Rotary Shaft Seals

What Is the Purpose of a Seal?

Today there is a wide selection of designs available for use in rotary applications. They range from the traditional single and double lip elastomeric configurations to PTFE-based designs. Even more complex designs incorporate multiple lips, differing materials and hybrid labyrinth designs. The purpose of this reference guide is to assist engineers and maintenance professionals in selecting the best design for a specific application based on service life requirements and cost objectives.

One of the most common purposes of a lip seal is to protect the bearing that is used to support a shaft in a rotating application. Retaining the bearing lubricant and keeping it clean ensures maximum bearing life and increases the overall service life of the equipment. Such applications include automotive wheels, electric motors, pumps, gearboxes and large rolls used in steel and paper manufacturing.

Radial lip seals are used throughout industries in a variety of other applications under a wide range of operating conditions. These conditions can vary from high-speed shaft rotation with light oil mist to low speed reciprocating shaft in muddy environments. Radial lip seals can be found sealing lube oil in high speed crankshaft applications for gasoline and diesel engines that operate from the tropics to the arctic, in submarines, oil tankers, spacecraft, windmills, steel mills, paper mills, refineries, farm tractors, appliances and automobiles. In fact, they can be found in anything that has a rotating shaft.
In rotating applications, a seal can also impact the service life of indirect components such as mechanical seals, couplings, pulleys or other in-line coupled equipment. If a seal allows the system lubricant to run below safe levels or allows foreign material to enter the bearing cavity, the bearing will soon begin to show signs of failure. As the bearing fails, vibration from excessive shaft runout will be transferred to all other in-line components and will shorten their service life as well.

The advantages of radial lip seals include: low cost, small space requirements, easy installation and an ability to seal a wide variety of applications.

In review, the primary purpose of the radial lip seal is to retain lubricants within a sump or cavity. The secondary purpose is to exclude contaminants from the system lubricant. Lip seals are also used to separate two different fluids, retain internal pressure or exclude an external pressure.

**History of Shaft Seals**

The earliest seals were rags and pieces of leather straps tied at the end of cart wheel axles to retain the animal fat or olive oil used at that time for lubrication. This slowly evolved to more complex sealing systems and lubricants, such as grease made with olive oil and lime.

The Industrial Revolution accelerated sealing innovations with bores in the wheel hubs to hold packings and ropes to seal rotating shafts. Higher shaft speeds increased operating temperatures and the development of thinner lubricants demanded constant improvements in seal design. This brought along better braided ropes made by specialists using different impregnations such as waxes and pitch.

In the 1930s, seals with beveled leather washers crimped in metal cases were produced. These assembled seals did not require adjustments and were easy to install and fit in much less space than the packings and stuffing boxes previously used. Leather inserts with taller flexible lips were also used because they were better able to follow the wobble of the shafts.

Springs were added to the leather lip seal in the 1940s. Leather was treated to reduce the seepage of lubricants through the sealing elements, but even with different coatings and impregnations, leather could only work slightly above the boiling point of water, so a better material was needed. The new material became synthetic rubber and was introduced as a lip material during World War II. During the war, copper coating and later chemical coats were used to bond the rubber to metal washers that were assembled in metal cases.

In the 1960s, the bonding became reliable enough that rubber lips were molded directly to the outer case. This eliminated possible leakage from between the assembled components. This was due to the components becoming loose from compression set of the rubber or distortion of the components from assembly into the bore. Leather also remained a common lip material throughout the 1970s.

Today, assembled seals made with leather or rubber are no longer recommended because of their high cost, internal leakage, and lack of dimensional control. Most manufacturers have converted small diameter seals to the bonded design; however, the need to use advanced materials such as thermoplastics (primarily PTFE) that can be difficult to bond to a metal case may still require an assembled case design. Large diameter seals have been much slower to move away from the antiquated assembled design, so extra care should be used when sourcing seals for large diameter applications.
How Do They Work?

Rotary shaft seals work by squeezing and maintaining the lubricant in a thin layer between the lip and shaft. Sealing is further aided by the hydrodynamic action caused by the rotating shaft, which creates a slight pumping action.

Rotary shaft seals provide protection by performing two critical functions. In most applications the primary function of the seal is to retain the bearing or system lubricant. There are thousands of different types of lubricants available today, but in general bearings are either oil or grease lubricated.

The second function of the seal is to exclude outboard material that can contaminate the system lubricant or directly damage the bearing. The type of contamination the seal will need to exclude is dependent on the application. The more common types are moisture and water, and dry materials including dust, sand, dirt or particulates such as those generated by manufacturing processes.

Typical petroleum oil has a useful life of thirty years at 86 °F (30 °C) if it is not contaminated with water or particulate matter, but the same oil has a life of only a month at 212 °F (100 °C). As little as 0.002% water in oil lubrication can reduce ball bearing life by 50%, primarily through hydrogen embrittlement. Solid particles cause more rapid damage to the bearing race through high-localized stresses and increased frictional heat.

The sliding contact between the seal lip and the shaft will generate friction, increasing the contact temperature beyond the temperature caused by the bearings and other sources. Heat accelerates the breakdown of the oil and starts forming a varnish on the hot spots. Over time, the varnish changes to carbon and builds in thickness as the surrounding oil loses its lubricity. How quickly this happens is dependent on temperature. The deposit can lift and abrade the lip, causing leakage. The time to reach each stage is cut in half for each 18 °F (10 °C) increase in temperature. The heat also accelerates the cure of the rubber, especially at the contact surface between the seal lip and the rotating shaft. Eventually the lip surface hardens, small cracks form and the surrounding rubber stiffens. The cracks get larger and the lip stiffer, until it can no longer follow the movement of the shaft or seal. In order to maximize seal life, it is critical to minimize the amount of frictional heat of the application.

The amount of frictional heat that is generated is a combination of many operating parameters. Shaft surface, internal pressure, operating speed, lubricant type, lubricant level, lip geometry and lip material are just a few of the conditions that need to be considered. It is important to note that these conditions are very interactive. For example, an increase in shaft speed will increase the sump temperature. If not vented, the temperature rise will increase the pressure inside the housing. The internal pressure will push on the seal lip and create additional force between the seal lip and the shaft. In turn, the operating temperature under the seal lip will see a significant rise in temperature and can cause premature seal failure within hours.

Figure 2-1. Rotary Shaft Seal at Work

The seal’s ability to retain the system lubricant and exclude contaminants plays a key role in the service life of equipment components such as bearings, gears and any other component that relies on the system lubricant. The seal can have a dramatic impact on the service life of the system lubricant by retaining the optimal level, reducing exposure to excessive frictional heat and excluding foreign matter.
It is easy to see why an understanding of rotating shaft seals is critical when trying to reduce the mean time between failure of rotating equipment. To better understand how rotary lip seals work, knowledge of basic seal components is needed.

**Seal Components**

Typical rotary shaft seal components include a rigid outer component and a flexible inner lip (see Figure 2-2). The seal lip can be springless or spring-loaded.

![Seal Components Diagram](image)

**Figure 2-2. Seal Components**

The outer rigid material can range from carbon steel, aluminum and stainless steel to a nonmetallic composite as pictured above. The purpose of the outer component is to position and retain the seal in the housing. The seal’s outer component must also be able to maintain a leak-free fit between the seal and the housing.

The outside diameter of the seal is larger than the seal housing to create a press fit. The actual seal diameter will depend on the size and material of the seal, the size and material of the housing and expected internal pressure and temperature. For general industry standards on OD press fit, see Tables 6-2 and 6-3 on Page 6-3.

The seal element is attached to the outer rigid material by bonding it as it is cured in a molding press or mechanically crimping a cured element between metal components. Designs that use high performance composite materials for the rigid outer section provide the advantages of a one-piece molded construction. One-piece molded designs and bonded designs should be used whenever possible. Assembled designs (small or large diameter) are easily damaged during handling and installation, causing the assembled components to loosen. This creates leak paths between the various components.

The sealing lip configuration will vary based on the type of service, speed, pressure and dynamic runout for which the seal is designed. The seal geometry may also include hydrodynamic pumping features which are normally molded into the lip element on its air side. Common hydrodynamic patterns are triangular and helical. They function by pumping oil that has passed by the primary lip back under the lip to reduce leakage, extending seal life. Refer to Section 4 for lip profile options.

The oil side of the seal lip has an angle in the range of 35 to 55 degrees. The air side has a much shallower angle and is typically 15 to 30 degrees. These angles determine the contact footprint of the lip on the shaft. Incorrect angles will form a footprint that cannot maintain a seal with the shaft and explains why heavy leakage occurs if a lip seal is installed backwards, or with the steep lip angle facing away from the oil side.
This also means that the primary function of single spring-loaded designs is dependent on the installation direction. While the seal will perform both retention and exclusion functions, they are not performed equally.

If the primary function is retention, the seal should be installed with the steep lip angle facing towards the lubricant. This is normally the open-faced side. If the primary function of the seal is to exclude, the steep angle needs to face toward the contaminant (see Figure 2-3).

Several spring types are used to energize the lip. The most common is a wound spring, often referred to as a garter spring. Finger springs are another option, although their loading is typically less uniform and they can be subject to severe distortion prior to or during installation, leading to areas of the lip that are not properly loaded. Other spring types used are cantilever, canted-coil and helical which are normally used in PTFE designs. In order for the spring to maintain the proper load over the life of the seal, the spring must be compatible with the fluids and the temperature of the application.

The dimensional relationship between the center of the spring and the lip contact point is called the R value. The leading edge of the lip should be toward the oil side, with the centerline of the spring slightly toward the air side. If the centerline of the spring is too far toward the air side (too positive R value) it will put too much of the lip (wide footprint) in contact with the shaft and cause excessive wear. A spring position that is too close to the lip contact point (negative R value) can cause the lip to become unstable or roll and dump the spring.

A spring-energized lip is required for positive oil retention, but not typically for grease retention (see Figure 2-4).

The rotary shaft seal is only one component in the sealing system. There are several key operating parameters that can work in unison to optimize seal life, or conversely, if misapplied, can reduce seal life to a few operational hours.

Figure 2-3. Installations Facing Lubricant and Contaminant

If both retention and exclusion are critical and the level of contaminants is heavy, one seal should be used to retain the lubricant, and exclusion capacity should be added using another lip seal, auxiliary excluders or by upgrading to a bearing isolator (see Page 2-27).

The purpose of the spring is to provide a constant, uniform load of the lip on the shaft for the life of the seal. The spring keeps the seal lip in contact with the shaft during higher shaft speeds and also overcomes compression set and wear of the lip material. Compression set of the lip material is normal as it is subjected to thermal cycles during operation.

Figure 2-4. Oil and Grease Seals
Lubricant Considerations

The contact lip is designed to run on a thin film of oil. Without the oil film, the seal lip will run directly on the rotating shaft and generate excessive friction and fail within hours. The lubricant selected needs to remain viable over the expected service life. If the underlip temperature exceeds the lubricant rating, carbonization of the oil will occur.

Abrasive carbonized oil particles will build up at the seal lip and accelerate lip and shaft wear. As the oil film becomes less than optimal, the lip friction increases, as does lip wear.

When selecting a lubricant keep the following in mind:
1. Do temperature limits of the lubricant match the underlip operating temperature of the seal?
2. Are the base oil and additives compatible with the lip material?
3. Does the oil level provide adequate lubrication and cooling at the seal lip?

Shaft Considerations

A proper shaft finish provides small pockets to hold the needed oil film between the lip and shaft, preventing direct contact that would otherwise cause friction and wear as the shaft rotates. The shaft surface must also be smooth enough to avoid peaks that are large enough to break through the lubrication film.

The optimal surface for elastomeric shaft seals is a plunge ground finish of 8 to 17 µin Ra (0.20 to 0.43 µm Ra) (0.010" [0.25 mm] cutoff) with a lead angle below 0.05 degrees. (See Table 2-5 on Page 2-21 for shaft finish requirements for PTFE seals.)

Recent studies show that the Ra measurement alone is insufficient to quantify a proper surface. The surfaces below have the same Ra finish, but the impact on seal performance will vary.

Two additional requirements are needed: Rz (the average peak to valley height) of 65 to 115 µin (1.65 to 2.90 µm), and RPM of 20 to 50 µin (0.5 to 1.25 µm), the average peak to mean height. For additional information, refer to Rubber Manufacturers Association Technical Bulletin OS-1-1.

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Figure 2-5. Sealing System

Table 2-5
When a shaft is turned to size, a continuous spiral groove is imparted on the shaft as the cutting bit traverses the shaft. This is called shaft lead.

If not removed by plunge grinding or other methods, the groove will act as an auger when the shaft rotates. The underlying groove will either pump oil past the seal lip or contaminants into the bearing housing, depending on the direction of the shaft rotation.

If a shaft is going to be plated, the machine lead must still be removed prior to the plating process.

**Testing for Machine Lead**

When lead is suspect and there is a need for verification in the field, the following field test can be performed:

1. Mount the shaft in a chuck and verify the shaft is level.
2. Lightly coat the shaft with silicone oil with a viscosity of 5 to 10 cps.
3. Drape a thread (unwaxed quilting thread 0.009 inches or 0.23 mm dia.) weighted with a one-ounce (30 g) weight around the shaft and tie the ends together so it is long enough to contact about 2/3 of the circumference of the shaft with the weight hanging. Position the thread so that the knot is not touching the shaft.
4. Rotate the shaft at slow speed, 60 RPM.

5. Place thread at both ends as well as center of shaft and observe for axial movement of the thread under BOTH CW and CCW rotation.
   - Movement of the thread in opposite directions, CW versus CCW rotation, indicates lead is present.
   - If the thread moves in the same direction under both CW and CCW rotation, verify that the shaft is level.
   - If the thread remains stationary when checking the ends and center of shaft under both CW and CCW rotation, significant lead is not present.

**Figure 2-6. Shaft Lead Testing**

Please note that this method does not guarantee the absence of lead as some patterns may go undetected using the string test. However, this simple test has been very successful in detecting if a significant lead is present.

The preferred material for the shaft-sealing surface is carbon steel (SAE 1035 or 1045) with a minimum hardness of Rockwell C30 (30 Rc). When heavy amounts of abrasive contamination are present, abrasive additives are used in the lip compound or high-pressure seal designs are going to be used, a minimum shaft hardness of 45 Rc is recommended to resist excessive shaft grooving. Softer materials such as bronze, aluminum or plastic will experience heavy wear (grooving), and should be avoided.
Shaft Tolerances

Shaft diameters should be held to the tolerances specified below:

<table>
<thead>
<tr>
<th>Table 2-1. Shaft Tolerance for Inch/Fractional</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft Diameter</td>
</tr>
<tr>
<td>Up to 4.000&quot;</td>
</tr>
<tr>
<td>4.001 – 6.000&quot;</td>
</tr>
<tr>
<td>6.001 – 10.000&quot;</td>
</tr>
<tr>
<td>Over 10.000&quot;</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 2-2. Shaft Tolerance for Metric*</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft Diameter</td>
</tr>
<tr>
<td>Up to 10 mm</td>
</tr>
<tr>
<td>Over 10 – 18</td>
</tr>
<tr>
<td>Over 18 – 30</td>
</tr>
<tr>
<td>Over 30 – 50</td>
</tr>
<tr>
<td>Over 50 – 80</td>
</tr>
<tr>
<td>Over 80 – 120</td>
</tr>
<tr>
<td>Over 120 – 180</td>
</tr>
<tr>
<td>Over 180 – 250</td>
</tr>
<tr>
<td>Over 250 – 315</td>
</tr>
<tr>
<td>Over 315 – 400</td>
</tr>
<tr>
<td>Over 400 – 500</td>
</tr>
</tbody>
</table>

*ISO Standard 286-2, h11

The leading edge of the shaft should have a burr-free chamfer to ease installation by preventing lip roll-back, spring dumping and damage (nicks or cuts) to the seal lip. Both ends of the chamfer should be free of sharp edges. Spring dumping can occur during seal installation when the lip rolls back on itself, causing it to fall out of the spring pocket. Heavy shock loads that can occur when installing a metal cased seal using a direct blow from a metallic driving tool can also force the spring out of the spring pocket and is also referred to as spring dumping.

Special precautions should be taken when replacing a seal over a used shaft because it is common for shafts to become grooved during service. Grooving is normally caused by either carbonized oil or an abrasive foreign matter getting trapped between the lip and the shaft. Over time, deep grooves can form.

Replacement seals should never be installed over a grooved shaft. Dressing the shaft with emery cloth is not recommended because it is extremely difficult to obtain an optimal finish and lead will normally be imparted. If the shaft is worn, it should either be re-ground or fitted with a shaft repair sleeve. See Section 7 for shaft repair options.

Underlip Operating Temperature

When selecting a seal design, lip material and system lubricant, the operating temperature under the seal lip should be used as the upper limit rather than using the sump temperature.

Underlip temperature can exceed sump temperature by 60 °F (33 °C) or more, dependent on shaft diameter, shaft speed, fluid type and level. The increased temperature can exceed the limits of both the lip material and lubricant that is selected based on the sump temperature alone.

<table>
<thead>
<tr>
<th>Table 2-2a Min. Chamfer Length</th>
</tr>
</thead>
<tbody>
<tr>
<td>English Shaft Dia Up To And Including</td>
</tr>
<tr>
<td>0.375</td>
</tr>
<tr>
<td>0.750</td>
</tr>
<tr>
<td>1.250</td>
</tr>
<tr>
<td>1.500</td>
</tr>
<tr>
<td>2.000</td>
</tr>
<tr>
<td>2.750</td>
</tr>
<tr>
<td>3.750</td>
</tr>
<tr>
<td>5.000</td>
</tr>
<tr>
<td>9.000</td>
</tr>
<tr>
<td>+18.000</td>
</tr>
</tbody>
</table>

See Table 2-2a for chamfer length.
Other operating parameters such as a rough shaft finish or internal pressure will drive the underlip temperature even higher. As a general rule, the °F increase in underlip temperature above the sump temperature can be estimated as the square root of the shaft speed in feet per minute. (Replace the feet per minute units with °F.) This would be 55 °F (30 °C) for a shaft running at 3000 fpm (15 m/s).

\[
\frac{\text{Shaft Seal}}{\text{in fpm}} = \frac{\text{Increase Underlip Temperature}}{\sqrt{3000 \text{ fpm}}} = 55 \text{ °F}
\]

As sump temperatures increase, the difference between sump and lip temperature decreases.

Figure 2-8 shows the relationship of shaft diameter, shaft speed and sump temperature and the impact they have on the temperature at the contact point of the seal lip and the shaft (underlip temperature).

### Seal Torque

The underlip temperature increase is due to the friction between the shaft and seal lip. Torque is the frictional force the shaft must overcome to rotate in the seal. The energy consumption of the seal can be determined when the torque and shaft speed are known. Different seal designs, rubber compounds, fluids, fluid levels, temperatures, shaft textures, pressures and time in service each affect friction, so there is no exact calculation to predict torque. However, the following can give an approximate value for elastomer shaft seals. When the torque value is critical for the application, testing should be performed.

Torque from a dry running seal is 2 to 3 times the above.

For example: Torque is about 90 in-ounces for a three-inch shaft rotating at 3600 revolutions per minute in 250 °F SAE 30 weight oil to the shaft center. The energy in kilowatts the seal uses is 7.395 x 10^{-7} x torque x revolutions per minute. In this case, 0.24 kW.

Bearing isolators are an excellent choice when low torque is required because they add virtually no torque to the system.

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**Figure 2-8. Example Shaft Conditions**

An easier but more crude estimate is 20 °F (6.7 °C) higher than the sump for each 1,000 RPM of shaft speed for sump temperatures about 75 to 210 °F (24 to 99 °C).
Internal Pressure

Most elastomeric lip seals are designed to work in vented applications with zero internal pressure but will provide satisfactory service with pressures up to 3 psi (0.20 bar). Higher pressure will force the lip against the shaft and cause excessive friction. Severe pressure will distort and force the airspace of the lip to contact the shaft and can cause massive failure within hours of operation. See Figure 2-9 below. Excessive pressure can also push the seal out of the housing.

Figure 2-9. Internal Pressure

Parker offers several designs for applications where high internal pressure cannot be avoided. Elastomeric designs include MP, HP, NTC, TDN, and depending on design, can handle service up to 300 psi (20 bar). Refer to Pages 5-13 and 6-11.

Most PTFE designs can handle pressure, some up to 10,000 psi (690 bar). See Tables 9-4, 10-4 and 11-4.

Shaft Speed

Shaft seals operate in a wide range of speeds. When shaft speeds increase, so does underlip temperature, wear and internal pressure, if oil sumps are not vented. To assure optimal performance, select the proper seal design and material to accommodate for these factors.

Most seal manufacturers rate the speed limit using surface feet per minute (or meters per second). This is a measurement of how many surface feet (meters) pass a given point at the seal lip per minute (second) in time. Since this method considers the shaft diameter in addition to speed, it is a better service indicator than RPM alone.

The formulas below can be used to determine the fpm (feet per minute) or m/s (meters per second) for metric applications.

Inch

\[
\text{Shaft Diameter} \times \text{RPM} \times 0.262 = \text{fpm}
\]

Metric

\[
\text{Shaft Diameter (mm)} \times \text{RPM} \times 0.000523 = \text{m/s}
\]

A typical seal design in NBR material can operate up to 3,000 fpm (15 m/s) assuming all other operating parameters are reasonable. If any of the other operating conditions are excessive, seal designs and material upgrades are available to improve performance. Parker FKM and PTFE seals can be used for applications approaching 6,000 fpm (30 m/s) and ProTech bearing isolators for even higher speeds.
Housing / Bore Considerations

Typical radial shaft seals are pressed into the bore to assure proper OD sealing and seal retention in the housing. The most commonly used materials for seal housings are steel and cast iron. Care must be taken when softer materials such as aluminum, bronze or plastics are used for the housing material. Aluminum has a thermal expansion rate almost double that of steel. Steel case designs can lose the required press fit in an aluminum housing when they go through thermal cycles.

A seal with an aluminum, composite or rubber covered OD should be used for aluminum housings. These materials help maintain the press fit in the housing during thermal cycles and reduce the possibility of galvanic corrosion. Plastic housings can also expand at rates that can create problems if a metal OD seal is used.

The following chart shows typical values of thermal expansion for common metals in inch/ inch°F.

<table>
<thead>
<tr>
<th>Item</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum</td>
<td>0.000013</td>
</tr>
<tr>
<td>Brass</td>
<td>0.000011</td>
</tr>
<tr>
<td>Carbon Steel</td>
<td>0.0000058</td>
</tr>
<tr>
<td>Cast Iron</td>
<td>0.0000059</td>
</tr>
<tr>
<td>Stainless Steel</td>
<td>0.000010</td>
</tr>
</tbody>
</table>

Table 2-3. Typical Values of Thermal Expansion

Fiber reinforced and rubber OD seals are more forgiving so their bore tolerance can be greater than for metal OD seals. Aluminum bores are typically smaller than steel bores for metal OD seals to compensate for some of the difference in thermal expansion. A finish range of 40 to 100 μin Ra (1.0 to 2.5 μm Ra) is recommended for service pressures up to 3 psi (0.20 bar). If the fluid is thick, such as a grease, a 125 μin Ra (3.17 μm Ra) finish would be acceptable with no system pressure.

The finish on aluminum bores is more sensitive and must be maintained to keep seals from spinning in the bore and should not be smoother than 60 μin Ra (1.5 μm Ra).

A lead-in chamfer is highly recommended for all seal housings. The chamfer aligns the seal during installation and helps prevent the seal from cocking. Both corners of the chamfer should be free of burrs and sharp edges.

Figure 2-10. Housing Profile

Shaft to Bore Misalignment (STBM)

When the center of the shaft rotation is not the same as the center of the bore, the shaft pushes against the lip on one side of the seal greater than the other. This can cause the lip to wear rapidly in one place and have inadequate contact on the opposite side.

Figure 2-11. STBM
Shaft Runout

When the shaft does not rotate around its center, it wobbles. This condition is called runout. The seal lip has to move back and forth to maintain contact. The life of a seal is shortened as the runout is increased, and when the runout exceeds the capability of the lip, it will leak.

Parker offers seals for misalignment conditions. See Pages 5-15 and 6-17.

![Diagram of Shaft Runout](image)

**Figure 2-12. Shaft Runout**

Shaft Seal Summary

In conclusion, because the seal is only one component of the sealing system, all the following operating factors need to be considered for optimal seal life:

**Lubrication:** A seal is designed to run on a film of oil. Without the film of oil, the sealing lip will harden and crack due to the heat generated by excessive friction. The lubricant must also be compatible with underlip temperatures to avoid the buildup of abrasive, carbonized particles at the seal lip.

**Shaft Finish:** A shaft finish that is too smooth will cause a stick slip flutter that will let the fluid escape under the lip and cause excessive heat that will harden the lip. Excessive roughness will penetrate the lubricant film, cause leakage and accelerate lip wear. Maintaining the desired surface finish is critical for maximizing the service life of any contact rotary lip seal.

**Shaft to Bore Eccentricity:** When the center of the shaft rotation is not the same as the center of the bore, the shaft pushes against the lip on one side of the seal greater than the other. This can cause the lip to wear rapidly in one place and have inadequate contact on the opposite side.

**Dynamic Shaft Runout:** When the shaft does not rotate around its own center, the lip has to move back and forth to follow it. In excess, the lip will be unable to maintain contact as the shaft rotates, causing leakage.

**Pressure:** Excessive pressure will force the lip against the shaft and cause excessive frictional heat and wear.

**Bore:** A bore finish that is too coarse can cause a leak path by itself. If it has burrs or other sharp edges, they can scar the metal diameter during assembly, causing a leak path on the seal OD.

**Speed:** Shaft speed causes the underlip temperature to increase in addition to elevating the overall sump temperature. Over time, the heat will harden the elastomeric lip and reduce the seal’s ability to maintain positive contact with the entire circumference of the shaft.

**Operating Temperature:** Controlling the temperature of the sealing system is key to maximizing seal life. The relationship between speed, sump temperature, underlip temperature, pressure and shaft finish need to be considered since these operating parameters are interactive and will determine the service life of both the lip material and system lubricant.

Shaft Seal Installation

1. Prior to installation the seal should be examined to ensure that it is clean, undamaged and the correct seal for the application.

2. Verify spring is present for spring-loaded seal designs.

3. Prelubricate the seal lip with a system-compatible lubricant. It is preferable to use the system lubricant.

4. For seals with a rubber outside diameter, lightly lubricate seal OD with a system compatible lubricant. **DO NOT LUBRICATE THE OD OF A CLIPPER OIL SEAL THAT HAS A COMPOSITE OD.**

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5. Verify the desired lip direction for the application (lip toward oil for best retention).

6. Examine the leading edge of the shaft. Shaft should be properly chamfered and free of nicks and burrs that could cut or nick the seal lip.

7. Examine the leading edge of the housing. The seal bore should be chamfered and free of nicks and burrs that could gouge the seal outside diameter or make the seal difficult to install into the seal housing.

8. Examine the shaft where the lip will make contact. This surface must be free of grooves from prior service. If shaft is damaged or worn in this area, dress shaft for proper finish or install a Quick Sleeve or wear sleeve. If using a Quick Sleeve, an oversized seal is not required. If using a standard wear sleeve, the replacement seal must have an inside diameter that is designed to be used with the wear sleeve's outside diameter.

9. If the seal lip must pass over keyways or splines on the shaft, use an installation sleeve to protect the seal lips as they pass over these areas. If an installation sleeve is not available, wrap masking tape around the shaft to form a protective barrier.

10. Slide the seal over the shaft to the seal housing. With finger pressure, start seal into housing with a slight rotating motion until seal has a light press fit in the housing. Be sure seal is square or perpendicular to the shaft. If the seal is crooked or cocked, continuing with installation will damage the seal.

11. Position the installation tool and drive the seal into the housing until it is flush with the housing or recessed into the bore the proper distance. Please note that a screwdriver, punch or hammer should not be used to install the seal. Refer to the diagrams at right for recommended installation tools.

12. When using a metal driver to install metal clad seals, extra care is needed to be sure the shock load does not dislodge the spring.

13. If the seal is cocked in the housing, remove seal and start over using a new seal. Attempts to square the seal in the housing using direct blows will damage the seal.

14. Inspect the seal to be sure it is straight and flush. Examine the face of the seal for damage. If it is dented from installation, the lip will be deflected and will normally cause premature failure.

**Counter Bore Installation — Flush Mount**

Tool bottoms out against machined face of housing to position seal.

**Counter Bore Installation**

Seal is positioned square by seating against counter bore.
Recessed Installation
Tool bottoms out against end of shaft to position seal in the housing.

Installation Sleeve
Use to install seal over keyways and splines.

Handling and Storage
1. Care should be taken when storing rotary shaft lip seals to ensure optimal performance.

2. Seals should be stored in a cool, dry area below 86 °F (30 °C) with an average relative humidity of 40 to 70%.

3. Rotating stock is important. If inventory is old, seals should be used on a “first in, first out” basis. Based on the relative low cost of a lip seal compared to the expense associated with a failed piece of equipment, a good practice is to discard aged inventory since old seals may have deteriorated lip materials.

4. Seals should be stored away from direct or reflected sunlight and electrical equipment to avoid UV and ozone aging of the lip material.

5. Avoid storing seals in damp areas or where high humidity is present. Excessive humidity will deteriorate some seal element materials. Metal cases and springs will also rust and corrode if exposed to high levels of moisture or humidity.

6. Seals should not be exposed to radiation.


8. Do not use wire or string to tag a seal. Wire can easily cut the seal lip. Wire or string can also deform the lip beyond the point of recovery and can lead to leakage at start-up.

9. Do not store seals on hooks, nails or pegboard. Over time the weight of the seal resting on the hook will deform the lip beyond recovery.

10. Avoid storing seals where high levels of fumes are present. Depending on the chemical and concentration, it can chemically interact with the lip material.
PTFE Shaft Seals

How Do I Choose the Right Profile for My Application?

Parker’s PTFE product line includes both standard designs for the most common applications and custom designs that our engineers can help you develop.

For the long term, we suggest that you familiarize yourself with the design elements in this Engineering section that are critical when choosing a FlexiLip™, FlexiCase™ or FlexiSeal®.

For quick reference and ease of sorting through the many standard designs, we have provided simple decision trees and placed them throughout this design guide. If it becomes apparent that you need a custom design to meet your unique needs, or if you just want us to confirm the standard seal choice you’ve made, please contact Parker’s PTFE Engineering team at 801-972-3000.

Parker designs and manufactures a complete line of PTFE seals for both reciprocating and rotary applications. This guide focuses on seals for rotary applications. For reciprocating applications please refer to publication EPS 5340 PTFE Lip Seal Design Guide.

PTFE lip seals are commonly used as an upgrade over elastomeric lip seals when conditions are severe. Common reasons for upgrading to a PTFE material include chemical compatibility, poor lubrication at the lip, high pressure, high speed or high temperature.

For rotary applications, Parker offers three primary design groups: FlexiLip, FlexiCase and FlexiSeal.

FlexiLip seals are available in the above basic profiles. Excluder lips and internal metal stabilizer bands can be added to each profile depending on application requirements. The main difference is the shape of the primary lip. Additional options are available for the lip and O-ring material for added design flexibility.

FlexiCase designs feature PTFE lip elements encased in a metal jacket and are available in the above basic profiles. FlexiCase designs can be used in the same applications as FlexiLip profiles where more bore retention is required. Excluder lips can be added for additional exclusion capacity. Additional options are available for the lip and case material for added design flexibility. Typical operating limits are 6,000 sfpm, 500 psi and 450 °F (30 m/s, 34 bar and 232 °C).

FlexiSeal rotary seals are spring-energized designs and are available in the basic profiles above. Three spring options are available for each profile: cantilever, canted-coil and helical. Shaft speeds are very limited (below 1,000 sfpm or 5 m/s) but they can provide positive sealing to 10,000 psi (690 bar). This is the preferred design for rotating unions as well as oscillating and slow rotating shafts under high pressure conditions.

LF = Mandrel Formed Lip
LE = Elf Toe Lip
LG = Lip With Garter Spring
LM = Machined Lip
LD = Dual Lip

FlexiLip seals are intended for continuous running rotary shafts under various operating conditions. An O-ring is used on the OD for positive static sealing and proper bore retention. Typical operating limits are up to 6,000 sfpm, 150 psi and 450 °F (30 m/s, 10 bar and 232 °C). See Table 9-4 on Page 9-10 for specific limits.
Spring Designs

FlexiSeal profiles utilize three different spring designs.

- **V Series — Cantilever**
- **C Series — Canted-Coil**
- **H Series — Helical**

The two elements to consider when selecting a spring design are its load value and its deflection range. The spring’s load affects the sealing ability, friction and wear rate. As the spring load is increased, the lips seal tighter, with friction and wear increasing proportionately. The spring’s deflection range affects the seal’s ability to compensate for variations in gland tolerances and for normal seal wear. Each spring size has a specific deflection range. The available deflection increases as the seal and spring cross-section increase; this could be a deciding factor in selecting one cross-section over another. Springs with a wide deflection range should be used when sealing surfaces are nonconcentric (see Page 2-25).

![Figure 2-13. Spring Loading](image1)

Spring Loading Provides Positive Sealing Contact

Spring-Loaded, Positive Contact

Not Loaded, Poor Contact

![Figure 2-14. FlexiSeal Spring Energizers](image2)

**Figure 2-14. FlexiSeal Spring Energizers**

Figure 2-14 shows a relative comparison of load vs. deflection curves for the three spring types. The ▪ signifies the typical deflection when the seal is installed. The hatch marks indicate the deflection range through which the seal will function properly. Notice that H Series has a much smaller deflection range than both the V and the C Series.
Cantilever Springs — V Series

The FlexiSeal Cantilever spring is made from flat metal strip stock of 300 Series stainless steel or Elgiloy® as an option. The strip stock is punched or chemically etched into a serpentine pattern and formed into a rounded "V" shape. It is available in either a light or medium load spring. The medium spring is suitable in most applications, but the light load spring can be used if having low friction is more important than sealability. The medium spring load deflection curve is depicted in Figure 2-14 on Page 2-16.

The cantilever spring is intended for dynamic applications involving rotary or reciprocating motion. It can also be used in static conditions when there is need for a higher deflection spring due to wide gland tolerance, excessive expansion and contraction, or lift-off due to high pressure.

The long beam leg design puts the spring load out at the leading edge of the seal, creating the best load location for the FlexiSeal to act as a scraper when the optional scraper lip is selected.

The geometry of the V Series cantilever spring provides flexibility by utilizing individual tabs, separated by small gaps. This shape allows the spring to flex into radial and axial seal designs. The spring tabs can overlap on the ID and spread apart on the OD when the cross-section is too large for the diameter.

Table 2-4 provides the minimum diameters for V Series springs for rod and piston seals, as well as internal and external pressure face seals. For diameters smaller than those listed, C or H Series spring designs are recommended.

Table 2-4. Minimum Diameters for V Series

<table>
<thead>
<tr>
<th>Nominal Cross-Section</th>
<th>Rod Shaft Dia.</th>
<th>Piston Bore Dia.</th>
<th>Internal Pressure (Seal OD)</th>
<th>External Pressure (Seal ID)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/16</td>
<td>0.125</td>
<td>0.250</td>
<td>0.750</td>
<td>0.500</td>
</tr>
<tr>
<td>3/32</td>
<td>0.187</td>
<td>0.375</td>
<td>1.250</td>
<td>0.875</td>
</tr>
<tr>
<td>1/8</td>
<td>0.375</td>
<td>0.625</td>
<td>1.750</td>
<td>1.125</td>
</tr>
<tr>
<td>3/16</td>
<td>0.875</td>
<td>1.250</td>
<td>2.250</td>
<td>2.000</td>
</tr>
<tr>
<td>1/4</td>
<td>1.625</td>
<td>2.125</td>
<td>3.500</td>
<td>3.000</td>
</tr>
</tbody>
</table>

Features
- V-shaped spring with moderate load vs. deflection
- Standard inch/fractional and MIL-G-5514 sizes
- Standard 300 series stainless steel springs
- NACE compliant Elgiloy springs available in medium spring load, -450 to 600 °F
- Scraper lip designs for abrasive medias
- Available as external & internal pressure face seals

Recommended Applications
- Reciprocating rods & pistons
- Rotary shafts <1000 sfpm
- Wide tolerance and misaligned glands (static)
- Abrasive medias (when scraper lip is designated)
- Dynamic applications above 450 °F

Figure 2-15. Installed State

*Elgiloy® is a registered trademark of Combined Metals of Chicago, LLC, Chicago, IL.
Canted-Coil Springs — C Series

The FlexiSeal C Series spring is made from round wire that is coiled and formed into a canted or slanted shape. The result is a radial compression spring with a very flat load versus deflection curve as illustrated in Figure 2-14 on Page 2-16. Both 302 stainless steel and Hastelloy® C-276 are available as standards in three different spring loads.

The canted-coil spring is intended for dynamic reciprocating and rotary applications. It is also used in static applications when wide gland tolerance or misalignment is present. The flat load curve of this design makes it an ideal choice for friction sensitive applications.

The C Series spring can be fit into small seal diameters without overlapping the individual spring coils. Because the ID coils tend to butt up to each other, the spring has very small gaps providing maximum spring contact. This geometry is well suited for dynamic rod seal applications less than 1/2” diameter.

The C Series spring is available in Light, Medium and Heavy load ranges.

- **Light**: Applications that require extremely low break-out and running friction when sealing ability is less important than friction.
- **Medium**: General application. Medium friction but reliable sealing capability. Normally the starting point for new applications. Balance functions of friction, sealing ability and dynamic wear.
- **Heavy**: Applications where optimum resilience is required due to hardware separation. Accelerated seal material wear in dynamic applications. Used when primary objective is sealing and friction and/or wear is secondary.

The C Series spring produces compression load near the center of the seal. The standard beveled lip seal geometry puts the point of contact slightly in front, forcing the spring back into the spring cavity. The lip design provides concentrated unit load at the sealing interface, and allows lubrication to the dynamic lip, increasing the wear life. Because of this geometry, the C Series is not the best choice for abrasive medias. For abrasive conditions the FlexiSeal V Series is recommended. See Page 2-17 for details.

**Features**
- Canted coil spring with flat load vs. deflections
- Light, medium and heavy load springs standard
- Standard inch/fractional and MIL-G-5514 sizes
- Standard 302 series stainless steel springs
- Hastelloy springs available
- Available as external & internal pressure face seals

**Recommended Applications**
- Friction sensitive applications
- Reciprocating rods & pistons
- Rotary shafts <1000 sfpm
- Wide tolerance and misaligned glands
- Dynamic applications above 450 °F
- Diameters <1/2” and cross-sections <3/32”

*Hastelloy® is a registered trademark of Haynes International, Inc., Kokomo, IN.*
**Helical Springs — H Series**

The H Series spring is made from flat ribbon metal strip stock that is formed into a helix shape. The standard material is 17/7 PH stainless steel, and Elgiloy® is offered as an option. The finished spring produces a very high load versus deflection curve as shown in Figure 2-14 on Page 2-16.

The helical spring design is intended for static applications due to the high unit load. It can be used in very slow or infrequent dynamic conditions when friction and wear are secondary concerns to positive sealing.

The H series spring produces evenly distributed load across each individual band, with very small gaps between the coils. This tight spacing provides near continuous load, reducing potential leak paths. This, combined with the high unit load, makes the H series well-suited for vacuum and cryogenic applications or when pressure is too low to energize the seal.

The load provided by the H Series spring is directly through its centerline. The lip design of the FBN-H profile is a full radius at the sealing interface, providing maximum load to the contact points to effect a tight seal. The spring is welded at the ends. When the seal is compressed into the hardware, the spring cavity is designed to allow axial spring growth.

The relatively small deflection range of the H Series spring prevents it from being used in applications having wide gland tolerances, eccentricity or misalignment. The V or C Series FlexiSeal should be considered for these conditions.

**Features**

- Helical wound ribbon spring with high load vs. deflection
- Standard inch/fractional and MIL-G-5514 sizes
- Standard 17/7 PH stainless steel springs
- NACE compliant Elgiloy springs available
- Available as external & internal pressure face seals

**Recommended Applications**

- Static rods & pistons
- Static internal & external pressure face seal applications
- Slow dynamic applications <200 sfpm
- Vacuum sealing
- Applications where sealing ability is critical
Lip Shapes

**Formed Lips** — The formed lip is the most common profile for general rotary service. After the lip is machined to size, a mandrel is used to form the lip to the desired interference to achieve the optimal lip load. Both FlexiLip and FlexiCase designs are available with single or multiple formed lips. Formed lips are used to retain lubricant, can handle pressure up to 150 psi (10 bar) and are rated for shaft speeds up to 5,000 fpm (25 m/s). Because the lip is not spring-loaded, its ability to handle misalignment and runout conditions is limited.

**Machined Lips** — The machined lip allows for tighter control of the lip interference and is used for special applications where the lip load is critical. Both FlexiLip and FlexiCase designs are available with a single machined lip for lubricant retention. Machined lips are also commonly used as an excluder lip in conjunction with a primary formed lip. The machined lip normally has a narrower contact footprint on the shaft so it is more sensitive to eccentricity and runout than the formed lip.

**Elf Toe Lips** — The elf toe lip profile has a sharp angle on the leading edge of the lip to help keep abrasive media from getting trapped under the lip. The pocket that is formed by this profile also allows a garter spring to be added. The addition of a garter spring allows the lip to maintain contact with the shaft under high misalignment conditions up to 0.020” (0.5 mm) Total Indicator Runout (TIR).

**Chamfered Lips** — The most common spring-energized lip shape is the chamfered or back-beveled design and is available with the V and C Series spring types. This design allows for ease of installation and permits lubrication to nest under the lip and feed through in reciprocating dynamic applications. The result is a microscopic film of lubrication that increases seal and hardware service life. Since the footprint of a chamfered lip is a single point, all of the sealing force is concentrated, yielding the highest sealability and lowest friction. The high lip contact force limits the use in rotary service to 1000 sfpm (5 m/s).

**Scraper Lips** — Applications often involve medias with abrasive particles that can get caught between the seal lip and the mating hardware. This increases wear to both the seal and the mating surface. The scraper lip contact point is positioned directly over the load point of the spring in each design for maximum scraping action. The scraper lip can be positioned on the ID, OD or both. The high spring-loaded contact point limits the use in rotary service to low speed applications.
Shaft Considerations

The shaft finish required for PTFE seals is just as critical as that for elastomeric lip seals (see Page 2-6).

Proper surface finish is critical to ensure positive sealing, and achieve the longest seal life possible in rotating applications. Rotating surfaces that are too rough can create leak paths and can be very abrasive to the seal. Unlike elastomer contact seals, PTFE-based Flexi designs can run on very smooth surfaces with or without lubrication. Due to the toughness and low coefficient of friction of PTFE, Flexi designs, unlike seals made of other materials, slip over the high points of the mating surface and resist abrasion. To maximize seal performance, the recommendations for surface roughness in Table 2-5 should be followed.

Dynamic surfaces with relatively rough finishes will result in higher wear rates, which decrease the seal life and may compromise performance. Additionally, dynamic surfaces which have a finish smoother than recommended may also decrease the seal’s effectiveness. The optimum surface roughness allows a film of the fluid being sealed to flow between the seal and the mating surface, which effectively lubricates and extends the life of the seal.

PTFE rotary seal applications require a hard running surface on the dynamic portion of the hardware. The harder surface allows the use of higher reinforced seal materials that will increase the seal and hardware life. Softer running surfaces must use lower wear resistant materials that will not damage the hardware and normally yield shorter seal life. A balance between seal material and dynamic surface hardness must be met to ensure that the seal remains the sacrificial component. Table 9-3 includes minimum recommended surface hardness for Parker materials in dynamic applications, based on temperature, motion and speed.

When the dynamic surface hardness is below 45 Rc, most seal materials will polish the running surface of the hardware and the seal. This initial break-in period will cause seal wear to taper off over a period of time, depending on the seal material, surface finish and PV of the application. When hardness exceeds 45 Rc, the initial surface finish is very important since the surface is much harder to polish and the time to achieve break-in is much longer. Surface hardness above 65 Rc will generally not polish and therefore the initial surface finish is even more critical to seal life. The hardness of the dynamic hardware surface affects the wear rate of the seal. Additionally, some seal lip materials are abrasive and will wear softer metal shafts or dynamic components. In general, higher surface finish results in better overall seal and hardware performance. The ideal hardness of the dynamic surfaces of the hardware is 50 to 60 Rockwell C. The actual hardness used is normally a balance between the additional cost associated with finishing harder materials versus the maximum seal life that will be achievable.

Table 2-5. Surface Roughness, Rₐ

<table>
<thead>
<tr>
<th>Media Being Sealed</th>
<th>Dynamic Surfaces</th>
<th>Static Surfaces</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>µ inch</td>
<td>µ m</td>
</tr>
<tr>
<td>Cryogenics</td>
<td>4 max.</td>
<td>0.1 max.</td>
</tr>
<tr>
<td>Helium Gas Hydrogen Gas Freon</td>
<td>6 max.</td>
<td>0.15 max.</td>
</tr>
<tr>
<td>Air Nitrogen Gas Argon Natural Gas Fuel (Aircraft and Automotive)</td>
<td>8 max.</td>
<td>0.2 max.</td>
</tr>
<tr>
<td>Water Hydraulic Oil Crude Oil Sealants</td>
<td>12 max.</td>
<td>0.3 max.</td>
</tr>
</tbody>
</table>

Figure 2-18. Shaft Profile
The leading edge of the shaft should have a burr-free chamfer to ease installation by preventing lip roll-back. Because PTFE lip seals are not as flexible as rubber lip seals they tend to be more difficult to install over the shaft. First time installers of PTFE lip seals normally destroy a few seals before realizing they are more difficult to start over the shaft than a rubber lip seal. When possible, use an installation sleeve to get the PTFE seal started over the shaft. The sleeve will also protect the lips from sharp edges common with keyways or splines.

**Figure 2-19. Housing Profile**

For pressurized rotary applications, additional precautions are needed to ensure the seal is not pushed out of the housing. If the seal is installed in an open bore, a snap ring or cover plate should be added to retain the seal. The specific pressure that requires additional retention is a function of the seal surface area, seal design, internal pressure and bore finish. As a general rule, retention devices should be used with FlexiLip designs in applications over 2 psi (0.15 bar) and FlexiCase designs over 30 psi (2 bar).

**Housing/Bore Considerations**

Typical FlexiLip and FlexiCase shaft seals are pressed into the bore to assure proper OD sealing and seal retention in the housing. The most commonly used materials for seal housings are steel and cast iron. Care must be taken when softer materials such as aluminum, bronze or plastics are used for the housing material. Aluminum has a thermal expansion rate almost double that of steel. Metal case designs can lose the required press fit in an aluminum housing when they go through thermal cycles due to the higher rate of thermal expansion of aluminum.

A finish range of 32 to 63 μin Ra (1.0 to 2.5 μm Ra) is recommended for service pressures up to 3 psi (0.20 bar). If the fluid is thick, such as a grease, a 125 μin Ra (3.17 μm Ra) finish would be acceptable with no system pressure.

A lead-in chamfer is highly recommended for all seal housings. The chamfer aligns the seal during installation and helps prevent the seal from cocking. Both corners of the chamfer should be free of burrs and sharp edges.
FlexiSeal flanged designs can be used in either static, rotary or reciprocating applications and are designed to be dynamic only on the ID. They excel in rotary applications because the flange can be clamped axially to prevent the seal from rotating with the shaft. This extra stability allows the flanged design to hold more pressure at higher surface speeds. The housing must be made in two pieces for installation purposes and the seal can be installed either lips-first or heel-first.

**Pressure and Shaft Velocity**

Unlike reciprocating applications, seals that ride on a rotating shaft have a contact point that is localized in only one small area where dynamic forces and energy are concentrated. In fact, much of the energy from the shaft is dissipated at the seal in the form of frictional heat and wear, both of which are detrimental to seal life. This effect is accentuated by increasing the shaft speed or by increasing the perpendicular force holding the lip against the shaft. Shaft speed can be measured in surface feet per minute and the lip force can be approximated by measuring the differential pressure across the seal in psi. Shaft velocity in surface feet per minute is calculated as follows:

\[
\text{Surface Velocity} = \frac{\text{Shaft Diameter}}{\text{Shaft RPM}} \times 0.262
\]

One way to estimate the exposure to these risks is to calculate the PV-value by multiplying the pressure held by the seal (P in psi) by the surface velocity of the shaft (V in surface feet per minute). The product of this multiplication provides the designer with a guide to aid in the choice of seal profile and material. Let us run through an example:

**Given:**
Pressure = 45 psi  
Shaft diameter = 1.25"  
Shaft rotational speed = 350 RPM

\[
\text{Surface Velocity} = \frac{1.25"}{350 \text{ RPM}} \times 0.262
\]

\[
= 115 \text{ sfpm}
\]

\[
\text{PV-value} = \text{Pressure} \times \text{Surface Velocity}
\]

\[
= 45 \text{ psi} \times 115 \text{ sfpm}
\]

\[
= 5175 \text{ ft. lb./in}^2 \text{ min.}
\]
Figure 2-21. Pressure — Velocity Chart

The PV graph in Figure 2-21 applies to un lubricated rotary applications using a stable rotary seal in a jacket material with a 4 or 5 wear resistance rating. As a rule of thumb, a PTFE rotary seal can be used in un lubricated applications with a PV of up to 150,000.

This information is intended to be used only as a guide since there are many other factors, such as sealing media, hardware material and surface finish, which affect the amount of heat generated and the wear life of the seal. In cases where the media being sealed is a lubricant, these seals can operate continuously at PV levels 10 to 20 times higher than those shown in Figure 2-21.

Lubrication

While Parker PTFE seals have a natural lubricity and can be used in un lubricated applications, it is always better to have lubrication present in rotary applications. A film of lubricant between the seal lip and the shaft reduces seal wear and frictional heat generation, makes higher surface speeds possible, and helps prevent the seal from wearing a groove in the shaft. When the lubricant splashes or flows past the seal area, it acts as a coolant, prolonging seal life.

Rotary PTFE Product Choice

While the black and white curves above attempt to draw the line between what can and cannot be done, they do not show which profiles work best within the limits of feasibility. The blue and brown curves above show which product lines work better with regard to pressure and surface speed assuming there is no lubrication. Rotary FlexiSeals can be used when pressures are high and speeds are low, while FlexiLip and FlexiCase profiles lend themselves more to applications with high surface speeds and low pressure.
Shaft Misalignment and Runout

Applications with rotating shafts come with their own set of common problems. Among these are those associated with the shaft not being aligned properly with the surrounding hardware. Misalignment most commonly manifests itself as Eccentricity and Runout. Every shaft has some degree of both as described in Figure 2-22.

Eccentricity of a rotating shaft creates two problems. One is that it forces the seal lip to follow a shaft that is not centered in the bore, wearing the lip more on one side. Because they are less elastic, PTFE seals are more susceptible to failure, misalignment and runout conditions than elastomeric lip seals. The second potential problem is that it enlarges the extrusion gap on one side, which could be detrimental if high pressure is involved. Extended heel designs will reduce seal extrusion.

Shaft Runout is when the shaft is spinning on an axis of rotation that is offset from the geometric center of the shaft at the point of seal lip contact. Runout can be caused by a bent shaft or by whirling deflection while spinning. The seal must be sufficiently compliant to maintain contact with the shaft despite being compressed and extended each revolution. It follows that shaft runout becomes more of a problem at high speeds.

Figure 2-23. FlexiSeal Eccentricity and Runout Limits

All rotating shafts have eccentricity and runout to some degree. The risk of failure increases significantly if a system has a considerable amount of both. Figure 2-23 shows the acceptable maximum for these parameters for all rotary FlexiSeal profiles except the FFN-H. Figure 2-24 shows the limits for FlexiLip and FlexiCase profiles.

Figure 2-24. FlexiLip and FlexiCase Eccentricity and Runout Limits

03/28/06
Rotary PTFE Seal Considerations

For all rotary seals — FlexiSeal Rotary, FlexiLip and FlexiCase — the designer must consider:

- pressure and shaft velocity
- lubrication
- shaft misalignment and runout
- shaft hardness and surface finish
- advantages of different lip shapes
- shaft lead
- temperature

For additional information on reciprocating applications, please refer to publication EPS 5340, *PTFE Lip Seal Design Guide*.

Alternate Housing Configurations

- Rotary Housing
- Flanged Rotary Housing
- FlexiLip with Retainer for Higher Pressure
- Banded FlexiLip with Retainer
- FlexiCase with Snap Ring
**Bearing Isolators**

**General Theory of Operation**

Engineered labyrinth type seals, also called “Bearing Isolators,” should be considered when increasing the Mean Time Between Failures (MTBF) is a primary objective for seal selection. Common equipment that uses labyrinth-type seals includes ANSI pumps, IEEE 841 rated electric motors, split pillow block bearings, turbines and gearboxes.

All bearing isolator designs consist of at least a rotor and a stator. An external O-ring at the stator OD maintains a press fit in the seal housing and provides a static seal for oil retention. The O-ring press fit allows for easy seal installation while also providing excellent bore retention. The press fit will withstand external forces to eliminate movement or spinning in the housing and has even been tested in the vertical down position to ensure the stator will not walk out of the seal bore.

The stator has a series of internal grooves to retain oil splash and return it to the sump. Before ProTech® seal designs were introduced, bearing isolators relied on a single inboard groove for oil retention.

To improve lubricant retention capability, Parker offers a double groove and a severe splash design. The double groove design works well for grease and mild oil splash. Parker developed and introduced the severe splash design for applications where oil splash is heavy. The severe splash design is also used in applications where outboard mounted fans are present and are pulling the lubricant through a simple groove design. Types “LS”, “LN” and “LB” feature a double groove. A severe oil splash groove is standard on types “LW”, “LX”, “ML” and “MN”.

An outboard drain port is machined in the stator. Contaminants that enter the assembled seal are expelled through this drain port. During installation it is imperative that the drain port is centered at the 6 o’clock position. Designs with multiple drain ports are available for applications when equipment mounting in the field cannot ensure the drain port will be positioned at 6 o’clock.

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The second component, the rotor, uses an external static O-ring at the seal ID to maintain a press fit on the shaft. Since the rotor spins with the shaft, it will not wear, groove or cause damage, so the costs associated with having to recondition the sealing surface are eliminated.

The wraparound profile of the ProTech rotor is no accident. Years of testing allowed Parker to refine this feature to provide optimal water exclusion. As a result of Parker’s testing and design leadership, the wraparound rotor profile is quickly becoming an industry standard.

The rotor and stator are assembled at the factory where they are permanently unitized by means of patented processes. Unitized, two-piece designs were pioneered and patented by Parker. Unitized designs allow for one-piece installation and maintain a minimal clearance between the rotor and stator interface for the life of the seal. This interface is the first line of defense against contamination. A unitized design maintains the seal's integrity by keeping high-pressure water spray, vibration or axial movement from separating or increasing the gap between the rotor and stator.

**Figure 2-25. Pressure Drops and Turbulence**

Bearing isolators are very effective at retaining grease or oil splash. Except for hybrid designs such as the ProTech 360, they are not designed to be used in “flooded” oil applications. A flooded condition occurs when the oil level is above the seal’s oil drain-back port during operation. Prior to installing a labyrinth-type seal, it is important to verify that the cavity between the seal and the bearing does not become flooded during operation.
Excessive oil pumping by the bearing, inadequate oil drain back design of the housing or excessive oil levels are the most common causes of unexpected flooded conditions. For oil lubrication, bearing isolators are designed for oil levels that are filled to the center of the bottom roller of the bearing.

ProTech Bearing Isolators excel in their ability to exclude contaminants such as dust, water and even high-pressure water spray. Because they are non-contact, the seals provide superior exclusion for the life of the bearing. They provide such a long service life because they do not form a dynamic seal at the shaft like traditional lip seals do. As an additional benefit, maintaining the optimal shaft finish is not required when using labyrinth seals.

ProTech designs are available that exceed ingress protection levels according to IEEE IP55, IP56, IP66 and IP69k. For additional information on IEEE levels of protection, see Table 8-2.

Across the industry, bearing isolators are available in a wide range of designs. Some use internal seals such as O-rings, locking rings, lip seals or other components as a primary seal and to lock the seal components together.

High performance designs like Parker ProTech rely on true, non-contact labyrinth technology (Type 1). The performance of true non-contact designs is more reliable over time because there are no internal components to wear out or become clogged by contamination. As a rule of thumb, avoid designs that rely on internal components for sealability or unitizing the rotor and stator (Types 2, 3 and 4). The presence of internal components is an indication that the labyrinth design may be inferior. The performance of such designs are reduced to zero when tested with the internal component removed. Consequently, as the internal seal wears, performance will decrease. Internal components that serve as the primary seal are often affected by temperature, centrifugal force and chemicals with unintended and undesirable results.

Bearing isolators are available in a wide range of materials including high performance PTFE, bronze, carbon steel, stainless steel and various other alloys. The primary considerations for material selection are chemical resistance, temperature and internal dusting. PTFE is the preferred material because it has the best chemical resistance, lowest friction and best balance of properties of any of the materials used.

Internal dusting is a common trait of bearing isolators. During initial break in, dusting of the seal material will occur. As the seal dust can enter the bearing cavity, a material that will not damage the bearing (such as PTFE) should be selected when possible.

Housing bore requirements regarding finish are the same as standard lips seals with dimensional tolerances of ±0.002" (±0.05 mm). Lead-in chamfers are also recommended to prevent damage during installation.

Because bearing isolators do not create a dynamic seal at the shaft, shaft finish is not as important as that required for a lip seal. Typical shaft finishes are 16 to 32 µin Ra (0.4 to 0.8 µm Ra) but up to 60 µin Ra (1.5 µm Ra) is acceptable.

Parker's numerous standard designs allow for retrofits without any equipment modifications for most applications. Custom designs are also readily available. For additional information on bearing isolators, refer to Section 8 of this catalog.
Testing and Validation

Finite Element Analysis (FEA) allows new concepts to be studied for weaknesses and compared to potential modifications before tooling and samples are made. Bench testing equipment meeting SAE J110 standards is used to verify design capabilities.

Utilizing sophisticated test equipment, Parker technicians accurately perform tests, controlling conditions such as shaft speed, temperature, misalignment, pressure and chemical compatibility. Tests evaluate fluid containment as well as water and contaminant exclusion. Custom fixtures are designed and built for unique situations, incorporating customer’s equipment to evaluate the entire sealing system of a specific application. When necessary, independent testing laboratories are used to verify results.