Click to Go to CATALOG Table of Contents

Engineering

Contents

Parker Fluid Power Seals for All Application Technologies

Seals have been used since ancient times and have evolved into a wide variety of shapes and materials. For those who are not familiar with sealing technology, the number of options available can be confusing. Selecting the most suitable product for a given application can be difficult. This engineering section will assist in product selection by explaining the fundamentals of seal design and material technology.

Sealing Theory

Static vs. Dynamic Sealing

Every seal, whether static or dynamic, must seal against at least two contacting surfaces. In static applications, both surfaces are non-moving relative to one another. In dynamic applications at least one surface is in motion relative to the other sealing surface(s). For example, in a standard hydraulic cylinder, the rod and piston seals would be classified as dynamic seals, while the seal between the bore and the head gland would be considered a static seal.

In both static and dynamic applications, a certain amount of squeeze or compression is required upon installation to maintain contact with the sealing surfaces and prevent fluid leakage. Dynamic applications in particular involve other variables and require that additional factors be evaluated to ensure proper system performance. These variables are discussed in this section.

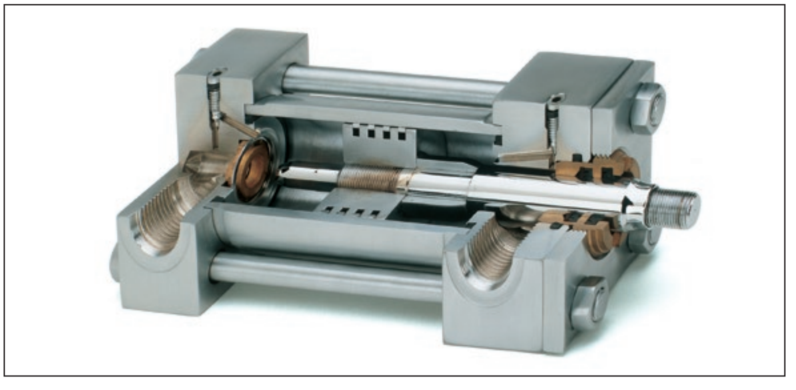


Fig. 2-1. Hydraulic cylinder

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

Click to Go to CATALOG Table of Contents
Click to Go to SECTION Table of Contents

Leakage Control

When choosing a sealing system, the desired result is ultimately leakage control. Seal design and material improvements have made it possible not only to have seal combinations that provide zero leakage, but also provide extended life in a variety of applications. Aside from the seals themselves, a thorough understanding of system parameters is necessary to obtain the best results.

Optimal sealing is best achieved by taking a systems approach to the seal package rather than considering components individually. Our profiles have been designed specifically to complement one another to create high performance systems. For example, pairing a Parker rod seal with a Parker wiper minimizes fluid leakage and maximizes contamination exclusion. Our rod seals are designed with knife-trimmed lips to ensure the best possible film breaking. This dry rod technology permits the wiper to be extremely aggressive, excluding contamination without building up oil leakage around the wiper. Another systems approach to effectively control leakage is to incorporate multiple sealing lips. Parker's BR buffer ring, BT u-cup and AH double-lip canned wiper are designed to work together to give optimized performance and the driest sealing available in the industry (see Figure 2-2).

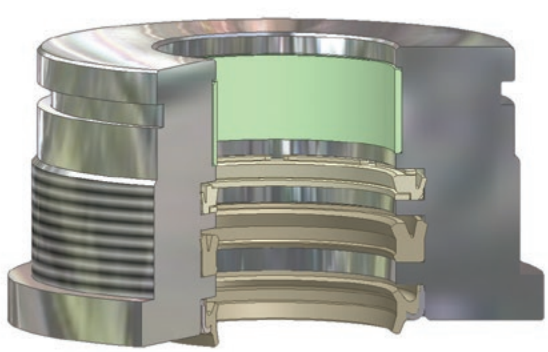


Figure 2-2. BR, BT, AH sealing system for leakage control

Even when appropriate seals are specified, it is still possible to experience leakage due to factors extending beyond the seals themselves. Examples are hardware considerations like surface finish, installation damage, seal storage, chemical wash downs, maintenance and contamination. Adhering to the design recommendations found herein not only for seals, but also for the mating hardware will provide the greatest likelihood of minimized leakage.

Lip vs. Squeeze Seals

The cross-sectional shape of a seal dramatically affects how it functions, especially at low pressure. The greatest trade-off in dynamic sealing is low friction performance vs. low pressure sealability. At low pressure, friction, wear and sealing ability are affected by whether or not the seal is a lip or squeeze profile (see Figure 2-3). With this in mind, seals are often categorized as either "lip seals" or "squeeze seals," and many fall somewhere in between. Lip seals are characterized by low friction and low wear; however, they also exhibit poor low pressure sealability. Squeeze seals are characterized by just the opposite: high friction and high wear, but better low pressure sealability.

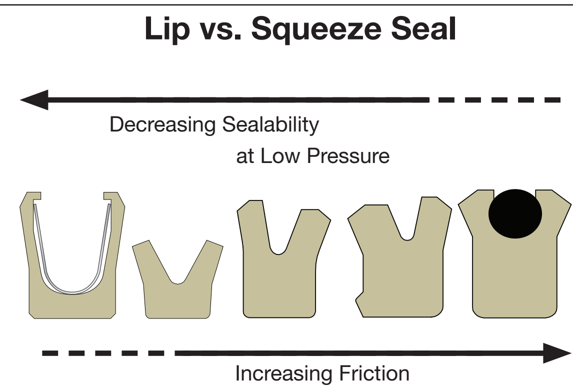


Figure 2-3. Lip seal vs. squeeze seal

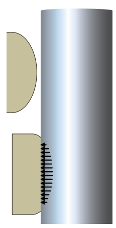
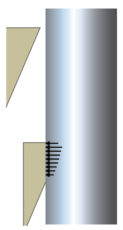
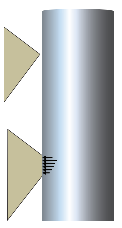
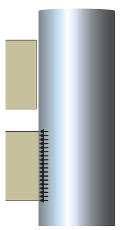
As described above, a squeeze type seal will generate much more sealing force than a lip type seal. The assumption here is that both seals are under zero or low pressure. However, as fluid pressure increases, the differences between seal types become insignificant due to the force from the fluid pressure overcoming the designed squeeze. Pressure generally improves leakage control, but increases friction and its associated heat, wear and potential for extrusion.

In pneumatic applications, low friction is of the utmost importance. As such, lip seals are an excellent choice for these low pressure applications. Conversely, in hydraulic cylinders, where high system pressures easily overcome frictional forces, squeeze seals are often the appropriate choice. An example of a hydraulic application in which a squeeze seal would not be appropriate is a gravity returned hydraulic ram. In this case, a lip type hydraulic seal would generate lower friction, allowing the gravity return to function properly.

Effects of Lip Geometries

Lip geometry will determine several functions of the seal. Force concentration on the shaft, film breaking ability, hydroplaning characteristics and contamination exclusion are all factors dependent on lip shape. Table 2-1 shows four different lip shapes and provides helpful insights for choosing an appropriate lip geometry.

Table 2-1. Seal Lip Contact Shape

Contact Shape	Rounded	Straight Cut	Beveled	Square
Seal Lip Shape Shape of Contact Force/ Stress Profile				
Film Breaking Ability	Low	High	Very High	Medium
Contamin-ation Exclusion	Low	Very High	Low	High
Tendency to Hydro-plane	High	Very Low	Low	Medium
Typical Uses	Pneu-matic U-cups	Wipers and Piston Seals	Rod Seals	Piston Seals

Friction

Friction is a function of the radial force exerted by the seal and the coefficient of friction between the seal and the dynamic sealing surface. Reducing friction is generally desirable, but not always necessary. Friction is undesirable because of heat generation, seal wear and reduced system efficiency.

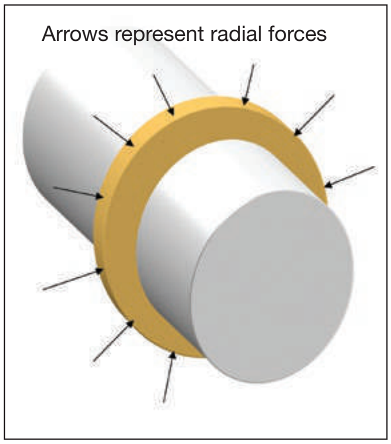


Figure 2-4. Radial force

Factors that affect the radial force are:
- Pressure
- Material modulus
- Temperature
- Lip geometry
- Squeeze vs. lip seal

Factors that affect the coefficient of friction are:
- Seal material
- Dynamic surface roughness
- Temperature
- Lubrication

When the proper seal selection is made, most seals will function such that friction is not a concern. However, when friction becomes critical, there are several ways to reduce it:
- Reduce the lip cross-section
- Decrease lip squeeze
- Change seal material
- Evaluate the hardware's surface finish
- Reduce system pressure
- Improve lubrication

Lowering friction increases seal life by reducing wear, increasing extrusion resistance, decreasing compression set and the rate of chemical attack.

> ***Breakaway friction*** *must be overcome for movement to begin. It is influenced by the duration in which an application remains stationary. The longer the duration, the more lubrication will be forced out from between the seal and the contacting surface. The seal material then conforms to the profile of the surface finish. These events increase breakaway friction.*

Stick-slip is characterized by distinct stop-start movement of the cylinder, and may be so rapid that it resembles severe vibration, high pitched noise or chatter. Seals are often thought to be the source of the stick-slip, but other components or hardware can create this issue.

Causes of stick-slip include swelling of wear rings or back-up rings, extreme side-loading, valve pulsation, poor fluid lubricity, external sliding surfaces or seal pressure trapping. This condition can be puzzling or difficult to resolve. Possible causes and trouble-shooting solutions are listed in the following Table 2-2.

Table 2-2. Stick-slip Causes and Troubleshooting Tips

Possible Causes	Troubleshooting Tips
Surface finish out of specification	Verify surface is neither too smooth or too rough
Poor fluid lubricity	Change fluid or use oil treatments or friction reducers
Binding wear rings	Check gland dimensions, check for thermal or chemical swell
Side loading	Review cylinder alignment, incorporate adequate bearing area
Seal friction	Use material with lower coefficient of friction
Cycle speed	Slow movement increases likelihood of stick-slip
Temperature	High temperature softens seals, expands wear rings, and can cause thermal expansion differences within hardware
Valve pulsation	Ensure valves are properly sized and adjusted
External hardware	Review system for harmonic resonance

Pressure Effects and Extrusion

Extrusion occurs when fluid pressure forces the seal material into the clearance gap between mating hardware. Dynamic motion further promotes extrusion, as surfaces in motion tend to pull material into the extrusion gap, generating additional frictional forces and heat. This can cause premature failure via several modes. Extruded seal material can break away and get caught underneath sealing lips, creating leak paths. As material continues to break away, seal geometry erodes, causing instability and eventual leakage. Additionally, heat generated from added friction will cause the seals to take a compression set, dramatically shortening their life.

Careful design considerations should be evaluated to prevent extrusion. For example, minimizing clearance gaps and selecting a proper material based on system temperature, pressure and fluid are both helpful in reducing the risk of extrusion. As clearance gaps increase, less pressure is required in order for extrusion to occur. Higher temperatures can also play a role in this effect by causing seal materials to soften, encouraging extrusion at lower pressures. If the seal material chosen is not suitable to be used in the system fluid, softening due to chemical attack can also decrease its ability to resist extrusion.

Figure 2-5. Extrusion damage

The following Table 2-3 lists possible causes of extrusion and troubleshooting tips for preventative or corrective measures.

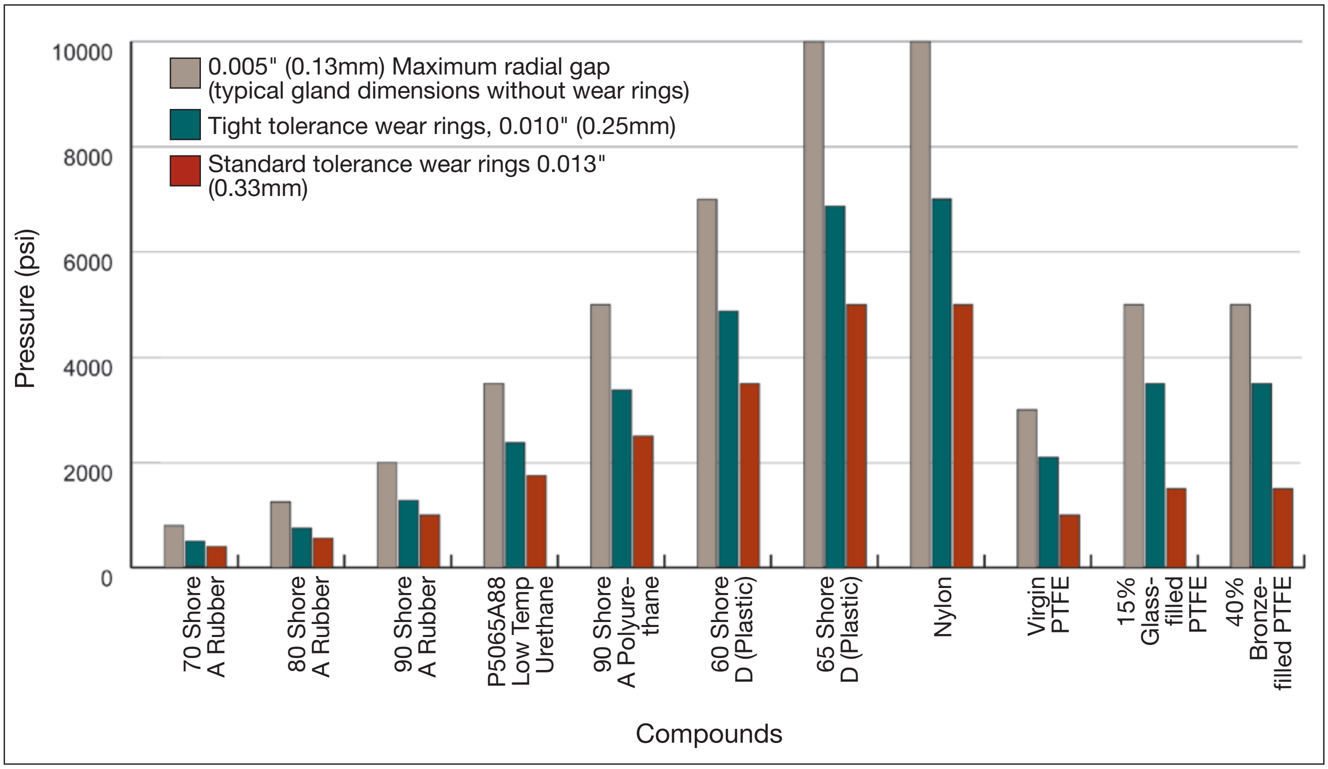
Table 2-3. Extrusion Causes and Troubleshooting Tips

Possible Causes
Large extrusion gaps
High operating temperature
Soft materials
High system pressure
Pressure spikes
Side loading
Wear rings
Chemical compatibility
Troubleshooting Tips
Reduce extrusion gaps
Check gland dimensions
Replace commercial grade wear rings with tight tolerance wear rings
Incorporate back-up rings
Evaluate size and positioning of wear rings for side load resistance
Consider harder, higher modulus and tensile strength compound
Match seal compound for pressure, temperature and fluid compatibility

By definition, the radial gap is one-half of the diametrical gap. The actual extrusion gap is often mistaken as the radial gap. This is too optimistic in most cases because side loading of the rod and piston will shift the diametrical clearance to one side. Often, gravity alone is sufficient for this to occur. Good practice is to design around worst case conditions so that extrusion and seal damage do not occur. Table 2-4 provides maximum *radial* extrusion gaps for various seal compounds.

As a general rule of thumb, *the pressure rating of dynamic seals will be approximately one-half that of static seals.*

Table 2-4. Typical Pressure Ratings for Standard Seal Compounds in Reciprocating Applications at +160°F (see Note)



Note: Pressure ratings are based upon a test temperature of +160°F (+70°C). Lower temperatures will increase a material's pressure rating. Higher temperatures will decrease pressure ratings. Maximum radial gap is equal to the diametrical gap when wear rings are not used. Wear rings keep hardware concentric, but increase extrusion gaps to keep metal-to-metal contact from occurring, thereby decreasing pressure ratings when used.

As noted in Table 2-4, pressure ratings decrease when wear rings are used due to the larger extrusion gaps required to eliminate metal-to-metal contact. If wear rings are used, be sure to consult Section 9 (Wear Rings) and Section 10 (Back-ups) for appropriate hardware dimensions. Wear ring hardware dimensions for the piston and rod throat diameters always supersede those dimensions called out for the seals themselves.

Seal Wear

Seals will inevitably wear in dynamic applications, but with appropriate design considerations, this can be minimized. The wear pattern should be even and consistent around the circumference of the dynamic lip. A small amount of even wear will not drastically affect seal performance; however, if the wear patterns are uneven or grooved, or if the amount of wear is excessive, performance may be dramatically reduced. There are many factors that influence seal wear, many of which are described in the following Table 2-5.

Table 2-5. Factors Influencing Seal Wear

Factors that Influence Seal Wear	
Rough surface finish	Excessive abrasion may occur above 12 µin Ra
Ultra smooth surface finish	Surface finishes below 2 µin Ra can create aggressive seal wear due to lack of lubrication
High pressure	Increases the radial force of the seal against the dynamic surface
High temperature	While hot, materials soften, thus reducing tensile strength
Poor fluid lubricity	Increases friction and temperature at sealing contact point
Tensile strength of seal compound	Higher tensile strength increases the material's resistance to tearing and abrading
Fluid incompatibility	Softening of seal compound leads to reduced tensile strength
Coefficient of friction of seal compound	Higher coefficient materials generate higher frictional forces
Abrasive fluid or contamination	Creates grooves in the lip, scores the sealing surface and forms leak paths
Extremely hard sealing surface	Sharp peaks on hard surfaces will not be rounded off during normal contact with the wear rings and seals, accelerating wear conditions

06/01/2014

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

Seal wear may be indicated by flattening out of the contact point, or, in extreme circumstances, may appear along the entire dynamic surface as shown in Figure 2-6.

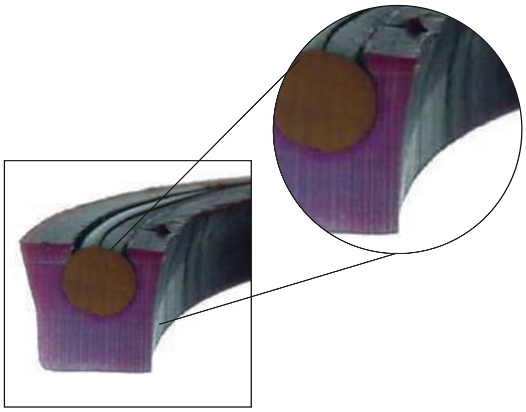


Figure 2-6. Seal wear on dynamic surface

Seal Stability

Dynamic stability is integral to a seal's performance, allowing the lip to effectively contact the sealing surface, eliminating rocking and pumping effects and promoting an even wear pattern at the sealing contact point. Instability can create leakage and seal damage. A typical instability malfunction known as "spiral failure" can occur when o-rings are used in reciprocating applications. Due to frictional forces that occur while the system is cycling, the o-ring will tend to roll or twist in the groove, causing leakage and even possible breakage. A square geometry will tend to resist this better than a round profile, but is not impervious to instability failure. Rectangular geometries provide the best stability in dynamic applications.

Other less obvious factors that influence the stability of a seal are:

- Percent gland fill
- Hardness or stiffness of the seal material

Fig. 2-7. Instability failure of a square profile piston seal

- Rough surfaces which create high friction
- Cross-section (larger is better)
- Design features of a seal (i.e. stabilizing lip, non-symmetrical design). Figure 2-8 illustrates how design features can make a seal more stable. In the first FEA plot, the seal is centered in the gland and does not incorporate a stabilizing lip. In the second plot, the seal is loaded against the static gland and includes a stabilizing lip. Stability has been enhanced by the design changes.

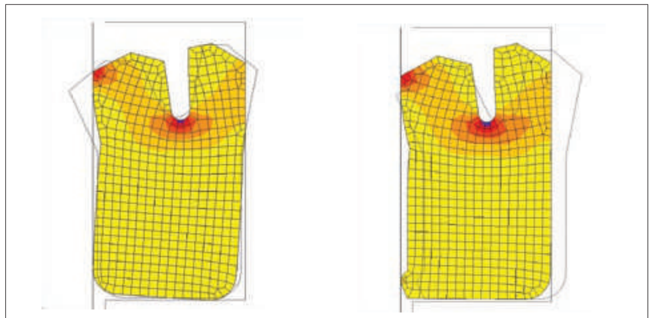


Figure 2-8. Design improvements for increased stability

Surface Speed

The surface speed of a reciprocating shaft can affect the function of a seal. Hydroplaning and frictional heat may occur with excessive speed, while stick-slip, discussed previously in the friction section, is most often associated with slow speed.

Hydroplaning occurs when hydrodynamic forces lift the sealing lip off of the dynamic surface, allowing fluid to bypass the seal. The lip geometry, as well as the overall force on the lip, will influence its ability to resist hydroplaning. Most hydraulic seals are rated for speeds up to 20 inches/second (0.5 m/second), but this may be too fast for certain lip geometries or when the seal has a lightly loaded design. Table 2-1 on page 2-3 shows which lip geometries are subject to hydroplaning. Straight cut and beveled lip geometries are the most effective at resisting hydroplaning so long as sufficient lip loading is present to overcome the hydrodynamic forces.

High surface speeds can create excessive frictional heat. This can create seal problems when the dynamic surface is continuously moving. The under-lip temperature of the seal will become much hotter than the system fluid temperature, especially when the seal is under pressure. If the heat being generated cannot be dissipated, the seal will experience compression set, wear, extrusion and/or increased chemical attack.

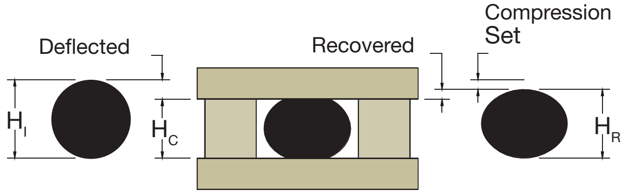
Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

Compression Set

Compression set is the inability of a seal to return to its original shape after being compressed. As defined by ASTM, it is the percent of deflection by which the seal fails to recover after a specific deflection, time and temperature. Compression set is calculated using the following equation:

$$\text{Compression Set} = \frac{H_I - H_R}{H_I - H_C} \text{ X } 100$$

where



H_I = Initial height
H_C = Compressed height
H_R = Recovered height

Compression set reduces sealing forces, resulting in poor low pressure sealability. It takes place primarily because of excessive exposure to a high temperature. A material's upper end temperature limit may give an indication of its compression set resistance. Although compression set always reduces the seal's dimensions, chemical swell or shrinkage can either positively or negatively impact the final geometry of the seal. If material shrinkage occurs due to the system fluid, the deflection of the seal will decrease, accelerating leakage. If chemical swell is present, it can negate or offset the negative effects of compression set. While it is true that swelling can offset compression set, extreme fluid incompatibility can break down the polymer's chemical structure and cause the material to be reformed in its compressed state. (See also page 3-9.)

Lip wear is also a dimensional loss, but is not related to compression set. Dimensional loss due to lip wear will increase the final compression set value.

The seal shown in Figure 2-9 exhibits nearly 100% compression set with minimal wear. Note how the lips flare out very little.

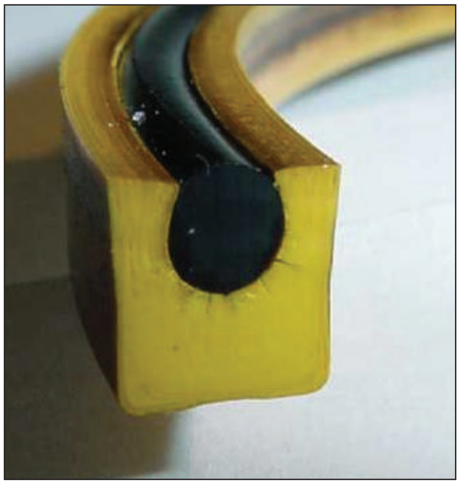


Figure 2-9. Seal exhibiting nearly 100% compression set

Influence of Temperature

All seal materials have a specified operating temperature range (see Section 3, Materials). These temperatures are provided as guidelines and should not be used as specification limits. It is wise practice to stay well within this range, knowing that physical properties are severely degraded as either limit is approached.

Temperature affects extrusion, wear, chemical resistance and compression set, which ultimately influences the sealing ability of a product. High temperatures reduce abrasion resistance, soften materials, allowing them to extrude at lower pressures, increase compression set and can accelerate chemical attack. Low temperatures can cause materials to shrink and harden, reducing resiliency and sealability. Some of these problems can be solved by using low temperature expanders or metal springs as a component of the seal selection (see Section 3, Materials).

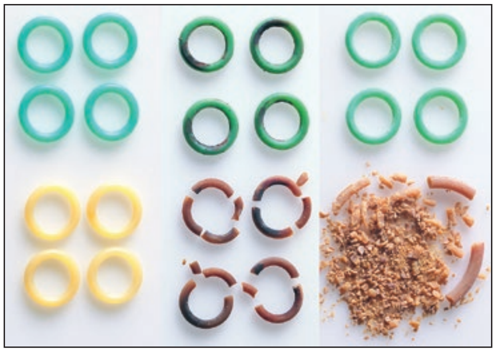


Figure 2-10. Progressive effect (hydrolysis) of high temperature water on standard urethane seals (yellow) vs. Parker Resilon® 4301 polyurethane seals (aqua).

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

General Guidelines for Hardware Design

For easy assembly and to avoid damage to the seal during assembly, Parker recommends that designers adhere to the tolerances, surface finishes, leading edge chamfers and dimensions shown in this catalog.

Table 2-6.

Installation Chamfer, Gland Radius, and Taper		
Seal Cross Section	"A" Dimension	"R" Dimension
1/16	0.035	0.003
3/32	0.050	0.015
1/8	0.050	0.015
5/32	0.070	0.015
3/16	0.080	0.015
7/32	0.080	0.015
1/4	0.080	0.015
9/32	0.085	0.015
5/16	0.085	0.015
11/32	0.085	0.015
3/8	0.090	0.015
13/32	0.095	0.015
7/16	0.105	0.030
15/32	0.110	0.030
1/2	0.120	0.030
17/32	0.125	0.030

Installation Chamfer, Gland Radius, and Taper		
Seal Cross Section	"A" Dimension	"R" Dimension
9/16	0.130	0.030
19/32	0.135	0.040
5/8	0.145	0.040
21/32	0.150	0.040
11/16	0.160	0.040
23/32	0.165	0.040
3/4	0.170	0.040
25/32	0.180	0.060
13/16	0.185	0.060
27/32	0.190	0.060
7/8	0.200	0.080
29/32	0.205	0.080
15/16	0.215	0.080
31/32	0.220	0.080
1	0.225	0.080

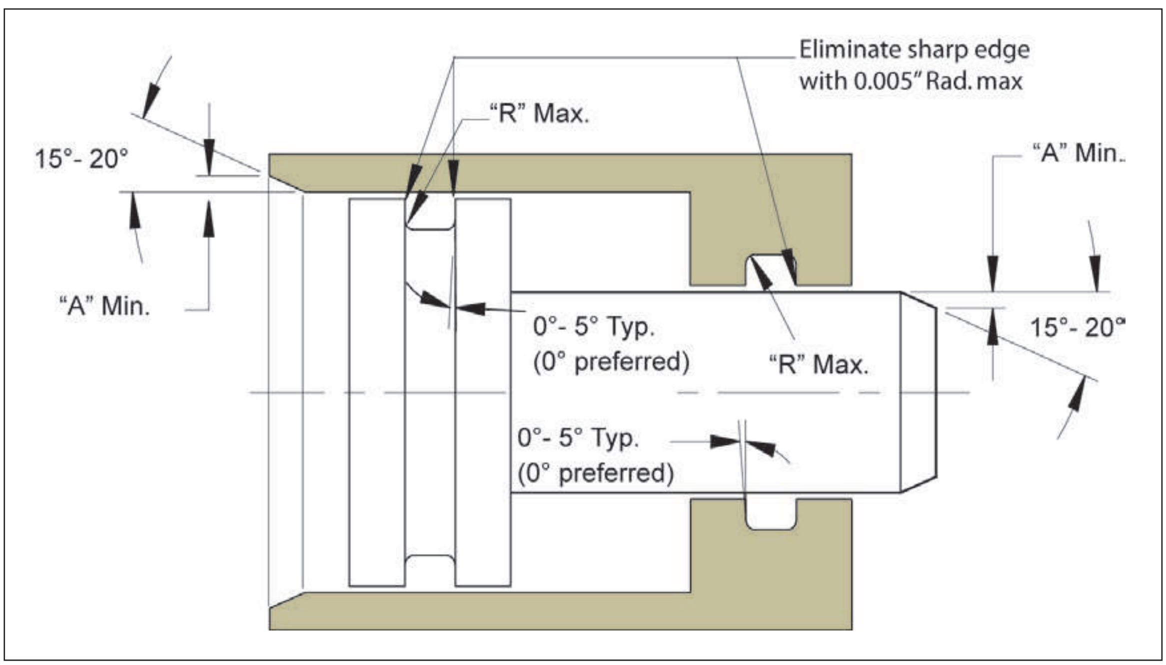


Figure 2-11.

Hardware Surface Finish

Understanding and applying the benefits of appropriate surface finish specifications can dramatically affect the longevity of a sealing system. In a dynamic surface, microscopic variations form recesses which hold an oil film between the seal lip and the moving surface. If the surface is too smooth, friction and seal wear will be high because this oil film will not be present. If the surface is too rough, the variations will create leak paths and accelerate lip wear. For these reasons, it is critical to have an in depth understanding of surface finishes as they pertain to dynamic sealing systems. As such, Parker recommends following the guidelines for surface finish as outlined below or conducting individual testing for specific applications to validate seal function and expected life.

Over the years, greater attention has been given to this subject as realizations about warranty savings and system life become more prevalent. As equipment required to measure and maintain a proper surface finish has evolved and improved, the subject of surface finish has become more complex. Traditional visual inspection gauges are no longer sufficient to effectively measure surface finish. Profilometers are now commonly used to achieve precise measurements with repeatable results. In the same way, the terms used to define a surface finish have also advanced.

For many years, a single surface parameter has often been used to quantify surface finish. RMS (also known as Rq) stands for Root Mean Square and has historically been the most typical value. In more recent years, the Arithmetic Average Roughness, Ra, has become more frequently specified. Using either of these parameters by itself is inadequate to define a proper reciprocating sealing surface. Figure 2-12 depicts why this parameter alone cannot accurately describe a surface finish.

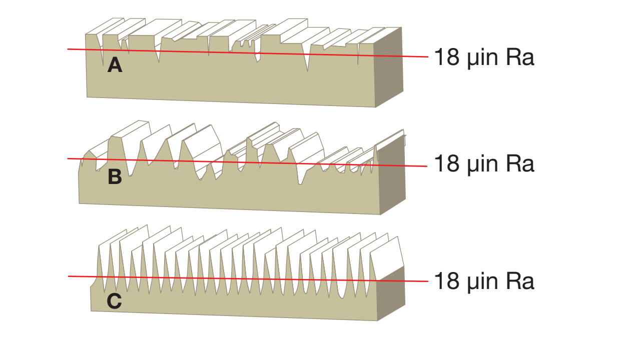


Figure 2-12. Different surface finishes yielding same Ra value

The three surface finishes shown in Figure 2-12 all have the same Ra value but very unique characteristics. The first profile (A) is an example of a proper surface finish for dynamic seals in which the sharp peaks have been minimized or removed. The second profile (B) will exhibit high wear characteristics because of the wide spacing between the peaks. The third profile (C) will also wear out the seals quickly because of its extremely sharp peaks.

RMS = Rq. The Root Mean Square (RMS) as defined by ISO 4287:1997 and other standards is often defined as Rq. These terms are interchangeable.

Rq ≠ Ra. Confusion has typically existed regarding these values, leading to misconceptions that they are interchangeable. Rq and Ra will never be equal on typical surfaces. Another misconception is that there is an approximate 11% difference between the two. Ground and polished surfaces can have Rq values that are 20 to 50 percent higher than Ra. The 11% difference would only occur if the surface being measured took the form of a true sine wave. A series of tests conducted at Parker has shown Rq to be 30% higher than Ra on average.

What's the Significance? Specifications previously based on a maximum surface finish of 16 µin RMS for ground and polished rods should specify a maximum finish of **12 µin Ra.**

Ra is sufficient to define the magnitude of surface roughness, but is insufficient to define a surface entirely in that it only describes the average deviation from the mean line, not the nature of the peaks and valleys in a profile. To obtain an accurate surface description, parameters such as Rp, Rz and Rmr (tp) can be used to define the relative magnitude of the peaks and the spacing between them. These parameters are defined in Table 2-7, and their combination can identify if a surface is too rough or even too smooth for reciprocating applications.

There are other parameters that can be considered for surface finish evaluation. For example, the limitation of Rt is that it considers only one measurement, while Rz, Rp and Rmr consider the full profile.

Table 2-7. Roughness Parameter Descriptions

Parameter Descriptions
Roughness parameters are defined per ISO 4287:1997 and ISO 4288:1996.
Ra* – Arithmetic average or mean deviation from the center line within a sampling length.
Rq* – Root mean square deviation from the center line within a sampling length.
Rp* – Maximum profile peak height within a sampling length. Also known as Rpm in ASME B46.1 – 2002.
Rv* – Maximum profile valley depth within a sampling length. Also known as Rvm in ASME B46.1 – 2002.
Rz* – Maximum height of profile within a sampling length (Rz = Rp + Rv).
NOTE: ISO 4287:1984, which measured five peaks and five valleys within a sampling length, is now obsolete. This value would be much lower because additional shorter peaks and valleys are measured. Over the years there have been several Rz definitions used. Care needs to be taken to identify which is used.
Rt – Maximum height of the profile within the evaluation length. An evaluation length is typically five sampling lengths.
Rmr – Relative material ratio measured at a given height relative to a reference zero line. Indicates the amount of surface contact area at this height. Also known as tp (bearing length ratio) in ASME B46.1 – 2002.
*Parameters are first defined over a sampling length. When multiple sampling lengths are measured, an average value is calculated, resulting in the final value of the parameter. The standard number of sampling lengths per ISO 4287:1997 and ISO 4288:1996 is five.

Figure 2-13 graphically represents Ra. The shaded area, which represents the average height of the profile, Ra, is equal to the area of the hatched portion. The mean line, shown in red, splits the hatched area in half and forms the center line for Ra. The graph also shows Rq, which is higher than Ra.

Figure 2-14 shows the actual surface profile of a polished chrome rod.

Upon examination of the profile, it can be seen that the polishing operation has removed or rounded the peaks producing a positive affect on the characteristics of the sealing surface, as described below by Ra, Rp, Rz and Rmr.

- Ra = 8.9 µin
- Rp = 14.8 µin (which is 1.7 x Ra, less than the 3x guideline)
- Rz = 62.9 µin (which is 7.1 x Ra, less than the 8x guideline)
- Rmr = 74%

Figure 2-14 also illustrates how Rp and Rz are calculated using the following equations:

$$Rp = \frac{Rp1 + Rp2 + Rp3 + Rp4 + Rp5}{5}$$

$$Rz = \frac{Rz1 + Rz2 + Rz3 + Rz4 + Rz5}{5}$$

NOTE: In the profile shown in Figure 2-14, Rt = Rz2 because the tallest peak and deepest valley occur in the same sampling length.

Figure 2-15 considers the same surface and illustrates how the Rmr value of 74% is determined. To accomplish this, locate the height of the curve at 5% material area (this is the reference line or “zero line”). From this height, move down a distance of 25% Rz and locate the new intersection point along the curve. This new intersection point is the actual Rmr value of 74%.

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

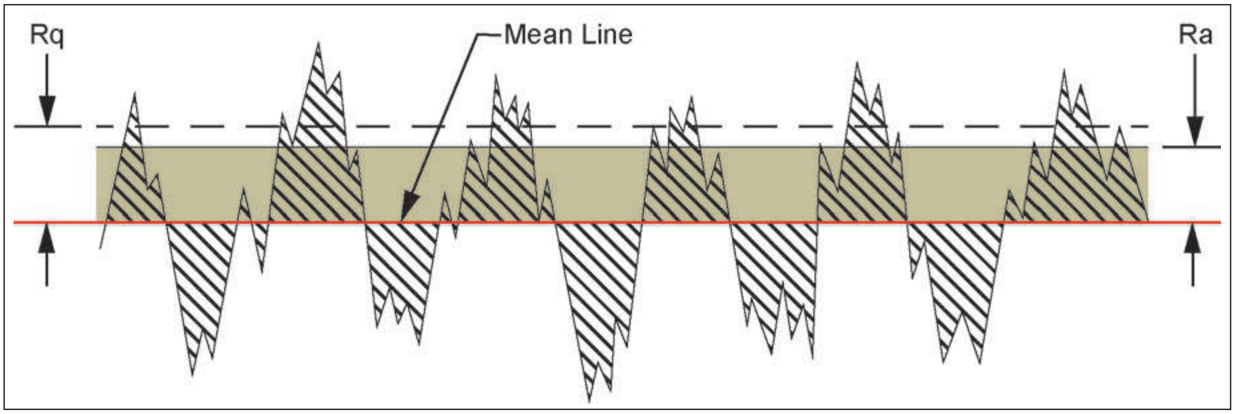


Figure 2-13.

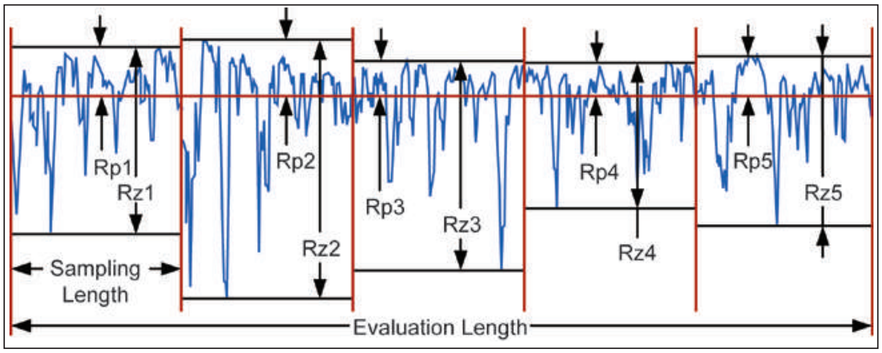


Figure 2-14.

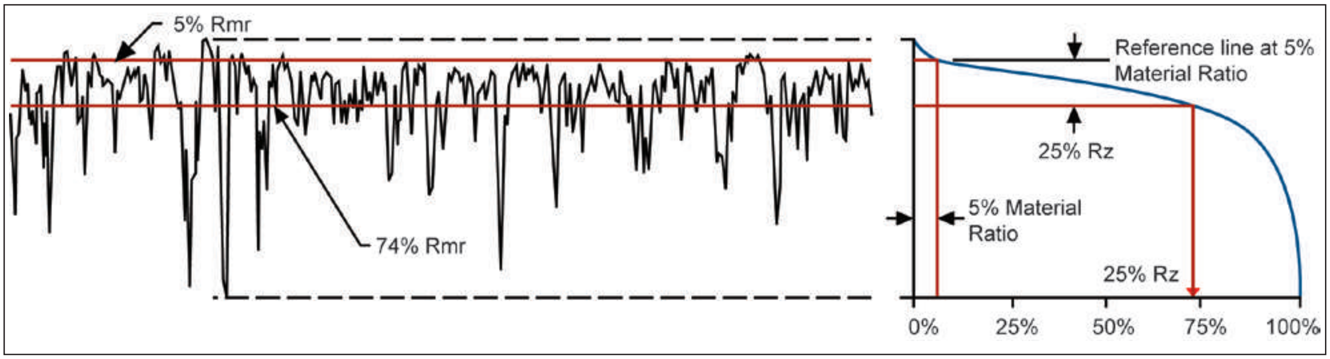


Figure 2-15.

Surface Finish Guidelines for Reciprocating Seals

Recommendations for surface roughness are different for static and dynamic surfaces. Static surfaces, such as seal groove diameters, are generally easier to seal and require less stringent roughness requirements; however, the type of fluid being sealed can affect the guidelines (see Table 2-8). It is important to remember that surface finish recommendations will vary depending upon the seal material of choice. PTFE seals require smoother finishes than seals made from polyurethane and most rubber compounds.

Four parameters have been selected to define a proper surface finish for hydraulic and pneumatic reciprocating applications. These parameters are Ra, Rp, Rz and Rmr. For descriptions of these parameters, please consult Table 2-8.

Grinding as a final process for dynamic sealing surfaces is rarely sufficient. *In order to obtain an acceptable Rmr value, the surface must often be ground* ***and*** *polished. If the surface is not polished in addition to being ground, the ratio of Rp and Rz to Ra will be too high or Rmr ratio too low.*

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

2

Table 2-8. Surface Finish Guidelines

<table>
<tr><th colspan="5">Ra Guidelines</th></tr>
<tr><th rowspan="2">Application</th><th colspan="2">Thermoplastic and Rubber Seals</th><th colspan="2">PTFE Seals</th></tr>
<tr><th>Dynamic Surfaces</th><th>Static Surfaces</th><th>Dynamic Surfaces</th><th>Static Surfaces</th></tr>
<tr><td>Cryogenics</td><td>—</td><td>—</td><td>4 µin (0.1 µm) maximum</td><td>8 µin (0.2 µm) maximum</td></tr>
<tr><td>Helium Gas
Hydrogen Gas
Freon</td><td>3 to 10 µin (0.08 to 0.25 µm)</td><td>12 µin (0.3 µm) maximum</td><td>6 µin (0.15 µm) maximum</td><td>12 µin (0.3 µm) maximum</td></tr>
<tr><td>Air
Nitrogen Gas
Argon
Natural Gas
Fuel (Aircraft and Automotive)</td><td>3 to 12 µin (0.08 to 0.3 µm)</td><td>16 µin (0.4 µm) maximum</td><td>8 µin (0.2 µm) maximum</td><td>16 µin (0.4 µm) maximum</td></tr>
<tr><td>Water
Hydraulic Oil
Crude Oil
Sealants</td><td>3 to 12 µin (0.08 to 0.3 µm)</td><td>32 µin (0.8 µm) maximum</td><td>12 µin (0.3 µm) maximum</td><td>32 µin (0.8 µm) maximum</td></tr>
<tr><th colspan="5">Rp Guidelines</th></tr>
<tr><th rowspan="2">Application</th><th colspan="2">Thermoplastic and Rubber Seals</th><th colspan="2">PTFE Seals</th></tr>
<tr><th>Dynamic Surfaces</th><th>Static Surfaces</th><th>Dynamic Surfaces</th><th>Static Surfaces</th></tr>
<tr><td rowspan="3">All media/fluids</td><td>If Ra ≥ 5 µin (0.13 µm), then Rp ≤ 3 × Ra</td><td rowspan="2">—</td><td>If Ra ≥ 5 µin (0.13 µm), then Rp ≤ 3 × Ra</td><td rowspan="2">—</td></tr>
<tr><td>If Ra < 5 µin (0.13 µm), then Rp ≤ 3.5 × Ra</td><td>If Ra < 5 µin (0.13 µm), then Rp ≤ 3.5 × Ra</td></tr>
<tr><td colspan="4">Example: If Ra = 4 µin, then Rp ≤ 14 µin.</td></tr>
<tr><th colspan="5">Rz Guidelines</th></tr>
<tr><th rowspan="2">Application</th><th colspan="2">Thermoplastic and Rubber Seals</th><th colspan="2">PTFE Seals</th></tr>
<tr><th>Dynamic Surfaces</th><th>Static Surfaces</th><th>Dynamic Surfaces</th><th>Static Surfaces</th></tr>
<tr><td rowspan="3">All media/fluids</td><td>Rz ≤ 8 × Ra and 70 µin (1.8 µm) maximum</td><td>Rz ≤ 6 × Ra</td><td>Rz ≤ 8 × Ra and 64 µin (1.6 µm) maximum</td><td>Rz ≤ 6 × Ra</td></tr>
<tr><td colspan="4">Example: If Ra = 4 µin, then Rz ≤ 32 µin (dynamic calculation)</td></tr>
<tr><td colspan="4">Note: Rz values above maximum recommendations will increase seal wear rate.</td></tr>
<tr><th colspan="5">Rmr Guidelines</th></tr>
<tr><th rowspan="2">Application</th><th colspan="2">Thermoplastic and Rubber Seals</th><th colspan="2">PTFE Seals</th></tr>
<tr><th>Dynamic Surfaces</th><th>Static Surfaces</th><th>Dynamic Surfaces</th><th>Static Surfaces</th></tr>
<tr><td rowspan="3">All media/fluids</td><td>45% to 70% (thermoplastic)</td><td rowspan="2">—</td><td rowspan="2">60% to 90%</td><td rowspan="2">—</td></tr>
<tr><td>55% to 85% (rubber materials)</td></tr>
<tr><td colspan="4">Rmr is measured at a depth of 25% of the Rz value based upon a reference level (zero line) at 5% material/bearing area.</td></tr>
</table>

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

Surface Finish FAQs

What is the difference between RMS (Rq) and Ra?

RMS which stands for Root Mean Square (and now known as Rq), is one way of quantifying the average height of a surface. The Arithmetic Average, Ra, quantifies the surface in a different manner, providing a true mean value. These parameters will almost always be different, but there is not an exact relationship between the two for a typical sealing surface of random peaks and valleys. If a surface were to perfectly resemble a sine wave, the result would place the RMS value 11% higher than Ra, but this is not a very realistic scenario. On various ground and polished surfaces, RMS has been observed to be as much as 50% higher than Ra, but on average, runs about 30% higher. If this 30% average difference is applied to a 16 µin RMS specification, the maximum recommended value would be 12 µin Ra.

Why are Rp and Rz specified as a function of Ra, and not simply a range?

Take a shaft with the minimum recommended value of Ra = 3 µin, for example. Using the formula for Rz, the maximum value would be calculated as 24 µin (8 x 3). If the requirement simply stated a range that allowed Rz values up to 70 µin, this large difference indicates that the surface profile could have many large, thin surface peaks which would abrade the seal quickly. By the same regard, a maximum Ra value of 12 µin would result in an Rz value of 96 µin (12 x 8), which is beyond the recommended maximum value of 70 µin. The same principle applies for Rp: peaks should be removed to reduce seal wear via a polishing process. Grinding without polishing can leave many abrasive surface peaks.

Why is Ry (also known as Rmax) not used in Parker's roughness specification?

Ry only provides a single measurement (a vertical distance from one peak to valley) within the whole evaluation length. In actuality, there may be several peaks and valleys of similar height, or there may only be one large peak or valley. Rp and Rz provide much more accurate results, showing the average of five peak to valley measurements (one measurement in each of the five sampling lengths). Furthermore, ISO 4287:1997 and ISO 4288:1996 standards no longer incorporate the use of Ry.

How can a dynamic surface finish be too smooth?

There are two areas of concern that have been observed on extremely smooth surfaces, the first being seal wear, the second being leakage. When surface finishes have been measured at or below 1 µin Ra, an extremely accelerated seal wear rate has been observed. A small jump to 1.8 to 2 µin Ra shows significant improvement, indicating that the extremely low range should be avoided. With higher values showing even greater life extension, the optimal range for Ra has been determined to be 3 to 12 µin.

Regarding leakage, some seal designs that function well with 6 to 12 µin Ra finishes begin to leak when the finish falls below 3 µin Ra. Due to technological advances, there are many suppliers who manufacture rods with finishes this smooth. It is always necessary to validate seal performance, especially if using an ultra-smooth dynamic surface.

When does a dynamic surface finish become too rough?

Although it is possible for some seals to function when running on rough finishes, there are always concerns with accelerated wear and leakage control. Certain seals have been able to function at 120 µin Ra finishes for short periods of time, but seal life in these cases can be reduced up to five or six times. On the contrary, some seals have failed at surface finishes as low as 16 µin Ra when pressure was insufficient to effectively energize the sealing lips as they rapidly wore out. Even though a rough finish is not a guaranteed failure mode, it is always best to stay within the recommended specifications. Remember that a proper finish also meets the recommendations for Rp, Rz and Rmr listed in the surface roughness guidelines.

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

Installation

Considerations

Installation techniques may vary considerably from case to case, depending on whether a seal is being replaced as a maintenance procedure or being installed in the original manufacture of reciprocating assemblies. Variations also arise from differences in gland design. A two-piece, split gland design, although rarely used, poses fewer problems than a "snap-in" groove positioned deep inside the body of a long rod gland. In production situations, or where frequent maintenance of similar or identical assemblies is performed, it is customary to utilize special tools to permit fitting a seal into its groove without overstressing it or subjecting it to nicks and cuts during insertion.

The common issues associated with all installation procedures are:

1. **Cleanliness.** The seal and the hardware it must traverse on its way into the groove, as well as the tools used to install the seal, must be cleaned and wiped with lint-free cloths.

2. **Nick and Cut Protection.** Threads, sharp corners and burrs can damage the seal. Care should be taken to avoid contact with these surfaces. Burrs must be removed, sharp corners should be blunted or radiused, and threads should be masked or shielded with special insertion tooling (see Figure 2-16). Although it is good practice to take extra care in the handling and manipulation of the seal, this is seldom sufficient and it usually requires either a safety tool or masking to protect the seal against such damage.

3. **Lubrication.** Both the seal and its installation path must be lubricated prior to insertion. The lubricant should be selected for its compatibility with the seal compound and the working fluid it will later encounter. Often, the working fluid itself can be used as the lubricant (see Table 2-9).

Table 2-9. Seal Installation Lubricants

Type	Temp. Range °F (°C)	Seal Use	Seal Material Compatibility
Petroleum base (Parker O Lube)	-20 to +180 (-29 to +82)	Hydrocarbon fluids; Pneumatic systems under 200 psi	Molythane®, Resilon®, Polymyte®, Nitroxile®, HNBR, NBR, FKM, (DO NOT use with EPR)
Silicone grease or oil (Parker Super O Lube)	-65 to +400 (-54 to +204)	General purpose; High pressure pneumatic	Molythane, Resilon, Polymyte, Nitroxile, HNBR, NBR, EPR, FKM
Barium grease	-20 to +300 (-29 to +149)	Pneumatic systems under 200 psi	Molythane, Resilon, Polymyte, Nitroxile, HNBR, NBR, FKM
Fluorocarbon fluid	-65 to +400 (-54 to +204)	Oxygen service	EPR

4. **Lead-in Chamfer.** A generous lead-in chamfer will act as a guide to aid in seal installation. With the proper lead-in chamfer, the seals can be installed without lip damage. Refer to Figure 2-17 below and Table 2-6 on page 2-8 for proper lead-in chamfer dimensions.

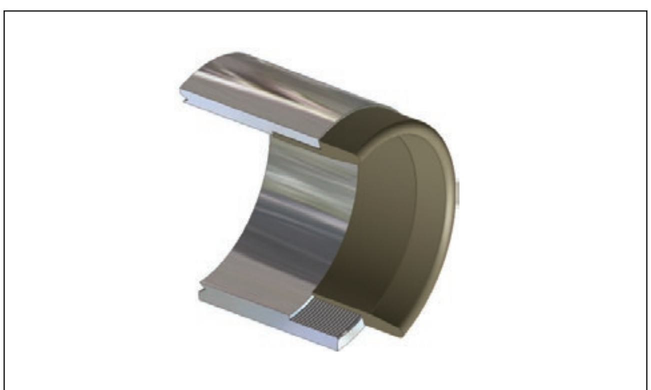


Figure 2-16. Thread protection installation tool cutaway view

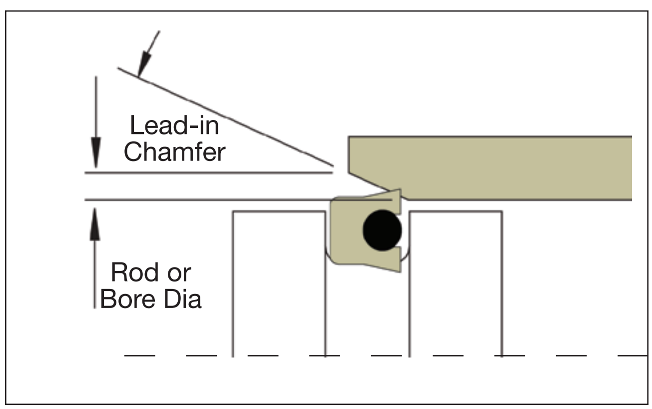


Figure 2-17. Seal installation lead-in chamfer

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

5. **Heating.** Where harder or fabric-reinforced compounds are used in snap-in applications, elasticity of the seal may fall short of that required for stretching or compressing onto (or into) the groove. Since seal compounds characteristically exhibit a high thermal coefficient of expansion, and tend to soften somewhat when heated, it is sometimes possible to "soak" the seals in hot lubricant to aid installation. Be sure to observe the compound temperature limits, and avoid heating the seals while stretched. Heating a seal while stretched will invoke the Gow-Joule effect and actually shrink the seal.

6. **Cross Section vs. Diameter.** Care must be taken to properly match a seal's cross-section to its diameter. If the cross-section is too large in relation to the diameter, it will be difficult to snap-in or stretch the seal into the groove. This condition is typically only associated with polyurethane, Polymyte® and other high modulus materials. The data shown in Table 2-10 may be used as a guide to determine this relationship for ease of installation.

Table 2-10. Seal Cross Section vs. Diameter Installation Guide

Installation Guide Cross Section vs. Diameter				
Cross Section	Minimum Diameter Rod Seal		Minimum Diameter Piston Seal	
	Poly-urethane	Polymyte	Poly-urethane	Polymyte
1/8"	.750 I.D.	1.000 I.D.	1.250 I.D.	1.750 I.D.
3/16"	1.000 I.D.	1.750 I.D.	1.750 I.D.	2.750 I.D.
1/4"	1.750 I.D.	2.750 I.D.	3.000 I.D.	4.500 I.D.
3/8"	3.000 I.D.	5.000 I.D.	6.000 I.D.	8.000 I.D.
1/2"	6.000 I.D.	8.000 I.D.	10.000 I.D.	12.000 I.D.
3/4"	8.000 I.D.	9.000 I.D.	15.000 I.D.	17.000 I.D.
1"	10.000 I.D.	10.000 I.D.	20.000 I.D.	25.000 I.D.

7. **Installation Tools.** Use installation tools as recommended (see pages 2-16 and 2-17).

8. **Itemize and Use a Check List.** All components required to complete a sealing assembly should be itemized and checked off as they are installed. The absence of any single component can cause the entire system to fail.

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

Installation Tools — Piston Seals

The installation of piston seals can be greatly improved with the use of installation tooling. Tooling not only makes the installation easier, but also safer and cost effective for high volumes as seals are less likely to be damaged when using proper tooling. For piston seal installation using tooling, use the following steps:

1. Inspect all hardware and tooling for any contamination, burrs or sharp edges. Clean, debur, chamfer, or radius where necessary. Make sure the piston and groove are undamaged.
2. If using a two-piece energized cap seal, install the o-ring or rubber energizer into the groove per vendor specifications.
3. Install the expanding mandrel onto the piston (Figure 2-18).
4. Light lubrication and/or warming (+140°F max) may aide installation. Use system compatible lubricant only.
5. Place the seal onto the expanding mandrel, and using hand pressure or a pusher, if necessary, gently push the seal along the taper until it snaps into place (Figure 2-19).
6. If back-up rings are to be used, install split versions into their proper location or use the mandrel method in Step 5 for non-split rings.
7. For PTFE cap seals, slide the resizing tool over the seal to compress the seal to its original diameter (Figures 2-20, 2-21).

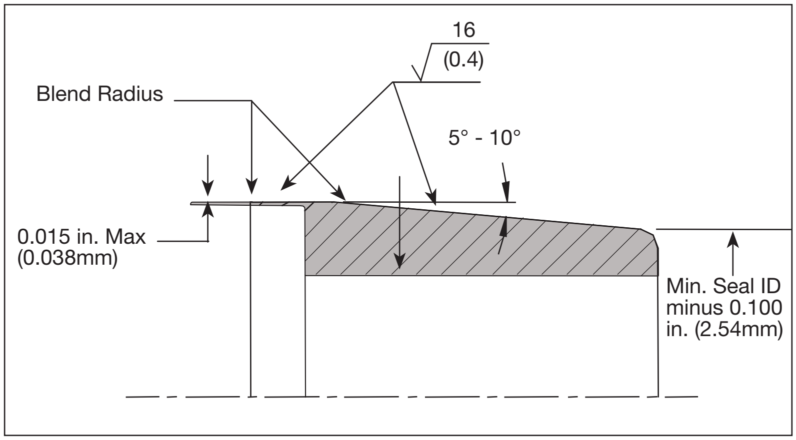


Figure 2-18. Expanding mandrel

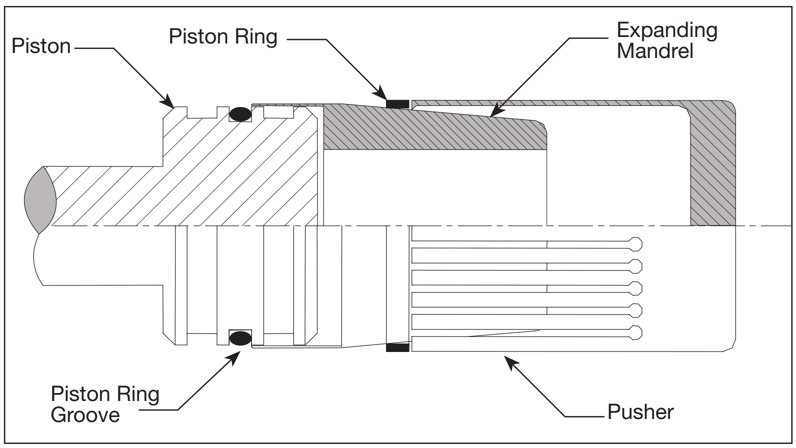


Figure 2-19. Installation of piston seal with tooling

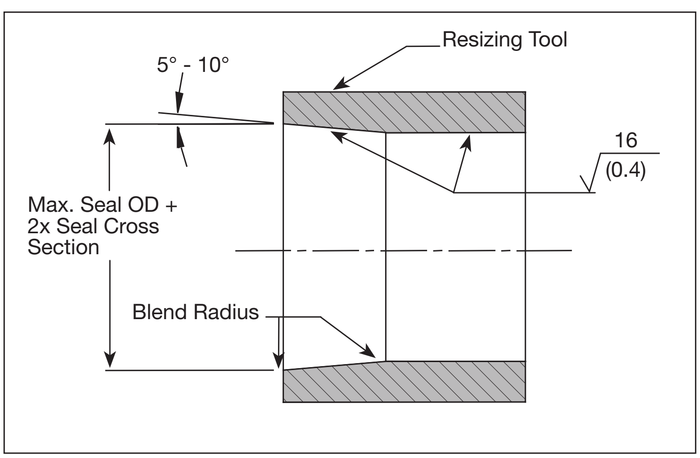


Figure 2-20. Resizing tool

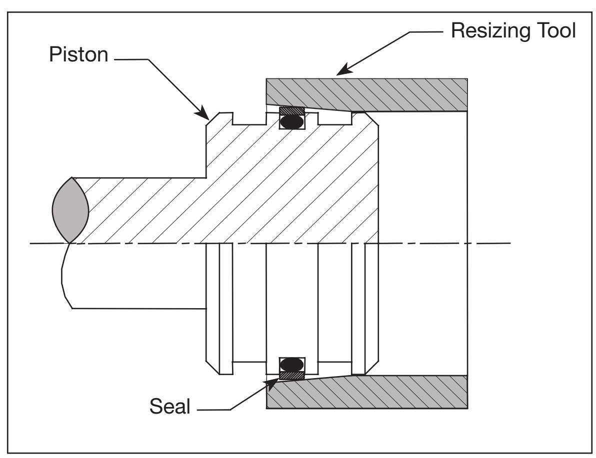


Figure 2-21. Resizing

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000

Installation Tools — Rod Seals

Many rubber, plastic and PTFE rod seals can be manipulated by hand for installation into the seal groove. Small diameter parts or parts with large cross sections may require a two piece (split) groove for installation. Special tooling can be utilized to help the installation process; however, PTFE and Polymyte® seals in particular require caution to ensure the sealing component is not nicked, dented or damaged. The following guidelines provide the steps for proper rod seal installation. If needed, please call your local Parker representative for recommendations.

1. Inspect all hardware and tooling for any contamination, burrs or sharp edges. Clean, debur, chamfer or radius where necessary. Make sure the bore, groove and rod are undamaged.

2. If using a two-piece, energized cap seal, first carefully install the o-ring or rubber energizer into the groove to ensure proper seating.

3. By hand, gently fold the seal into a kidney shape (Figure 2-22) and install into the groove. For rubber and polyurethane seals, the use of a three-prong installation tool can be helpful for folding the seal and installing it into the groove (Figure 2-23).

4. Unfold the seal into the groove, and using your finger, feel the inside diameter of the seal to make sure it is properly seated.

5. For PTFE seals, after unfolding the seal in the groove, use a resizing tool (Figure 2-24) to re-expand the seal.

6. If a back-up ring is to be used with the rod seal, position the seal toward the internal side of the groove to allow space for the back-up ring installation.

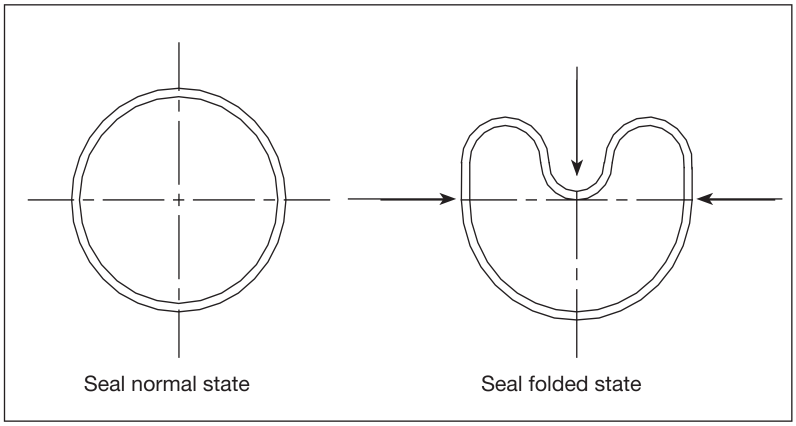


Figure 2-22. Rod seal folding

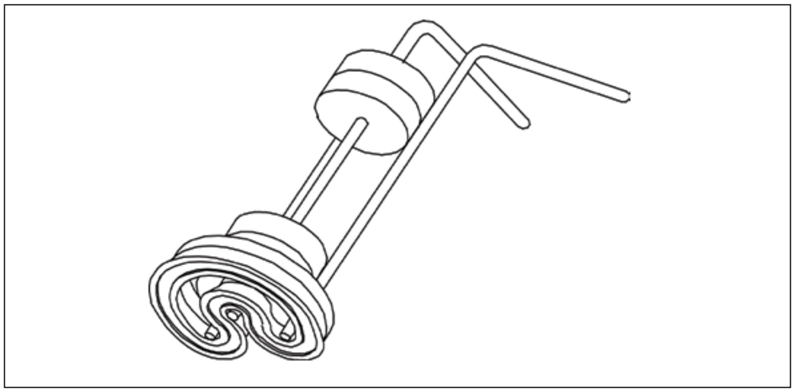


Figure 2-23. Three-leg installation tool for polyurethane and rubber seals

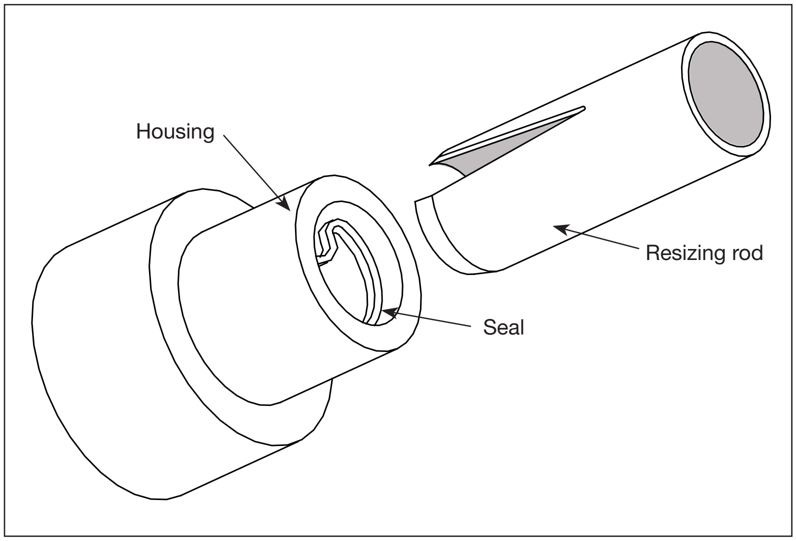


Figure 2-24. Rod seal installation

Click to Go to CATALOG Table of Contents
Click to Go to SECTION Table of Contents

Finite Element Analysis

Finite Element Analysis (FEA) is a powerful computer simulation tool that allows engineers to evaluate product designs and materials and to consider "what if" scenarios in the development phase. FEA helps minimize time and cost by optimizing a design early in the process, reducing pre-production tooling and testing. Within the simulation program, the product being evaluated is divided into "finite elements," and model parameters such as pressure and seal lip squeeze are defined. The program then repeatedly solves equilibrium equations for each element, creating an overall picture of seal deformation, stress and contact forces (see Figure 2-25). These results can then be linked to application testing to predict performance.

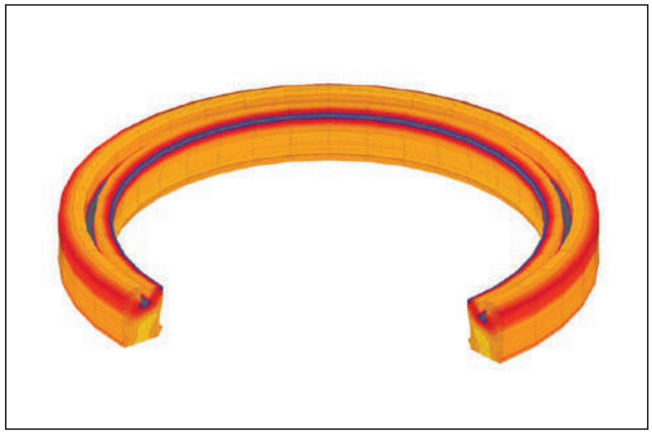


Figure 2-25.

Precise material characterization is an essential component of accurately modeling elastomeric products with FEA. Due to the complex nature of elastomers, multiple tests must be performed in order to determine their behavior under stress and strain. Figure 2-26 shows the typical nonlinear stress-strain curves for elastomers compared to the linear property of steel. These nonlinear complexities make performing FEA for elastomers much more difficult than for metal materials. Advances in material characterization are continually being made to improve the ability to capture and predict thermoviscoelastic effects of elastomers.

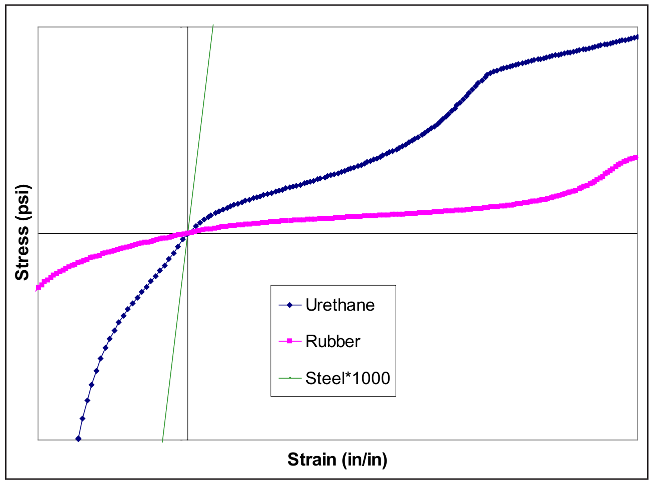


Figure 2-26. Stress/Strain relationship of steel vs. elastomers

FEA results must be linked with lab and field testing to create a baseline to predict seal performance. Once this baseline is established, design iterations can be performed within FEA until the desired results are achieved and an optimum design is predicted. This evaluation process enables engineers to anticipate the performance of new seal designs by minimizing the time and cost associated with prototype tooling investments (see Figure 2-27).

Like any computer simulation, FEA has its limitations. The cost of performing FEA should always be justified by its results. FEA can provide relative information on leakage performance and wear life, but cannot give concrete answers to questions like, "Will this seal leak, and if so, how much?" and "How many cycles can be expected before failure occurs?"

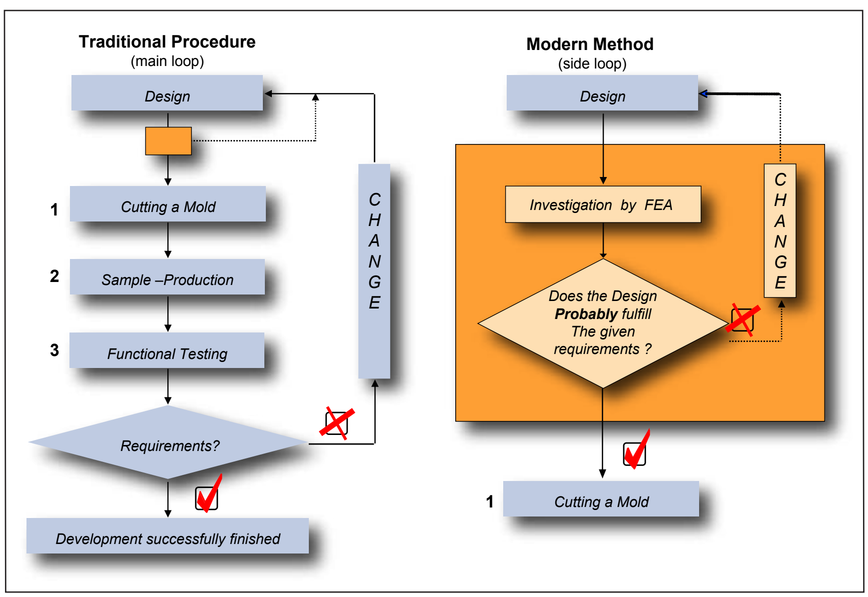


Figure 2-27. Traditional process vs. modern seal development process using FEA

Parker Hannifin Corporation
Engineered Polymer Systems Division
Phone: 801 972 3000