Chapter D4
Shaft deflection, runout, vibration, and axial motion

Revision 2  October 8, 2014

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**

Some Kalsi Seal applications have a significant amount of lateral shaft displacement that must be properly addressed for satisfactory seal performance. Examples of lateral shaft displacement are dynamic runout, deflection, and vibration. This chapter describes several types of lateral shaft displacement, and provides guidance for addressing the issue when implementing rotary seals.

2. **Dynamic runout of the rotary shaft**

The term “dynamic runout” can be visualized as eccentric rotation (Figure 1). Measurement of runout is typically performed using an instrument such as a dial indicator (Figure 2), and readings are expressed as Total Indicator Reading (T.I.R.) or Full Indicator Movement (F.I.M.). Dynamic runout under actual operating conditions is often difficult or impossible to measure, and is typically greater than the measurements gathered while rotating the shaft slowly in a shop setting.

Assuming the same magnitude of T.I.R., the type of runout that is illustrated on the left side of Figure 1 is more damaging to rotary seals, because the runout occurs twice per revolution. As a result, the seal accumulates more compression-relaxation cycles, and experiences accelerated extrusion damage in high pressure service.

![Figure 1](image)

**Figure 1**

**Simple examples of runout**

This figure provides two simple examples of runout to illustrate what Total Indicator Reading (T.I.R.) means. Solid lines represent one shaft position, and phantom lines represent another. In the left hand image, the runout is caused by an out of round shaft. In the right hand image, the runout is caused by eccentric rotation. These factors, and other factors such as bearing clearance and load induced shaft flexure, can combine to produce complex lateral shaft motion.
Measuring shaft runout with a dial indicator

A dial indicator mounted on a magnetic base is being used to measure shaft runout as the shaft is being turned slowly. The runout measurement is reported in terms of the total movement of the indicator needle. Runout measurements in actual operating conditions may be impractical to measure, and are likely to be far greater than measurements taken while rotating the shaft slowly, without actual operational loads. When space is restricted, a dial test indicator can be used in place of the illustrated dial indicator.


**Sleeves add to runout**

One easy way to reduce runout and improve heat transfer is to avoid the use of sleeves as seal running surfaces. Such sleeves often use a slip fit instead of a press fit, because a press fit may crack hard surface coatings, such as tungsten carbide. Slip fits can cause the eccentricity conditions that are shown in Figure 3 (clearance is exaggerated). In the left hand side of Figure 3, the sleeve is egg-shaped, as may happen with thin, large diameter sleeves. This produces the runout condition illustrated schematically on the left hand side of Figure 1. In the right hand Figure 3 image, the sleeve is offset to the extent permitted by the sleeve to shaft clearance. This produces the runout condition illustrated schematically on the right hand side of Figure 1.

Even when sleeves are press fit to the shaft, they still interrupt heat transfer from the dynamic sealing interface to the shaft. They also increase the tolerance on the sealing surface diameter, because the tolerances of three different surfaces accumulate to influence the size of the sealing surface.

![Figure 3](image)

**Figure 3**

**To reduce runout and seal temperature, avoid sleeves**

Sleeves often have clearance with the shaft, to facilitate assembly, and to prevent cracking of the hard surface coatings that are typically used in abrasive service. Sleeves subject the rotary seal to unnecessary runout, and also interrupt heat transfer from the sealing interface to the shaft.
**Bearing internal and mounting clearances influence shaft runout**

Side loads cause the shaft to articulate within mounting clearances, resulting in lateral shaft deflection at the seal location. The potential range of such clearance-related lateral deflection can be calculated based on component mounting clearances and tolerances. This lateral deflection can contribute to runout if the shaft wobbles back and forth within the available clearance (Figure 4). Although Figure 4 illustrates a journal bearing for the sake of simplicity, many rolling element bearings also have internal and mounting clearances. Some types of bearings, such as angular contact and tapered roller bearings, have little or no internal clearance, which helps to minimize clearance-related shaft runout.

![Figure 4](GA1818.2)

**Figure 4**

Rotary shaft runout due to wobble within bearing clearance

This schematic illustrates how shaft wobble due to bearing clearance influences dynamic runout. For the sake of simplicity, journal bearings are illustrated. When rolling element bearings are used, the total clearance may include both internal bearing clearance and bearing mounting clearance. Runout due to bearing clearance can be minimized through the use of bearings that minimize or eliminate internal clearance, such as angular contact and tapered roller bearings. Runout at the rotary seal can be minimized by placing the seal close to the outboard radial bearing. The distance between bearings can also influence lateral deflection.
Dynamic runout exposes the seal to highly repetitive radial gland and extrusion gap dimensional changes that can:

- Cause accelerated extrusion damage to pressurized rotary seals,
- Cause abrasion between the seal and the groove wall,
- Eventually exceed the remaining seal compression, accelerating the onset of compression set related lubricant leakage, and causing accelerated abrasive invasion of the dynamic sealing interface,
- Cause increased seal, shaft, and seal housing wear, due to particle entrapment in the extrusion gap,
- Cause metal-to-metal contact between the rotary shaft and the seal housing, resulting in component damage and seal overheating, and
- In very high rotary speed applications and very low temperature applications, exceed the ability of the seal to follow the radial motion of the shaft, resulting in increased lubricant leakage. High-speed dynamic runout may also have environmental exclusion implications, but the topic is unexplored.

Some of the variables that contribute to runout are bearing internal clearance, bearing mounting clearance, the shaft being out of round, manufacturing eccentricity, and shaft loading conditions.

**How much dynamic runout is okay?**

The answer to the question “How much dynamic runout is okay?” depends on such factors as sealing life expectancy, speed, initial compression, differential pressure, temperature, seal type, and more. An enormous amount of testing would be necessary, in order to have an answer for every customer’s operating conditions and goals. The fundamental truth is this: Less runout is always better than more.

Our routine laboratory tests typically have shaft runout in the range of \( \leq 0.002'' \) (0.05 mm) T.I.R.

We have, however, successfully tested 2.75” (69.85 mm) Kalsi Seals against drilling fluid at 480 rpm using about 0.007” (0.18 mm) T.I.R. runout. The lubricant pressure was 1,000 psi (6.89 MPa), the lubricant temperature was maintained at 162°F (72.2°C), and the diametric extrusion gap clearance was 0.020” (0.51 mm). The duration of the test was 272.8 hours, and the seals were worn but still effective at the conclusion of the test. The 0.007” (0.18 mm) runout corresponds to the loosest clearance typically encountered when guiding a moderate diameter shaft with journal bearings.

Our tests also evaluated 2.75” (69.85 mm) Kalsi Seals against drilling fluid at 400 rpm using axial spring loads ranging from very light to relatively high, with dynamic runout in the range of 0.0075” to 0.0120” (0.19 to 0.30 mm) T.I.R. The tests, performed
with a 162°F (72.2°C) lubricant temperature, were terminated at 93 and 87.6 hours respectively, to free up the test fixtures for other tests. The seals looked nearly new at the end of the test, except for environment end scuffing from millions of back and forth radial sliding motions against the groove wall.

3. Shaft deflection

Another basic type of lateral shaft motion is deflection resulting from side loads on the rotary shaft (Figure 5). In addition to shaft articulation as described in Section 2, side loads can cause the shaft to deflect elastically. Elastic deflection estimates are possible, when side loads are quantifiable, by using Roark and Young type calculations, or finite element analysis. If the angular location of the side load is constant, then the angular orientation of the shaft deflection is also constant. Although this condition, known as static offset, does influence local seal compression changes (increased on one side, decreased at the opposite side), it does not cause runout, per se. If the angular location of the side load moves, then the resulting shaft deflection does contribute to runout.

Shaft deflection reduces rotary seal extrusion resistance by causing a larger extrusion gap at one side of the shaft (Figures 6 and 7). Shaft deflection also can cause damaging metal-to-metal contact between the seal housing and the shaft (Chapter D7).

![Figure 5](GA1819.1)

**Shaft deflection due to side load**

This schematic illustrates how an overhung side load produces lateral shaft deflection through flexure of the shaft. When the location of the side load moves, the deflection can impact runout. Shaft deflection has to be considered when designing seal grooves and extrusion gap clearance. Deflection can sometimes be minimized by stiffening the shaft between the bearings. Designs exist where shaft deflection is minimized in severe service conditions by providing a journal bearing along a significant length of the shaft.¹

---

¹ For example, see U.S. Patent 6,416,225.
Eccentricity increases seal high-pressure extrusion damage

In high differential pressure sealing applications, static offset between the seal carrier and the rotary shaft increases extrusion damage on the half of the rotary seal that is exposed to the increased extrusion gap. When the extrusion gap size varies dynamically, the rate of seal damage increases, because the changing gap causes the extruded material to experience high strain.

Figure 6

Example of high-pressure seal damage in eccentric conditions

This Wide Footprint Seal, tested in our lab, illustrates the effect of an eccentric extrusion gap in high differential pressure conditions. The portion of the seal facing the reduced extrusion gap is in nearly perfect condition. The portion of the seal facing the maximum extrusion gap shows the maximum extrusion related damage. Because of the wide dynamic sealing lip, plenty of material remains usable for additional rotary operation.
4. **Designing to minimize runout and deflection**

The machine designer should take steps to minimize the effects of shaft runout, deflection and misalignment, because rotary seal life depends to a large degree on the quality of the implementation. Shaft runout, misalignment, and articulation related deflection can be minimized by paying careful attention to the clearances and tolerances between relevant components, including the bearing to shaft interface, the bearing to housing interface, the housing to seal carrier interface, axial and radial bearing internal clearances, etc. Shaft deflection can be minimized by:

- Using suitably spaced radial bearings,
- Positioning the radial loads near a radial bearing, and
- Making the shaft as stiff (large) as possible through & between the radial bearings.

Preloaded rolling element bearings are recommended whenever minimal runout is critical, because they eliminate internal bearing clearance. Journal bearings facilitate the use of larger diameter, stiffer shafts, and are often used to limit shaft deflection in equipment with tight dimensional constraints, such as oilfield downhole mud motors.

**Oilfield tools**

Some applications, such as oilfield rotary control devices (RCDs)\(^2\), inherently have unacceptable levels of shaft runout, deflection, and misalignment that simply cannot be avoided. In such applications, hydraulically force balanced, laterally translating seal carrier arrangements\(^3\) (Chapter D16) or backup rings (Chapter D17) should be used.

Shaft deflection can be extreme in downhole drilling tools, particularly in short radius drilling. Deflections far exceeding calculated values can be encountered. As shown and described elsewhere in this handbook, in mud motor sealed bearing assemblies:

- A barrier compensation piston (Chapters D10 and D14) can be located below the fixed location seal to limit shaft deflection at the fixed location Kalsi Seal, and
- Journal bearings (Chapter D15) such as DU bushings\(^4\) can be used in lieu of rolling element bearings to achieve increased shaft diameter and stiffness.

It may also be possible to incorporate a hydraulically force balanced, laterally translating backup ring arrangement (Figure 8) into a mud motor for high pressure sealing or high temperature sealing. To take the best advantage of available radial space, locate the backup ring arrangement in the vicinity of the thrust bearings.

---

\(^2\) Also known as rotating heads, rotary blowout preventers, rotating blowout preventers, rotating drilling heads, rotary bop’s, and rotating diverters.

\(^3\) For examples of hydraulically force balanced, laterally translating seal carrier arrangements, see U.S. Patents 5,195,754 and 6,227,547, and U.S. Patent Application Publication 2011/0127725.

\(^4\) For examples of the use of journal bearings in mud motor assemblies, see U.S. Patents 5,727,641 and 6,416,225.
5. Shaft following high pressure sealing arrangements

Handbook chapters D16 and D17 describe several high pressure sealing mechanisms that move laterally to accommodate shaft deflection, and allow a small extrusion gap without danger of heavy metal-to-metal contact. Of these, we believe the Chapter D17 arrangements provide the best conditions for high pressure sealing, provided that the low pressure side of the rotary seal is a clean environment. Figures 8 and 9, below, show these patent pending arrangements.

![Diagram of high pressure sealing arrangement](image)

**Figure 8**

**A floating backup ring arrangement for smaller levels of runout and misalignment**

In this patent pending seal cartridge, an axially force balanced metal backup ring is free to float laterally to align on the shaft. The backup ring is also radially pressure balanced, allowing for the smallest practical extrusion gap, to achieve the maximum high pressure extrusion resistance. Because runout and misalignment affect the compression of the Kalsi Seal, this arrangement is best suited for applications with relatively small levels of misalignment and runout, such as washpipe assemblies. We performed a 950 hour test of a 2.75” (69.85mm) diameter version of this arrangement at 5,000 psi with a shaft runout of 0.010” (0.254mm) and a shaft surface speed 252 ft/minute. The PN 655-4-106 Kalsi-brand rotary seals were in good condition at the conclusion of the test. The lubricant was an ISO 320 viscosity grade synthetic hydrocarbon lubricant, and the bulk lubricant temperature was maintained at 120 to 130°F to simulate an oilfield washpipe assembly. Contact Kalsi Engineering, Inc. for licensing information.
A floating backup ring seal cartridge for higher levels of runout and misalignment

In this patent pending cartridge arrangement, the Kalsi-brand rotary seal is mounted in an axially force balanced metal backup ring that. The axial force balance frees the backup ring to float laterally, following shaft misalignment and runout. This arrangement isolates the Kalsi Seal from large changes in radial compression, and is preferred for applications having large levels of lateral shaft deflection. The backup ring is radially pressure balanced, which allows for the smallest practical extrusion gap, to provide maximum high pressure extrusion resistance. Contact Kalsi Engineering, Inc. for licensing information.
5. **Vibration**

Due to the expense of vibrating test fixtures and shaker tables, we have not tested Kalsi Seals in vibrating conditions. Kalsi Seals, however, are routinely used in the high vibration conditions encountered in elevated temperature downhole oilwell and coil tubing drilling. The ability to follow rapid shaft lateral motion diminishes in arctic temperature startup conditions, owing to the increased stiffness of the sealing material at low temperatures.

6. **Abrasive ingestion due to axial shaft motion**

Axial shaft motion contributes significantly to shaft and rotary seal wear in abrasive environment applications. Axial motion can drag abrasives into the dynamic sealing interface, and can also damage the seal by causing contact with damaged or contaminated portions of the shaft. Kalsi Seals with wider dynamic sealing lips are preferred in rotary applications where minor axial motion cannot be avoided.

If possible, the thrust bearing implementation of the host machinery should permit very little axial shaft motion. Preloading the thrust bearings with heavy disk springs can sometimes be used help to reduce axial motion. Some rolling element thrust bearing manufacturers actually recommend that their thrust bearings be implemented with at least a minimum level of thrust load.

In applications that warrant the expense, the seal carrier itself can incorporate small cross section rolling element bearings that couple it to the rotary shaft, so that radial and axial motion is absorbed by the carrier to housing seal\(^5\), rather than by the rotary seal (Figures 9 and 10). This arrangement causes the sliding motion to be separated from the rotary motion, so that the rotary seal operates only in a rotary mode, and so that the sliding seal only operates in a sliding mode, and also absorbs the radial motion between the shaft and the housing.

In the arrangement shown in Figure 10, the bearing set has to be robust enough to withstand any differential pressure that may act over the area between the sealing diameters of the sliding seal and the Kalsi Seal. In oilfield downhole applications, due to several possible causes of reversing pressure differential, the area between the sealing diameters of the sliding seal and the Kalsi Seal should be minimized to the extent possible. Figure 11 shows an arrangement where the area between the static and rotary sealing diameters is zero. This means that differential pressure imposes no axial force on the seal carrier and its bearings.

---

\(^5\) Various types of carrier to housing seals are possible. For example, expired U.S. Patent 5,014,998 shows the use of an flexible annular diaphragm to allow a rotary seal carrier to accommodate moderate seal carrier motion.
Figures 10 and 11 are schematic in nature, and are intended only to illustrate how a seal carrier can be bearing mounted to follow radial and axial shaft motion. These figures do not represent a seal implementation that is ideal for reversing pressure conditions. For reversing pressure conditions, redundant rotary seals (Chapter D10) are often recommended. In oilfield downhole drilling applications, the pressure of the lubricant filled region between such redundant seals should be balanced to the pressure of the drilling fluid environment.

**Figure 10**

*Isolating the rotary shaft seal from axial shaft motion in low DP service*

In low differential pressure applications with axial shaft motion, the seal carrier can be mounted to the shaft with bearings so that the axial motion is absorbed by a sliding seal, rather than by the Kalsi Seal. This improves the ability of the Kalsi Seal to withstand environmental abrasives. If preferred, an anti-rotation tang can be used in lieu of anti-rotation seals.
Isolating the rotary seal from axial motion while isolating the bearings from DP

If the relative axial motion is relatively small, the housing to seal carrier sliding seal can be incorporated on a reduced diameter extension of the seal carrier as shown here, to eliminate differential pressure induced thrust loads on the seal carrier bearings (Expired U.S. Patent 5,195,754). For longer stroke applications, the length of the reduced diameter extension can be increased, and the anti-rotation tang can be oriented radially on the seal carrier, engaging a longitudinal slot in the bore of the housing (or vice versa).

7. Rotating housing vs. rotating shaft

It is preferred that the shaft rotate, rather than the housing that holds the rotary seal. In some applications, such as roller reamers, it is more practical to rotate the housing due to factors such as shaft fatigue.

In rotating housing applications, any static housing to shaft offset causes a once per revolution change in the gland and extrusion gap radial dimensions (Figure 12). The repetitive changes in gland dimensions accelerate the effects of rotary seal compression set. The repetitive extrusion gap dimensional fluctuations promote extrusion damage in pressurized applications. In high rpm rotating housing applications, higher leak rates may occur because the rubber must respond dynamically to the rapidly fluctuating radial gland dimension.

Rotating housing applications should, if possible, be provided with precise radial bearings to minimize gland and extrusion gap radial dimension fluctuations, and high rotary speeds should be avoided. While not an optimal mechanical arrangement, Kalsi Seals can be used successfully in applications that incorporate low speed rotating housings.
When sealing high differential pressure, the floating backup ring arrangements of Figures 8 and 9 can be adapted to rotating housing applications.

**Figure 12**  
**Rotating shaft vs. rotating housing**

When the seal rotates with the housing around a stationary shaft, any static offset causes the radial gland depth at any seal location to change from maximum to minimum once per revolution. The same thing is true of the radial extrusion gap size. The radial gland depth change accelerates the effect of compression set, and the extrusion gap change accelerates extrusion damage. When the shaft rotates inside of a stationary seal, any static offset between the housing and the shaft does not cause the local radial gland depth and radial extrusion gap to change on a once per revolution basis.
This page intentionally left blank.