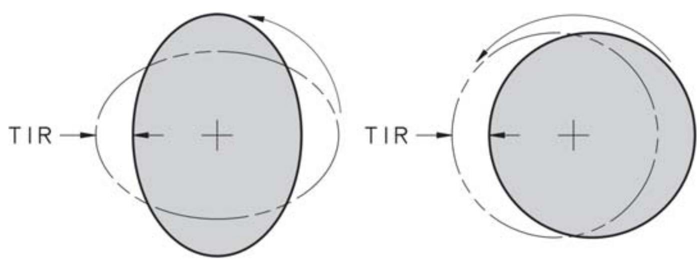
Chapter D4

Shaft deflection, runout, vibration, and axial motion



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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 https://www.kalsi.com/seal-handbook/.

1. Introduction

Some rotary 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.

Whether a Kalsi Seal™ is being used for high or low pressure service, it will benefit from being isolated from shaft lateral displacement as much as possible. This chapter describes several types of lateral shaft displacement, and provides guidance for addressing the issue when implementing Kalsi-brand rotary seals.

2. What is dynamic runout?

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.

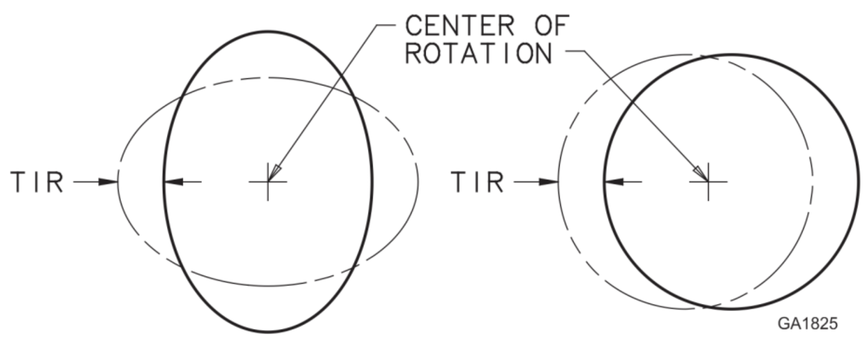

Figure 1

Total indicated runout examples

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 during rotation.

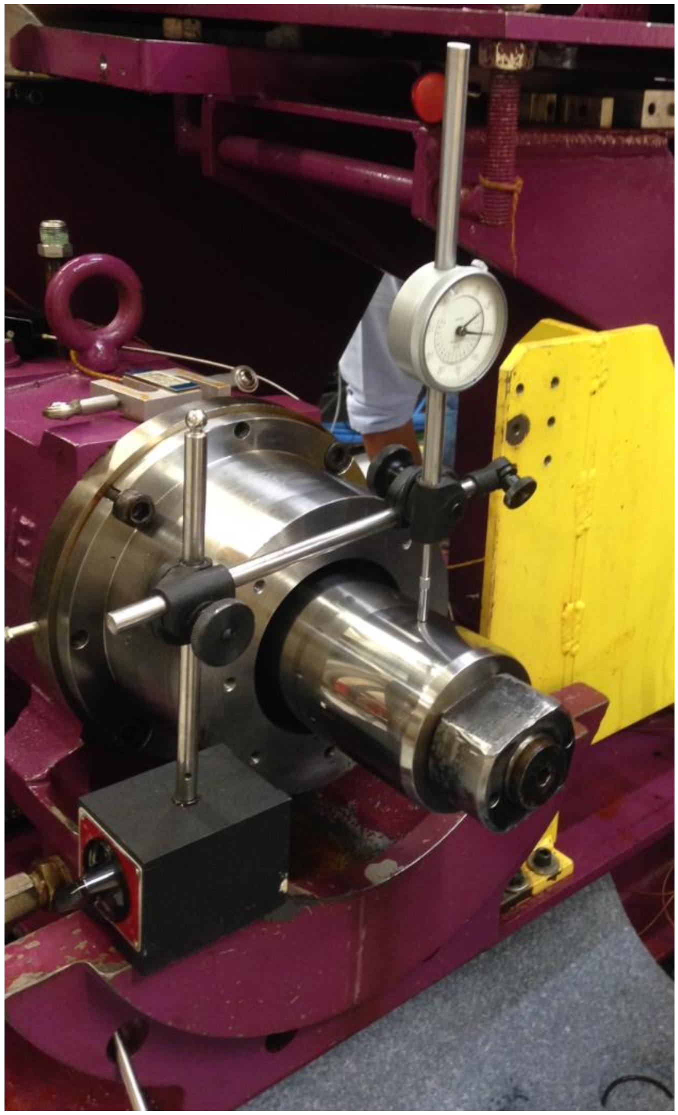

Figure 2

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. On the right-hand side of Figure 3, 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.

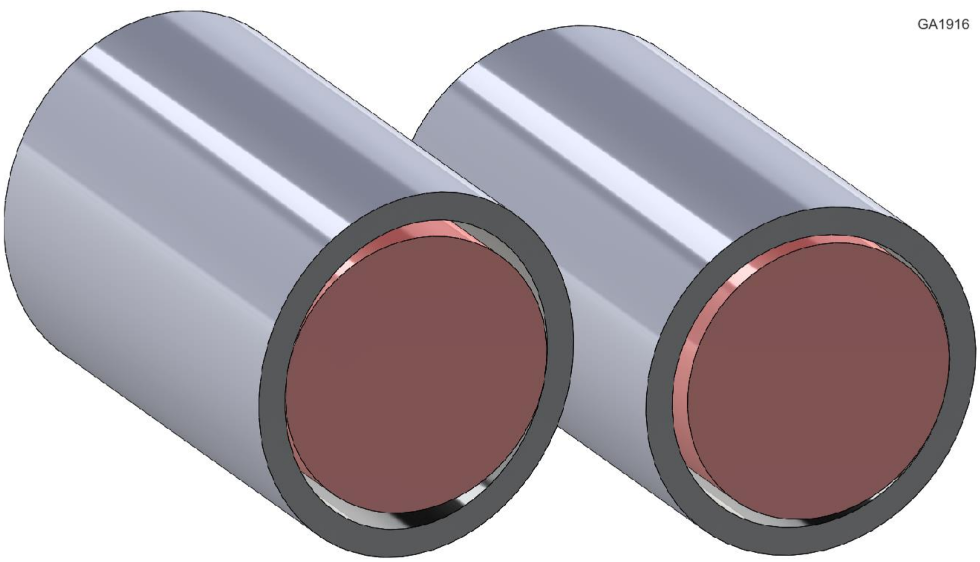

Figure 3

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 critical 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 location of the rotary seal. 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, X-type (four-point contact), and tapered roller bearings, have little or no internal clearance, which helps to minimize clearance-related shaft runout.

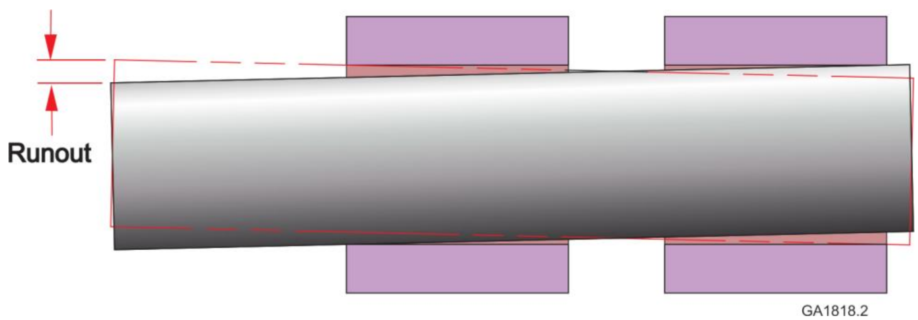

Figure 4

Dynamic 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 wear of groove wall (Figure 5) and mating seal surface,
- 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[1] and very low temperature applications,[2] 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 ≤ 0.002” (0.05 mm) T.I.R.

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1 The basis for this statement is a low-pressure seal test that was run in the 1,750 to 2,750 rpm range (902 to 1,418 ft/min) with 0.0035” runout, compared to a test that was run with 0.0012” runout.

2 The ability of an elastomeric rotary seal to follow rapid shaft lateral motion diminishes in arctic temperature startup conditions, owing to the increased stiffness of the sealing material at low temperatures.

Figure 5

Groove wall wear due to dynamic runout

This photo shows wear of the environment-side wall of a seal groove that occurred during extensive operation at 346 ft/min with 0.010” dynamic runout. The wear, which is limited to the inner half of the groove wall, is caused by the radial motion of the seal as it followed the dynamic runout of the shaft.

3. Lateral shaft deflection

Another basic type of lateral shaft motion is deflection resulting from side loads on the rotary shaft (Figure 6). In addition to shaft articulation as described in Section 2, side loads can cause the shaft to deflect elastically. Elastic deflections can be estimated 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 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 extrusion resistance of a rotary shaft seal by causing a larger extrusion gap at one side of the shaft (Figures 7 and 8). Shaft deflection also can cause damaging metal-to-metal contact between the seal housing and the shaft (Chapter D7).

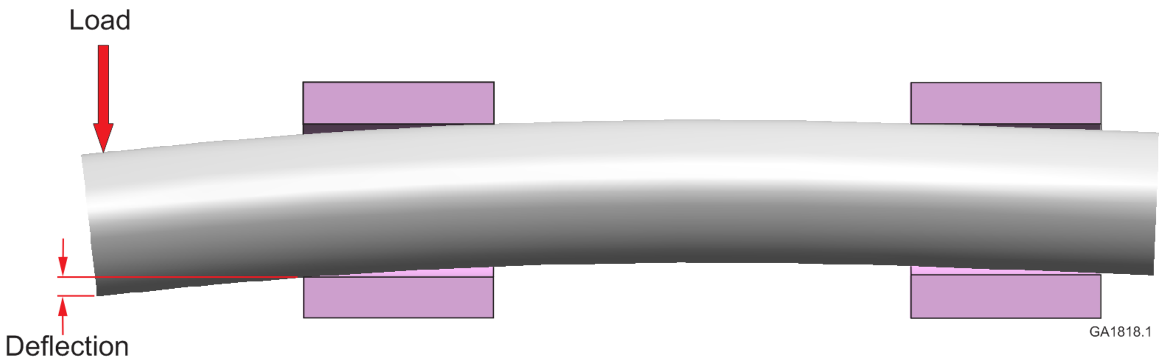

Figure 6

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.[3]

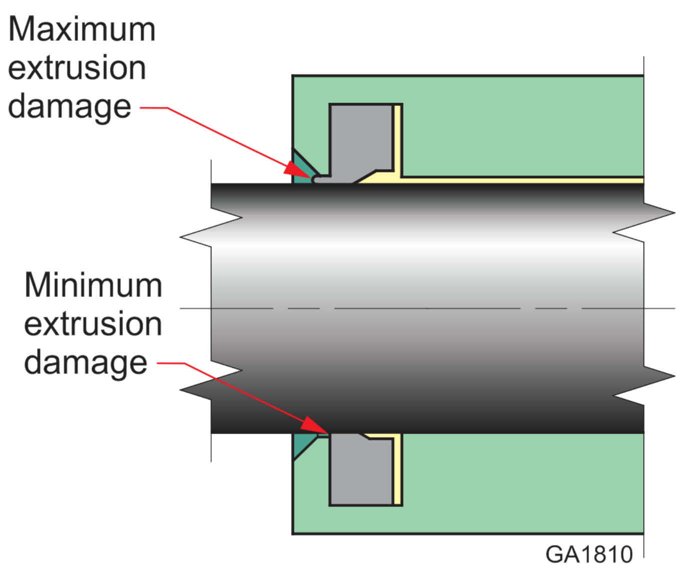

Figure 7

Eccentricity increases seal high-pressure extrusion damage

Static offset between the seal carrier and the shaft causes the extrusion gap to vary from a minimum clearance location to a maximum clearance location. In high differential pressure sealing applications, the extrusion damage to the rotary seal is typically minimum at the minimum clearance location, and maximum at the maximum clearance location. When the extrusion gap clearance varies dynamically due to factors such as runout and vibration, the rate of damage to the rotary seal typically increases, because the extruded seal material experiences increased strain.

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[3] For example, see U.S. Patent 6,416,225.

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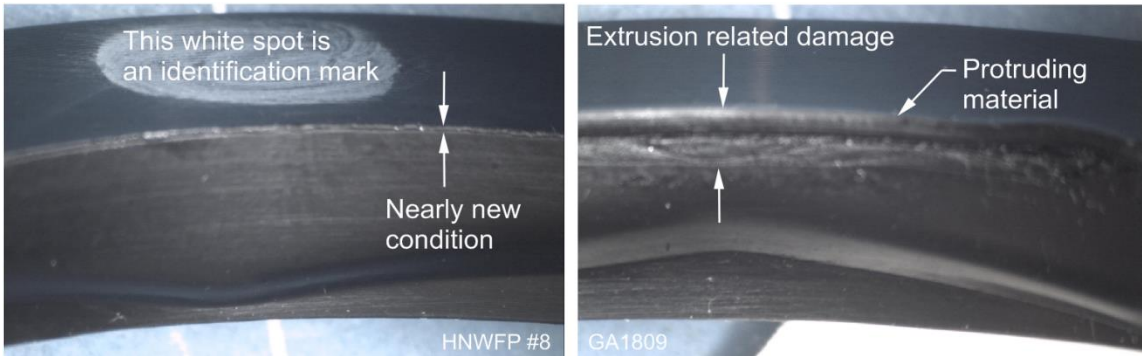

Figure 8

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 location is in nearly perfect condition. The portion of the seal facing the maximum extrusion gap location shows the maximum extrusion related damage. Because of the wide dynamic sealing lip, plenty of material remains usable for additional rotary operation.

4. Static lateral misalignment

Static lateral misalignment between the seal housing and the shaft increases seal compression and the risk of seal and shaft damaging metal-to-metal contact on one side. It also reduces seal compression and increases the extrusion gap clearance on the other side. The locally reduced compression makes the seal less able to withstand shaft deflection and runout. The locally increased extrusion gap clearance makes the seal less able to withstand differential pressure.

One cause of static misalignment is locating the seal groove in a housing that is separate from the bearing housing. This means that lateral offset can be created by the mounting clearance between the seal housing and the bearing housing. Additional lateral offset can occur due to machining eccentricity between housing-to-housing pilot surfaces and other critical surfaces, such as the housing bore that locates the bearing.

Floating arrangements are described in the chapter sections below that are positioned laterally by the shaft. Such arrangements are not always practicable due to factors such as economics and limited space.

When floating arrangements are not practical to use, the bore that receives the radial bearing should be cut directly into the seal housing, to achieve maximum concentricity between the bore that locates the bearing and the bore that defines the extrusion gap clearance with the shaft (Figure 9).

When seal carriers that are separate from the bearing housing are required, the best piloting practices should be employed. In general, the axial engagement length should be very short (Figure 10) so that the pilot clearance can be very tight without risk of causing the "sticky drawer effect" that is discussed elsewhere in this handbook.

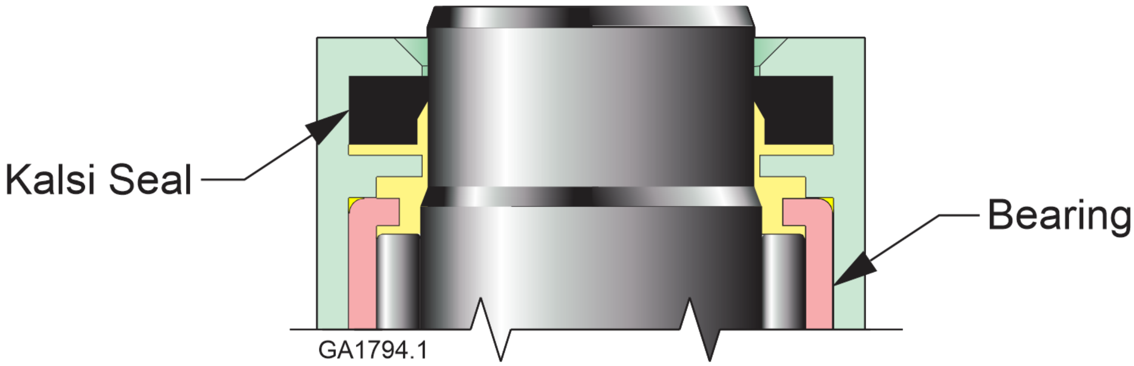

Figure 9

Achieving maximum concentricity

Maximum concentricity can be achieved between the shaft and the seal groove, and between the shaft and the bore that defines the extrusion gap, by incorporating the bearing directly into the seal housing.

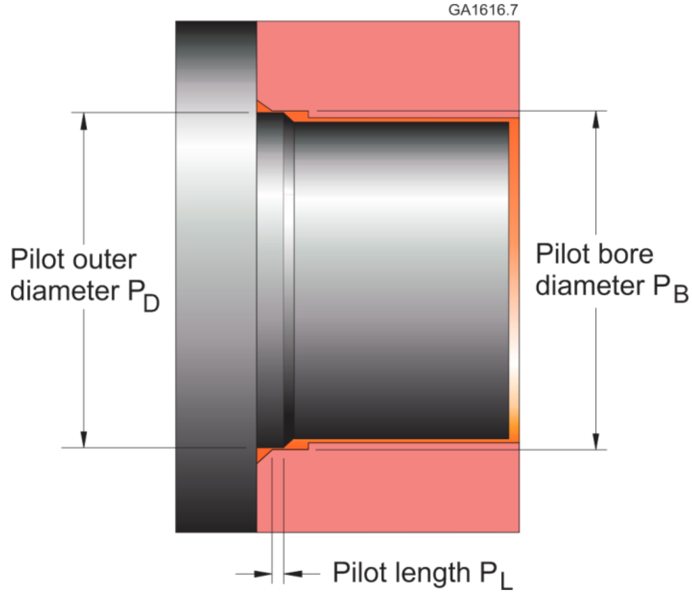

Figure 10

Use a short pilot length between the seal housing and the bearing housing

When a seal housing has to be separate from a bearing housing, the tightest fit between the pilot bore diameter P_B of the bearing housing and the pilot outer diameter P_D of the seal housing can be achieved when the pilot engagement length P_L is very short. The short engagement length prevents the "sticky drawer effect" that often occurs with tight pilot fits, because the corner contact that causes sticking cannot occur.

How much lateral misalignment is okay?

We are sometimes asked how much lateral misalignment between the shaft and the seal housing is too much. We view the performance effect of lateral misalignment as being a continuum, rather than something that presents an obvious reason to set a bright-line limit.

We have not intentionally varied lateral misalignment to evaluate performance effect. Based on a typical lack of metal-to-metal contact when using seal carriers with bores that are nominally 0.0195" larger than the shaft diameter, our test fixtures rarely approach 0.010" lateral misalignment.

5. Vibration

Kalsi-brand shaft seals are used in the high vibration conditions encountered in oilwell and coil tubing drilling. Due to the expense of vibrating test fixtures and shaker tables, we have not performed laboratory testing of Kalsi Seals® in vibrating conditions. We have, however, fielded downhole tools to get direct feedback on seal performance.

One cause of vibration in oilwell drilling is the *"stick-slip"* phenomenon. During the *"stick"* phase, twisting of the drill string above the stuck point stores energy, acting like a long torsion bar spring. During the *"slip"* phase, the stored energy is suddenly released, causing temporarily accelerated rotation and violent vibration. Because of the length and resulting torsional flexibility of the drill string, torsional vibration also occurs from non-constant torsional resistance. Because of the length and resulting axial flexibility of the drill string, the cutting action of the drill bit can cause severe axial vibration.

From a seal life standpoint, we believe the beast way to deal with extreme vibration is to minimize relative radial motion at the rotary seal location by the use of tight journal bearing clearance. In some cases, this may involve an annular component (i.e., a seal carrier, a backup ring, or a compensation piston) that positions the rotary shaft seal, and follows lateral shaft motion.

6. 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)[4], inherently have significant levels of shaft runout, deflection, and misalignment that simply cannot be

[4] Also known as rotating heads, rotary blowout preventers, rotating blowout preventers, rotating drilling heads, rotary bop's, and rotating diverters.

avoided. In such applications, hydraulically force balanced, laterally translating seal carrier arrangements[5] (Chapter D16) or backup rings (Chapter D17) should be considered.

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[6] 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 11) 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.

7. 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 11 and 12, below, show these patented arrangements. Contact us for licensing information.

5 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.

6 For examples of the use of journal bearings in mud motor assemblies, see U.S. Patents 5,727,641 and 6,416,225.

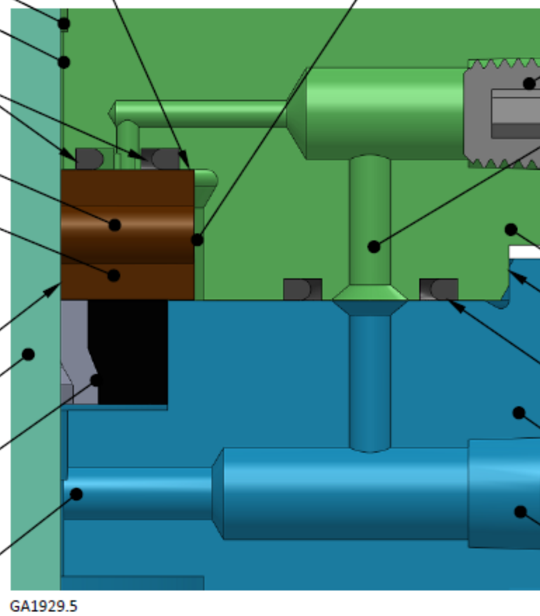

Figure 11

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

In this patented arrangement, 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. 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. Contact Kalsi Engineering, Inc. for licensing information.

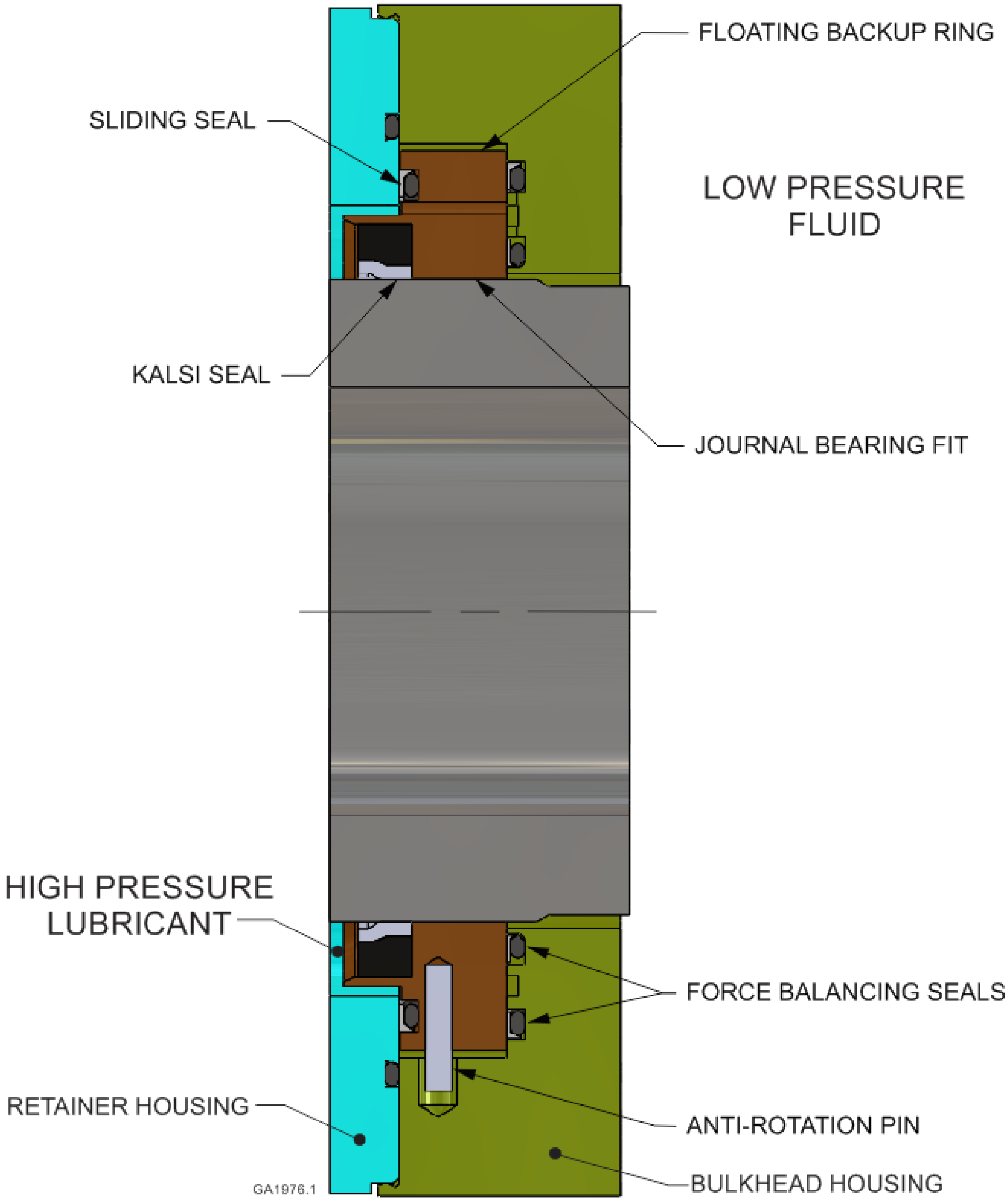

Figure 12

A floating backup ring for higher levels of runout and misalignment

In this patented arrangement, the rotary seal is mounted in an axially force balanced backup ring. Axial force balance frees the backup ring to float laterally, following shaft misalignment and runout. This isolates the Kalsi Seal from large compression changes and is preferred for applications having large levels of 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.

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8. Shaft following low pressure sealing arrangements

Kalsi-brand rotary shaft seals are often used to partition an abrasive environment, such as oilfield drilling fluid, from a seal and bearing lubricant. The pressure of the seal lubricant is typically equal to, or slightly greater than, the pressure of the environment. Our testing indicates that third body wear due to environmental abrasives increases with increasing shaft runout. Our tests were at 480 rpm with <=0.002” runout and about 0.010” runout. Many tests with <=0.002” runout had little or no third body wear. Many tests with about 0.010” runout had significant third body wear.

Rotary shaft seals that are exposed to only low differential pressure can be mounted in floating seal carriers that are guided laterally by a journal-bearing type fit with the shaft (Figure 13). Since the seal carrier follows lateral shaft motion, the rotary shaft seal is largely isolated from the negative effects of lateral shaft motion.

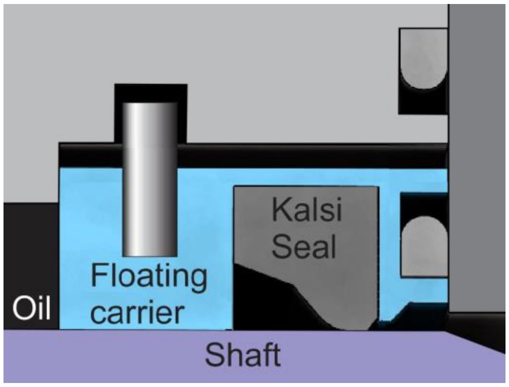

Figure 13

A partially balanced seal carrier for low levels of differential pressure

In this arrangement, the Kalsi-brand rotary shaft seal is mounted in a seal carrier that defines a journal bearing fit with the shaft, and follows lateral shaft motion. When differential pressure acts across the rotary shaft seal, the axially acting hydraulic forces are unequal, but the resulting axial force imbalance is negligible when the magnitude of differential pressure is low. Because the seal carrier is aligned laterally by the shaft, the rotary shaft seal is isolated from large changes in radial compression.

9. 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[7], rather than by the rotary seal (Figures 14 and 15). 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 14, 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 15 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.

Figures 14 and 15 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

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7 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.

filled region between such redundant seals should be balanced to the pressure of the drilling fluid environment.

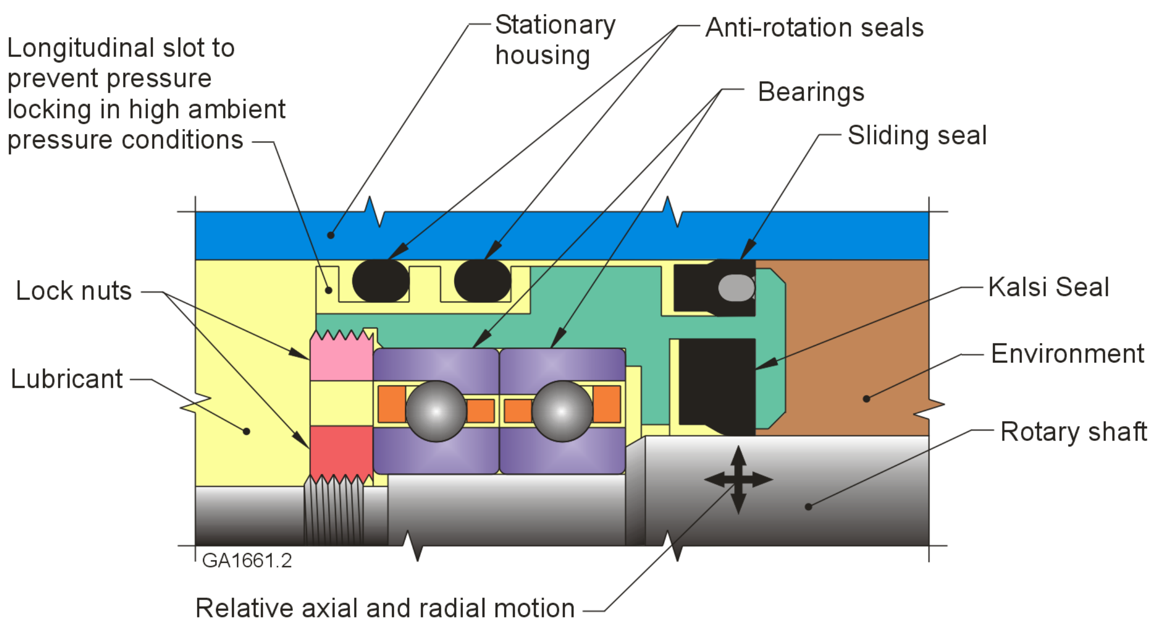

Figure 14

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.

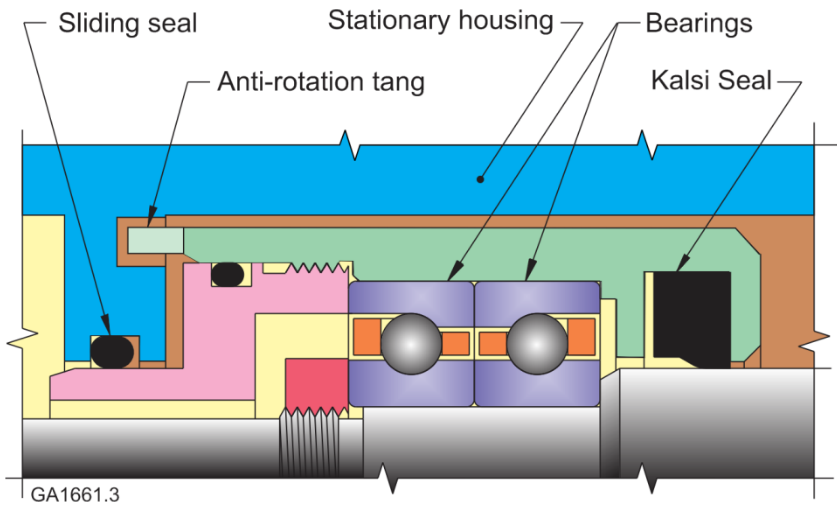

Figure 15

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).

10. 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 16). 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.[8] In high rpm rotating housing applications, higher leak rates may occur because the rubber must respond dynamically to the rapidly fluctuating radial gland dimension.

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8 High differential pressure causes a small portion of the rotary shaft seal to protrude into the extrusion gap clearance between the rotating seal housing and the shaft. Any static lateral offset between the shaft and the housing means the extrusion gap is minimum at one angular location, and maximum at another location. If the seal and seal housing rotate around the shaft, the protruding seal material is exposed to minimum and maximum extrusion gap clearance once per revolution. This heats and fatigues the protruding material, leading to accelerated seal extrusion damage. It also applies additional torque to the seal, encouraging the seal to slip circumferentially with respect to the housing.

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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 11 and 12 can be adapted to rotating housing applications.

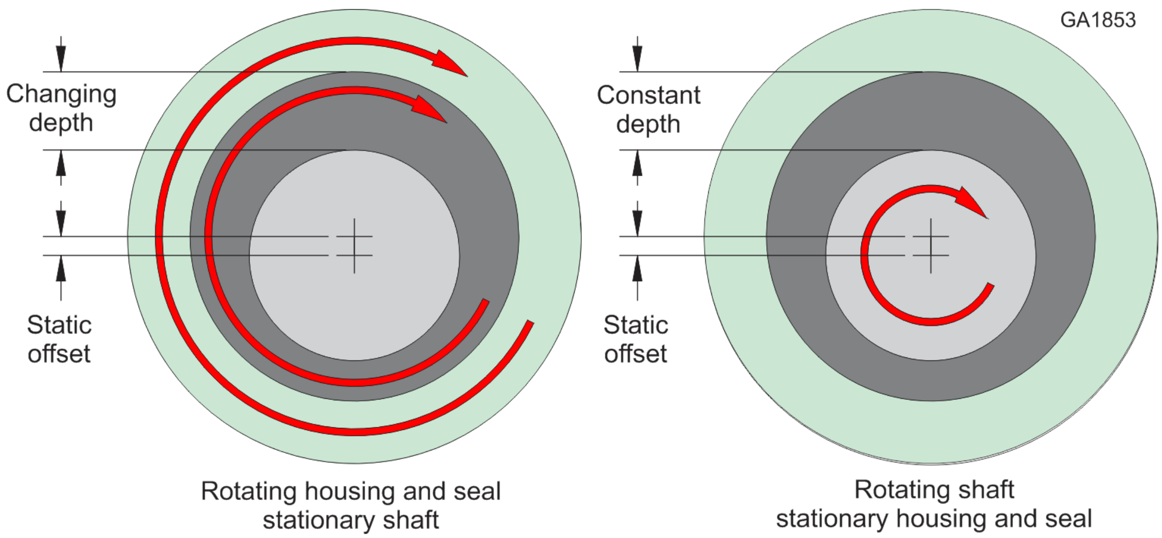

Figure 16

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.

11. Dynamic sealing at the housing bore

In all of the examples in this chapter, the dynamic interface (the location of slippage) is between the inner surface of the rotary seal and the outer surface of the mating shaft. While it is possible to design seals to slip against a housing bore, such designs are not ordinarily recommended. For one reason, mechanical means are needed to keep the seal from slipping relative to the shaft. For another reason, with any given shaft and bore size, slipping against the housing bore increases surface speed, which increases seal-generated heat.