dimensions.

Although these dimensions are suitable for most reciprocating designs, it is often desirable, or even necessary, to deviate from them. Other portions of this handbook on Basic O-Ring Elastomers (Section II) and O-Ring Applications (Section III) are helpful in determining when such special designs are indicated and provide useful data for such modified designs.

Procedures for using Design Table 5-2 are outlined in Design Guide 5-2. See Section X, Table 10-6 for installation guidelines.

Gland Dimensions for Industrial Reciprocating O-Ring Seals

Groove Diameter (Rod Gland) Tolerance	
-	.000 for all sizes
+	.002 for sizes 2-006 through 2-324
+	.004 for sizes 2-325 through 2-460
Groove Diameter (Piston) Tolerance	
+	.000 for all sizes
-	.002 for sizes 2-006 through 2-324
-	.004 for sizes 2-325 through 2-460

Design Guide 5-2a: Gland Dimensions for Industrial Reciprocating O-Ring Seals

Guide for Design Table 5-2

If Desired Dimension is Known for	Select Closest Dimension in Column	Read Horizontally in Column	To Determine Dimension for
Bore Dia of cylinder	A	B-1	Groove Dia of piston
		C	OD of piston
		G	Groove width
OD of piston	C	A	Bore Dia of cylinder
		B-1	Groove Dia of piston
		G	Groove width
OD of rod	B	A-1	Groove Dia for rod
		D	Bore ID for rod
		G	Groove width
Bore Dia for rod	D	A-1	Groove Dia for rod
		B	OD of rod
		G	Groove width

Design Guide 5-2b: Guide For Design Table 5-2

After selecting gland dimensions, read horizontally to determine proper O-ring size number. Specify compound.

Industrial Reciprocating O-Ring Packing Glands

O-Ring 2-Size AS568A-	W Cross-Section Nominal	W Cross-Section Actual	L Gland Depth	Squeeze Actual	Squeeze %	E(a) Diametral Clearance	G-GrooveWidth No Parbak Ring(G)	G-GrooveWidth One Parbak Ring(G_1)	G-GrooveWidth Two Parbak Rings(G_2)	R Groove Radius	Max. Eccentricity (b)
006 through 012	1/16	.070 ± .003	.055 to .057	.010 to .018	15 to 25	.002 to .005	.093 to .098	.138 to .143	.205 to .210	.005 to .015	.002
104 through 116	3/32	.103 ± .003	.088 to .090	.010 to .018	10 to 17	.002 to .005	.140 to .145	.171 to .176	.238 to .243	.005 to .015	.002
201 through 222	1/8	.139 ± .004	.121 to .123	.012 to .022	9 to 16	.003 to .006	.187 to .192	.208 to .213	.275 to .280	.010 to .025	.003
309 through 349	3/16	.210 ± .005	.185 to .188	.017 to .030	8 to 14	.003 to .006	.281 to .286	.311 to .316	.410 to .415	.020 to .035	.004
425 through 460	1/4	.275 ± .006	.237 to .240	.029 to .044	11 to 16	.004 to .007	.375 to .380	.408 to .413	.538 to .543	.020 to .035	.005

(a) Clearance (extrusion gap) must be held to a minimum consistent with design requirements for temperature range variation.
(b) Total indicator reading between groove and adjacent bearing surface.

Design Chart 5-2-a: Design Chart for Industrial Reciprocating O-Ring Packing Glands

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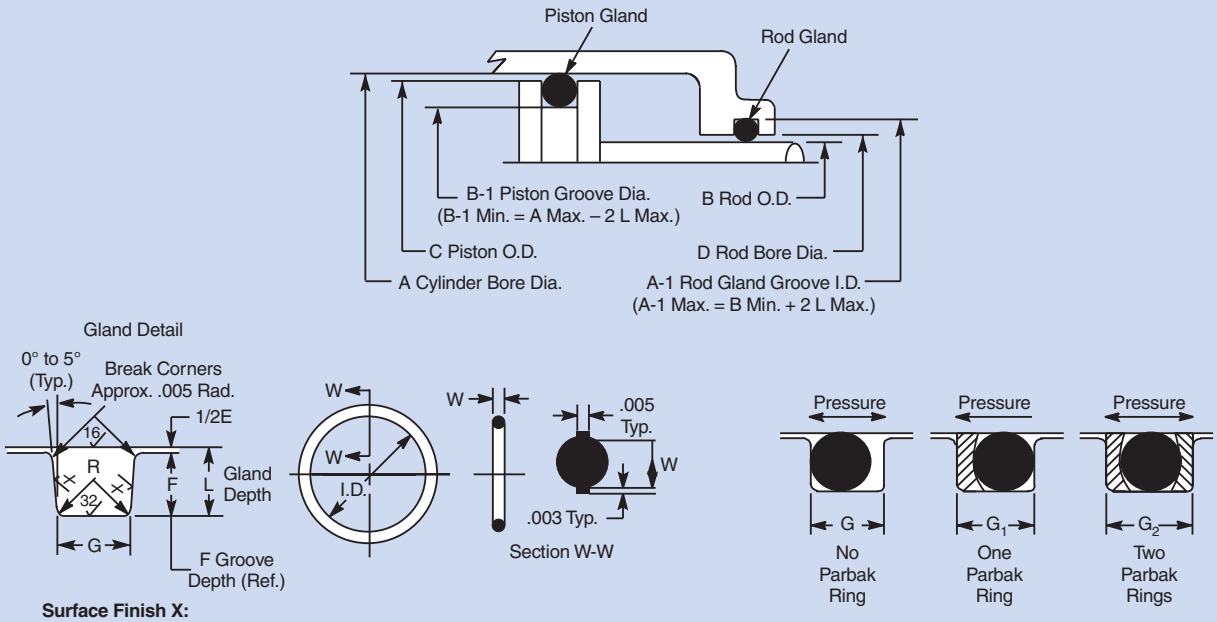
Industrial Reciprocating O-Ring Packing Glands



Surface Finish X:
32 Without Parbak Ring
63 With Parbak Ring

Finishes are RMS values

Refer to Design Chart 5-2 (below) and Design Table 5-2 for dimensions.

Industrial Dynamic Metric Cross-Sections

Industrial Dynamic Metric Cross-Sections

W Cross-Section	L Gland Depth	G-Grove Width No Parbak Ring (G)	R Groove Radius
1.50	1.30	1.90	0.20 to 0.40
1.80	1.45	2.40	0.20 to 0.40
2.00	1.70	2.60	0.20 to 0.40
2.50	2.10	3.30	0.20 to 0.40
2.65	2.20	3.60	0.20 to 0.40
3.00	2.60	3.90	0.40 to 0.80
3.55	3.05	4.80	0.40 to 0.80
4.00	3.50	5.30	0.40 to 0.80
5.00	4.45	6.70	0.40 to 0.80
5.30	4.65	7.10	0.40 to 0.80
6.00	5.40	8.00	0.40 to 0.80
7.00	6.20	9.50	0.40 to 0.80

(a) Dimensions are in mm. The ISO/DIN recommendations are preferred.
(b) Parbaks are not available in standard for metric sizes.

Design Chart 5-2-b: Design Chart for Industrial Dynamic Metric Cross-Sections

Gland Dimensions for Industrial Reciprocating O-Ring Seals, 103.5 Bar (1500 psi) Max.†

O-Ring Size Parker No. 2-	I.D.	Dimensions ±	Dimensions W	Mean O.D. (Ref)	A Bore Dia. (Cylinder) +.002 -.000	A-1 Groove Dia. (Rod Gland) +.002 -.000	B OD (Rod) +.000 -.002	B-1 Groove Dia. *(Piston) +.000 -.002	C OD (Piston) +.000 -.001	D Bore Dia. (Rod) +.001 -.000	G Width Groove +.005 -.000
006	.114	.005		.254	.249	.234	.124	.139	*.247	.126	
007	.145	.005		.285	.280	.265	.155	.170	*.278	.157	
008	.176	.005		.316	.311	.296	.186	.201	*.309	.188	
009	.208	.005	.070	.348	.343	.328	.218	.233	*.341	.220	.093
010	.239	.005	±.003	.379	.374	.359	.249	.264	*.372	.251	
011	.301	.005		.441	.436	.421	.311	.326	*.434	.313	
012	.364	.005		.504	.499	.484	.374	.389	*.497	.376	
104	.112	.005		.318	.312	.300	.124	.136	*.310	.126	
105	.143	.005		.349	.343	.331	.155	.167	*.341	.157	
106	.174	.005		.380	.374	.362	.186	.198	*.372	.188	
107	.206	.005		.412	.406	.394	.218	.230	*.404	.220	
108	.237	.005		.443	.437	.425	.249	.261	*.435	.251	
109	.299	.005		.505	.499	.487	.311	.323	*.497	.313	
110	.362	.005	.103	.568	.562	.550	.374	.386	*.560	.376	.140
111	.424	.005	±.003	.630	.624	.612	.436	.448	*.622	.438	
112	.487	.005		.693	.687	.675	.499	.511	*.685	.501	
113	.549	.007		.755	.749	.737	.561	.573	*.747	.563	
114	.612	.009		.818	.812	.800	.624	.636	.810	.626	
115	.674	.009		.880	.874	.862	.686	.698	.872	.688	
116	.737	.009		.943	.937	.925	.749	.761	.935	.751	
201	.171	.005		.449	.437	.427	.185	.195	*.434	.188	
202	.234	.005		.512	.500	.490	.248	.258	*.497	.251	
203	.296	.005		.574	.562	.552	.310	.320	*.559	.313	
204	.359	.005		.637	.625	.615	.373	.383	.622	.376	
205	.421	.005		.699	.687	.677	.435	.445	.684	.438	
206	.484	.005		.762	.750	.740	.498	.508	.747	.501	
207	.546	.007		.824	.812	.802	.560	.570	.809	.563	
208	.609	.009		.887	.875	.865	.623	.633	.872	.626	
209	.671	.009		.949	.937	.927	.685	.695	.934	.688	
210	.734	.010		1.012	1.000	.990	.748	.758	.997	.751	
211	.796	.010	.139	1.074	1.062	1.052	.810	.820	1.059	.813	.187
212	.859	.010	±.004	1.137	1.125	1.115	.873	.883	1.122	.876	
213	.921	.010		1.199	1.187	1.177	.935	.945	1.184	.938	
214	.984	.010		1.262	1.250	1.240	.998	1.008	1.247	1.001	
215	1.046	.010		1.324	1.312	1.302	1.060	1.070	1.309	1.063	
216	1.109	.012		1.387	1.375	1.365	1.123	1.133	1.372	1.126	
217	1.171	.012		1.449	1.437	1.427	1.185	1.195	1.434	1.188	
218	1.234	.012		1.512	1.500	1.490	1.248	1.258	1.497	1.251	
219	1.296	.012		1.574	1.562	1.552	1.310	1.320	1.559	1.313	
220	1.359	.012		1.637	1.625	1.615	1.373	1.383	1.622	1.376	
221	1.421	.012		1.699	1.687	1.677	1.435	1.445	1.684	1.438	
222	1.484	.015		1.762	1.750	1.740	1.498	1.508	1.747	1.501	

* These designs require considerable installation stretch. If assembly breakage is incurred use a compound having higher elongation or use a two-piece piston.
† This groove width does not permit the use of Parbak rings. For pressures above 103.5 Bar (1500 psi), consult Design Chart 5-2 for groove widths where Parbak rings must be used.

Design Table 5-2: Gland Dimensions for Industrial Reciprocating O-Ring Seals, 103.5 Bar (1500 psi) Max.†

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Gland Dimensions for Industrial Reciprocating O-Ring Seals, 103.5 Bar (1500 psi) Max.† (Continued)

O-Ring Size Parker No. 2-	Dimensions I.D.	±	W	Mean O.D. (Ref)	A Bore Dia. (Cylinder) +.002 -.000	A-1 Groove Dia. (Rod Gland) +.002 -.000	B OD (Rod) +.000 -.002	B-1 Groove Dia. *(Piston) +.000 -.002	C OD (Piston) +.000 -.001	D Bore Dia. (Rod) +.001 -.000	G Width Groove +.005 -.000
309	.412	.005		.832	.812	.805	.435	.442	*.809	.438	
310	.475	.005		.895	.875	.868	.498	.505	*.872	.501	
311	.537	.007		.957	.937	.930	.560	.567	*.943	.563	
312	.600	.009		1.020	1.000	.993	.623	.630	.997	.626	
313	.662	.009		1.082	1.062	1.055	.685	.692	1.059	.688	
314	.725	.010		1.145	1.125	1.118	.748	.755	1.122	.751	
315	.787	.010		1.207	1.187	1.180	.810	.817	1.184	.813	
316	.850	.010		1.270	1.250	1.243	.873	.880	1.247	.876	
317	.912	.010		1.332	1.312	1.305	.935	.942	1.309	.938	
318	.975	.010		1.395	1.375	1.368	.998	1.005	1.372	1.001	
319	1.037	.010		1.457	1.437	1.430	1.060	1.067	1.434	1.063	
320	1.100	.012		1.520	1.500	1.493	1.123	1.130	1.497	1.126	
321	1.162	.012		1.582	1.562	1.555	1.185	1.192	1.559	1.188	
322	1.225	.012		1.645	1.625	1.618	1.248	1.255	1.622	1.251	
323	1.287	.012		1.707	1.687	1.680	1.310	1.317	1.648	1.313	
324	1.350	.012		1.770	1.750	1.743 **+.004 -.000**	1.373	1.380 **+.000 -.004**	1.747	1.376	
325	1.475	.015		1.895	1.875	1.868	1.498	1.505	1.872	1.501	
326	1.600	.015		2.020	2.000	1.993	1.623	1.630	1.997	1.626	
327	1.725	.015		2.145	2.125	2.118	1.748	1.755	2.122	1.751	
328	1.850	.015		2.270	2.250	2.243	1.873	1.880	2.247	1.876	
329	1.975	.018	.210	2.395	2.375	2.368	1.998	2.005	2.372	2.001	.281
330	2.100	.018	±.005	2.520	2.500	2.493	2.123	2.130	2.497	2.126	
331	2.225	.018		2.645	2.625	2.618	2.248	2.255	2.622	2.251	
332	2.350	.018		2.770	2.750	2.743	2.373	2.380	2.747	2.376	
333	2.475	.020		2.895	2.875	2.868	2.498	2.505	2.872	2.501	
334	2.600	.020		3.020	3.000	2.993	2.623	2.630	2.997	2.626	
335	2.725	.020		3.145	3.125	3.118	2.748	2.755	3.122	2.751	
336	2.850	.020		3.270	3.250	3.243	2.873	2.880	3.247	2.876	
337	2.975	.024		3.395	3.375	3.368	2.998	3.005	3.372	3.001	
338	3.100	.024		3.520	3.500	3.493	3.123	3.130	3.497	3.126	
339	3.225	.024		3.645	3.625	3.618	3.248	3.255	3.622	3.251	
340	3.350	.024		3.770	3.750	3.743	3.373	3.380	3.747	3.376	
341	3.475	.024		3.895	3.875	3.868	3.498	3.505	3.872	3.501	
342	3.600	.028		4.020	4.000	3.993	3.623	3.630	3.997	3.626	
343	3.725	.028		4.145	4.125	4.118	3.748	3.755	4.122	3.751	
344	3.850	.028		4.270	4.250	4.243	3.873	3.880	4.247	3.876	
345	3.975	.028		4.395	4.375	4.368	3.998	4.005	4.372	4.001	
346	4.100	.028		4.520	4.500	4.493	4.123	4.130	4.497	4.126	
347	4.225	.030		4.645	4.625	4.618	4.248	4.255	4.622	4.251	
348	4.350	.030		4.773	4.750	4.743	4.373	4.380	4.747	4.376	
349	4.475	.030		4.895	4.875	4.868	4.498	4.505	4.872	4.501	

* These designs require considerable installation stretch. If assembly breakage is incurred use a compound having higher elongation or use a two-piece piston.
† This groove width does not permit the use of Parbak rings. For pressures above 103.5 Bar (1500 psi), consult Design Chart 5-2 for groove widths where Parbak rings must be used.

Design Table 5-2: Gland Dimensions for Industrial Reciprocating O-Ring Seals, 103.5 Bar (1500 psi) Max.†

Gland Dimensions for Industrial Reciprocating O-Ring Seals, 103.5 Bar (1500 psi) Max.† (Continued)

O-Ring Size Parker No. 2-	Dimensions I.D.	±	W	Mean O.D. (Ref)	A Bore Dia. (Cylinder) +.002 -.000	A-1 Groove Dia. (Rod Gland) +.002 -.000	B OD (Rod) +.000 -.002	B-1 Groove Dia. *(Piston) +.000 -.002	C OD (Piston) +.000 -.001	D Bore Dia. (Rod) +.001 -.000	G Width Groove +.005 -.000
425	4.475	.033		5.025	5.002	4.971	4.497	4.528	4.998	4.501	
426	4.600	.033		5.150	5.127	5.096	4.622	4.653	5.123	4.626	
427	4.725	.033		5.275	5.252	5.221	4.747	4.778	5.248	4.751	
428	4.850	.033		5.400	5.377	5.346	4.872	4.903	5.373	4.876	
429	4.975	.037		5.525	5.502	5.471	4.997	5.028	5.498	5.001	
430	5.100	.037		5.650	5.627	5.596	5.122	5.153	5.623	5.126	
431	5.225	.037		5.775	5.752	5.721	5.247	5.278	5.748	5.251	
432	5.350	.037		5.900	5.877	5.846	5.372	5.403	5.873	5.376	
433	5.475	.037		6.025	6.002	5.971	5.497	5.528	5.998	5.501	
434	5.600	.037		6.150	6.127	6.096	5.622	5.653	6.123	5.626	
435	5.725	.037		6.275	6.252	6.221	5.747	5.778	6.248	5.751	
436	5.850	.037		6.400	6.377	6.346	5.872	5.903	6.373	5.876	
437	5.975	.037		6.525	6.502	6.471	5.997	6.028	6.498	6.001	
438	6.225	.040		6.775	6.752	6.721	6.247	6.278	6.748	6.251	
439	6.475	.040		7.025	7.002	6.971	6.497	6.528	6.998	6.501	
440	6.725	.040		7.275	7.252	7.221	6.747	6.778	7.248	6.751	
441	6.975	.040		7.525	7.502	7.471	6.997	7.028	7.498	7.001	
442	7.225	.045	.275	7.775	7.752	4.721	7.247	7.278	7.748	7.251	.375
443	7.475	.045	±.006	8.025	8.002	7.971	7.497	7.528	7.998	7.501	
444	7.725	.045		8.275	8.252	8.221	7.747	7.778	8.248	7.751	
445	7.975	.045		8.525	8.502	8.471	7.997	8.028	8.498	8.001	
446	8.475	.055		9.025	9.002	8.971	8.497	8.528	8.998	8.501	
447	8.975	.055		9.525	9.502	9.471	8.997	9.028	9.498	9.001	
448	9.475	.055		10.025	10.002	9.971	9.497	9.528	9.998	9.501	
449	9.975	.055		10.525	10.502	10.471	9.997	10.028	10.498	10.001	
450	10.475	.060		11.025	11.002	10.971	10.497	10.528	10.998	10.501	
451	10.975	.060		11.525	11.502	11.471	10.997	11.028	11.498	11.001	
452	11.475	.060		12.025	12.002	11.971	11.497	11.528	11.998	11.501	
453	11.975	.060		12.525	12.502	12.471	11.997	12.028	12.498	12.001	
454	12.475	.060		13.025	13.002	12.971	12.497	12.528	12.998	12.501	
455	12.975	.060		13.525	13.502	13.471	12.997	13.028	13.498	13.001	
456	13.475	.070		14.025	14.002	13.971	13.497	13.528	13.998	13.501	
457	13.975	.070		14.525	14.502	14.471	13.997	14.028	14.498	14.001	
458	14.475	.070		15.025	15.002	14.971	14.497	14.528	14.998	14.501	
459	14.975	.070		15.525	15.502	15.471	14.997	15.028	15.498	15.001	
460	15.475	.070		16.025	16.002	15.971	15.497	15.528	15.998	15.501	

* These designs require considerable installation stretch. If assembly breakage is incurred use a compound having higher elongation or use a two-piece piston.
† This groove width does not permit the use of Parbak rings. For pressures above 103.5 Bar (1500 psi), consult Design Chart 5-2 for groove widths where Parbak rings must be used.

Design Table 5-2: Gland Dimensions for Industrial Reciprocating O-Ring Seals, 103.5 Bar (1500 psi) Max.†

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5.31.3 O-Ring Glands for Pneumatic Floating Piston Ring Seals

Design Chart 5-3 provides the basis for calculating gland dimensions. For standard O-ring sizes, these dimensions have been calculated and are listed in Design Table 5-3. The procedures for the use of Design Table 5-3 are outlined in Design Guide 5-3.

After selecting gland dimensions, read horizontally to determine proper O-ring size number. Specify compound.

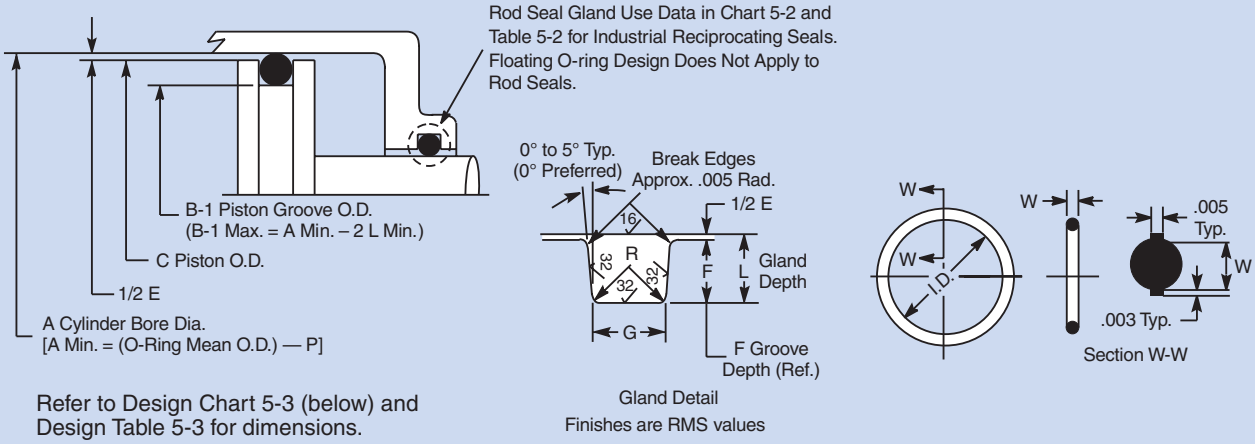
Refer to 8-2, Military Fluid Specification Description, Fluid Compatibility Tables, or Table 3-1 to select the proper compound. The effective part number for an O-ring consists of both a size number and a compound number.

Guide for Design Table 5-3

If Desired Dimensionis Known for	Dimension in Column	Select Closest Horizontally in Column	Read to Determine Dimension for
Bore Dia of cylinder	A	B-1	Groove Dia of piston
		C	OD of piston
		G	Groove width
		A	Bore Dia of cylinder
OD of piston	C	B-1	Groove Dia of piston
		G	Groove width

Design Guide 5-3: Guide For Design Table 5-3

Floating Pneumatic Piston Ring Seal Glands 13.8 Bar (200 psi) Max.



Gland Detail

Finishes are RMS values

Refer to Design Chart 5-3 (below) and Design Table 5-3 for dimensions.

Floating Pneumatic Piston Ring Seal Glands

O-Ring Size (a) Parker No. 2-	W Cross Section Nominal	W Cross Section Actual	P (c) Peripheral Squeeze (Variable)	L Gland Depth	G Groove Width	E Diametral Clearance	Eccentricity Max. (b)	R Groove Radius
006 through 012	1/16	.070 ±.003	.035 to .042	.072 to .076	.075 to .079	.002 to .010	.002	.005 to .015
104 through 116	3/32	.103 ± .003	.038 to .062	.105 to .109	.111 to .115	.002 to .010	.002	.005 to .015
201 through 222	1/8	.139 ± .004	.061 to .082	.143 to .147	.151 to .155	.003 to .011	.003	.010 to .025
309 through 349	3/16	.210 ± .005	.084 to .124	.214 to .218	.229 to .233	.003 to .011	.004	.020 to .035
425 through 460	1/4	.275 ± .006	.140 to .175	.282 to .286	.301 to .305	.004 to .012	.005	.020 to .035

(a) Only sizes listed are recommended for this design.
(b) Total indicator reading between groove and adjacent bearing surface.
(c) Use to calculate A^{min} diameter.

Design Chart 5-3: Design Chart for Floating Pneumatic Piston Ring Seal Glands

Floating Pneumatic Piston Ring Seal Gland Dimensions

O-Ring Size Parker No. 2-	Dimensions I.D.	±	W	Mean O.D. (Ref)	A Bore Dia. (Cylinder) +.004 -.000	B-1 Groove Dia. (Piston) +.000 -.004	C OD (Piston) +.000 -.004	G Groove Width +.004 -.000	P Peripheral Squeeze
006	.114	.005		.254	.219	.075	*.217		.035
007	.145	.005		.285	.249	.105	*.247		.036
008	.176	.005		.316	.279	.135	*.277		.037
009	.208	.005	.070	.348	.309	.165	*.307	.075	.039
010	.239	.005	±.003	.379	.339	.195	*.337		.040
011	.301	.005		.441	.400	.256	*.398		.041
012	.364	.005		.504	.462	.318	.460		.042
104	.112	.005		.318	.280	.070	*.278		.038
105	.143	.005		.349	.309	.099	*.307		.040
106	.174	.005		.380	.338	.128	*.336		.042
107	.206	.005		.412	.368	.158	*.366		.044
108	.237	.005		.443	.397	.187	*.395		.046
109	.299	.005		.505	.457	.247	*.455		.048
110	.362	.005	.103	.568	.518	.308	*.516	.111	.050
111	.424	.005	±.003	.630	.578	.368	*.576		.052
112	.487	.005		.693	.639	.429	.637		.054
113	.549	.007		.755	.699	.489	.697		.056
114	.612	.009		.818	.760	.550	.758		.058
115	.674	.009		.880	.820	.610	.818		.060
116	.737	.009		.943	.881	.671	.879		.062
201	.171	.005		.449	.388	.102	*.385		.061
202	.234	.005		.512	.450	.164	*.447		.062
203	.296	.005		.574	.511	.255	*.508		.063
204	.359	.005		.637	.573	.287	.570		.064
205	.421	.005		.699	.634	.348	.631		.065
206	.484	.005		.762	.696	.410	.693		.066
207	.546	.007		.824	.757	.471	.754		.067
208	.609	.009		.887	.819	.533	.816		.068
209	.671	.009		.949	.880	.594	.877		.069
210	.734	.010		1.012	.942	.656	.939		.070
211	.796	.010	.139	1.074	1.003	.717	1.000	.151	.071
212	.859	.010	±.004	1.137	1.065	.779	1.026		.072
213	.921	.010		1.199	1.126	.840	1.123		.073
214	.984	.010		1.262	1.188	.902	1.185		.074
215	1.046	.010		1.324	1.249	.963	1.246		.075
216	1.109	.012		1.387	1.311	1.025	1.308		.076
217	1.171	.012		1.449	1.372	1.086	1.369		.077
218	1.234	.012		1.512	1.434	1.148	1.431		.078
219	1.296	.012		1.574	1.495	1.209	1.492		.079
220	1.359	.012		1.637	1.557	1.271	1.554		.080
221	1.421	.012		1.699	1.618	1.332	1.615		.081
222	1.484	.015		1.762	1.680	1.394	1.677		.082
309	.412	.005	.210	.832	.748	.320	.745	.229	.084
310	.475	.005	±.105	.895	.810	.382	.807		.085
311	.537	.007		.957	.871	.443	.868		.086

* These designs require considerable installation stretch. If assembly breakage is incurred use a compound having higher elongation or use a two-piece piston.

Design Table 5-3: Floating Pneumatic Piston Ring Seal Gland Dimensions

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Design Table 5-3 — Floating Pneumatic Piston Ring Seal Gland Dimensions (Continued)

O-Ring Size Parker No. 2-	Dimensions I.D.	±	W	Mean O.D. (Ref)	A Bore Dia. (Cylinder) +.004 -.000	B-1 Groove Dia. (Piston) +.000 -.004	C OD (Piston) +.000 -.004	G Groove Width +.004 -.000	P Peripheral Squeeze
312	.600	.009		1.020	.933	.505	.930		.087
313	.662	.009		1.082	.994	.566	.991		.088
314	.725	.010		1.145	1.056	.628	1.053		.089
315	.787	.010		1.207	1.117	.689	1.114		.090
316	.850	.010		1.270	1.179	.751	1.176		.091
317	.912	.010		1.332	1.240	.812	1.237		.092
318	.975	.010		1.395	1.302	.874	1.299		.093
319	1.037	.010		1.457	1.363	.935	1.360		.094
320	1.100	.012		1.520	1.425	.997	1.422		.095
321	1.162	.012		1.582	1.486	1.058	1.483		.096
322	1.225	.012		1.645	1.548	1.120	1.545		.097
323	1.287	.012		1.707	1.609	1.181	1.636		.098
324	1.350	.012		1.770	1.671	1.243	1.668		.099
325	1.475	.015		1.895	1.795	1.367	1.792		.100
326	1.600	.015		2.020	1.919	1.491	1.916		.101
327	1.725	.015		2.145	2.043	1.615	2.040		.102
328	1.850	.015		2.270	2.167	1.739	2.164		.103
329	1.975	.018	.210	2.395	2.291	1.863	2.288	.229	.104
330	2.100	.018	±.005	2.520	2.415	1.987	2.412		.105
331	2.225	.018		2.645	2.539	2.111	2.536		.106
332	2.350	.018		2.770	2.663	2.235	2.660		.107
333	2.475	.020		2.895	2.787	2.359	2.784		.108
334	2.600	.020		3.020	2.911	2.483	2.908		.109
335	2.725	.020		3.145	3.035	2.607	3.032		.110
336	2.850	.020		3.270	3.159	2.731	3.156		.111
337	2.975	.024		3.395	3.283	2.855	3.280		.112
338	3.100	.024		3.520	3.407	2.979	3.404		.113
339	3.225	.024		3.645	3.531	3.103	3.528		.114
340	3.350	.024		3.770	3.655	3.270	3.652		.115
341	3.475	.024		3.895	3.779	3.351	3.776		.116
342	3.600	.028		4.020	3.903	3.475	3.900		.117
343	3.725	.028		4.145	4.027	3.599	4.024		.118
344	3.850	.028		4.270	4.151	3.723	4.148		.119
345	3.975	.028		4.395	4.275	3.847	4.272		.120
346	4.100	.028		4.520	4.399	3.971	4.396		.121
347	4.225	.030		4.645	4.523	4.095	4.520		.122
348	4.350	.030		4.773	4.647	4.219	4.644		.123
349	4.475	.030		4.895	4.771	4.343	4.768		.124
425	4.475	.033		5.025	4.885	4.321	4.881		.140
426	4.600	.033		5.150	5.009	4.445	5.005		.141
427	4.725	.033		5.275	5.133	4.569	5.129		.142
428	4.850	.033	.275	5.400	5.257	4.693	5.253	.301	.143
429	4.975	.037	±.006	5.525	5.381	4.817	5.377		.144
430	5.100	.037		5.650	5.505	4.941	5.501		.145
431	5.225	.037		5.775	5.629	5.065	5.625		.146
432	5.350	.037		5.900	5.753	5.189	5.749		.147

* These designs require considerable installation stretch. If assembly breakage is incurred use a compound having higher elongation or use a two-piece piston.

Design Table 5-3: Floating Pneumatic Piston Ring Seal Gland Dimensions

Design Table 5-3 — Floating Pneumatic Piston Ring Seal Gland Dimensions (Continued)

O-Ring Size Parker No. 2-	Dimensions I.D.	±	W	Mean O.D. (Ref)	A Bore Dia. (Cylinder) +.004 -.000	B-1 Groove Dia. (Piston) +.000 -.004	C OD (Piston) +.000 -.004	G Groove Width +.004 -.000	P Peripheral Squeeze
433	5.475	.037		6.025	5.877	5.313	5.873		.148
434	5.600	.037		6.150	6.001	5.437	5.997		.149
435	5.725	.037		6.275	6.125	5.561	6.121		.150
436	5.850	.037		6.400	6.249	5.685	6.245		.151
437	5.975	.037		6.525	6.373	5.809	6.369		.152
438	6.225	.040		6.775	6.622	6.058	6.618		.153
439	6.475	.040		7.025	6.871	6.307	6.867		.154
440	6.725	.040		7.275	7.120	6.556	7.116		.155
441	6.975	.040		7.525	7.369	6.805	7.365		.156
442	7.225	.045	.275	7.775	7.618	7.054	7.614	.301	.157
443	7.475	.045	±.006	8.025	7.867	7.303	7.863		.158
444	7.725	.045		8.275	8.116	7.552	8.112		.159
445	7.975	.045		8.525	8.365	7.801	8.361		.160
446	8.475	.055		9.025	8.864	8.300	8.860		.161
447	8.975	.055		9.525	9.363	8.799	9.359		.162
448	9.475	.055		10.025	9.862	9.298	9.858		.163
449	9.975	.055		10.525	10.361	9.797	10.357		.164
450	10.475	.060		11.025	10.860	10.296	10.856		.165
451	10.975	.060		11.525	11.359	10.795	11.355		.166
452	11.475	.060		12.025	11.858	11.294	11.854		.167
453	11.975	.060		12.525	12.357	11.793	12.353		.168
454	12.475	.060		13.025	12.856	12.292	12.852		.169
455	12.975	.060		13.525	13.355	12.791	13.351		.170
456	13.475	.070		14.025	13.854	13.290	13.850		.171
457	13.975	.070		14.525	14.353	13.789	14.349		.172
458	14.475	.070		15.025	14.852	14.288	14.848		.173
459	14.975	.070		15.525	15.351	14.787	15.347		.174
460	15.475	.070		16.025	15.850	15.286	15.846		.175

* These designs require considerable installation stretch. If assembly breakage is incurred use a compound having higher elongation or use a two-piece piston.

Design Table 5-3: Floating Pneumatic Piston Ring Seal Gland Dimensions

Guide For Design Table 5-4

If Desired Dimension is Known for	Select Closest Dimension in Column	Read Horizontally in Column	To Determine Dimension for
OD of shaft	B	A-1	Groove Dia. for shaft
		D	Throat Dia.
		G	Groove width
Throat Dia.	D	A-1	Groove Dia. for shaft
		B	OD of shaft
		G	Groove width

Design Guide 5-4: Guide For Design Table 5-4

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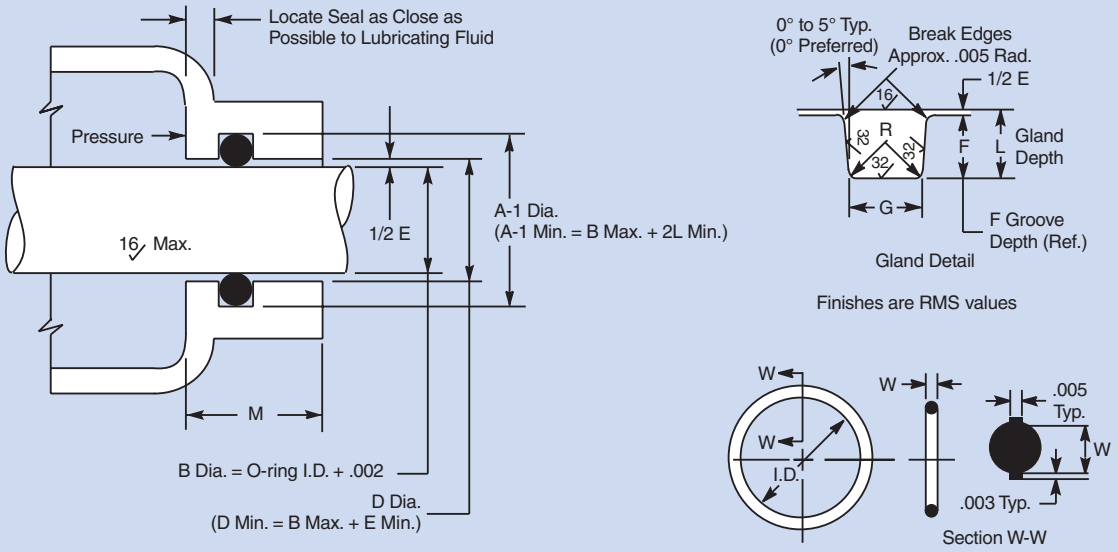
5.31.4 O-Ring Glands for Rotary Seals

Design Chart 5-4 provides the basis for calculating gland dimensions. For standard O-ring sizes, these dimensions have been calculated and are listed in Design Table 5-4. The procedures for the use of Design Table 5-4 are outlined in Design Guide 5-4 below.

After selecting gland dimensions, read horizontally to determine proper O-ring size number. Specify compound. Refer to the discussion on rotary seals earlier in this chapter for the selection of the proper compound The effective part number for an O-ring consists of both a size number and a compound number.

Rotary O-Ring Seal Glands, 55.2 Bar (800 psi) Max.

See Rotary Seal discussion in Dynamic Seals section before using this chart.



Note: Due to effect of centrifugal force, do not locate groove in shaft. Refer to Design Chart 5-4 (below) and Design Table 5-4 for dimensions.

Rotary O-Ring Seal Glands, 55.2 Bar (800 psi) Max. (c)
(Below 200 FPM, Use Design Chart 5-2)

O-Ring Size Parker No. 2-	W Cross Section Nominal	W Cross Section Actual	Maximum Speed FPM (a)	Squeeze %	L Gland Depth	G Groove Width	E (c) Diametral Clearance	Eccentricity Max (b)	M Bearing Length Min. (c)	R Groove Radius
004 through 045	1/16	0.070 ±.003	200 to 1500	0-11	0.065 to 0.067	0.075 to 0.079	0.012 to 0.016	0.002	0.700	0.005 to 0.015
102 through 163	3/32	0.103 ±.003	200 to 600	1-8 1/2	0.097 to 0.099	0.108 to 0.112	0.012 to 0.016	0.002	1.030	0.005 to 0.015
201 through 258	1/8	0.139 ±.004	200 to 400	0-7	0.133 to 0.135	0.144 to 0.148	0.016 to 0.020	0.003	1.390	0.010 to 0.025

(a) Feet per minute = 0.26 X Shaft Diameter (inches) X rpm.
(b) Total indicator reading between groove OD, shaft, and adjacent bearing surface.
(c) If clearance (extrusion gap) must be reduced for higher pressures, bearing length M must be no less than the minimum figures given. Clearances given are based on the use of 80 shore durometer minimum O-ring for 55.2 Bar (800 psi) max. See Figure 3-2 for recommended clearances when pressures exceed 55.2 Bar (800 psi).

Design Chart 5-4: Design Chart for Rotary O-Ring Seal Glands

Rotary O-Ring Seal Gland Dimensions

O-Ring Size Parker No. 2-	Dimensions I.D.	±	W	Mean O.D. (Ref)	B OD (Shaft) +.000 -.001	A-1 Groove Dia. (Gland) +.003 -.000	D Throat Dia. +.003 -.000	G Groove Width +.004 -.000
004	.070	.005		.210	.072	.202	.084	
005	.101	.005		.241	.103	.233	.115	
006	.114	.005		.254	.116	.246	.128	
007	.145	.005		.285	.147	.277	.159	
008	.176	.005		.316	.178	.308	.190	
009	.208	.005		.348	.210	.340	.222	
010	.239	.005		.379	.241	.371	.253	
011	.301	.005		.441	.303	.433	.315	
012	.364	.005		.504	.366	.496	.378	
013	.426	.005		.566	.428	.558	.440	
014	.489	.005		.629	.491	.621	.503	
015	.551	.007		.691	.553	.683	.565	
016	.614	.009		.754	.616	.746	.628	
017	.676	.009		.816	.678	.808	.693	
018	.739	.009	.070	.879	.741	.871	.753	.075
019	.801	.009	±.003	.941	.803	.933	.815	
020	.864	.009		1.004	.866	.996	.878	
021	.926	.009		1.066	.928	1.058	.940	
022	.989	.010		1.129	.991	1.121	1.003	
023	1.051	.010		1.191	1.053	1.183	1.065	
024	1.114	.010		1.254	1.116	1.246	1.128	
025	1.176	.011		1.316	1.178	1.308	1.190	
026	1.239	.011		1.379	1.241	1.371	1.253	
027	1.301	.011		1.441	1.303	1.433	1.315	
028	1.364	.013		1.504	1.366	1.496	1.378	
029	1.489	.013		1.629	1.491	1.621	1.503	
030	1.614	.013		1.754	1.616	1.746	1.628	
031	1.739	.015		1.879	1.741	1.871	1.753	
032	1.864	.015		2.004	1.866	1.996	1.878	
033	1.989	.018		2.129	1.991	2.121	2.003	
034	2.114	.018		2.254	2.116	2.246	2.128	
035	2.239	.018		2.379	2.241	2.371	2.253	
036	2.364	.018		2.504	2.366	2.496	2.378	
037	2.489	.018		2.629	2.491	2.621	2.503	
038	2.614	.020		2.754	2.616	2.746	2.628	
039	2.739	.020		2.879	2.741	2.871	2.753	
040	2.864	.020		3.004	2.866	2.996	2.878	
041	2.989	.024		3.129	2.991	3.121	3.003	
042	3.239	.024		3.379	3.241	3.371	3.253	
043	3.489	.024		3.629	3.491	3.621	3.503	
044	3.739	.027		3.879	3.741	3.871	3.753	
045	3.989	.027		4.129	3.991	4.121	4.003	
102	.049	.005		.255	.051	.245	.063	
103	.081	.005	.103	.287	.083	.277	.095	.108
104	.112	.005	±.003	.318	.114	.308	.126	
105	.143	.005		.349	.145	.339	.157	
106	.174	.005		.380	.176	.370	.188	

† For pressures over 55.2 Bar (800 psi), consult Design Chart 5-4 and the design sections of this Handbook.

Design Table 5-4: Rotary O-Ring Seal Gland Dimensions

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Rotary O-Ring Seal Gland Dimensions (Continued)

O-Ring Size Parker No. 2-	Dimensions I.D.	±	W	Mean O.D. (Ref)	B OD (Shaft) +.000 -.001	A-1 Groove Dia. (Gland) +.003 -.000	D Throat Dia. +.003 -.000	G Groove Width +.004 -.000
107	.206	.005		.412	.208	.402	.220	
108	.237	.005		.443	.239	.433	.251	
109	.299	.005		.505	.301	.495	.313	
110	.362	.005		.568	.364	.558	.376	
111	.424	.005		.630	.426	.620	.438	
112	.487	.005		.693	.489	.683	.501	
113	.549	.007		.755	.551	.745	.563	
114	.612	.009		.818	.614	.808	.626	
115	.674	.009		.880	.676	.870	.688	
116	.737	.009		.943	.739	.933	.751	
117	.799	.010		1.005	.801	.995	.813	
118	.862	.010		1.068	.864	1.058	.876	
119	.924	.010		1.130	.926	1.120	.938	
120	.987	.010	.103	1.193	.989	1.183	1.001	.108
121	1.049	.010	±.003	1.255	1.051	1.245	1.063	
122	1.112	.010		1.318	1.114	1.308	1.126	
123	1.174	.012		1.380	1.176	1.370	1.188	
124	1.237	.012		1.443	1.239	1.433	1.251	
125	1.299	.012		1.505	1.301	1.495	1.313	
126	1.362	.012		1.568	1.364	1.558	1.376	
127	1.424	.012		1.630	1.426	1.620	1.438	
128	1.487	.012		1.693	1.489	1.683	1.501	
129	1.549	.015		1.755	1.551	1.745	1.563	
130	1.612	.015		1.818	1.614	1.808	1.626	
131	1.674	.015		1.880	1.676	1.870	1.688	
132	1.737	.015		1.943	1.739	1.933	1.751	
133	1.799	.015		2.005	1.801	1.995	1.813	
134	1.862	.015		2.068	1.864	2.058	1.876	
135	1.925	.017		2.131	1.927	2.121	1.939	
136	1.987	.017		2.193	1.989	2.183	2.001	
137	2.050	.017		2.256	2.052	2.246	2.064	
138	2.112	.017		2.318	2.114	2.308	2.126	
139	2.175	.017		2.381	2.177	2.371	2.189	
140	2.237	.017		2.443	2.239	2.433	2.251	
141	2.300	.020		2.506	2.302	2.496	2.314	
142	2.362	.020		2.568	2.364	2.558	2.376	
143	2.425	.020		2.631	2.427	2.621	2.439	
144	2.487	.020		2.693	2.489	2.683	2.501	
145	2.550	.020		2.756	2.552	2.746	2.564	
146	2.612	.020		2.818	2.614	2.808	2.626	
147	2.675	.022		2.881	2.677	2.871	2.689	
148	2.737	.022		2.943	2.739	2.933	2.751	
149	2.800	.022		3.006	2.802	2.996	2.814	
150	2.862	.022		3.068	2.864	3.058	2.876	
151	2.987	.024		3.193	2.989	3.183	3.001	
152	3.237	.024		3.443	3.239	3.433	3.251	
153	3.487	.024		3.693	3.489	3.683	3.501	

† For pressures over 55.2 Bar (800 psi), consult Design Chart 5-4 and the design sections of this Handbook.

Design Table 5-4: Rotary O-Ring Seal Gland Dimensions

Rotary O-Ring Seal Gland Dimensions (Continued)

O-Ring Size Parker No. 2-	Dimensions I.D.	±	W	Mean O.D. (Ref)	B OD (Shaft) +.000 -.001	A-1 Groove Dia. (Gland) +.003 -.000	D Throat Dia. +.003 -.000	G Groove Width +.004 -.000
154	3.737	.028		3.943	3.739	3.933	3.751	
155	3.987	.028		4.193	3.989	4.183	4.001	
156	4.237	.030		4.443	4.239	4.433	4.251	
157	4.487	.030	.103	4.693	4.489	4.683	4.501	.108
158	4.737	.030	±.003	4.943	4.739	4.933	4.751	
159	4.987	.035		5.193	4.989	5.183	5.001	
160	5.237	.035		5.443	5.239	5.433	5.251	
161	5.487	.035		5.693	5.489	5.683	5.501	
162	5.737	.035		5.943	5.739	5.933	5.751	
163	5.987	.035		6.193	5.989	6.183	6.001	
201	.171	.005		.449	.173	.439	.189	
202	.234	.005		.512	.236	.502	.252	
203	.296	.005		.574	.298	.564	.314	
204	.359	.005		.637	.361	.627	.377	
205	.421	.005		.699	.423	.689	.439	
206	.484	.005		.762	.486	.752	.502	
207	.546	.007		.824	.548	.814	.564	
208	.609	.009		.887	.611	.877	.627	
209	.671	.009		.949	.673	.939	.689	
210	.734	.010		1.012	.736	1.002	.752	
211	.796	.010		1.074	.798	1.064	.814	
212	.859	.010		1.137	.861	1.127	.877	
213	.921	.010		1.199	.923	1.189	.939	
214	.984	.010	.139	1.262	.986	1.252	1.002	.144
215	1.046	.010	±.004	1.324	1.048	1.314	1.064	
216	1.109	.012		1.387	1.111	1.377	1.127	
217	1.171	.012		1.449	1.173	1.439	1.189	
218	1.234	.012		1.512	1.236	1.502	1.252	
219	1.296	.012		1.574	1.298	1.564	1.314	
220	1.359	.012		1.637	1.361	1.627	1.377	
221	1.421	.012		1.699	1.423	1.689	1.439	
222	1.484	.015		1.762	1.486	1.752	1.502	
223	1.609	.015		1.887	1.611	1.877	1.627	
224	1.734	.015		2.012	1.736	2.002	1.752	
225	1.859	.018		2.137	1.861	2.127	1.877	
226	1.984	.018		2.262	1.986	2.252	2.002	
227	2.109	.018		2.387	2.111	2.377	2.127	
228	2.234	.020		2.512	2.236	2.502	2.252	
229	2.359	.020		2.637	2.361	2.627	2.377	
230	2.484	.020		2.762	2.486	2.752	2.502	
231	2.609	.020		2.887	2.611	2.877	2.627	
232	2.734	.024		3.012	2.736	3.002	2.752	
233	2.859	.024		3.137	2.861	3.127	2.877	
234	2.984	.024		3.262	2.986	3.252	3.002	
235	3.109	.024		3.387	3.111	3.377	3.127	
236	3.234	.024		3.512	3.236	3.502	3.252	
237	3.359	.024		3.637	3.361	3.627	3.377	

† For pressures over 55.2 Bar (800 psi), consult Design Chart 5-4 and the design sections of this Handbook.

Design Table 5-4: Rotary O-Ring Seal Gland Dimensions

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Rotary O-Ring Seal Gland Dimensions (Continued)

O-Ring Size Parker No. 2-	I.D.	Dimensions ±	Dimensions W	Mean O.D. (Ref)	B OD (Shaft) +.000 -.001	A-1 Groove Dia. (Gland) +.003 -.000	D Throat Dia. +.003 -.000	G Groove Width +.004 -.000
238	3.484	.024		3.762	3.486	3.752	3.502	
239	3.609	.028		3.887	3.611	3.877	3.627	
240	3.734	.028		4.012	3.736	4.002	3.752	
241	3.859	.028		4.137	3.861	4.127	3.877	
242	3.984	.028		4.262	3.986	4.252	4.002	
243	4.109	.028		4.387	4.111	4.377	4.127	
244	4.234	.030		4.512	4.236	4.502	4.252	
245	4.359	.030		4.637	4.361	4.627	4.377	
246	4.484	.030	.139	4.762	4.486	4.752	4.502	.144
247	4.609	.030	±.004	4.887	4.611	4.877	4.627	
248	4.734	.030		5.012	4.736	5.002	4.752	
249	4.859	.035		5.137	4.861	5.127	4.877	
250	4.984	.035		5.262	4.986	5.252	5.002	
251	5.109	.035		5.387	5.111	5.377	5.127	
252	5.234	.035		5.512	5.236	5.502	5.252	
253	5.359	.035		5.637	5.361	5.627	5.377	
254	5.484	.035		5.762	5.486	5.752	5.502	
255	5.609	.035		5.887	5.611	5.877	5.627	
256	5.734	.035		6.012	5.736	6.002	5.752	
257	5.859	.035		6.137	5.861	6.127	5.877	
258	5.984	.035		6.262	5.986	6.252	6.002	

† For pressures over 55.2 Bar (800 psi), consult Design Chart 5-4 and the design sections of this Handbook.

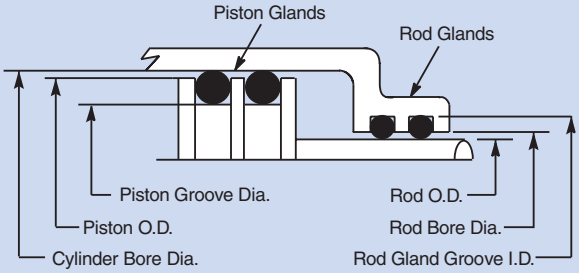
Design Table 5-4: Rotary O-Ring Seal Gland Dimensions

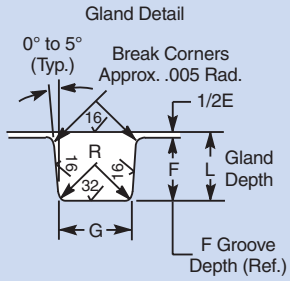
5.32 Dynamic Vacuum Sealing

There is very little data available on dynamic vacuum seals, but reasonably low leak rates have been reported using two O-rings seals designed according to the standard radial dynamic design dimensions for reciprocating seals which are shown in Design Chart 5-5.

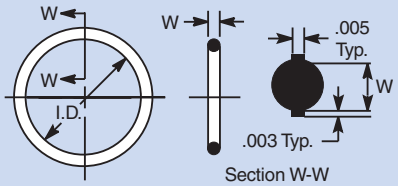
In sealing gases and vacuum, it is quite feasible to use two O-ring seals in separate grooves. (In reciprocating hydraulic applications, however, such redundant seals are not recommended because of the danger of creating a pressure trap between the two seals.)

For Reciprocating Vacuum Packing Glands





Finishes are RMS values



Refer to Design Chart 5-5 (below) for dimensions.

Reciprocating Vacuum Packing Glands

O-Ring Size AS568A-	W Cross-Section Nominal	W Cross-Section Actual	L Gland Depth	Squeeze Actual	Squeeze %	E Diametral Clearance	G Groove Width	R Groove Radius	Max.* Eccentricity
006 through 012	1/16	.070 ±.003	.055 to .057	.010 to .018	15 to 25	.002 to .005	.093 to .098	.005 to .015	.002
104 through 116	3/32	.103 ±.003	.088 to .090	.010 to .018	10 to 17	.002 to .005	.140 to .145	.005 to .015	.002
201 through 222	1/8	.139 ±.004	.121 to .123	.012 to .022	9 to 16	.003 to .006	.187 to .192	.010 to .025	.003
309 through 349	3/16	.210 ±.005	.185 to .188	.017 to .030	8 to 14	.003 to .006	.281 to .286	.020 to .035	.004
425 through 460	1/4	.275 ±.006	.237 to .240	.029 to .044	11 to 16	.004 to .007	.375 to .380	.020 to .035	.005

*Total indicator reading between groove and adjacent bearing surface.

Design Chart 5-5: Design Chart for Reciprocating Vacuum Packing Glands

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