Chapter D7
Extrusion gap considerations

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1. **Introduction**

The environment-side clearance between the shaft and the seal housing is referred to as the “extrusion gap”, as shown in Figure 1. Service conditions, such as the magnitude of differential pressure and the presence or absence of abrasives, are important considerations in determining the extrusion gap clearance and width, as discussed in detail in this chapter. As a general rule, in the absence of environmental abrasives, the smaller the extrusion gap, the better, so long as heavily loaded contact between the seal housing and the shaft cannot occur. Extrusion gap size recommendations appear in Figure 9 in diametric values; i.e., two times the radial extrusion gap. Eccentric extrusion gaps decrease extrusion resistance, because the extrusion gap clearance is larger on one side.

2. **High-pressure seal extrusion damage explained**

When exposed to a lubricant pressure that is higher than the environment pressure, the resulting differential pressure forces the Kalsi Seal against the environment side groove wall (Figure 1). Since the environmental end of the seal has generally the same flat shape as the environment side groove wall, the wall provides support against the pressure differential at all points except the extrusion gap. When the differential pressure is high, a small portion of the seal bulges (extrudes) into the extrusion gap. Various phenomena, such as shaft deflection, shaft runout, and pulsating pressure can flex and overstress the extruded material, causing pieces to break away from the seal. For an example of high-pressure extrusion damage, see Figure 2.

In some cases, the extrusion damage stabilizes, while in other cases, it may continue until breaching the dynamic sealing lip. In extreme cases of extrusion related material loss, the hydraulic force of the lubricant pressure occasionally holds the seal in sealing engagement, even though the seal is no longer in radial compression, due to material loss. In such cases, the extrusion damage can cause a local portion of the cross section to roll within the groove, as more and more material is consumed by the extrusion mechanism.

A small extrusion gap is necessary for proper seal support in high differential pressure conditions, so the rotary shaft seal can bridge the gap without suffering excessive extrusion damage. Wider dynamic sealing lips, and higher modulus materials, provide better resistance to extrusion related failure, provided the lip is adequately lubricated. Increased interfacial lubrication also increases seal extrusion resistance.
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Figure 1
Extrusion gap nomenclature

Surface finish is in microinches (µin); multiply by 0.0254 to obtain micrometers (µm).

Figure 2
Example of high-pressure extrusion damage

This photo shows a Kalsi Seal with material loss from nibbling type high-pressure extrusion damage. Extrusion damage is caused by cyclic stressing of the seal material which protrudes into the extrusion gap, which ultimately causes the protruding material to fatigue and break away from the sealing element.
Some factors that affect seal extrusion damage are:

- Magnitude of the differential pressure.
- Radial extrusion gap clearance, and any increase to that clearance that results from abrasive wear or eccentricity due to shaft deflection or misalignment.
- Defects at the extrusion gap corner, such as nicks, burrs, or poor surface finish, that can cut or abnormally constrain the seal.
- Repetitive extrusion gap dimensional fluctuations related to factors such as shaft dynamic runout, shaft vibration, and pressure breathing (pressure related dimensional changes) of the seal carrier and/or the shaft.
- Elastomer modulus at the extrusion gap.
- Axial width of the dynamic sealing lip.
- Temperature related softening of the seal and reduction of tensile strength, including overheating of the seal that can occur because of inadequate heat transfer, inadvertent heavily loaded housing-to-shaft contact at the extrusion gap, etc.
- Circumferential seal slippage, particularly when extrusion gap corner damage is present.
- Lubricant pressure fluctuations or pulsations that produce cyclic stress-induced extrusion damage by causing repetitive fluctuations in the magnitude of extrusion.
- Adequacy of interfacial lubrication.
- Profile of the extrusion gap “corner”.
- Hydraulic pressure shock.

3. **Heavily loaded metal-to-metal contact causes seal damage**

Inadvertent heavily loaded metal-to-metal contact at the extrusion gap can cause significant damage to the seal, the seal carrier, and the shaft. When such heavily loaded contact occurs, the seal carrier receives loads intended for the radial bearings, and rapid frictional heat buildup occurs near the rotary seal. This heat softens the elastomer, making it less extrusion resistant. In severe cases, localized seal melting and severe compression set can occur, as shown in Figure 3. Such contact can also damage the seal

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1 For examples of tensile strength variation with temperature, see Robert Flitney's 2007 *Seals and Sealing Handbook* (Elsevier B.V.).
carrier and the shaft (Figure 4), and such damage accelerates seal extrusion damage. Metal-to-metal contact between the shaft and the seal carrier typically relates to heavy shaft side loads. The contact is often due to a combination of shaft deflection, shaft articulation within mounting clearances, and tolerance accumulation.

An additional contributing cause of inadvertent heavily loaded metal-to-metal contact at the extrusion gap is misalignment between machine housings caused by ill-designed threaded connections between the housings. The use of good square mating shoulders, and the use of close fitting pilot diameters may help to avoid such misalignment problems. Ideally, the seal groove and at least one of the radial shaft bearings will be located in the same housing. Beware that, if the radial bearings that guide the shaft are in separate housings, housing misalignment can cause the bearings to bind the shaft (Chapter D20).

**Figure 3**

*Seal damage from shaft-to-seal carrier rubbing*

If heavily loaded contact occurs at the extrusion gap between the rotary shaft and the seal housing, then the resulting frictional heat may damage the seal. This figure shows localized melting, which is accompanied by significant local compression set. In high differential pressure applications, the seal is likely to suffer extra extrusion damage on the non-contacting side, where the maximum extrusion gap clearance occurs.
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Heavily loaded metal-to-metal contact at the extrusion gap damages the seal carrier, the Kalsi Seal, and the shaft hard coating. For high-pressure sealing applications, the designer’s challenge is to determine the smallest possible extrusion gap clearance that presents no danger of heavily loaded metal-to-metal contact.

Avoiding metal-to-metal contact at the extrusion gap

Inadvertent metal-to-metal contact at high-pressure extrusion gaps can sometimes be avoided by careful selection of fits and tolerances, in conjunction with placing the rotary seal close to a radial bearing which is mounted directly in the seal carrier. However, in applications with heavy overhanging side loads, such as downhole mud motors, extra measures are necessary to limit large shaft deflections at the extrusion gap. One way to avoid metal-to-metal contact at the high-pressure extrusion gap in such applications is to interpose a journal bearing between the rotary seal and the overhanging load. In other words, situate the rotary seal between two radial bearings. The seal should be distanced from the outboard journal bearing to isolate it from bearing generated heat. One way to employ an outboard journal bearing is to mount it in an axially movable barrier compensation piston that allows the primary equipment bearings to receive most of the load, but limits peak shaft deflection, as shown in Figure 5.
When space is available, laterally translating sealing assemblies can be employed to avoid heavily loaded metal-to-metal contact at the extrusion gap while minimizing extrusion gap clearance; see Chapters D16 and D17. In such assemblies, the component that defines the extrusion gap is hydraulically force balanced in the axial direction. This allows the component to move laterally in unison with shaft motion, avoiding heavily loaded contact with the shaft. Dual Durometer Kalsi Seals can tolerate somewhat larger extrusion gaps, if harder materials make up the dynamic sealing lip.

Figure 5
Limiting elastic shaft deflection with a barrier compensation piston

In this oilfield mud motor seal arrangement, the primary radial bearings mount directly in the high-pressure seal carrier for maximum concentricity with the bore that defines the high-pressure extrusion gap. The journal bearing in the barrier compensation piston limits elastic shaft deflection, so that metal-to-metal contact does not occur at the high-pressure extrusion gap. The primary radial bearings react most of the side load, and the barrier compensation piston only receives the portion of the load not absorbed by the elastic deflection of the shaft.

4. Environmental abrasion considerations with extrusion gaps

Environmental abrasion considerations with high-pressure extrusion gaps

Some applications may have high or low differential pressure acting from the lubricant-side of the rotary seal at various times in the operating cycle, and therefore require a relatively small extrusion gap. If such an extrusion gap is exposed to an abrasive environment, the axial width of the extrusion gap should be kept very short (Figure 6) to minimize seal wear in the low pressure conditions. If feasible, a barrier seal (Chapter D10) should also be used.
If the differential pressure of a high-pressure application drops occasionally to as low as 100 psi (0.69 MPa) during rotation, then the extrusion gap width can be kept at about 0.040” (1.02 mm) without causing undue seal wear. If the differential pressure drops to as low as 15 psi (103 kPa) during rotation, the extrusion gap width should be kept at about 0.020” to 0.040” (0.51 to 1.02 mm) if possible (0.020” is better than 0.040”), and a barrier seal is strongly recommended. Dimension the axial width of the bore that defines extrusion gap width, rather than dimensioning the size of the chamfer (Figure 7).

The testing basis for recommending a narrow extrusion gap width

We tested single durometer -11 HNBR Kalsi Seals against abrasive oilfield drilling fluid using seal carriers with 0.010” (0.25 mm) radial extrusion gap clearance and a small amount of shaft runout. With 15 psi (103 kPa) lubricant over-pressure and a large extrusion gap width, the seal experienced rapid abrasive wear. With the same 15 psi (103 kPa) lubricant over-pressure, but a 0.020” (0.51 mm) extrusion gap width, the abrasive wear of the seal was significantly diminished. In parametric testing with 15 psi lubricant over-pressure, the 0.020” (0.51 mm) extrusion gap width was significantly better than a 0.040” (1.02 mm) extrusion gap width. With 100 psi (0.69 MPa) lubricant over-pressure and a 0.040” (1.02 mm) extrusion gap width, abrasive wear of the seal was negligible. With 300 psi (2.07 MPa) lubricant over-pressure, abrasive wear of the seal was negligible, even with a large extrusion gap width.

We also tested seals of the same type against drilling fluid using 15 psi (103 kPa) lubricant over-pressure and a 0.020” (0.51 mm) radial extrusion gap clearance (intended for low differential pressure only). The abrasive wear of the seals was negligible. Although we did not test for sensitivity to extrusion gap width with the 0.020” radial extrusion gap clearance, we recommend use of a narrow extrusion gap width, when practical, based on the tests with the 0.010” (0.25 mm) radial extrusion gap clearance.
Short extrusion gap widths reduce hydraulic effects that cause seal wear

Laboratory tests of standard width -11 HNBR Kalsi Seals were performed with drilling fluid, a 0.01” (0.25 mm) radial extrusion gap clearance, and lubricant over-pressures of 15, 100, and 300 psi. At 15 psi, abrasive wear of the seal was less when the axial width of the extrusion gap was minimized. The level of over-pressure also influenced abrasive wear. Seals with 100 and 300 psi over-pressure experienced less abrasive wear, compared to seals with 15 psi over-pressure.

Dimension the length of the extrusion gap bore, rather than the chamfer size.

The accelerated third-body induced abrasion of rotary seals that are exposed to low differential pressure and environmental abrasives with a small extrusion gap is believed to be a hydraulic effect induced by shaft runout. Runout has the effect of rapidly changing the local radial dimension of the extrusion gap. This change displaces some of the abrasive-laden fluid toward the seal. Some of the pressure created when the fluid is displaced must be reacted by the seal. An analogy would be clapping your hands together when they are submerged under water. You can feel water jetting from between your hands, just as the fluid in a small extrusion gap does in response to lateral shaft motion. If you stop the clap while the hands are still one inch apart, not nearly as much water is
displaced, and that is analogous to the benefit provided by a radially large extrusion gap. If you only clap two fingers together, not nearly as much water is displaced, and that is analogous to the benefit provided by an axially short extrusion gap width.

The seal that had 300 psi (2.07 MPa) lubricant over-pressure excluded abrasives well, despite the 0.010” (0.25 mm) radial extrusion gap clearance and a wide extrusion gap width, because the differential pressure causes the exclusion edge of the seal to bite down harder, and exclude abrasives better. Likewise, a seal that had 100 psi lubricant over-pressure, a 0.010” radial extrusion gap clearance, and a 0.040” extrusion gap width excluded abrasives better than a seal exposed to 15 psi differential pressure and the same extrusion gap dimensions. Even if your differential pressure is in the 100 to 300 psi range (or greater), a small extrusion gap width is recommended, because your application may have more runout than our test fixture, and therefore may experience a more pronounced hydraulic effect.

**Figure 8**

*Long, tight extrusion gaps cause seal, shaft and seal carrier wear*

Lateral shaft motion crushes abrasives in a high-pressure extrusion gap, causing shaft and seal carrier wear that damages the seal. The seal wear occurs due to exposure to the worn shaft surface, and due to hydraulic effects resulting from shaft runout and vibration. Minimize the axial width of the extrusion gap to minimize the hydraulic effect, and protect the high-pressure extrusion gap with a barrier seal when possible.
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Figure 9
Diametric extrusion gap clearance recommendations

(Divide by 2 for radial clearance)

This graph shows environment-side housing-to-shaft diametric extrusion gap recommendations that are based on testing -11 HNBR seals. As a general rule, in the absence of environmental abrasives, the smaller the extrusion gap, the better, so long as it does not cause heavily loaded contact between the seal housing and the shaft. At lower differential pressures in abrasive environments, the gap must be kept large to minimize seal and shaft abrasion. At higher differential pressures, the gap must be kept small to minimize seal extrusion damage. When transient pressures are encountered, the gap size should be governed by the highest anticipated differential pressure.

In abrasive environments, a high-pressure extrusion gap can also pack up with solids and abrade the shaft and seal carrier. The bore that defines the extrusion gap can become excessively large from abrasive wear, and should be inspected on a regular basis because it may need periodic repair (by plating or hard coating) or replacement to restore the original diameter. Wider extrusion gap widths are believed to accelerate such wear problems because the hydraulic effect makes it more difficult for particles to escape the extrusion gap clearance when runout and vibration temporarily reduce the extrusion gap.
clearance. The shaft wear can undermine and wear the rotary seal (Figure 8), especially if any relative axial shaft motion is present. Keep your extrusion gap width short, and if possible, protect the high-pressure extrusion gap from abrasives by using a barrier seal arrangement (Chapter D10).

**Environmental abrasion considerations with low pressure extrusion gaps**

When high-pressure extrusion damage is not an issue, the extrusion gap clearance can be kept relatively large (see Figure 9) to minimize seal wear in abrasive environments. As described above, low differential pressure rotary seals that are exposed to an abrasive environment will experience accelerated abrasive wear if used with the small extrusion gaps that are required for high-pressure rotary seals. In unpressurized tests and in 15 psi (103 kPa) tests simulating mud motor low differential pressure seal positions, a radial extrusion gap of 0.020” (0.51 mm) has been used with excellent results to minimize seal and shaft abrasion. As with high-pressure rotary seals, we believe the extrusion gap width should be minimized to reduce the potential for packing related seal and shaft wear. An extrusion gap width of about 0.040” to 0.060” (1.02 to 1.52 mm) is recommended for 0.020” (0.51 mm) radial extrusion gap clearance. Barrier seals (Chapter D10) can also be used to help to promote maximum low pressure rotary seal life.

**Low pressure seals located near a shoulder**

When a low pressure seal faces a shoulder in an abrasive environment, seal wear can be reduced by incorporating a narrow step in the extrusion gap (Figure 10). The narrow step helps to prevent the shoulder from packing abrasives against the seal due to any unavoidable end play.

**Minimizing wear of the air-side seal in surface equipment**

In outdoor surface equipment, various contaminants can enter the air-side extrusion gap. Examples of such contaminants are dust, grimy road splash, and drippings from other equipment. In vertical shaft applications, use of a rotating debris shield (shown in Figure 11) can minimize such contamination. Arrangements are possible that provide a level of protection even in certain horizontal shaft applications; see our U.S. Patent 7,798,496.
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Figure 10
A stepped extrusion gap minimizes end play related seal wear

End play between the seal and the shoulder tends to pack abrasives against the seal, causing wear in equipment such as roller reamers. The stepped extrusion gap helps to reduce packing related wear.

Figure 11
Rotating debris shield

In vertical shaft surface equipment, a rotating debris shield can be used to shield the air-side extrusion gap from contamination. This helps to preserve the life of the air-side seal. Configurations are possible that provide some protection in horizontal shaft applications; see the cement pump cartridge in U.S. patent 7,798,496. (Ensure a complete lubricant fill in vertical shaft equipment, so the upper seal is submerged in lubricant, rather than exposed to an air pocket.)
5. **Seal carrier pressure breathing**

In large diameter or thin wall components, the size of the extrusion gap may change significantly in response to pressure induced component stress. This phenomenon is often called “pressure breathing”, and can cause the extrusion gap to increase or decrease, depending on the pressure location.

If the extrusion gap increases significantly in response to pressure, the rotary seal may be inadequately supported and may experience excessive extrusion damage. If the extrusion gap decreases substantially in response to pressure, metal-to-metal contact between the shaft and seal carrier may occur, and cause seal and shaft damage (Figures 3 and 4). If system pressure changes frequently, the resulting frequent changes in extrusion gap clearance can potentially cause seal extrusion damage (Figure 2) due to the repeated working of any seal material protruding into the extrusion gap.

One way to minimize the effect of pressure breathing is to make the parts suitably stiff. Another potential way is to balance the interior and exterior pressures in a way that minimizes pressure breathing. Figure 12 is an example of a radially pressure balanced seal carrier that is immune to pressure breathing.

![Figure 12](image)

**Radial force balance**

The aligned placement of the static seal and rotary seal causes this seal carrier to be radially pressure balanced, and immune to pressure breathing.
Depending on seal carrier shape and pressure exposed surface area, it may be possible to estimate the potential for pressure breathing by using conventional hand calculations for thin or thick wall pressure vessel stress-induced deformation. For best results, finite element analysis should be employed – particularly when the pressure breathing of a seal carrier must be finely matched to the simultaneous pressure breathing of a mating shaft.

6. **Threaded connection influence**

As shown in Figure 13, the location of highly torqued threaded connections can influence bearing and extrusion gap clearance. In Figure 13, component stress from threaded connection torque will cause both the journal bearing clearance and extrusion gap clearance to be significantly reduced.

Torque related deformation can be estimated based on thread flank angle, shoulder diameter, estimated thread and shoulder friction, and thread torque. The thread calculation\(^2\) predicts the clamping load, and then the thread flank angle is used to predict the hoop force and resulting hoop stress. The hoop stress is used to estimate deformation.

**Figure 13**

*Threaded connection location influences extrusion gap clearance*

Oilfield mud motor threaded connections are tightened to very high torque values to prevent thread loosening in the high vibration downhole drilling environment. When the threads shown here are tightened, the resulting hoop stress may cause enough dimensional change to bind the journal bearing against the shaft, and to significantly reduce the extrusion gap clearance.

\(^2\) For shouldered thread calculations, see "Mechanical Engineering Design" by Shigley and Mischke (McGraw-Hill).
7. **Lubricant-side housing-to-shaft clearance**

Sizing of the housing-to-shaft clearance $L_C$ (Figure 1) on the lubricant side of the seal groove depends on the specifics of the application. When the clearance is intended to serve as a radial bearing, the lubricant-side clearance is dictated by the principles of journal bearing design (Chapter D15). If a small extrusion gap clearance is needed, the extrusion gap tolerancing technique shown in Figure 14 can be used. In applications where the seal may be exposed to high-pressure acting from the environment side of the seal (such as rotary steerable tools and certain hydraulic swivel designs), the housing-to-shaft clearance needs to be appropriate to the anticipated level of differential pressure.

In cases where the seal will not be exposed to high-pressure acting from the environment side of the seal, and the lubricant-side clearance is not intended to serve as a radial bearing, the clearance is not critical. The lubricant-side clearance should be large enough to prevent any undesirable contact between the shaft and the housing, in order to prevent over constraint, unnecessary friction, or damage to the shaft surface. The lubricant-side clearance should not be so grossly oversized that the seal is inadequately supported during installation onto the shaft. Larger lubricant-side clearances are easier to fill with lubricant, and transmit pressure changes more quickly (Chapter D11).

![Figure 14](image)

**Figure 14**

**Extrusion gap dimensioning option for laterally translating carriers**

When the seal carrier defines a journal bearing bore, the extrusion gap bore and the journal bearing bore can be machined in the same setup. If a small extrusion gap clearance is desired, the extrusion gap bore can be defined by a radial step dimension and tolerance, as shown here. This tolerancing approach takes advantage of the accuracy of the lathe, while helping to avoid contact between the shaft and the extrusion gap bore.

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3 We recommend avoiding reverse pressure in hydraulic swivel design by using the concepts shown in Chapter E2.
8. **Hydraulic pressure shock**

In some applications, Kalsi Seals are used to define chambers to conduct hydraulic fluid from a stationary member to a rotating member. Sudden, repetitive stoppage of hydraulically actuated equipment can cause repeated pressure spikes well above nominal system pressure that may promote seal extrusion damage. Extrusion gaps for such applications should be sized accordingly. In terms of hydraulic system design, mechanical brakes may be useful to decelerate hydraulically actuated equipment slowly to minimize such pressure spikes, and pressure relief valves may also be useful.

9. **Higher lubricant pressure with 0.020” radial extrusion gap clearance**

Much of our rotary seal testing with a 0.020” radial extrusion gap clearance has been performed with 15 psi lubricant overpressure. Some of our initial 15 psi testing was done to confirm that 15 psi differential pressure was adequate to prevent skew-induced wear in tests with a 162°F bulk lubricant temperature.

No systematic testing has been performed to determine how much lubricant overpressure a given seal design can withstand with a 0.020” radial extrusion gap clearance. We do know that the Wide Footprint Seal can handle more differential pressure when exposed to a 0.020” radial extrusion gap clearance, compared to a standard width Kalsi Seal.

**80 durometer -8 FEPM testing with a 0.020” radial extrusion gap**

We have tested standard width -8 FEPM seals against a 0.020” radial extrusion gap at temperatures up to 375°F (190.6°C) with 38 psi lubricant pressure and a spring force that was calculated to be equivalent to about 45 psi at 400°F (204.4°C). These seals had noticeable force-related protrusion into the extrusion gap clearance, but were otherwise in good condition. The testing was performed with drilling fluid, and the protrusion seemed to have no negative consequences. This testing is also applicable to the -200 composite material, which uses the -8 material to form the dynamic sealing lip.

**80 durometer -30 FKM testing with a 0.020” radial extrusion gap**

We have tested standard width -30 FKM seals against a 0.020” radial extrusion gap at 300°F with 15 psi lubricant pressure and a spring force that was calculated to be equivalent to about 45 psi at 400°F (204.4°C). These tests indicate that the -30 material is compatible with the usual range of axial spring loading at the temperatures that HNBR is normally used.

We have run a number of tests with PN 462-49-30 axially constrained seals at bulk lubricant temperatures up to the 340°F range. The axial constraint and exclusion edge...
chamfer cause this seal design to have relatively high interfacial contact pressure. Seals exposed to higher differential pressure also experience such increased interfacial contact pressure. The tests of the 462-49-30 seal show that, unlike some FKM materials, the -30 FKM material is tolerant of the asperity contact that can occur with increased differential pressure.

87 durometer -11 HNBR testing with a 0.020” radial extrusion gap
We have tested standard width -11 HNBR seals against a 0.020” radial extrusion gap at 162°F with 70 psi lubricant pressure and a spring force that was calculated to be equivalent to 29 psi. The testing was performed against drilling fluid, and was uneventful.

The true maximum lubricant overpressure value would presumably be related to the pressure-related abrasion resistance benefit being canceled out by detrimental factors such as extrusion damage. Such a maximum would vary from application to application, and would be influenced by factors such as temperature and shaft runout.

No general rule has been established for maximum lubricant overpressure with a 0.020” radial extrusion gap clearance. We have tested -11 Wide Footprint Seals with a 0.020” radial extrusion gap at 200°F and 500 psi lubricant pressure with an air environment. The hydrodynamic pumping related leakage was reportedly clean, suggesting reasonable lubrication. We went on to test these seals at higher lubricant overpressure that produced dark hydrodynamic pumping related leakage and seal damage, and unfortunately we did not get to observe seal condition after the 500 psi leg of the test.