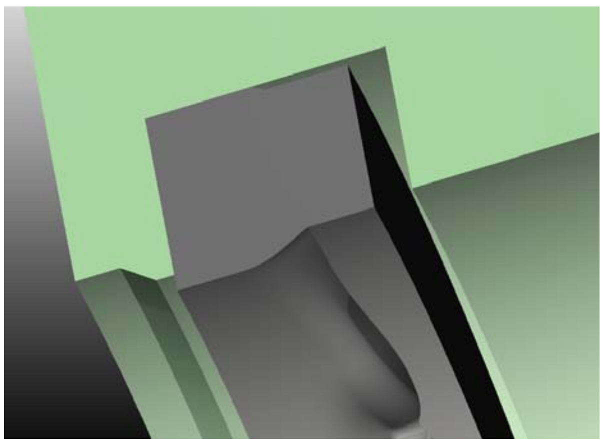
Chapter D5

Kalsi Seal gland guidelines



Revision 7 December 5, 2019

Individual chapters of the Kalsi Seals Handbook™ are periodically updated. To determine if a newer revision of this chapter exists, please visit https://www.kalsi.com/seal-handbook/.

1. Introduction and scope

The “gland” is the enclosed space defined by the housing groove and the rotary shaft. The gland locates the seal axially and compresses the seal radially. This chapter addresses gland design practices for solid cross-section direct compression-type single and Dual Durometer Seals, Plastic Lined Seals, BDRP Seals™, Filled Seals™, and Grooved Seals™. This chapter does not address gland design practices for KLS Seals, washpipe packing, and Plastic Backup Rings.

The direction of differential pressure influences gland design. Specific Kalsi Seal™ designs have been developed for different pressure conditions. Many Kalsi Seals® are designed for lubricant overpressure applications, where the pressure of the seal lubricant is greater than the pressure of the environment. The Axially Constrained Seal™ is designed for applications where the pressure of the seal lubricant is roughly the same as the pressure of the environment and can vary from slightly less to slightly more than the pressure of the environment. The BDRP Seal™ was developed to partition two lubricants, with differential pressure acting in either direction. Some Kalsi Seals are configured for applications where the pressure of the environment is greater than the pressure of the seal lubricant and incorporate tangs on the environment end of the seal to prevent circumferential slippage.

2. Gland nomenclature

Figure 1 identifies the nomenclature used to describe gland features and dimensions. The dimensions and relative proportions of the illustrated gland features may vary significantly, depending on seal design, equipment design, and application specifics. For example, the extrusion gap width and the pressure side gap width may be very short as shown or may be relatively long. For another example, many small diameter seals and plastic-elastomer composite seals require a removable groove wall to permit seal insertion. For such reasons, Figure 1 should be regarded as schematic in nature, rather than as a visual template for your equipment design.

Figure 1 orients the features relative to the direction of differential pressure. For example, the groove wall on the low-pressure side supports the seal against differential pressure, and the seal is required to bridge the radial extrusion gap between the seal housing and shaft. With seals that are exposed to reversing pressure, such as ACS and BDRP Seals, both groove walls undergo service periods where they support the seal against differential pressure, and both housing-to-shaft gaps undergo service periods where they function as extrusion gaps.

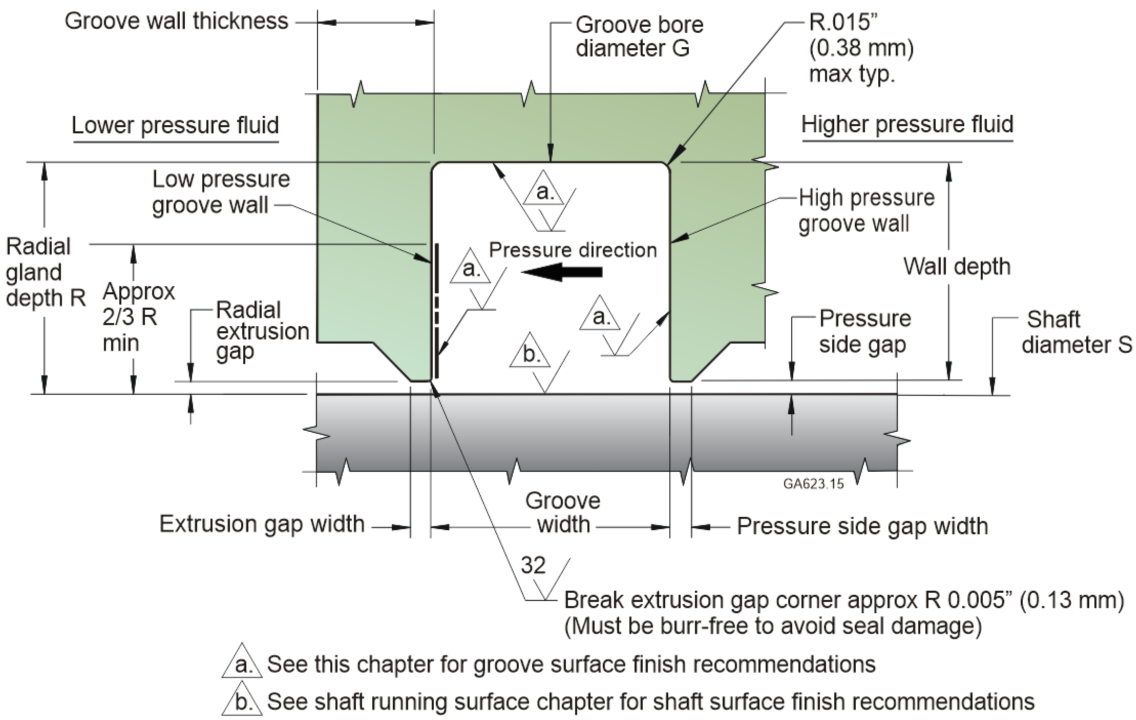


Figure 1
Kalsi Seal gland dimensions and nomenclature

For groove bore diameter G use Equation 1. See the Kalsi Engineering website for groove width and radial gland depth R. The radial gland depth R is the radial distance from the shaft diameter S to the groove bore diameter G. The radial gland depth R is **NOT** the depth of cut when machining the seal groove. Very small diameter elastomeric seals and many plastic-elastomer composite seals require a removable groove wall to permit seal insertion. Don’t forget to specify a position or concentricity requirement for groove bore diameter G and the bore that defines the extrusion gap clearance.

3. Groove bore diameter

The groove bore diameter G determines the amount of radial seal compression. This compression establishes sealing contact pressure with the groove bore and the shaft, and physically blocks the leakage path. Under static conditions, Kalsi Seals establish static sealing with respect to the groove bore and the shaft. During rotation, the seal hydroplanes on a thin film of lubricant. Excessive compression may adversely affect rotary seal performance by increasing interfacial contact pressure, which can increase seal running torque and seal generated heat and reduce seal lubrication. If compression is inadequate, tolerances and elastomer compression set may cause higher leakage and environmental invasion, especially with eccentrically running shafts, or shafts that are subject to large deflection. When severe shaft deflection is encountered, a laterally translating seal carrier

(Chapter D16), or a laterally translating backup ring with integral seal groove (Chapter D17), may be necessary.

Due to compression considerations, the Kalsi Seal radial cross-sectional dimension is held to tighter tolerances than are typically used in other molded rubber products. The radial tolerances of most Kalsi Seals are provided on our website.

Determining the groove bore diameter G

Determine the nominal shaft diameter S_{nom} in accordance with Chapter D2. Determine the nominal groove bore diameter G_{nom} using the Equation 1 and the nominal radial gland depth R_{nom} from the website:[1]

Equation 1, Nominal groove bore diameter:

$$G_{nom} = S_{nom} + 2 \text{ x } R_{nom}$$

Example 1, where S_{nom} = 2.75", R_{nom} = 0.309":

$$G_{nom} = 2.75" + 2 \text{ x } 0.309" = 3.368"$$

Applying tolerance to the groove bore diameter G

After sizing the nominal groove bore diameter, determine the minimum and maximum groove bore diameter using Equations 2 and 3 and the suggested groove tolerance G_t from Table 1:

Equation 2, Minimum groove bore diameter:

$$G_{min} = G_{nom} - G_t$$

Example 2, where G_{nom} = 3.369", G_t = 0.002":

$$G_{min} = 3.368 - 0.002 = 3.366$$

Equation 3, Maximum groove bore diameter:

$$G_{max} = G_{nom} + G_t$$

Example 3, where G_{nom} = 3.369", G_t = 0.002":

$$G_{min} = 3.368 + 0.002 = 3.370$$

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[1] The website provides the recommended radial gland depth for typical service.

Table 1 **Suggested bilateral groove tolerance G_t**		
Seal radial cross-section depth D (inches)	**Groove diameter bilateral (±) tolerance G_t, (inches)**	**Groove diameter bilateral (±) tolerance, G_t (millimeters)**
0.145	0.0005	0.013
0.186	0.0010	0.030
0.212	0.0010	0.030
0.270	0.0015	0.040
0.300	0.0015	0.040
0.335 & 0.345	0.0020	0.050
0.415 & 0.450	0.0030	0.080

Determining worst case eccentric radial gland depth

It can be useful to determine the worst-case eccentric radial gland depth resulting from lateral shaft deflection under load, accumulated tolerances, and shaft articulation within mounting clearances. Nominal concentric radial gland depth can be verified with Equation 4. Minimum and maximum radial gland depth in eccentric conditions can be determined using Equations 5 and 6 in conjunction with Figure 2. (Equation variables are defined in Appendix 3).

Equation 4, Nominal radial Kalsi Seal gland dimension:

$$R_{nom} = \frac{G_{nom} - S_{nom}}{2}$$

Example 4, where G_{nom} = 3.369", S_{nom} = 2.750":

$$R_{nom} = \frac{3.368 - 2.750}{2} = 0.309$$

Equation 5, Minimum radial Kalsi Seal gland dimension:

$$R_{\min} = \frac{G_{\min} - S_{\min}}{2} - \frac{M_{\max}}{2} - \frac{E_{\max}}{2} - L_{\max}$$

Example 5, where G_{min} = 3.366", S_{min} = 2.7495", M_{max} = 0.006", E_{max} = 0.0015", and L_{max} = 0.001":

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$$R_{\min} = \frac{3.366 - 2.7495}{2} - \frac{0.006}{2} - \frac{0.0015}{2} - 0.001 = 0.3035$$

Equation 6, Maximum radial Kalsi Seal gland dimension:

$$R_{\max} = \frac{G_{\max} - S_{\min}}{2} + \frac{M_{\max}}{2} + \frac{E_{\max}}{2} + L_{\max}$$

Example 6, where G_{max} = 3.370, S_{min} = 2.7495", M_{max} = 0.006", E_{max} = 0.0015", and L_{max} = 0.001":

$$R_{\min} = \frac{3.370 - 2.7495}{2} + \frac{0.006}{2} + \frac{0.0015}{2} + 0.001 = 0.315$$

Determining the range of seal compression

To determine average, minimum and maximum radial Kalsi Seal compression, use Equations 7, 8, and 9 respectively (Equation variables are defined in Appendix 3 and seal cross-sectional tolerances are provided on the website).

Equation 7, Nominal radial Kalsi Seal Compression:

$$C_{nom} = D_{nom} - R_{nom}$$

Example 7, where D_{nom} = 0.335", R_{nom} = 0.309":

$$C_{nom} = 0.335 - 0.309 = 0.026"$$

Equation 8, Minimum Radial Kalsi Seal Compression:

$$C_{\min} = D_{\min} - R_{\max}$$

Example 8, where D_{min} = 0.330", R_{max} = 0.315":

$$C_{\min} = 0.330 - 0.315 = 0.015$$

Equation 9, Maximum Radial Kalsi Seal Compression:

$$C_{\max} = D_{\max} - R_{\min}$$

Example 9, where D_{max} = 0.340", R_{min} = 0.3035":

$$C_{\max} = 0.340 - 0.3035 = 0.0365$$

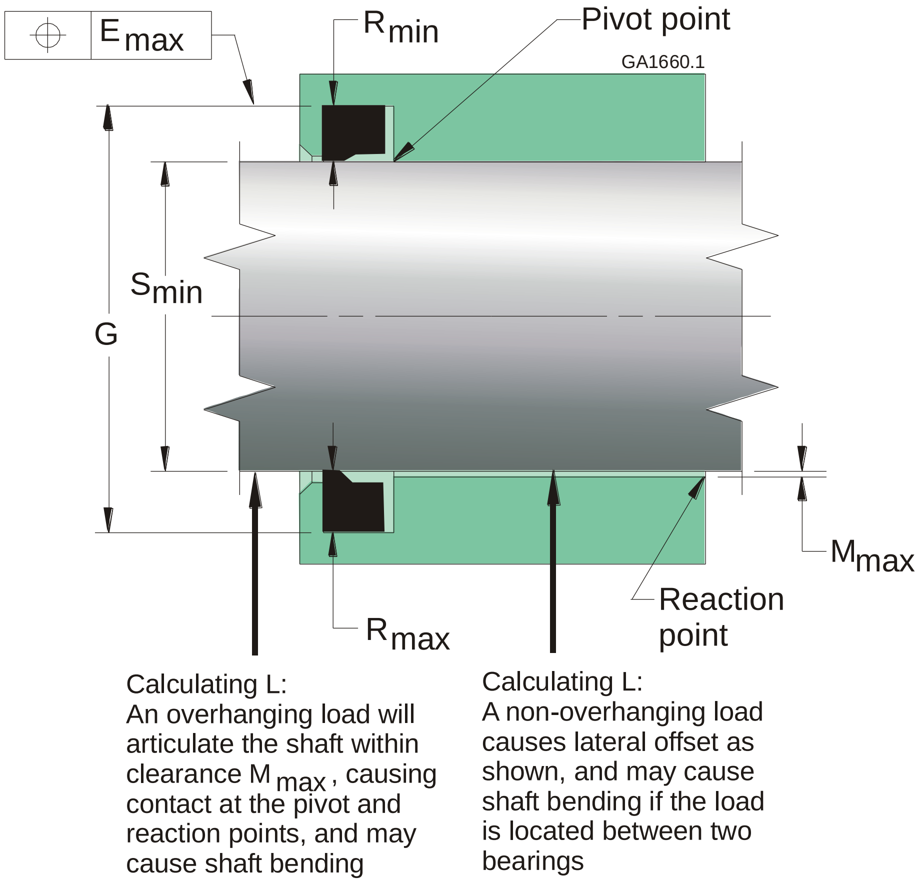

Figure 2
Minimum and maximum radial gland dimension

This figure schematically illustrates the variables used in Equations 3 and 4 to determine the minimum and maximum radial gland dimensions R_{min} and R_{max}. The equations take into account tolerances, groove to bearing eccentricity, lateral offset due to clearances, articulation due to side load, and deflection due to side load. Although a journal bearing is illustrated, the equations are equally applicable to other bearing arrangements. M_{max} is the accumulative diametric mounting clearance of the seal groove relative to the rotary shaft. For rolling element bearings, M_{max} includes bearing mounting clearances, bearing internal clearance, seal carrier to bearing housing mounting clearance, etc. The variable L is the maximum angulation related lateral shaft deflection at the dynamic sealing lip due to shaft articulation within clearances and due to side load-induced shaft bending. An overhanging side load will cause the shaft to pivot about the pivot point shown schematically above, until stopped by the reaction point provided by the bearing or bearings, resulting in an angulation related lateral shaft deflection L at the rotary seal. A non-overhanging side load will, if located between two bearings, also cause shaft bending that can cause an angulation related lateral shaft deflection L at the seal. To prevent seal-damaging heat from metal to metal contact at the shaft to seal carrier extrusion gap, the extrusion gap should be designed so that the lateral shaft deflection L cannot cause the shaft to contact the seal carrier at the extrusion gap.

If desired, worst-case tolerance accumulation can be analyzed statistically rather than with the simple arithmetic method used in Equations 4 to 9. One simple statistical assumption for tolerance accumulation is the Root Sum Squared method, where the likely total maximum tolerance accumulation is assumed to be the square root of the sum of the square of each individual tolerance[2]. The subject of statistical tolerance analysis is, however, beyond the scope of this handbook.

Qualitatively, higher values for C_{min} are more desirable for typical abrasive service applications. This simple truth is a good reason to use larger cross-section seals that permit higher nominal compression. It is also a good reason to design equipment that minimizes the effect that shaft deflection and runout has on seal compression. This is accomplished by designing to minimize deflection and runout at the rotary seal location, or by incorporating seal carriers that move laterally with shaft runout and deflection. With such designs, a C_{min} of at least 0.015” for seals with cylindrical static sealing surfaces and at least 0.025” for seals with a 0.010” static lip taper should be achievable.

Special purpose compression limits for elastomeric Kalsi Seals

The recommendations for the radial gland depth R that are provided on the website are not inviolable; they are merely based on knowledge of approximate minimum and maximum concentric compression that has been used in successful lab tests of elastomeric Kalsi Seals.

The radial gland depth R can sometimes be adjusted to achieve specific goals, particularly with larger radial cross section seals. For example, the radial gland depth R can be increased to compensate for the expanded radial seal depth that is caused by axial spring loading of a Kalsi Seal, in order to prevent the increased interfacial contact pressure and torque that are ordinarily associated with axial spring loading.

In uncooled applications where environmental exclusion is not an issue and shaft guidance is nearly perfect, nominal initial interference as low as 0.015” (0.38 mm) has been used successfully with 0.335” (8.51 mm) solid radial cross section elastomeric Kalsi Seals to reduce torque and seal generated heat. In a 65 psi (448.2 kPa) test with 0.335” (8.51 mm) Kalsi Seals and an ISO 320 viscosity grade lubricant, the seal running torque at 7.5% compression was 13% greater than it was at 4% compression at 175 rpm (126 ft/min), and 52% greater at 350 rpm (252 ft/min). With Dual Durometer Seals™ in a true-running application, reducing initial radial interference to 0.021” (0.53 mm) reduced torque and seal generated heat significantly (life was ultimately limited by compression set).

As another example, 0.031” (0.79 mm) initial compression has been used with 0.335” (8.51 mm) radial cross section Standard Kalsi Seals™ without apparent harm to provide

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2 See “**Mechanical Tolerance Stackup and Analysis**” by Bryan R. Fischer (CRC Press, 2011).

more initial interference to accommodate seal wear and compression set[3]. This practice is not recommended with Axially Constrained Kalsi Seals™.

In general, seals with larger radial cross-sections are preferred over seals with smaller cross-sections. At the same level of dimensional compression, a larger cross-section seal has less interfacial contact pressure than a smaller cross-section seal and will run cooler. At the same percentage of compression, a larger cross-section seal has more dimensional compression than a smaller cross-section seal, providing a more advantageous C_{min} value.

Large diameter seal-to-gland fit considerations

With large diameter rotary seals (typically over 6.30” (160.02 mm diameter), the manufacturing tolerance may result in a seal that has a loose fit within the groove prior to installation over the shaft. This condition can cause assembly difficulty by allowing the seal to miss the shaft installation chamfer. Using a seal inside diameter tolerance assumption of ±0.0065 inches per inch of diameter, and the seal cross sectional bilateral (±) tolerance from the website, determine whether or not the tolerance stackup can cause the seal to have a loose fit relative to the groove bore diameter G. If a loose fit can occur, make sure that the installation chamfer size and the assembly procedure can accommodate the variable seal position permitted by the loose fit, to assure that the seal will not be cut or twisted during installation.

The circumferential compression effect on the radial cross section

The radial compression of a Kalsi Seal causes circumferential compression, because radial compression also reduces the mean diameter of the seal. Circumferential compression is greater with smaller diameter rotary seals.

Finite element analysis of a 0.394” (10.00 mm) ID seal model was performed to evaluate the effect of circumferential compression on radial and axial seal dimensions. The simulation used a 0.730” (18.54 mm) diameter groove, with and without a 0.394” (10.00 mm) diameter shaft, at room temperature.

Without the shaft in place, the radial cross section increased by 0.0016” (0.04 mm), and the axial cross section increased by 0.0096” (0.24 mm). This shows that the cross section dimensional effect of the circumferential compression of small diameter seals is significant in the axial direction and relatively insignificant in the radial direction.

With the shaft in place, causing the seal model to be compressed radially, the predicted axial width of the seal increased by 0.0158” (0.40 mm) over its uncompressed width.

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3 Laboratory testing of 0.335" (8.51 mm) radial cross section Standard Kalsi Seals suggests that increasing the radial compression from 0.026" to 0.031" (0.66 to 0.79 mm) may improve abrasion resistance.

Inspecting the seal groove bore diameter

The use of a computer-controlled coordinate measurement machine is the best way to inspect seal groove diameters but is prohibitively expensive for many organizations. When cost is an issue, a parallel movement type groove measurement gage (Figure 3) and groove measurement calipers (Figure 4) are inexpensive alternatives when properly used by competent inspectors.

Figure 3
Groove measurement gage

A parallel movement type groove measurement gage is an inexpensive method of inspecting Kalsi Seal grooves. This model is produced by the Mueller Gages Company. Before use, the gage is set at the nominal groove diameter using an inside micrometer. During use, the dial indicator indicates how much the diameter of each groove deviates from the nominal diameter. Results depend on the operator's experience and skill.

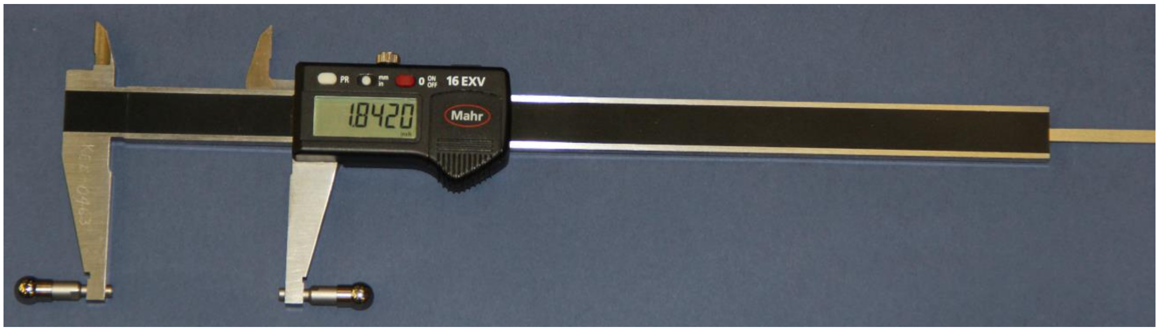

Figure 4
Groove measurement calipers

Groove measurement calipers are useful for inspecting seal grooves that are located near an end of a housing. The ball ends prevent groove corner radii from influencing the measurement.

4. How to determine Kalsi Seal groove width

Recommended groove widths are provided in the tables on our website. The basis for the groove width recommendations are described by footnotes to the tables. The website typically provides groove widths that are based on the temperature range of HNBR. For operation in higher temperatures, wider grooves are normally required; contact our technical support staff for assistance.

Groove width significance

The groove has to accommodate the installed width of the rotary seal, which is affected by these primary factors:

1. Seal material displaced axially by radial compression.
2. Seal material displaced axially by thermal expansion.
3. Volumetric swelling due to media exposure.
4. Seal tolerances.

If the groove width is too small, the seal-to-shaft interfacial contact pressure can increase. This may adversely affect seal lubrication, especially in high differential pressure sealing applications. To prevent seal roll-over during installation, and to minimize seal misalignment and wear in applications having little or no differential pressure or low levels of reversing pressure, the groove width should not be excessive.

Custom groove width information

For those customers who design groove widths for special conditions, Appendix 2 shows width predictions for Standard Kalsi Seals as a function of temperature, diameter, and tolerance. To a certain extent, the results can be interpolated for seals with non-standard cross sections. Size the groove width for the highest anticipated seal temperature.

The information in Appendix 2 is the result of extensive finite element analysis, not all of which could reasonably be included in this handbook. Call Kalsi Engineering, Inc. when additional width prediction information is needed.

5. Groove wall thickness

The LMC thickness of the groove wall on the low pressure side of a Kalsi Seal should be designed using typical pressure vessel design practices[4], to enable it to safely withstand the forces to which it will be exposed, including differential pressure loads and mechanical loads. Consider equipment failure modes and any unusual transient pressure conditions, and design for resulting loads. This can be accomplished with finite element analysis, or with closed-form solutions such as those found in the book "**Roark's Formulas for Stress and Strain**"[5].

6. Do I need a removable groove wall?

Many Plastic Lined Seals™, BDRP seals, and small diameter elastomeric seals require a removable groove wall (Figure 5) to permit installation of the seal into the groove. As shown in Figure 5, many Plastic Lined Seals and BDRP Seals also require an installation chamfer at the mouth of the groove bore, to ease the seal into the groove.

Factors controlling the installability of elastomeric seals into one-piece grooves are explained in handbook chapter D6. For information regarding the installability of plastic-elastomer composite seals in once-piece grooves, see the handbook chapters about those seals, or contact Kalsi Engineering.

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4 For example, see the ASME Boiler and Pressure Vessel Code.

5 Published by McGraw-Hill; available with companion software.

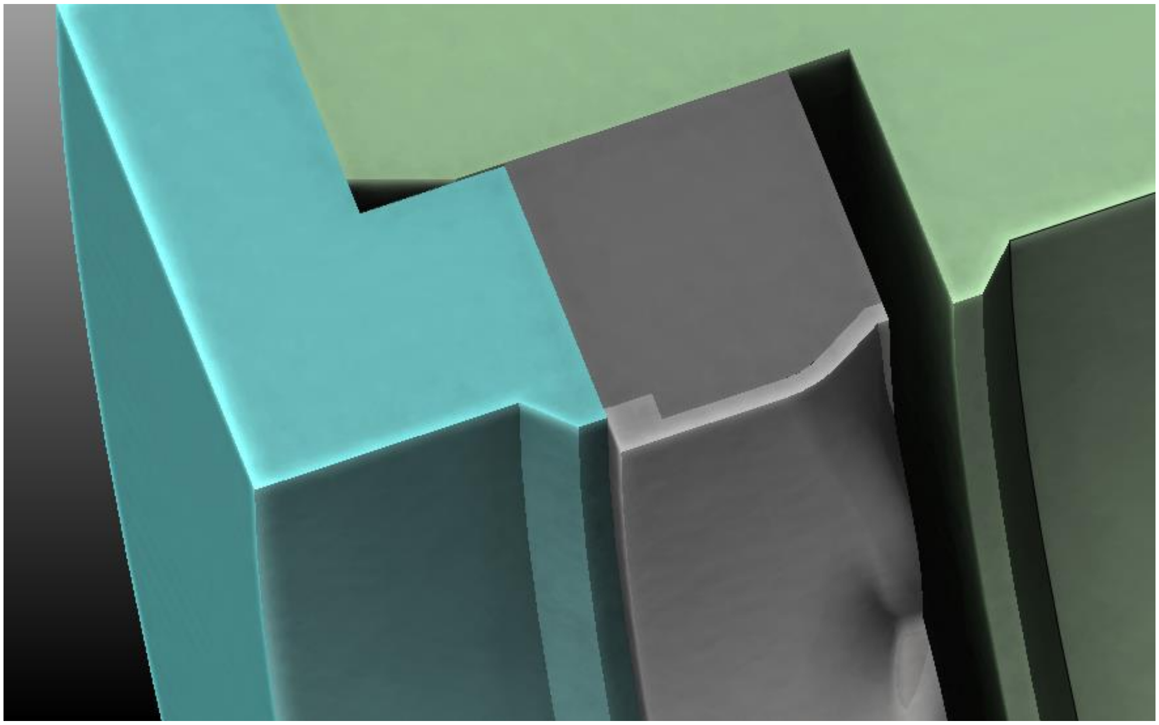

Figure 5
Example of a removable groove wall

Many Plastic Lined Seals, BDRP seals, and small diameter elastomeric seals require a removable groove wall to enable installation of the seal into the housing groove. The chamfer at the mouth of the groove bore is required with smaller diameter BDRP and Plastic Lined Seals and eases the seal into the groove.

7. Special considerations for BDRP Seals

Because of the stiffness of the plastic liner, most sizes of BDRP Seals require a removable groove wall (Figure 6) to permit the seal to be installed into the groove without damaging the liner. Smaller diameter BDRP Seals require an installation chamfer at the open end of the groove bore, as shown in Figure 6, to ease the seal into the groove. With larger diameter BDRP Seals the installation chamfer may not strictly be required but is a significant convenience.

BDRP Seals can be used for one direction of lubricant pressure or for reversing lubricant pressure. With one direction of pressure, only one groove wall serves as a low pressure groove wall. With reversing lubricant pressure, the groove walls take turns serving as the low pressure groove wall, as shown in Figure 6. Likewise, with reversing lubricant pressure, both housing-to-shaft gaps take turns serving as extrusion gaps, as shown in Figure 6, and are sized accordingly.

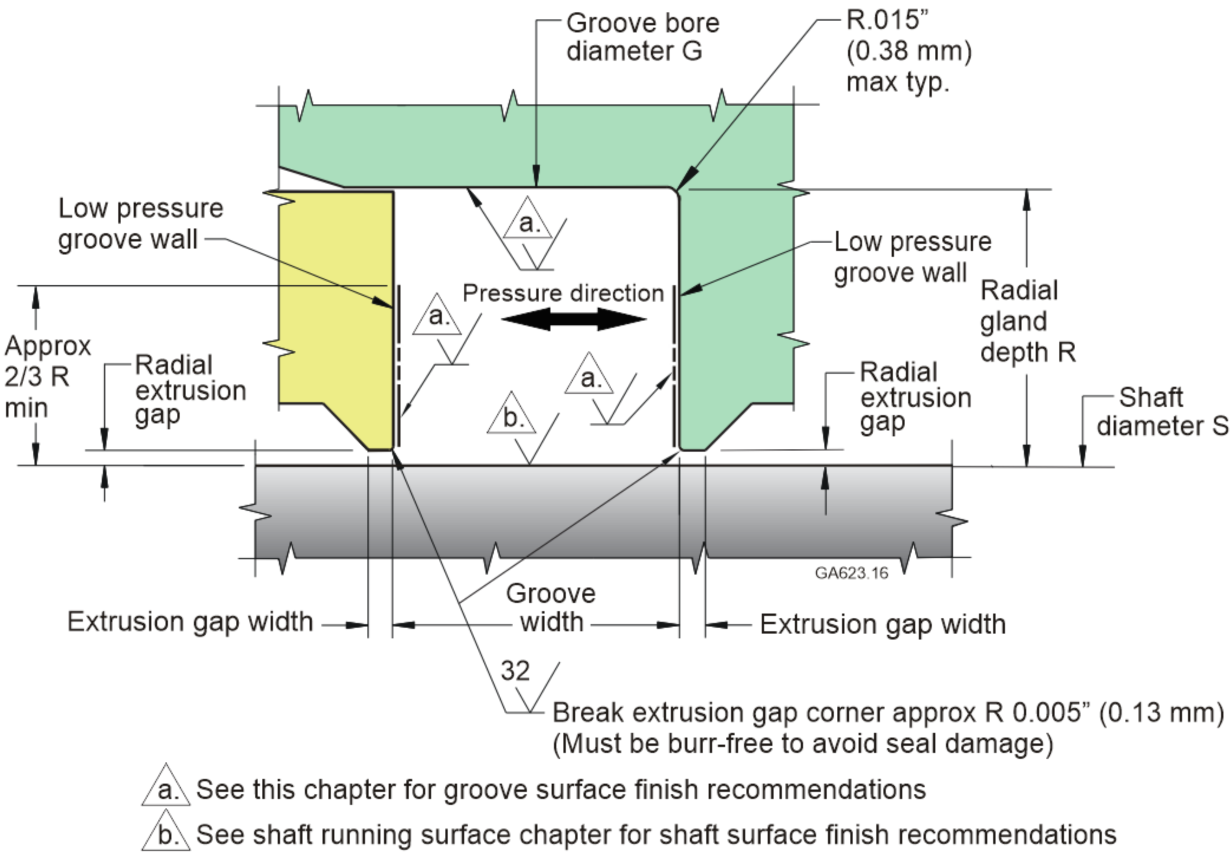

Figure 6
Special gland considerations for a BDRP Seal

Due to the stiffness of the plastic liner, most sizes of BDRP Seals require a removable groove wall. On smaller diameter BDRP Seals, an installation chamfer is required at the mouth of the groove bore to ease the seal into the bore. The radial depth of the installation chamfer should be no less than the radial compression of the seal. If a BDRP Seal is used for reversing differential pressure service conditions, both groove walls serve as low pressure groove walls, and both housing-to-shaft gaps serve as extrusion gaps.

8. Groove wall surface finish

Figure 1 shows a groove wall on the low pressure side of the groove, and a groove wall on the high pressure side of the groove. As shown by Figure 6, in BDRP Seal implementations with reversing pressure, each groove wall serves intermittently as a low pressure groove wall and should be treated as such in terms of surface finish.

Surface finish of a low pressure groove wall

Differential pressure acting across the rotary seal forces the seal against the low pressure groove wall, and dynamic shaft runout causes the seal to slide radially with respect to the low pressure groove wall. The inner 2/3 of a low pressure groove wall should be smooth so this radial sliding can freely occur. A surface finish roughness of 32 microinches[6] AA or smoother is recommended.[7] The outer third of a low pressure groove wall can be rougher if desired, because less sliding occurs there.

Surface finish of the high pressure groove wall

In applications with non-reversing differential pressure, the seal typically only contacts the low pressure groove wall. In such cases, surface finish of the high pressure groove wall is irrelevant to seal performance.

Special considerations for Axially Constrained Seals™

Axially Constrained Kalsi Seals contact both groove walls simultaneously and are often subjected to low levels of reversing pressure. With such seals, the groove wall on the environment side of the seal should be treated as a low pressure groove wall when determining surface finish. With Axially Constrained Seals, the effect the surface finish of the groove wall on the lubricant side of the seal has not been evaluated.

Avoid extrusion gap corner roughness

Care should be taken to prevent damage to the extrusion gap corner (see Figures 1 and 6) that is located between a low pressure groove wall and the bore that defines an extrusion gap. A rough extrusion gap corner can significantly accelerate pressure-related extrusion damage and slipping related damage to the rotary seal. A corner radius of approximately 0.005” (0.13 mm) is normally recommended.

9. The groove inside corner radius

The 0.015” inside corner radius recommendation that is provided in Figure 1 is based on finite element analysis using Axially Constrained Kalsi Seals, which contact both gland walls simultaneously. The analysis indicated that the 0.015” corner radius did not affect the stability of the Axially Constrained Seals.

With seals that do not contact the lubricant side groove wall, the radius of the lubricant side inside corner radius can be somewhat larger than 0.015”.

10. Housing-to-shaft clearance and length

In Figure 1, the housing-to-shaft clearance on the low pressure side of the seal groove is referred to as the extrusion gap, and the clearance on the high pressure side is referred to as the pressure side gap. The extrusion gap gets its name from the fact that high differential

6 Multiply microinches by 0.0254 to obtain micrometers (µm).

7 The surface roughness of the various surfaces of the seal groove can be difficult or impossible to inspect with a profilometer. The surface roughness can, however, be estimated with reasonable accuracy by visual comparison with electro-formed surface roughness comparison standards.

pressure tends to extrude seal material into the shaft-to-housing clearance. Information on sizing the extrusion gap clearance and extrusion gap length are provided in a separate chapter. In BDRP™ applications with reversing pressure directions, the gaps on both sides of the groove serve intermittently as extrusion gaps and are treated as such from a design standpoint.

In general, the size and width of both housing-to-shaft clearance gaps depends on their function. For example, if the gap is on the low pressure side of a seal, then smaller clearance typically results in less extrusion damage to the seal in high differential pressure conditions, and larger clearance results in less third body abrasion damage to the seal and shaft in low differential pressure conditions if the low pressure fluid contains abrasives. On the other hand, if the clearance is so small that housing-to-shaft metal-to-metal contact occurs, seal and housing damage may occur, and enough heat may be generated to destroy the seal, especially if the gap width is short.

If the gap is on the lubricant side of the groove, then larger clearance makes the groove easier to fill during the lubricant filling process and allows better seal cooling. If there is a potential operating condition that can cause high differential pressure to temporarily force the seal against the lubricant side groove wall (such as the wellbore annulus blockage that can occur with rotary steerable oilfield tools), then this potential operating condition may dictate that the gap be small, to help the seal resist extrusion damage from the temporary pressure condition.

If it is critical that lubricant pressure be communicated through a gap to the rotary seal as quickly as possible, then the width of the gap should be minimized. A narrower gap width is also preferable if the gap is exposed to abrasives, to minimize hydraulic effects that can accelerate third body abrasion of the seal. A narrower gap width is also desirable from a shaft cooling standpoint.

If the bore that establishes the gap serves as the journal bearing of a laterally movable seal housing, then journal bearing design practices dictate the clearance and width of the gap. This journal bearing usage does not obviate other potential concerns, such as ease of filling and rapid transmission of lubricant pressure to the rotary seal. As such, a pressure distribution groove may be required near the seal groove, with pressure communicated to the distribution groove via axial or radial passageways.

In summary, when designing gap clearance and width, observe Sullivan’s dictum: Form follows function. For assistance, contact our engineering support staff.

11. How seal groove location influences maintenance

During equipment maintenance, the used seals must be extracted without damaging the seal housing, and the seal groove must be wiped dry before new seals are installed. The ease of seal extraction and insertion is influenced by how far the groove is from the end of the housing, and the proportions of the rotary seal.

If the groove is located any significant distance from the end of the housing, and/or if the inside diameter of the seal is small, a custom seal extraction tool may be required; plan accordingly. Since the Kalsi Seal is ordinarily not reused, the extraction tool can intentionally damage the seal in order to engage and extract it. For one example, elements of the extraction tool can jab into the seal to grip it for extraction purposes. For another example, the extraction tool can cut partly through the seal to enable the seal to fold more readily for ease of extraction.

12. Circumferential seal slippage is undesirable

Circumferential seal slippage can sometimes occur, particularly in applications where differential pressure is low, or with large diameter rotary seals. Such slippage can result in wear to the seal and the groove, while generating unnecessary heat[8]. In high differential pressure conditions, circumferential seal slippage can also accelerate extrusion damage, particularly if the extrusion gap corner is not smooth.

Reasons why circumferential seal slippage may occur

Circumferential seal slippage may be related to:

- A smooth finish on the groove bore diameter,
- Lubricating the groove and/or the outside diameter of the seal prior to seal installation into the groove (Figure 7),
- Low lubricant viscosity,
- A rough shaft surface finish, and/or
- Wear of the dynamic sealing lip, which causes additional torque.

Circumferential seal slippage may also be influenced by the sudden accelerations that occur with drill string stick-slip[9] in oilfield applications like mud motors and rotary steerable tools, however the well annulus mud flow helps to prevent slippage related seal overheating.

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8 Such slippage related housing heat is believed to cause fewer problems in applications where the seal housing is actively cooled (such as mud motors, which are exposed to both bore and annulus flow).

9 For information on stick-slip, see SPE 145910, "**Drill Pipe Measurements Provide Valuable Insight into Drill String Dysfunctions**". In addition to acceleration, the slip phase creates severe drillstring vibration.

Influence of the seal installation lubricant

Seals installed with a higher viscosity seal installation lubricant, such as an ISO 1,000 viscosity grade lubricant, are less likely to slip than seals installed with a low viscosity lubricant such as hydraulic oil because less of the oil squeezes out of the seal-to-shaft interface over time. In room temperature startup testing, we observed less instances of slippage when we applied Valvoline multi-service grease to the inner surface of the dynamic lip as an installation lubricant.

Do not lubricate the seal or the groove prior to seal insertion

Do not lubricate the seal or the groove prior to insertion of the seal into the groove, because the lubricant will dramatically lower seal resistance to slippage (Figure 7).

Figure 7
Do not lubricate the outside diameter of the Kalsi Seal!

Avoid using soft metals for seal carriers

Ideally, to prevent slippage related seal carrier wear, the groove should not be machined into soft metals such as brass. Circumferential seal slippage causes more wear to soft metals than it does to hard metals. Soft metals may also wear more rapidly at the extrusion gap, if abrasives are present in the environment.

Influences of seal design

Circumferential seal slippage becomes more of a problem with larger diameter rotary seals, because the ratio between the inside and outside diameters becomes relatively small. In small diameter seals, this ratio is relatively large, and the static sealing lip therefore has a greater mechanical advantage over the dynamic sealing lip (friction times radius) and can more effectively resist slippage. Seals with smaller radial cross-sections are also more prone to slippage.

Seal designs that provide increased lubrication, such as Hybrid and Enhanced Lubrication Seals are demonstrably less prone to circumferential slippage, compared to other Kalsi Seal designs. Seals that have been treated with our low friction surface treatment also tend to slip less.

Surface finish of the groove bore

To inhibit circumferential seal slippage, the surface finish of the groove bore (see variable G on Figure 1) should be significantly rougher than the sealing surface of the shaft. One way to achieve a rough surface on the cylindrical bore of the groove is to roughen the forming tool that is used to cut the groove.

Grit blasting to inhibit circumferential seal slippage

One way to inhibit circumferential seal slippage is grit blasting of the groove bore to achieve a surface profile of 0.0010” to 0.0015” (0.025 to 0.038 mm) peak to peak having a roughness of approximately 110 µin (2.79 µm) AA (see Figure 8). This helps the seal achieve a mechanical interlock with the surface texture, which discourages slippage.

Blasting media can be any angular abrasive particle such as steel grit, aluminum oxide, sand, etc. Shot or glass bead blasting will not produce acceptable results. The correct surface profile can be achieved using steel grit with a hardness range of Rockwell C 55/62 and an SAE G50 grit size (0.30 to 0.71 mm). SAE G50 steel grit is shown in Figure 9. After grit blasting, thoroughly clean the component to avoid contaminating the seal lubricant with remnants of the abrasive grit. A grit blasting cabinet can be used to contain the grit during the blasting process (Figure 10). Nitrided surfaces cannot be roughened by grit blasting.

Do not apply the grit blast to the low pressure groove wall. The seal must be free to slide radially against the low pressure groove wall in response to lateral shaft motion (such as dynamic runout). Smoothness is particularly important at the inner two-thirds of the low pressure groove wall, and at the extrusion gap. (Most of the radial sliding between the seal and the low pressure groove wall occurs at the inner two-thirds of the groove wall.) During the grit blasting process, mask the low pressure groove wall to preserve its smoothness (Figure 11).

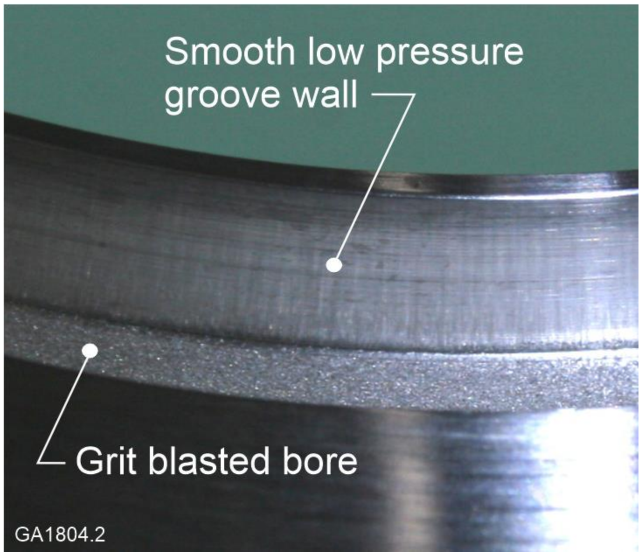

Figure 8
Example of a grit blasted groove bore

Grit blasting can be used to increase friction between the rotary seal and the groove. Keep at least the inner two-thirds of the low pressure groove wall smooth, so the seal can slide radially with respect to the groove wall in response to shaft runout. Clean the part thoroughly after the grit blast operation to avoid contaminating the seal and bearing lubricant.

Figure 9
SAE G50 steel grit

Figure 10
A grit blasting cabinet contains the grit blasting media

A grit blasting cabinet contains the grit blasting media and allows the blasting process to be performed indoors.

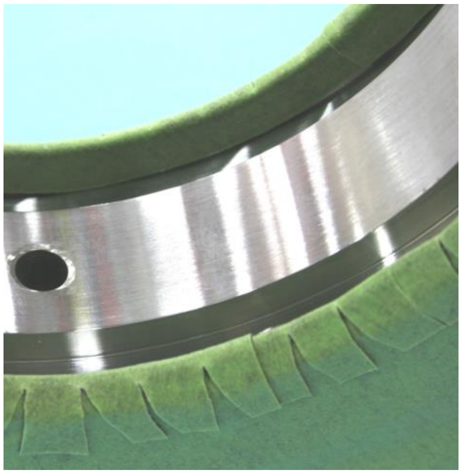

Figure 11
Masking of the low pressure groove wall prior to grit blasting

This photo shows the double layer technique we use to mask the low pressure groove walls of our test fixture seal carriers prior to grit blasting. This tape was selected for its ability to be easily photographed. Ordinarily we use a black colored polyurethane abrasion-resistant surface-protection tape.

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Masking is required to grit blast grooves on existing seal carriers

In existing equipment, masking is required during the grit blasting process to protect the low pressure groove wall. We mask our seal carriers using a double layer of polyurethane abrasion-resistant surface-protection tape. Figure 10 shows our masking technique, using a different type of tape which is easier to photograph.

Masking can be avoided when manufacturing new seal carriers

In equipment that is being newly manufactured, it is permissible to machine the groove bore to finished dimensions but leave extra material at the low pressure groove wall. After grit blasting, finish machine the low pressure groove wall to the final groove width dimension (Figure 12).

Grit blasting is not recommended for reversing pressure conditions

Grit blasting is not ordinarily recommended for abrasive applications where reversing pressure may occur because it encourages temporary twisting of the seal cross section as the pressure reverses, which is detrimental to environmental exclusion. Grit blasting is also not recommended when Filled Kalsi Seals™ are employed, because leakage may occur as a result of the low lip contact force and relatively stiff seal body material.

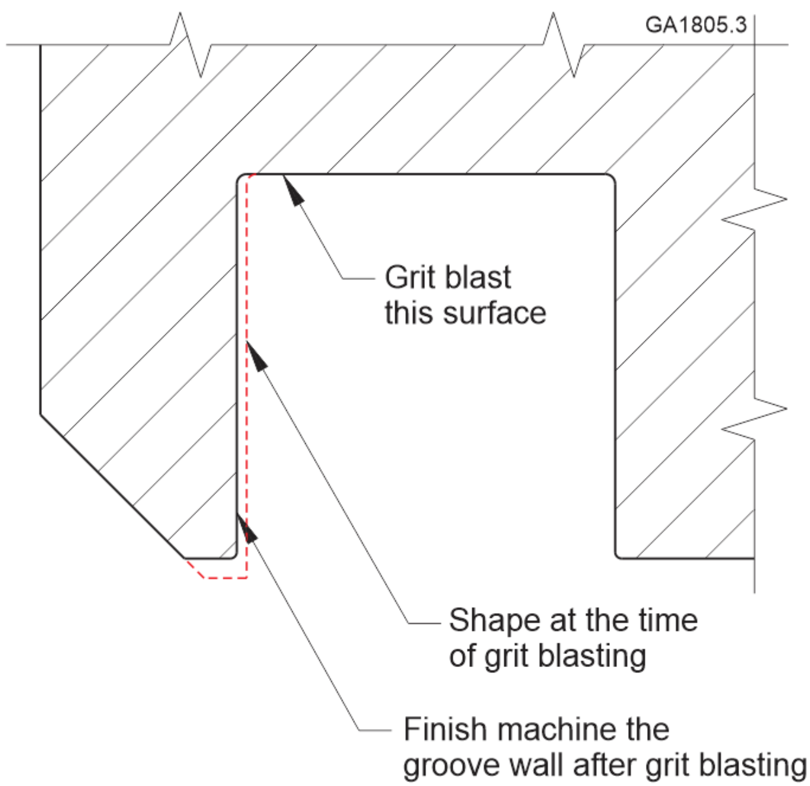

Figure 12
Method to avoid masking during component manufacturing

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Knurling seal grooves to address slippage

Testing has shown that a 0.0005” to 0.0020” (0.013 to 0.051 mm) deep 19 to 28 pitch[10] axially oriented straight knurl on the groove bore inhibits circumferential slippage of direct compression-type Kalsi Seals, even in reversing pressure conditions. The straight knurl (Figure 13) allows the seal to shuttle axially in response to the reversing pressure, while providing a mechanical grip for the static sealing lip in the circumferential direction.

Knurl depth accuracy (which is difficult to achieve and measure) is critical to leak prevention in cases where the knurl extends to the low pressure groove wall. For this reason, knurling is only recommended for seals that have a lip width equal to or wider than “Wide Footprint” Seals™.[11] Leave a 0.06” (1.52 mm) wide unknurled band near the low pressure groove wall to ensure a sealing band. With seals having a relatively wide static lip width, even if the seal shuttles to the lubricant side groove wall due to reversing pressure, part of the static lip still engages the unknurled surface. Test for sealing integrity with air pressure before employing vacuum filling techniques. Figure 14 shows a suitable way to indicate knurling on a part drawing.

Knurling is not recommended when filled Kalsi Seals are employed, due to relatively low contact force between the seal and the groove bore.

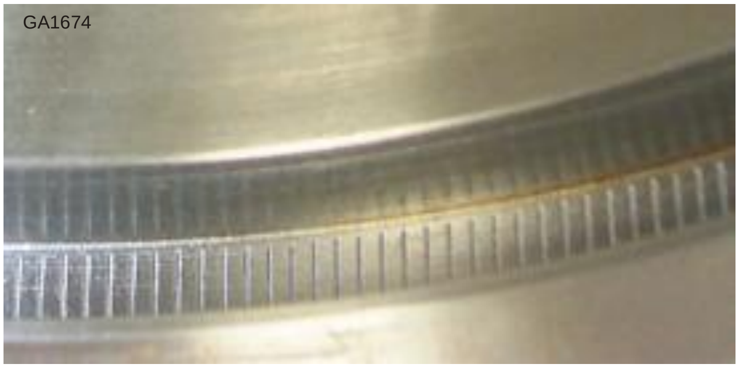

Figure 13
Knurling for Wide Footprint Seals

Seal slippage can be minimized by using a light 0.0005” to 0.0020” (0.013 to 0.051 mm) deep straight knurl. Stop the knurl 0.06” (1.52 mm) from the low pressure groove wall (this photo is looking at the high pressure groove wall). Knurling is preferred over grit blasting in applications where reversing pressure acts across the seal. Pressure test after assembly to verify sealing integrity. Do not use knurling with seals having a static lip width less than 0.11” (2.79 mm).

10 19 to 28 knurls per inch of circumference.

11 Knurling is not appropriate for Axially Constrained Seals or for seals that have the standard dynamic lip width, because these seals have a relatively narrow static lip.

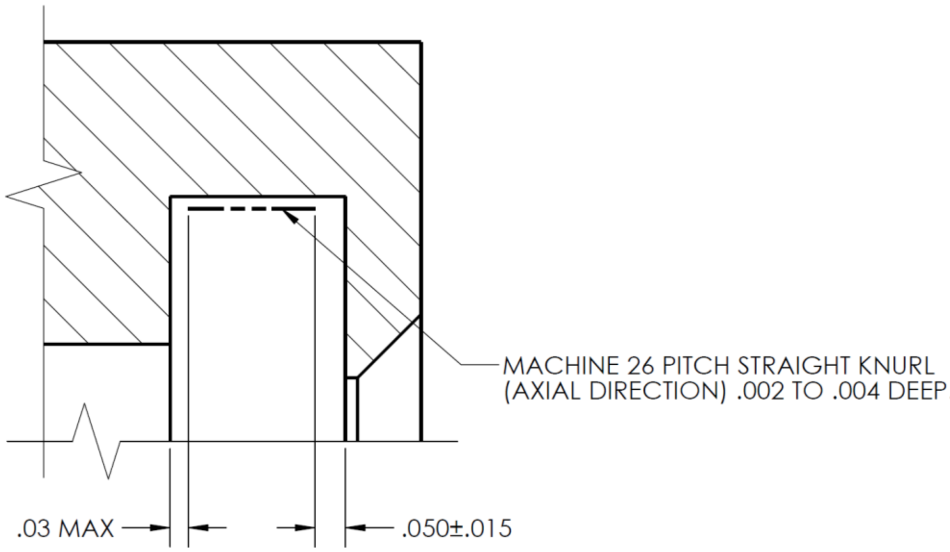

Figure 14
Drafting detail for straight knurl

Other methods of inhibiting seal slippage

Lubricant pressure and/or axial spring loading (Chapter D9) can be used to increase friction on the flank(s) of the seal to inhibit circumferential slippage. Lubricant pressure and/or seal spring loading also significantly reduces the abrasive wear of the dynamic sealing lip that increases torque and promotes seal slippage. With Axially Constrained Seals (Chapter C4), which contact both groove walls simultaneously in service, an anti-rotation benefit may be realized by grit blasting the low pressure groove wall.

If the seal is axially spring loaded, grit blasting or other surface roughening techniques can also be used on the face of the spring follower washer to inhibit slippage. In such cases it may be desirable to key the spring follower washer to the housing to prevent the backup washer from rotating in response to rotary seal torque.

Anti-rotation tangs for non-reversing pressure applications

In applications where no significant reversing pressure occurs, seals can optionally be provided with molded anti-rotation tangs that engage mating anti-rotation features in the groove to prevent seal slippage. Figures 15 and 16 show anti-rotation tangs engaging radially oriented anti-rotation pins mounted within the groove. The seal in Figure 15 is for applications where the lubricant pressure is greater than the environment pressure. The seal in Figure 16 is for applications where the environment pressure is moderately greater than the lubricant pressure. Although the groove anti-rotation features are illustrated as radial pins in Figures 15 and 16, the features can take other forms, such as recesses in the groove wall that receive the anti-rotation tangs of the seal.

If you select a Kalsi Seal with anti-rotation tangs, contact our staff for gland design details. If you are not sure you need anti-rotation tangs, leave enough axial length (0.140”) in your design to accommodate them.

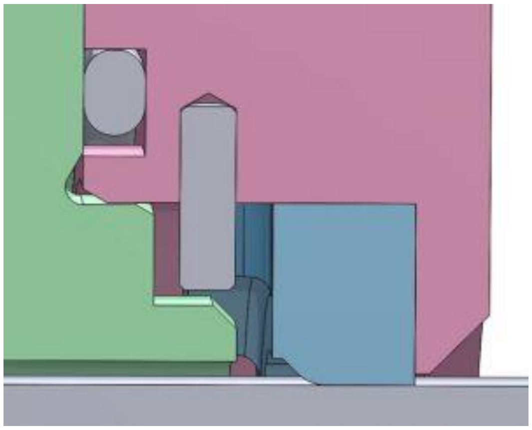

Figure 15
Lubricant-side anti-rotation tangs

In applications where the pressure of the environment never exceeds the pressure of the lubricant, lubricant-side anti-rotation tangs can be employed that engage mating features in the groove to prevent rotational slippage. In these images, the anti-rotation tangs of the seal engage radially oriented pins in the groove bore (patent pending). If desired, recesses in the groove wall can be used instead of the pins (patented).

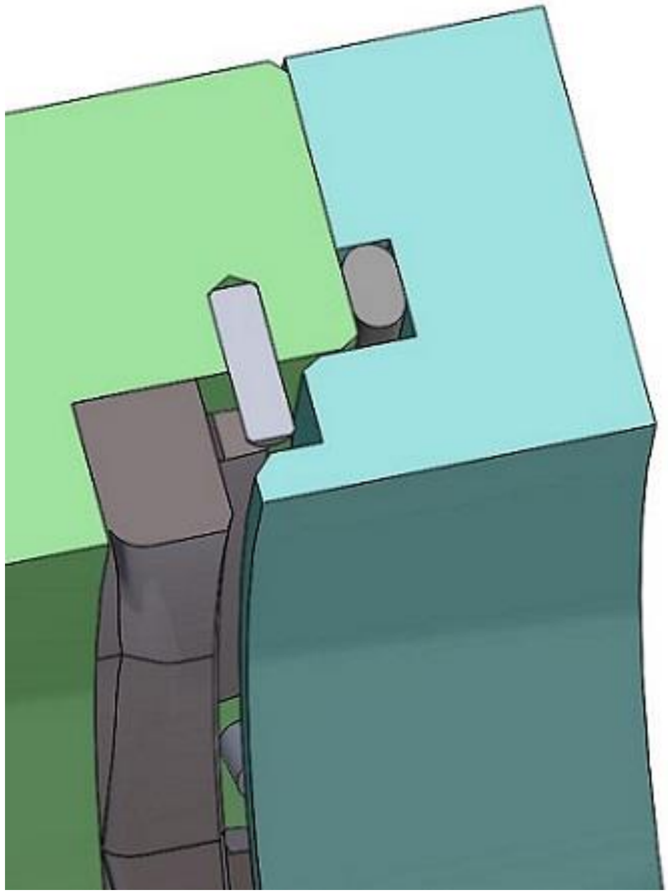

Figure 16
Environment-side anti-rotation tangs

In applications where the pressure of the environment is moderately greater than the pressure of the environment, environment-side anti-rotation tangs can be used that engage radial pins to prevent rotational slippage (patent pending).