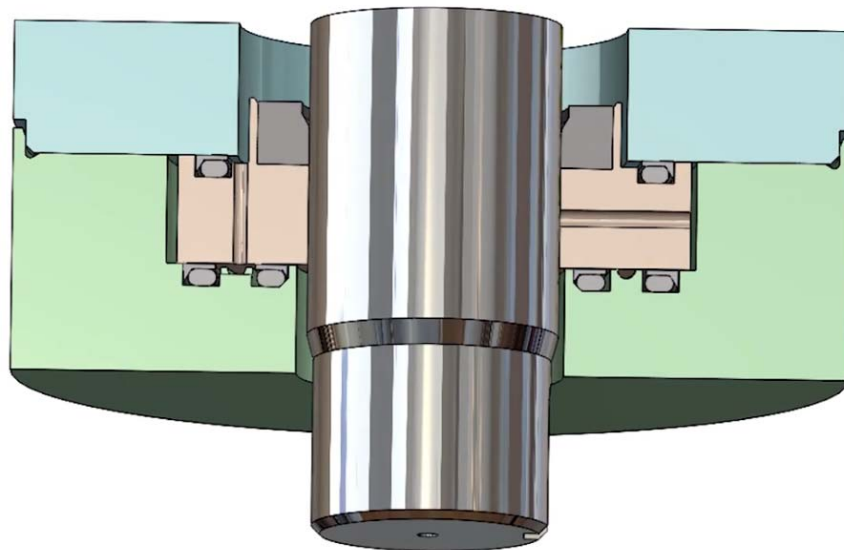


## Chapter D17

Axially force balanced, laterally floating backup rings



### Revision 5 June 29, 2020

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/>.

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## 1. Introduction

One factor that sets Kalsi Engineering apart from other seal companies is our ability to provide detailed seal implementation advice, based on decades of mechanical design and seal testing experience. This enables our customers to take a systems approach to design, achieving maximum rotary seal performance.

Over the years, Kalsi Engineering has conducted research to determine the best ways to employ Kalsi-brand rotary seals in high pressure sealing applications that involve runout, misalignment, and pressure-induced component deformation. The result has been a series of mechanical improvements,<sup>1,2,3,4</sup> culminating in the patented<sup>5</sup> floating backup ring arrangements that are described in this chapter. These backup ring arrangements are axially force balanced, and radially pressure balanced. These features allow the smallest practical extrusion gap to be used, while accommodating shaft runout and misalignment. By minimizing the extrusion gap, the rotary seals can withstand higher pressures for longer periods of time.

### ***Licensing and technical assistance***

The floating backup ring arrangements are applicable to many different types of fluid sealing equipment, including rotating control devices, hydraulic swivels, cement swivels, coring swivels, etc. [Contact Kalsi Engineering](#) to obtain technical assistance and discuss licensing options, which can range from a per-unit royalty to incorporating the royalty into the seal price.

## 2. Description of the problem

Rotary seals are sometimes used to retain high pressure lubricant within a housing by sealing the clearance between the housing and a bearing-guided rotatable shaft. The clearance is often referred to as the extrusion gap. High pressure, combined with eccentric shaft motion, is a significant challenge for any rotary seal. The seal can withstand higher pressure if the extrusion gap clearance between the housing and the shaft is small, but the clearance must be large enough to accommodate the full potential range of shaft misalignment, runout, and load-related deflection. Inadequate clearance between the

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<sup>1</sup> U.S. Patent 5,195,754, March 23, 1993, Laterally translating seal carrier for a drilling mud motor sealed bearing assembly.

<sup>2</sup> U.S. Patent 6,007,105, December 28, 1999, Swivel seal assembly.

<sup>3</sup> U.S. Patent 6,227,547, May 8, 2001, High pressure rotary shaft sealing mechanism.

<sup>4</sup> U.S. Patent 9,316,319, April 19, 2016, Pressure-balanced floating seal housing assembly and method.

<sup>5</sup> U.S. Patent 9,429,238, August 30, 2016, Dynamic backup ring assembly (other patents pending).

housing and the shaft results in metal-to-metal contact. This can produce seal-damaging heat, and damage the shaft and housing in ways that destroy the rotary seal.

High pressure expands the housing, which produces additional shaft misalignment and runout by increasing the mounting clearance of the bearings. The expansion also increases extrusion gap clearance.

The rotary seal is prone to extrusion damage at the extrusion gap, and motional wear at the sliding interface. Pressure extrudes seal material into the extrusion gap, and this material is nibbled away by cyclic runout-related stress. Friction at the sliding interface heats the seal and reduces its modulus, which makes it less extrusion resistant. Floating backup rings and Kalsi-brand rotary seals work together to mitigate these problems.

### 3. Overview of the floating backup ring arrangement

Figure 1 shows the key aspects of the preferred floating backup ring design, which is located axially by a simple stacked housing arrangement. The bulkhead and retainer housings are clamped together with a circular pattern of bolts that is not shown.

The backup ring has a journal bearing type fit with the rotary shaft and is prevented from rotating with the shaft by a radially oriented anti-rotation pin. The journal bearing fit defines the extrusion gap clearance for the Kalsi-brand rotary seal and causes the backup ring to align on the shaft. The journal bearing region is lubricated by the hydrodynamic pumping related leakage of the Kalsi Seal™.

The high pressure of the lubricant, acting over the sealed area on one side of the backup ring, produces a first axial force that acts on the backup ring. In Figure 1, the sealed area is defined by the rotary seal and a face-type sliding seal.

The high pressure is conducted axially through the backup ring, allowing it to act over the sealed area of the face-type force balancing seals that are shown in Figure 1. This produces a second axial force that acts on the backup ring in a direction that is opposite to the first axial force. Because the sealed areas are substantially equal, the first and second axial forces are substantially equal, and the net axial thrust on the backup ring is negligible. This axial force balance situation leaves the backup ring free to translate laterally to accommodate shaft eccentricity resulting from misalignment, dynamic runout, and lateral deflection.

The bulkhead housing defines clearances that accommodate lateral translation of the shaft and the backup ring. Radial overlap between the backup ring and the bulkhead housing prevents the lateral translation from exposing the face-type seal grooves.

The backup ring is located axially by a pocket created by the oppositely facing shoulders of the bulkhead and retainer housings. This pocket is sized to be only a few thousands of an inch longer than the backup ring, assuring minimal face type extrusion gap clearances at the three face-type extrusion gaps.

The backup ring is separate from the bulkhead and retainer housings, so it is immune from pressure-related expansion. This helps to minimize the extrusion gap clearance between the backup ring and the shaft.

The high-pressure rotary seal is compressed between the shaft and a thin axial extension of the floating backup ring. By being mounted in this manner, the seal is substantially isolated from compression changes related to shaft runout. The hydrodynamic interfacial lubrication provided by Kalsi-brand rotary seals reduces friction, wear, and seal generated heat, and the cooler operation promotes high pressure extrusion resistance.

The hydraulic forces that act axially on the backup ring are radially offset, which imparts a cross-sectional twisting action on the backup ring. The axial location of the rotary seal within the backup ring provides a radially outwardly acting countervailing force that controls the cross-sectional twisting action.<sup>6</sup> The ability to counteract twisting allows the clearance between the backup ring and the shaft to be very small, to facilitate maximum high pressure seal performance.

Smaller diameter backup rings can be constructed of entirely of a suitable copper-based bearing alloy. Larger diameter backup rings should be constructed primarily of steel, with a bearing alloy liner, to minimize pressure-related deformation and differential thermal expansion and contraction.<sup>7</sup>

#### ***Plastic backup rings for extreme pressure sealing***

Figure 2 is an enlargement of Figure 1 that shows the 712-Series plastic backup rings<sup>8</sup> that are used in extreme high pressure rotary sealing to bridge the gap between the floating backup ring and the housings that locate the backup ring axially. These plastic backup rings also provide a significant reduction in friction, which makes it easier for the metal backup ring to follow lateral shaft motion.

#### ***An outboard rotary seal option***

If desired, a low-pressure rotary seal can be mounted outboard of the floating backup ring, to protect the journal bearing of the backup ring from environmental contaminants, and to

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<sup>6</sup> U.S. Patent 9,845,879, "High Pressure Dynamic Sealing Arrangement".

<sup>7</sup> U.S. Patent 10,330,203, "High pressure dynamic sealing device".

<sup>8</sup> U.S. Patent 10,330,203, "High pressure dynamic sealing device".

retain lubricant for the journal bearing. If desired, this lubricant can be circulated outboard of the backup ring for improved shaft cooling. If the high-pressure rotary seal has a high hydrodynamic pumping related leak rate, the outboard low pressure seal can serve as a leakage collection seal, allowing the leakage to be conducted back to a lubricant reservoir.

### ***Pressure staging***

Pressure staging (Handbook Chapter D18) is the term for dividing fluid pressure among more than one rotary seal, to handle higher differential pressure. For example, with two pressure stages, the lubricant for the inboard, first stage Kalsi Seal is at pressure  $P_1$ , while the lubricant between the first and second stage Kalsi Seals<sup>®</sup> is at  $P_1/2$ . In such an arrangement, the first and second stage seals each retain a pressure differential equal to  $P_1/2$ . Any desired number of pressure stages may be used.

When pressure staging is used, the annular region between the backup ring and the bulkhead is vented to the pressure that is present in the journal bearing region of the backup ring. This creates a radial pressure balance situation, so that the backup ring is largely immune from pressure-related radial deformation.

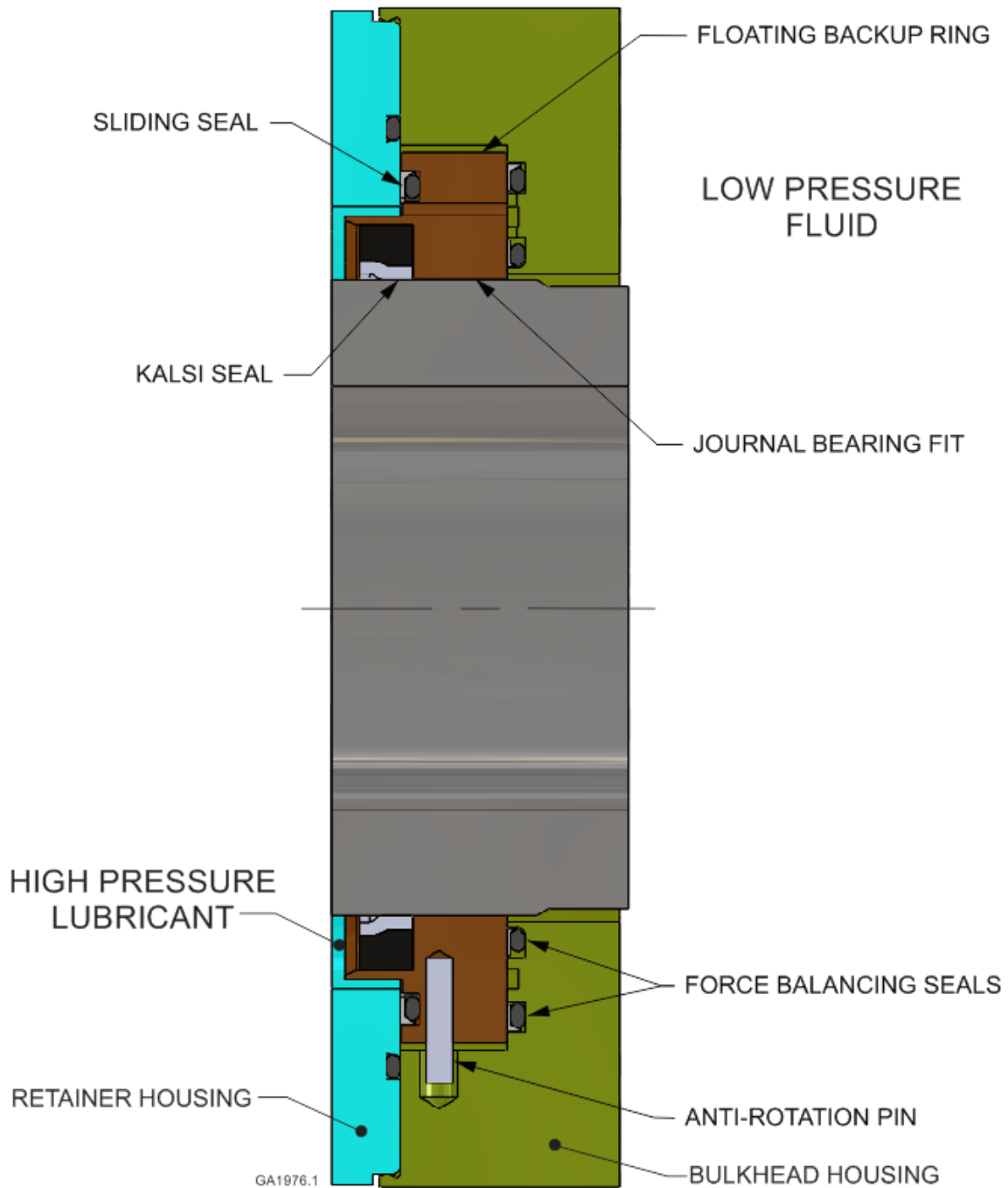
## **4. Using a pair of Kalsi Seals to retain a non-lubricating fluid**

Figure 3 shows how to use a pair of Kalsi-brand rotary seals to retain a high-pressure fluid that is not a lubricant. As with the Figure 1 arrangement, in Figure 2, the housings are clamped together with a circular pattern of bolts that is not shown.

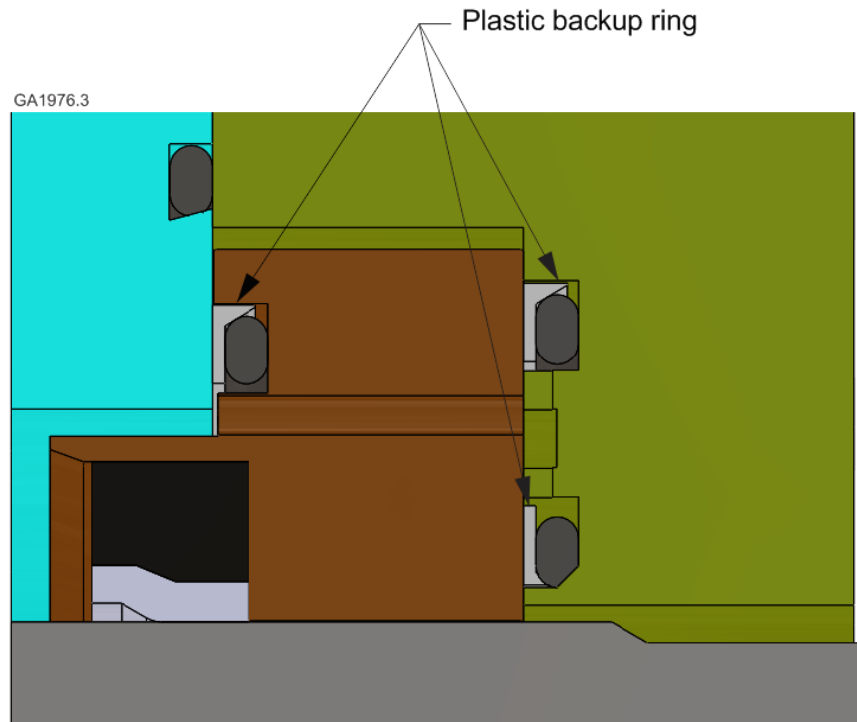
A partitioning seal is mounted in a partially balanced seal carrier (Chapter D16) and separates a high-pressure fluid (such as oilfield drilling fluid) from the seal lubricant. The partially balanced seal carrier has a journal bearing type relationship with the shaft, and translates laterally to accommodate shaft eccentricity resulting from misalignment, dynamic runout, and lateral deflection. This protects the partitioning seal from compression changes that would otherwise occur due to shaft eccentricity.

The seal lubricant is pressurized to a value that is somewhat greater than the pressure of the high-pressure fluid by a lubricant supply (not shown). For information on lubricant supplies, see Chapter D11.

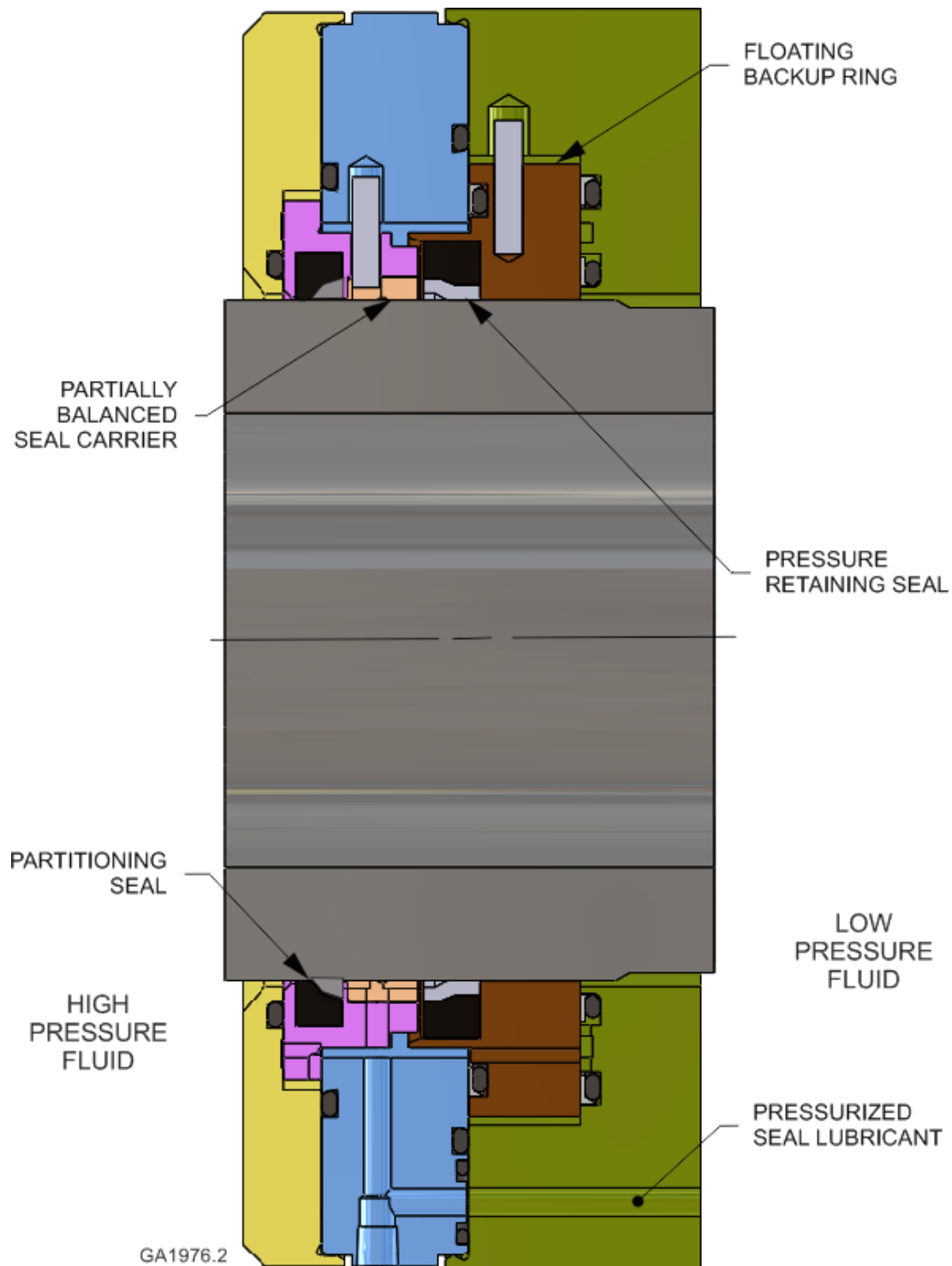
The pressure-retaining seal is mounted in a floating backup ring and retains the pressurized seal lubricant. The floating backup ring and the partially balanced seal carrier are both located axially by the outer housings. The partially balanced seal carrier serves as a removable gland wall for the pressure-retaining seal.

**Figure 1****A floating backup ring arrangement for higher levels of runout and misalignment**

In this stacked housing arrangement, the backup ring provides the smallest practical extrusion gap for the Kalsi-brand rotary seal, for maximum high pressure extrusion resistance. The rotary seal is isolated from runout-related compression changes by being mounted within an extension of the floating backup ring.

**Figure 2****Enlargement showing plastic backup rings**

The plastic backup rings are used in extreme high pressure rotary sealing, to prevent extrusion damage to the face-type O-rings that define the pressure areas. The plastic backup rings also dramatically reduce friction, which reduces the load on the journal bearing of the metal backup ring.

**Figure 3****Using two Kalsi-brand rotary seals to retain a non-lubricating process fluid**

In this arrangement, one rotary seal separates a non-lubricating process fluid from the pressurized seal lubricant, and another rotary seal retains the pressurized lubricant.



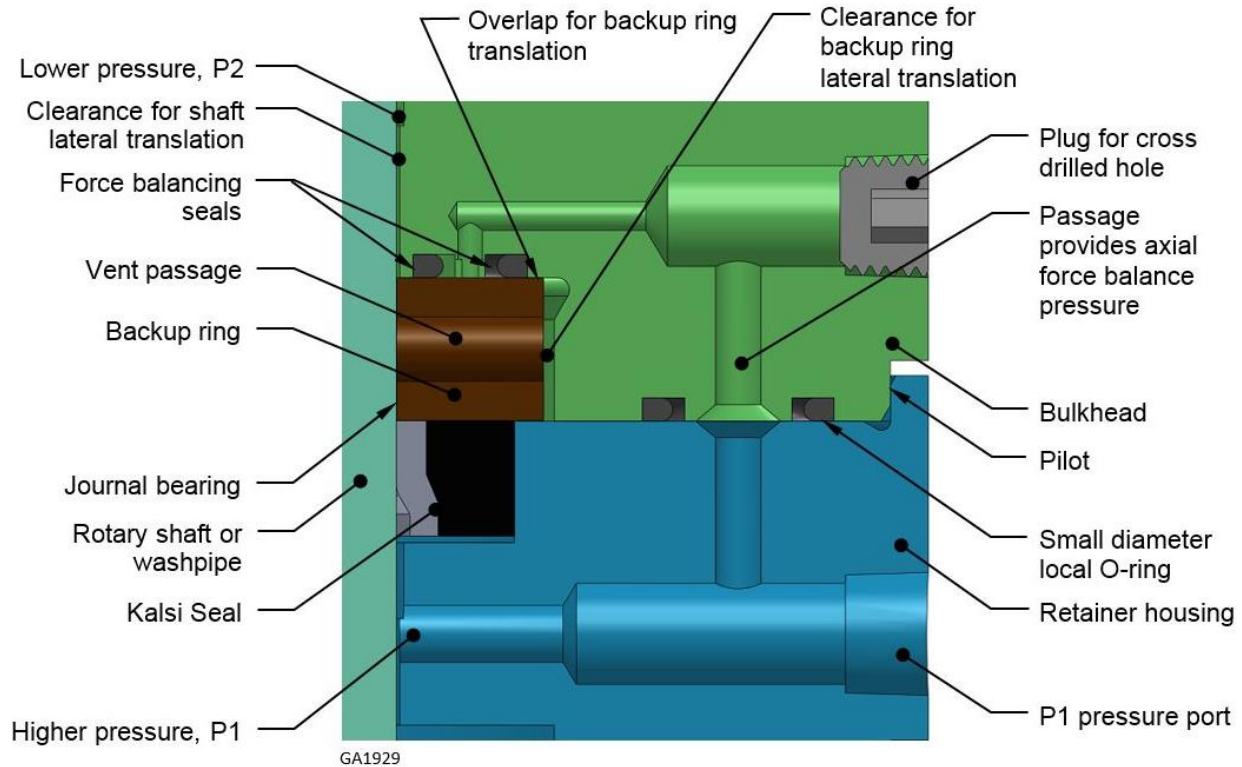
## 5. A floating backup ring for small diameters and minimal eccentricity

Figure 4 shows a floating backup ring for relatively small diameter applications having little eccentricity. As with Figure 1, in Figure 4 a metal backup ring is retained within and located axially by a stacked housing assembly that includes bulkhead and retainer housings. The stacked housing assembly is held together by a circle of bolts that are not shown. Unlike Figure 1, the rotary seal is compressed between the retainer housing and the shaft and bridges the gap between the bulkhead and retainer housings.

The principal advantage of the Figure 4 arrangement is that the backup ring is radially compact. The principal disadvantages, compared to the Figure 1 arrangement, are:

- The inability to easily counteract cross-sectional twisting of the metal backup ring. This becomes more critical as shaft diameter increases.
- Shaft runout and misalignment cause compression variation of the rotary seal, which means the arrangement can only tolerate modest levels of shaft eccentricity.
- The interface between the bulkhead and retainer housings is sealed by the rotary seal, at a location where the rotary seal is typically formed of elastomer. This makes the arrangement less capable of handling extreme pressure, compared to one that utilizes a face sealing O-ring and a plastic backup ring to seal the housing interface.
- Lubricant pressure must be conveyed between the bulkhead and retainer housings to reach the sealed region between the force balancing seals. The required cross-drilled holes fit between the bolts that secure the bulkhead and retainer housings together.

Despite these comparative disadvantages, the Figure 4 arrangement is useful in certain applications, such as small diameter hydraulic swivels, where shaft eccentricity is minor, and compactness is a virtue.

**Figure 4**

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

In this stacked housing arrangement, a metal backup ring is axially force balanced, and free to float laterally to follow misalignment and runout of the rotary shaft. Because misalignment and runout affect the compression of the Kalsi Seal, this arrangement is best suited for applications with relatively small levels of misalignment and runout, such as high pressure hydraulic swivels.

### **6. Evaluating pressure capacity through testing**

The pressure capacity of a rotary seal implemented with a floating backup ring is dependent on the combination of operating conditions. The most accurate way to determine the pressure capacity of a floating backup ring sealing system is to evaluate it in the actual equipment at intended conditions. If laboratory testing is to be used for determining pressure capacity, the test setup should be designed to reasonably simulate the expected thermal conditions, thermal system response, and mechanical dynamics expected in operation. For example, if the equipment has active cooling, design the test fixture with active cooling. Several tests of laterally floating backup rings are summarized below. The tests include both Plastic Lined Seals™ and seals constructed entirely of elastomer.

## 7. Testing for high pressure swivel conditions

### ***Introduction***

Various Kalsi Engineering seal types have been tested with metal backup rings at pressures and speeds directed at simulating the operating conditions of oilfield coaxial mud swivels and side entry cement swivels. Speeds have ranged from 28 to 252 ft/minute (0.14 to 1.28 m/s) and duration testing has been performed at pressures as high as 10,000 psi (68.95 MPa).

### ***A 10,000 psi, 368-hour duration test at 141 to 206 ft/minute***

A 4.50" (114.3mm) PN 750-1-318 Plastic Lined<sup>TM</sup> swivel seal was tested in a washpipe assembly (Figure 5) with a floating backup ring for 368 hours at 10,000 psi (68.95 MPa) and then disassembled for seal inspection. The seal was in excellent condition at the conclusion of the test (Figure 6). The test was conducted at 120 rpm (141.4 ft/minute) for 165-hours, 150 rpm (176.7 ft/minute) for 165-hours, and 175 rpm (206.2 ft/minute) for 38-hours using an ISO 68 viscosity grade seal lubricant. Cooling was provided by coolant circulating through a U-shaped tube inside the rotating washpipe at about eight gpm. The temperature of the fluid near the seal was about 103°F (39.4°C) at 120 rpm, 110°F (43.3°C) at 150 rpm, and 115°F (46.1°C) at 175 rpm.



**Figure 5**

### **10,000 psi rotary seal test**

The 368-hour, 10,000 psi (68.95 MPa) test of a PN 750-1-318 high pressure swivel seal was performed in this washpipe assembly. The seal was supported by a floating metal backup ring.



**Figure 6**

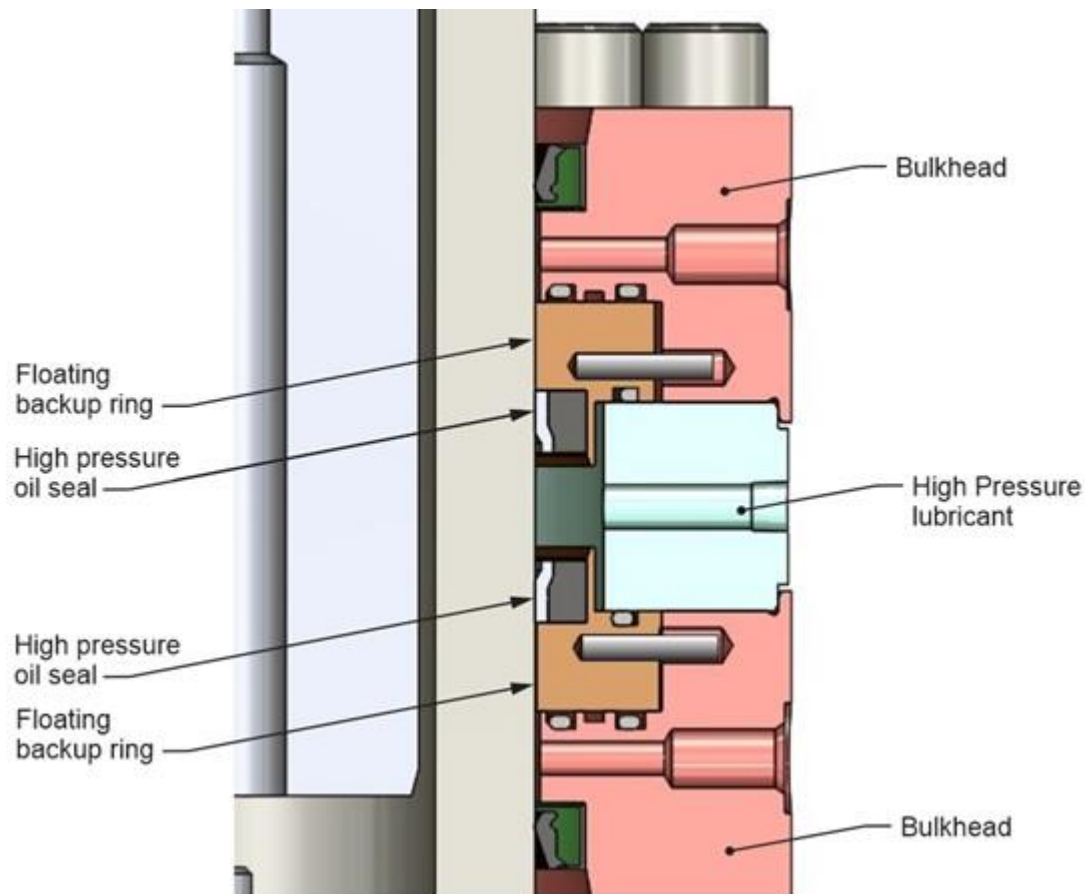
**The used high pressure seal after the 10,000 psi, 368-hour test**

The PN 750-1-318 high pressure swivel seal was still in good condition at the conclusion of the 368-hour, 10,000 psi (68.95 MPa) test. The floating metal backup ring is one of the keys to achieving this level of seal performance.

***A 9,500 to 9,800 psi, 320-hour duration test at 141 ft/minute***

A pair of 4.50" (114.3mm) PN 750-1-318 Super Wide Plastic Lined Seals™ were tested in floating backup rings for 320 hours at 9,500 to 9,800 psi (65.5 to 67.57 MPa) with an ISO 68 viscosity grade lubricant and then disassembled for seal inspection. Dynamic runout was 0.0045" to 0.006" FIM. Lubricant temperature was maintained at 100 to 120°F (37.8 to 48.9°C).

The fixture used in this high pressure seal test is illustrated in Figure 7. As typical to this test fixture, the used upper seal was in better condition than the used lower seal, but both seals were functional and capable of continued operation at the end of the test. The test demonstrated that the Super Wide Seal™ design is significantly more pressure capable, compared to the Extra Wide Plastic Lined Seals™ that were previously tested. The hydrodynamic pumping-related leak rate was approximately 5 ml/hour per seal.

**Figure 7****Test setup for the 9,500 to 9,800 psi, 320-hour test**

The test fixture for the 9,500 to 9,800 psi (65.5 to 67.57 MPa) test used a pair of oppositely facing PN 750-1-318 seals mounted in floating backup rings. The fixture simulates a pair of high pressure hydraulic seals in a single circuit hydraulic swivel. The pressurized ISO 68 viscosity grade lubricant was maintained at 100 to 120°F (37.8 to 48.9°C) by a modest amount of coolant circulation within the rotating shaft. Outboard lip seals are incorporated to capture the hydrodynamic pumping related lubricant leakage of the high pressure oil seals.



**Figure 8**

**The upper seal at the conclusion of the 9,500 to 9,800 psi, 320-hour test**

The upper high pressure seal was in excellent condition at the conclusion of the 320-hour 9,500 to 9,800 psi (65.5 to 67.57 MPa) test. Thanks to the super wide dynamic lip, the lower seal was still capable of continued operation as a high pressure hydraulic seal despite damage.

***A 7,500 psi, 1,000-hour duration test of Plastic Lined Seals at 252 ft/minute***

A pair of 2.75" (68.85mm) PN 682-5-303 Extra Wide Plastic Lined Seals were tested for 1,000 hours at 7,500 to 7,800 psi (51.71 to 53.78 MPa) lubricant pressure and a surface speed of 252 ft/minute (1.28 m/s). This surface speed is equal to 200 rpm on a 4.875" (123.83mm) OD washpipe. The seal lubricant was an ISO 150 viscosity grade synthetic hydrocarbon lubricant that was maintained at a temperature of 130°F (54.44°C).

The seal test fixture is shown in Figure 9. This early test used a radially pressure balanced backup rings and a floating washpipe to achieve a small extrusion gap clearance. The test was an important step in the development of floating metal backup rings. The excellent performance confirmed the benefit of radial pressure balancing and provided a strong incentive to continue our development of the floating backup ring.

The seals were in excellent condition at the conclusion of the 1,000-hour test (Figure 10) and could have kept running for much longer. A conventional washpipe packing only lasts a few hours under such extreme conditions, and an entire set may only last a day or two.





**Figure 9**

**7,500 psi rotary seal test fixture**

The 1,000 hour, 7,500 to 7,800 psi (51.71 to 53.78 MPa) test of Plastic Lined Kalsi Seals™ was performed in this washpipe-based test fixture. The floating washpipe was guided by radially pressure balanced backup rings. The rotary seals were in excellent condition after the test, demonstrating the advantage provided by the radially pressure balanced backup rings.



**Figure 10**

**The Plastic Lined Seals were in excellent condition after 1,000 hours at 7,500 psi**

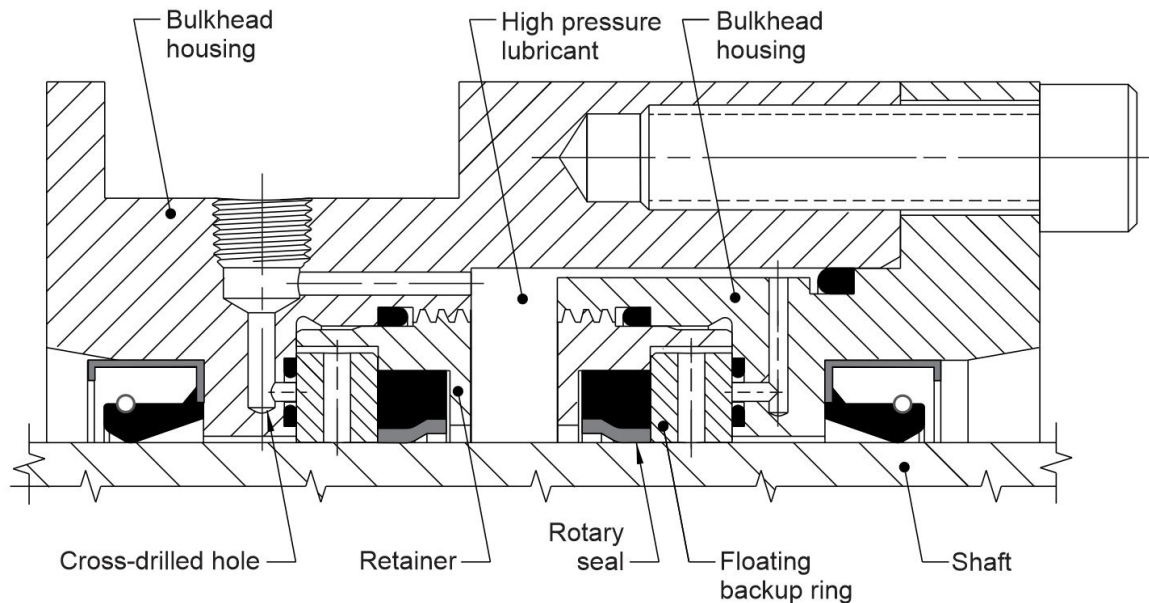
These photos were taken after a 1,000 hour, 7,500 to 7,800 psi (51.71 to 53.78 MPa) test of Plastic Lined Kalsi Seals under simulated oilfield washpipe operating conditions. The speed was equivalent to a 4.875" washpipe rotating at 200 rpm. The floating washpipe was guided by radially pressure balanced backup rings. The rotary seals were in excellent condition after the test, as can be seen in these photos. The upper test seal is shown in the upper photo, and the lower test seal is shown in the lower photo. Both seals are shown with the wavy side of the dynamic lip facing downward.

***A 5,000 psi, 950-hour test of HNBR seals at 252 ft/minute***

A pair of 2.75" (69.85mm) PN 655-4-114 Dual Durometer™ Kalsi-brand rotary seals were tested at 5,000 psi (34.47 MPa) lubricant pressure for 950 hours on a shaft rotating at 252 feet/minute with 0.01" (0.25mm) dynamic runout. The test used an ISO 320 VG lubricant maintained at 130°F. The seals were in good condition at the conclusion of the test and



could have continued running for much longer.<sup>9</sup> The configuration of this high pressure rotary seal test is shown in Figure 11.



**Figure 11**

**The fixture used for the 5,000 psi rotary seal test**

In this fixture, the rotary seals were still performing well after 950 hours of 5,000 psi (34.47 MPa) operation at 252 feet/minute with 0.010" (0.25mm) dynamic runout. Our experiences with this difficult-to-assemble test fixture led to significant improvements in bulkhead housing and retainer design.

***Slow speed testing of 4.50" seals at ~10,000 psi with floating backup rings***

A pair of 4.50" (114.30 mm) PN 682-7-318 Type F seals supported by floating backup rings were tested for 6.5 hours at 24 rpm with a lubricant pressure of 9,800 psi (67.57 MPa). The test was designed to replicate the operating conditions of an oilfield cementing swivel. The seal lubricant was an ISO 68 viscosity grade synthetic lubricant. Because of the interfacial lubrication provided by the Kalsi Seals, the bulk lubricant temperature did not exceed 96°F (35.6°C), and the seals were in excellent condition after the test.

[Contact](#) our staff to discuss high pressure swivel seals and the benefits provided by floating backup rings.

<sup>9</sup> OTC-26899-MS, "Advancements in Extreme Pressure Rotary Sealing".

## 8. Testing for RCD operating conditions

### ***Testing for RCD sealing conditions in a 2.75" fixture***

We performed 200 and 300-hour tests of -303 Extra Wide Plastic Lined Kalsi Seals™ mounted in a floating backup ring assembly simulating the extrusion gap clearance that is obtainable in an oilfield rotary control device (RCD). The purpose of the testing was to gain insight regarding how Plastic Lined Seals in floating backup rings would perform in a typical RCD dynamic pressure rating test.

The seals were tested with 3,000 psi (20.68 MPa) lubricant pressure and 0.010" (0.25mm) dynamic shaft runout, FIM. The speed was 750 rpm on a 2.75" (69.85mm) shaft, which is equivalent to 200 rpm on a 10.375" (263.53mm) shaft. At 750 rpm, however, the 2.75" test fixture subjects the rotary seals to 3.75 times more runout cycles, compared to a 10.375" shaft at 200 rpm. The seals were tested with an ISO 150 viscosity grade synthetic lubricant that was maintained at 200°F (93.33°C).

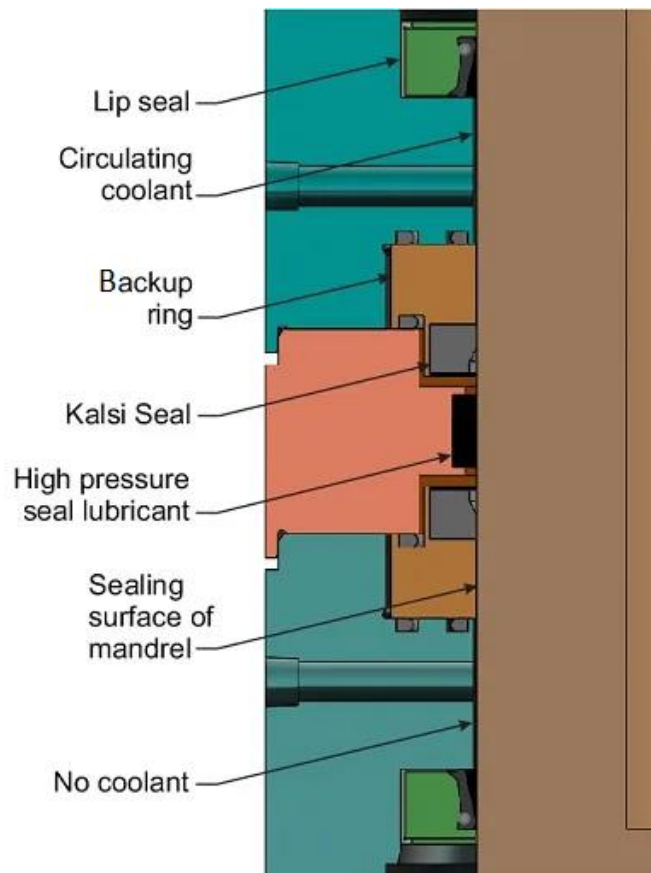
### ***Testing 10.50" RCD seals at 1,500 psi in floating backup rings***

A pair of 10.50" (266.7mm) PN 381-35-11 RCD seals were tested at 1,500 psi (10.34 MPa) with an ISO 320 viscosity grade lubricant and 0.0035" (0.09mm) to 0.0100" (0.25mm) dynamic runout (FIM) in floating backup rings. An objective of the test was to evaluate the performance of a full-scale RCD seal with a low hydrodynamic pumping related leak rate when used in a floating backup ring.

The test arrangement is shown by Figure 12. A water-glycol mixture was circulated above the upper seal at three to four gallons a minute. This circulation was intended to simulate the cooling provided in an actual RCD by exposure of a short portion of the mandrel to the circulating drilling fluid. Because of its intentional axial separation from this coolant circulation, the lower seal was the designated test seal. The coolant temperature ranged from 108 to 114°F (42.4 to 45.5°C), and the seal lubricant temperature above the lower seal, ranged from 210 to 235°F (98.9 to 112.8°C).

The test included four hours at 90 rpm (247.4 ft/minute), 17.8 hours at 150 rpm (412.3 ft/minute), and 120.65 hours at 190 rpm (522.3 ft/minute). The water-glycol mixture was unable to properly lubricate the journal bearing portion of the upper backup ring, resulting in damage to the mandrel that caused failure of the upper seal at 142.45 hours. The lower backup ring was adequately lubricated by the hydrodynamic pumping-related lubricant leakage of the lower seal, and the lower seal was still in excellent condition at the conclusion of the test (Figure 13).

We offer a wide range of RCD sealing solutions, including seals that do not require pressurized lubricant or a floating backup ring. [Contact](#) our staff to discuss the choices that are applicable to your RCD sealing requirements.



**Figure 12**

**Full size RCD seal test fixture**

The 10.50" (266.7mm) RCD seal test fixture utilizes two rotary seals to eliminate hydraulic thrust on the bearings that support and guide the mandrel. The seals can be mounted in floating backup rings or non-floating seal carriers, depending on the objectives of the test. Outboard lip seals define chambers where coolant can be circulated if desired.



**Figure 13**

**The test seal was in excellent condition**

Because of the relatively small extrusion gap clearance provided by the floating backup ring, the 10.50" (266.7mm) test seal was still in excellent condition after 142.45 hours of operation at 1,500 psi (10.34 MPa) and 90 to 190 rpm (247.4 to 522.3 ft/minute). These excellent results suggest that floating backup rings can provide a significant benefit in RCD applications that have requirements for minimal lubricant leakage.