Chapter D5

Kalsi Seal gland guidelines

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

NOTICE: The information in this chapter is provided under the terms and conditions of the Offer of Sale, Disclaimer, and other notices provided in the front matter of this handbook.
1. **Introduction**

The Kalsi Seal “gland” is the enclosed space defined by the housing groove and the rotary shaft. Figure 1 identifies gland dimensions and surface finishes.

![Figure 1: Kalsi Seal gland detail](image)

Surface finishes in this figure are in microinches (µin). Multiply microinches by 0.0254 to obtain micrometers (µm). For groove bore diameter G use Equation 1 and Table 1. For groove width and tolerance see the Kalsi Engineering website. Very small diameter seals require a removable groove wall to permit seal insertion, particularly when the seals are made from harder elastomers. Note that 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. Don’t forget to specify a position or concentricity requirement for groove bore diameter G and the bore that defines the extrusion gap clearance.

The gland determines the amount of radial and axial compression the seal will experience in operation. Radial seal compression physically blocks the leakage path. Under static conditions, Kalsi Seals function in a manner similar to conventional O-rings. During rotation, the seal hydroplanes on a thin film of lubricant, as described in Chapter A1. 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. Kalsi Seal radial tolerances are provided on our website.

2. **Determining the groove bore diameter**

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 Table 1:

**Equation 1**, Nominal groove bore diameter:

$$G_{nom} = S_{nom} + 2 \times R_{nom}$$

**Example 1**, where $S_{nom} = 2.75''$, $R_{nom} = 0.309''$:

$$G_{nom} = 2.75'' + 2 \times 0.309'' = 3.368''$$

<table>
<thead>
<tr>
<th>Seal radial cross-section depth D (inches)</th>
<th>Radial gland depth R (inches) Min</th>
<th>Nom</th>
<th>Max</th>
<th>Radial gland depth R (millimeters) Min</th>
<th>Nom</th>
<th>Max</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.145</td>
<td>0.123</td>
<td>0.125</td>
<td>0.127</td>
<td>3.125</td>
<td>3.175</td>
<td>3.225</td>
</tr>
<tr>
<td>0.186</td>
<td>0.164</td>
<td>0.166</td>
<td>0.168</td>
<td>4.166</td>
<td>4.216</td>
<td>4.266</td>
</tr>
<tr>
<td>0.212</td>
<td>0.186</td>
<td>0.189</td>
<td>0.192</td>
<td>4.720</td>
<td>4.800</td>
<td>4.880</td>
</tr>
<tr>
<td>0.270</td>
<td>0.238</td>
<td>0.244</td>
<td>0.250</td>
<td>6.048</td>
<td>6.198</td>
<td>6.348</td>
</tr>
<tr>
<td>0.300</td>
<td>0.268</td>
<td>0.274</td>
<td>0.280</td>
<td>6.810</td>
<td>6.960</td>
<td>7.110</td>
</tr>
<tr>
<td>0.335 &amp; 0.345</td>
<td>0.303</td>
<td>0.309</td>
<td>0.315</td>
<td>7.699</td>
<td>7.849</td>
<td>7.999</td>
</tr>
<tr>
<td>0.415</td>
<td>0.372</td>
<td>0.380</td>
<td>0.388</td>
<td>9.452</td>
<td>9.652</td>
<td>9.852</td>
</tr>
</tbody>
</table>
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 2:

**Equation 2,** Minimum groove bore diameter:

\[ G_{\text{min}} = G_{\text{nom}} - G_t \]

**Example 2,** where \( G_{\text{nom}} = 3.369" \), \( G_t = 0.002" \):

\[ G_{\text{min}} = 3.368 \cdot 0.002 = 3.366 \]

**Equation 3,** Maximum groove bore diameter:

\[ G_{\text{max}} = G_{\text{nom}} + G_t \]

**Example 3,** where \( G_{\text{nom}} = 3.369" \), \( G_t = 0.002" \):

\[ G_{\text{min}} = 3.368 + 0.002 = 3.370 \]

<table>
<thead>
<tr>
<th>Seal radial cross-section depth D (inches)</th>
<th>Suggested bilateral groove tolerance ( G_t ) (inches)</th>
<th>Groove diameter bilateral (±) tolerance ( G_t ) (inches)</th>
<th>Groove diameter bilateral (±) tolerance, ( G_t ) (millimeters)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.145</td>
<td>0.0005</td>
<td>0.013</td>
<td>0.030</td>
</tr>
<tr>
<td>0.186</td>
<td>0.0010</td>
<td>0.030</td>
<td>0.040</td>
</tr>
<tr>
<td>0.212</td>
<td>0.0010</td>
<td>0.030</td>
<td>0.040</td>
</tr>
<tr>
<td>0.270</td>
<td>0.0015</td>
<td>0.040</td>
<td>0.050</td>
</tr>
<tr>
<td>0.300</td>
<td>0.0015</td>
<td>0.040</td>
<td>0.050</td>
</tr>
<tr>
<td>0.335 &amp; 0.345</td>
<td>0.0020</td>
<td>0.050</td>
<td>0.070</td>
</tr>
<tr>
<td>0.415</td>
<td>0.0030</td>
<td>0.080</td>
<td>0.100</td>
</tr>
</tbody>
</table>

After sizing the groove bore diameter, perform a tolerance and clearance stackup using Equations 4, 5, and 6 to verify that the minimum and maximum size of the radial gland depth \( R \) falls within the range shown in Table 1. If higher or lower radial
compression is desired, see the “Special purpose compression limits for Kalsi Seals” information below.

The minimum and maximum values for radial gland depth R in Table 1 do not represent the limits of gland depth variation resulting from tolerance accumulation alone, with an assumption of concentric components. Rather, they represent the worst-case eccentric gland depth resulting from lateral shaft deflection under load, accumulated tolerances, and articulation within mounting clearances. To determine nominal, minimum and maximum radial gland depth, use Equations 4, 5, and 6 respectively in conjunction with Figure 2 (equation variables are given 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:

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

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

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

**Equation 6**, Maximum radial Kalsi Seal gland dimension:

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

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

\[
R_{max} = \frac{3.370 - 2.7495}{2} + \frac{0.006}{2} + \frac{0.0015}{2} + 0.001 = 0.315
\]

\(R_{max}\) and \(R_{min}\) fall within the recommended values given in Table 1. In instances where the calculated values for \(R_{max}\) and \(R_{min}\) do not fall within the recommended values given in Table 1, the designer needs to reconsider shaft and groove tolerances, bearing clearances, etc.
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 $R_{\text{min}}$ and $R_{\text{max}}$. 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. $M_{\text{max}}$ is the accumulative diametric mounting clearance of the seal groove relative to the rotary shaft. For rolling element bearings, $M_{\text{max}}$ 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.


Special purpose compression limits for Kalsi Seals

The recommendations for the radial gland depth R that are given in column 2 of Table 1 are not sacrosanct; they are merely based on knowledge of approximate minimum and maximum concentric compression that has been used in successful lab tests. The interpretation of the tests is that if the compression values worked in nominally concentric conditions, they will also work when the minimum and maximum compression occurs only locally due to an eccentric condition. Another point of reference in establishing the radial gland depth recommendations in certain cases is the degree of rotary shaft guidance that is practicably achievable.

The radial gland depth R can 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 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 more initial interference to accommodate seal wear and compression set1.

As another example, Kalsi Engineering can provide Hybrid or zigzag HNBR Wide Footprint Kalsi Seals in a non-standard 0.359” x 0.230” cross-sectional size that functions well in a conventional groove having 0.309” radial gland depth and a 0.289” groove width. Such seals, which use an HNBR that is compatible with the high compression, provide extra compressive margin to help deal with the high runout that is found in some applications, such as oilfield sealed bearing mud motors.

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1 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.
To determine average, minimum and maximum radial Kalsi Seal compression, use Equations 7, 8, and 9 respectively (Equation variables are defined in Appendix 3).

**Equation 7.** Average radial Kalsi Seal Compression:

\[
C_{\text{nom}} = D_{\text{nom}} - R_{\text{nom}}
\]

*Example 7*, where \(D_{\text{nom}} = 0.335"\), \(R_{\text{nom}} = 0.309"\):

\[
C_{\text{nom}} = 0.335 - 0.309 = 0.026"
\]

**Equation 8.** Minimum Radial Kalsi Seal Compression:

\[
C_{\text{min}} = D_{\text{min}} - R_{\text{max}}
\]

*Example 8*, where \(D_{\text{min}} = 0.330"\), \(R_{\text{max}} = 0.315"\):

\[
C_{\text{min}} = 0.330 - 0.315 = 0.015"
\]

**Equation 9.** Maximum Radial Kalsi Seal Compression:

\[
C_{\text{max}} = D_{\text{max}} - R_{\text{min}}
\]

*Example 9*, where \(D_{\text{max}} = 0.340"\), \(R_{\text{min}} = 0.3035"\):

\[
C_{\text{max}} = 0.340 - 0.3035 = 0.0365"
\]

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.

**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 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.

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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 circumferential compression is significant in the axial direction in small diameter seals, but is relatively insignificant in the radial direction.

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

**Large diameter seal-to-gland fit considerations**

With large diameter seals (typically over 9” (228.6 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 (see Chapter D3).

### 3. Determining 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 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 seal width predictions 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.

### 4. Groove wall thickness

The LMC thickness of the environment side groove wall should be designed using typical pressure vessel design practices\(^3\), 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”\(^4\).

### 5. Environment side groove wall surface finish

The environment side groove wall (see Figure 1) should be relatively smooth over the first 2/3 of the gland wall nearest the extrusion gap so that the seal can slide radially with ease at that location, in response to dynamic runout.\(^5\) The remaining portion of the groove wall can be rougher to combat circumferential seal slippage, as described below.

Care should be taken to prevent damage to the extrusion gap corner (see Figure 1) that is located between the environment side groove wall and the extrusion gap bore. A rough extrusion gap corner can significantly accelerate extrusion damage of the rotary seal. A corner radius of approximately 0.005” (0.13 mm) is normally recommended.

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\(^3\) For example, see the ASME Boiler and Pressure Vessel Code.

\(^4\) Published by McGraw-Hill; available with companion software.

\(^5\) 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.
6. **Circumferential seal slippage is undesirable**

Circumferential seal slippage can sometimes occur, particularly in low differential pressure seal installations, or with large diameter rotary seals. Such slippage can result in wear to the seal and the groove, while generating unnecessary heat\(^6\). In high differential pressure conditions, circumferential seal slippage can also accelerate extrusion damage, particularly if the extrusion gap corner is not smooth.

**Causes of circumferential seal slippage**

Circumferential seal slippage is usually related to:

- An excessively smooth finish on the groove bore diameter,
- Lubricating the groove and/or the seal prior to seal installation into the groove (Figure 11),
- Inadequate lubricant viscosity, particularly in high differential pressure applications,
- An excessively 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 drillstring stick-slip\(^7\) in oilfield applications like mud motors and rotary steerable tools, however the well annulus mud flow helps to prevent slippage related seal overheating.

**Influences of seal design**

Circumferential seal slippage becomes more of a problem with larger diameter 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. This advantage can be increased by only treating the inside diameter of the seal, which is more expensive than an all-over treatment.

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\(^6\) 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).

\(^7\) 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.
**Surface finish of the groove bore**

To inhibit circumferential seal slippage, the surface finish of the cylindrical bore of the groove (see Figure 1 variable G) should be significantly rougher than the rotary shaft\(^8\). 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.

7. **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 3). The seal achieves 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 4. 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 5).

**Keep the environment side groove wall smooth**

Do not apply the grit blast to the environment side groove wall. The seal must be free to slide radially against the environment side groove wall in response to lateral shaft motion (such as dynamic runout). Smoothness is particularly important at the inner two-thirds of the environment side groove wall, and at the extrusion gap. (Most of the radial sliding between the seal and the environment side groove wall occurs at the inner two-thirds of the groove wall.)

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\(^8\) The shaft should be ground and polished to a 4 microinch (0.1 µm) AA or less surface finish (see Figure 1), and a 2 microinch (0.05 µm) AA or less surface finish is preferred.
Figure 3
Example of a grit blasted groove bore

Grit blasting can be used to increase friction between the seal and the groove. Keep at least the inner two-thirds of the environment side 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 4
SAE G50 steel grit
A grit blasting cabinet contains the grit blasting media, and allows the blasting process to be performed indoors.

This photo shows the double layer technique we use to mask the environment side 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.

**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 environment side groove wall. We mask our seal carriers using a double layer of polyurethane abrasion-resistant surface-protection tape. Figure 6 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 environment side groove wall. After grit blasting, finish machine the environment side groove wall to the final groove width dimension (Figure 7).

**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.

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**Figure 7**

Method to avoid masking during component manufacturing
8. Knurling Wide Footprint 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\(^9\) axially-oriented straight knurl on the groove bore (i.e. the cylindrical static sealing surface) inhibits circumferential seal slippage, even in reversing pressure conditions. The straight knurl (Figure 8) 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 environment side groove wall. For this reason, knurling is only recommended for seals that have a wide static sealing lip, such as the Wide Footprint Seal. Leave a 0.06” (1.52 mm) wide unknurled band near the environment side groove wall.\(^{10}\) Even if the Wide Footprint 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 9 shows a suitable way to indicate knurling on a part drawing.

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

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\(^9\) 19 to 28 knurls per inch of circumference.

\(^{10}\) Knurling is not appropriate for Axially Constrained Seals or for seals that have the standard lip width, because these seals have a relatively narrow static lip.
9. **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 can be realized by grit blasting the lubricant side 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 reverse pressure is present\(^\text{11}\), filled and solid cross section seals can optionally be provided with molded anti-rotation tangs to engage recesses in the lubricant side groove wall to prevent seal slippage. For examples of anti-rotation tangs, see Figure 10. If you are not sure you need anti-rotation tangs, leave room in your design for tang recesses in case you need to add them in the future.

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\(^{11}\) Molded tangs are not appropriate for applications where the environment pressure may sometimes be greater than the lubricant pressure, because the seal will deform into the recesses in the lubricant side groove wall.
In applications where the pressure of the environment never exceeds the pressure of the lubricant, anti-rotation tangs can be employed that engage recesses in the lubricant side groove wall to prevent rotational slippage. This particular seal is used on the air side of the sealing assembly.

*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 11).

*Figure 10*

Anti-rotation tangs

*Figure 11*

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. 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.

10. 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.

11. Lubricant side clearance $L_c$

In Figure 1, the housing-to-shaft clearance on the lubricant side of the seal groove is referred to as the lubricant side clearance $L_c$. The size and axial length of the lubricant side clearance $L_c$ depends on its function. If the bore that establishes the clearance $L_c$ serves as a journal bearing, then journal bearing design practices dictate the clearance. 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 clearance $L_c$ be small, to help the seal resist extrusion damage. If it is critical that lubricant pressure be communicated to the rotary seal as quickly as possible, then the axial length of any region of close clearance must be minimized. In summary, when designing the clearance $L_c$, observe Sullivan’s dictum: Form follows function.
12. **Inspecting the seal groove 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 12) and groove measurement calipers (Figure 13) are inexpensive alternatives when properly used by competent inspectors.

![Figure 12](image)

**Figure 12**

**Groove measurement gage**

A parallel movement type groove measurement gage is an inexpensive method of inspecting Kalsi Seal grooves. This particular 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 13
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.

13. 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”.