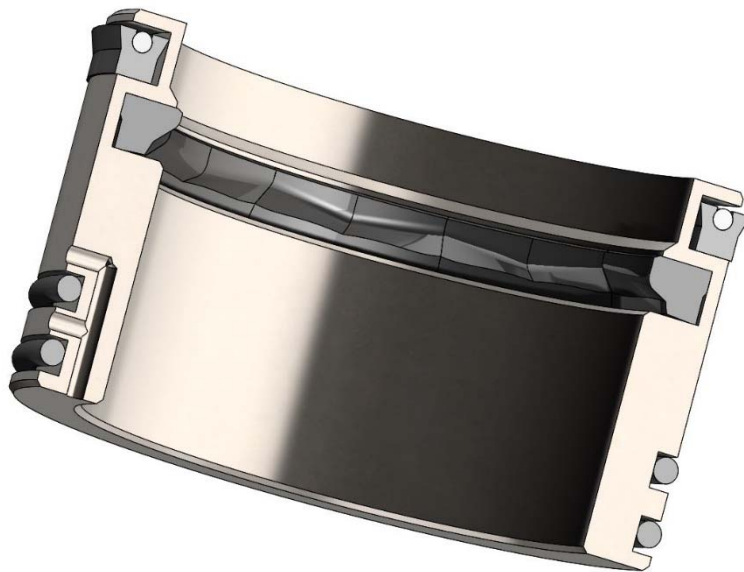


Chapter D14

Shaft guided compensation pistons



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. Compensation piston overview

In equipment that is submerged in a high-pressure environment, such as oilfield downhole drilling tools, the lubricant must be pressure balanced to the environment to protect the rotary seals from excessive differential pressure. In other words, the pressure of the lubricant within a downhole tool must be balanced to the ambient downhole pressure, to avoid creating a differential pressure across the rotary seals that is equal to the ambient downhole pressure. Such pressure balancing can be done with various devices, including bladders and diaphragms, but is often accomplished with an annular, shaft guided compensation piston¹.

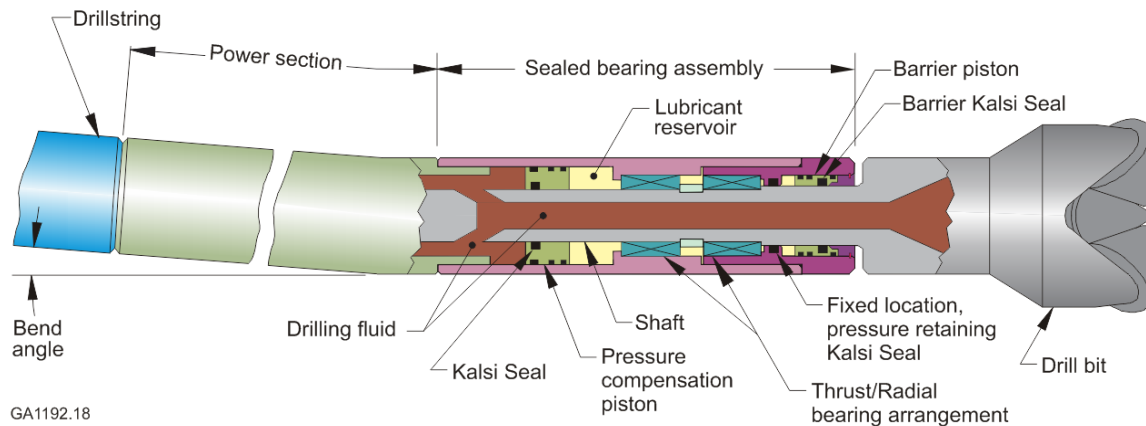


Figure 1

Schematic of a typical oilfield sealed bearing downhole drilling mud motor

Figure 1 is a schematic of an oilfield sealed bearing mud motor. It shows the use of two axially movable annular pistons, a “pressure compensation piston”, and a “barrier compensation piston”. The pressure compensation piston balances the bearing lubricant pressure to the pressure of the drilling fluid within the drillstring bore. The stroke of the pressure compensation piston defines the lubricant reservoir for the Kalsi Seal™ in the pressure compensation piston and the fixed location Kalsi Seal. The barrier compensation piston provides a clean lubricant environment for the fixed location Kalsi Seal, and balances that lubricant pressure to the well annulus pressure, and its stroke defines a lubricant reservoir for the barrier seal. The barrier compensation piston also limits the deflection of the shaft, so that the fixed location Kalsi Seal can have a relatively small extrusion gap. (Not all sealed bearing mud motors use a barrier compensation piston.)

When the surface mud pumps are running, the drilling fluid flows through the drillstring, exiting into the annulus of the well through the nozzles (not shown in

¹ A "compensation piston" is sometimes referred to as a "compensating piston".

Figure 1) of the drill bit. The annulus pressure is lower than the drillstring bore pressure due to the pressure drop that occurs as the drilling fluid passes through the drill bit nozzles. The fixed location Kalsi Seal has to withstand the pressure difference between the lubricant and the annulus (i.e., the pressure drop across the bit).

A compensation piston accomplishes six important tasks in a mud motor:

1. Partitioning the lubricant from the drilling fluid environment,
2. Substantially balancing the lubricant pressure to the drilling fluid environment,
3. Accommodating lubricant thermal expansion,
4. Providing a lubricant reservoir to accommodate hydrodynamic pumping related seal leakage,
5. Limiting the deflection and stress of the rotary shaft, and
6. Isolating the piston mounted rotary seal from the effects of shaft runout and deflection, for improved abrasion resistance.

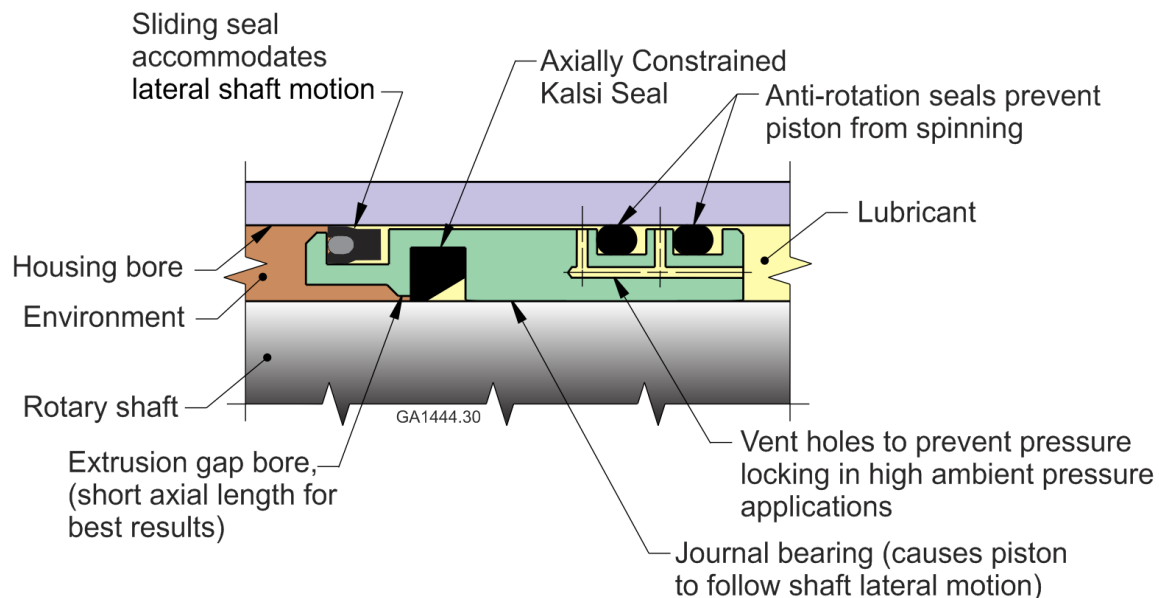


Figure 2

A typical mud motor compensation piston

The compensation pistons in an oilfield mud motor are shaft guided by a journal bearing relationship with the shaft. This minimizes the compression changes and the lateral sliding motion at the environment-side gland wall that the rotary seal experiences due to lateral shaft movement. A larger clearance is provided between the piston and the housing, to accommodate lateral shaft misalignment and deflection without overloading the journal bearing or binding the piston. The extrusion gap clearance of the mud-exposed sliding and rotary seals is relatively large, to minimize wear (see Figure 8).

Journal bearing and extrusion gap sizing

A typical mud motor compensation piston is shown in Figure 2. The Axially Constrained Kalsi Seal™ runs on the shaft, and the sliding seal runs on the housing bore. A journal bearing is defined on the lubricant side of the Kalsi Seal, and an extrusion gap bore is defined on the environment side. Guidelines for designing the length and clearance of the journal bearing are provided in Chapter D15.

The extrusion gap bore diameter should be about 0.040” (1.02 mm) larger than the shaft diameter to minimize abrasion of the Kalsi Seal and the shaft. Likewise, from an abrasion resistance standpoint one should keep the axial length of the extrusion gap width very short. An axial extrusion gap width in the range of 0.02 to 0.04” (0.51 to 1.01 mm) is recommended. Extrusion gap considerations are discussed in Chapter D7.

Flow restrictors

Some sealed bearing mud motor assemblies incorporate mud flow restrictors with flow bypass orifices² to minimize the pressure differential acting across the fixed location Kalsi Seal. Flow restrictor arrangements are particularly relevant in high temperature wells, because the high temperature reduces the modulus and extrusion resistance of the fixed location rotary seal.

Swab pressure

When a mud motor is lowered in a well, it displaces drilling fluid and produces swab pressure below the drill bit³. The fluid flows up through the drill bit nozzles, through the mud motor, and into the drillstring bore. The pressure drop across the drill bit nozzles causes a differential pressure across the fixed location rotary seal, acting from the drilling fluid side. This “reverse pressure” situation distorts the fixed location Kalsi Seal, making it more prone to abrasive invasion. For this reason, a barrier seal is recommended outboard of the fixed location Kalsi Seal. In Figure 1, the barrier seal is mounted in an axially movable compensation piston that balances the pressure of the barrier lubricant to the drilling fluid environment in the annulus of the well. Chapter D10 describes other, more compact arrangements for incorporating a barrier seal outboard of a Kalsi Seal, and balancing the pressure of the barrier lubricant to the environment.

2. Compensation piston rotary seal groove evaluation

For groove details such as surface finish, corner radii, etc., see Chapter D5. For groove bore diameter and tolerance, see the [website](#) and Chapter D5. After sizing the groove diameter, perform a tolerance and clearance stackup calculation using the equations in

² For examples of flow restrictors with bypass orifices, see expired US Patents 3,857,655 and 3,894,818.

³ For a description of swab and surge pressure, see U.S. Patent 6,220,087.

Chapter D5 to verify that the minimum and maximum Kalsi Seal radial gland dimension falls within the allowable range when the piston is laterally offset to the maximum amount permitted by the various clearances and tolerances.

3. Piston to housing fit

In an oilfield mud motor, the compensation piston OD should ordinarily establish a relatively large clearance with the housing bore so that the piston can move laterally with shaft deflection without overloading the journal bearing (see Figure 2). The primary equipment bearings should react most of the shaft side load, so that the journal bearing of the pressure compensation piston only carries part of the intermittent peak side loads.

In applications such as mud motors, where a compensation piston is sometimes mounted in a separate housing that is threaded to the main bearing housing, the clearance between the housing bore and the piston OD also helps to prevent outright shaft binding related to housing to housing threaded connection misalignment. To minimize the potential for such shaft binding, consider implementing piloting diameters between the housings, and pay attention to shoulder squareness.

4. Compensation piston sliding and anti-rotation seals

The sliding seal serves as a sealed partition between the lubricant and the environment. The anti-rotation seals provide additional friction to prevent piston rotation that can otherwise occur due to seal and bearing torque and drillstring stick-slip⁴. The sliding and anti-rotation seals may be any suitable type, such as O-ring energized lip seals or O-rings. Larger cross-sections are generally preferred, because they can more easily accommodate lateral offset between the piston and mating bore. O-ring energized lip seals may be preferable for the sliding seal because they are immune to spiral failure⁵, and because the lips define scraping edges that help to exclude the environment. The body of an O-ring energized lip seal should be made from elastomer rather than plastic, so the seal can accommodate large compression changes.⁶ The sliding seal must continue to seal while exposed to large compression variations, and it must have excellent compression set

⁴ For information on stick-slip, see SPE 145910, “**Drill Pipe Measurements Provide Valuable Insight into Drill String Dysfunctions**”. Severe vibration occurs during the slip phase.

⁵ For more information on spiral failure and its prevention, see Leonard J. Martini's 1984 book “**Practical Seal Design**”.

⁶ Always check for availability before specifying O-ring energized lip seals. Be aware that the clearance recommendations provided in catalogs for O-rings and O-ring energized lip seals are typically for high pressure service. Such clearance recommendations may not be appropriate for compensation pistons, and may lead to binding of the piston when non-concentric conditions occur.

resistance and adequate initial compression. Materials with poor compression set resistance, inadequate temperature range, low friction, or incompatibility with environmental fluids should be avoided.

Enough anti-rotation seals must be employed to prevent piston rotation. In the experience of Kalsi Engineering, two 0.210" (5.33 mm) 90 Shore A O-rings prevented spinning of the piston type shown in Figure 2. The more anti-rotation seals there are, the less likely the piston is to spin⁷, but the greater the pressure buildup before the piston moves in the axial direction—which can be hard on the rotary seal from an abrasive exclusion standpoint.

Preventing pressure locking of the sliding seals

In equipment exposed to high ambient pressure (such as oilfield downhole mud motors), only the sliding seal can be permitted to achieve a sealed relationship with the housing bore. The sealing function of the anti-rotation seals must be defeated so that the seals cannot trap atmospheric pressure at the time of assembly, and then become pressure locked by the high ambient downhole pressure. Pressure locking (Figure 3) puts the full ambient downhole pressure across the seals, which tends to bind the piston and exposes the Kalsi Seal unnecessarily to high differential pressure.

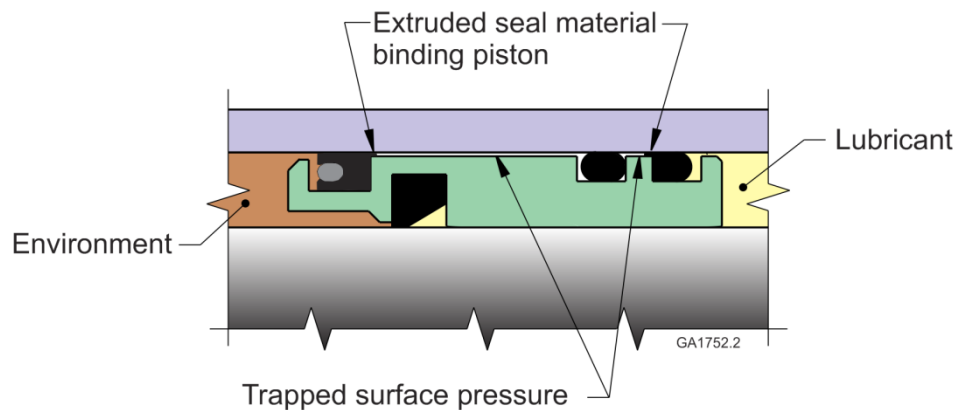


Figure 3

Do not allow pressure locking of the anti-rotation seals

If the regions between the sliding and anti-rotation seals are not vented to the lubricant, the differential pressure across the outboard seals is equal to the environment pressure, because atmospheric pressure is trapped between the seals. This causes significant friction when the unit is exposed to a high-pressure environment. The friction binds the piston, which in turn increases the differential pressure across the rotary seal.

⁷ Rotational slippage of the piston can also be related to an excessively rough shaft surface finish generating excessive rotary seal torque.

The best way to defeat the sealing function of the anti-rotation seals is to incorporate cross-drilled holes, as shown in Figure 2. Another way is to simply drill radial vent holes completely through the wall of the piston, as shown in Figure 4. If such radial vent holes are used, the hole breakout location at the journal bearing must be deburred to prevent inadvertent interference with the shaft, as shown in Figure 4. A ball end mill or a high-speed hand grinder with a spherical grinding stone can be used for such deburring.

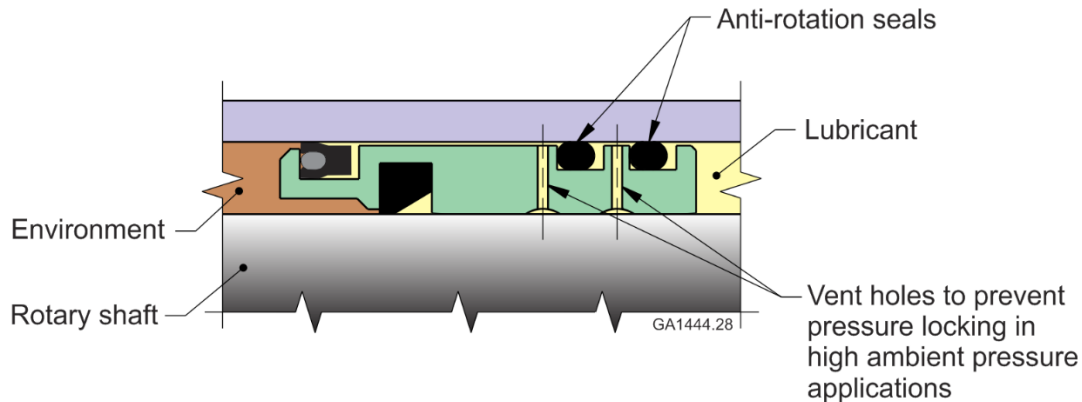


Figure 4

Using radial vent holes to prevent pressure locking

Radial vent holes can be used to prevent pressure locking of the seals that are located on the outside of the piston. The hole breakout locations at the journal bearing have been deburred with a high-speed hand grinder, using a spherical grinding stone.

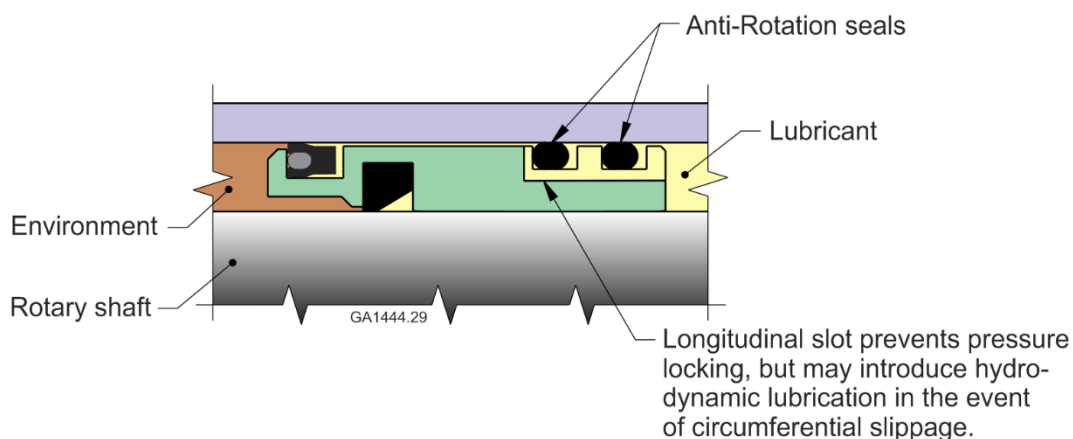


Figure 5

Longitudinal venting slot may increase the risk of circumferential seal slippage

Although a longitudinal slot prevents pressure locking, it may increase the risk of circumferential seal slippage for two reasons. Firstly, it removes some of the seal from compressive engagement with the housing bore. Secondly, it may introduce lubrication in the event of circumferential seal slippage.

A less desirable way to defeat the sealing function of the anti-rotation seals is to mill a longitudinal slot across the anti-rotation seal grooves, as shown in Figure 5. The drawback to this method is that the longitudinal slot removes a portion of the seal from compression, and therefore risks introducing hydrodynamic interfacial lubrication in the event of circumferential seal slippage.

Dimensioning the groove diameters of the sliding seals

Do not make the beginner's mistake of dimensioning the seal groove of a piston sliding seal by its radial depth from the outer diameter of the piston. Seal grooves for radially compressed seals are dimensioned by their **diameter**, not by their radial depth. The radial groove depth that is referenced in seal catalogs is a typically a reference dimension, and typically refers to the radial distance between the cylindrical groove surface and the mating bore of the housing.

Evaluating sliding seal compression

The sliding seal must be large enough to accommodate worst case lateral motion without losing compression. Perform a tolerance and clearance stackup to be sure that sufficient compression remains under worst case conditions to retain suitable sealing interference⁸. The sliding seal groove should be wide enough to accommodate the maximum local compression that will occur if the piston is forced into contact with the housing bore.

To determine average radial gland depth, and minimum and maximum radial depth of the sliding seal gland assuming the piston has moved laterally into contact with the housing bore, use the equations below (Equation variables are given in Figure 6, and in Appendix 3).

Equation 1, Average Radial Sliding Seal Radial Gland Dimension:

$$W_{avg} = \frac{H_{avg} - K_{avg}}{2}$$

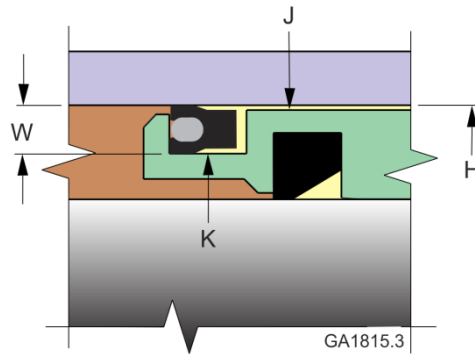
Equation 2, Minimum Radial Sliding Seal Radial Gland Dimension:

$$W_{min} = \frac{J_{min} - K_{max}}{2} - \frac{Ec_{max}}{2}$$

Equation 3, Maximum Radial Sliding Seal Radial Gland Dimension:

$$W_{max} = \frac{H_{max} - K_{min}}{2} + \frac{H_{max} - J_{min}}{2} + \frac{Ec_{max}}{2}$$

⁸ Kalsi Engineering can provide O-ring groove design software to customers who are designing compensation pistons, or other hardware that employs O-rings.

**Figure 6****Sliding seal groove equation variables**

The variables are in diameter format. E_c (not shown) is the eccentricity tolerance that can affect the position of the sliding seal groove relative to the piston outer diameter J . H is the housing bore diameter. K is the sliding seal groove diameter. W is the radial gland dimension of the sliding seal.

Understanding the size variations of sliding seals

The actual radial cross-sectional dimension of a newly installed sliding seal or an anti-rotation seal varies as a function of:

- Manufacturing tolerances (available from the sliding seal manufacturer),
- How much the installed seal is stretched, and
- How much the installed seal is thermally expanded.

The variations in cross sectional size should be understood to achieve adequate compression, and to allow adequate room within the seal groove to accommodate the thermal expansion of the seal.

The effect of sliding seal stretch

Sliding seals are typically stretched diametrically when installed, to ensure that they will fit past the housing installation chamfer without bunching and cutting. Another reason for installed stretch is because off-the-shelf seals are typically used, and they are only available in select sizes that may not be an exact fit for the housing bore that is being used. At any given temperature, the volume of a seal remains constant regardless of the amount of stretch, so as a result the cross section of the seal changes. Information on the observed reduction in cross section as a function of stretch are available from some sliding seal manufacturers.

With O-rings, the amount of stretch related cross-sectional reduction can be estimated by:

1. Based on the diameter of the O-ring cross section, calculate the area of the O-ring cross section using the conventional circle cross sectional area formula,
2. Multiply the cross-sectional area times the mean circumference of the relaxed O-ring to determine O-ring volume (Pappus Rule)⁹,
3. Determine the mean circumference of the O-ring gland,
4. Divide the O-ring volume by the mean circumference of the gland to determine the approximate cross-sectional area of the stretched O-ring, and
5. Reverse the circle cross sectional area formula to determine the diameter of the cross section of the stretched O-ring.

The effect of sliding seal thermal expansion

The thermal expansion associated with any dimension of a sliding seal (in its relaxed, uninstalled state) can be determined by multiplying the dimension times the linear coefficient of thermal expansion of the elastomer.

Compression-induced O-ring length change

Ideally, to avoid potential seal damage and increased sliding friction resulting from seal over confinement, the sliding piston to housing seal should fit in a groove that accommodates its compressed thermally expanded MMC length, including any swelling that may be present. If the sliding seal is installed with considerably more compression than normally recommended by the seal manufacturer (to accommodate lateral offset, or to increase sliding friction) the fit of the seal with the groove may have to be evaluated. If desired, the fit of the sliding seal can be predicted with FEA.

With concentrically compressed O-rings, the compressed seal width can be calculated by hand, with good correlation to FEA results, by using the method shown in Figure 7.

⁹ “Machinery's Handbook”, 23rd edition.

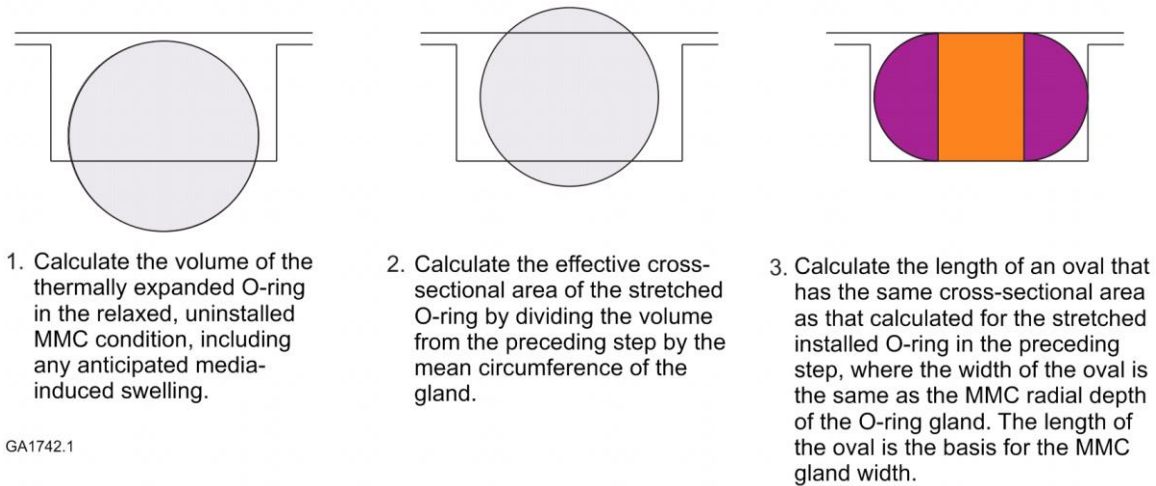


Figure 7
Fitting an O-ring gland

The approximate MMC width of an installed O-ring can be determined using this method.

Protecting the housing bore from abrasive crushing related damage

The piston diameter that defines the environment side extrusion gap for the sliding seal (Figure 8) should be kept smaller than the piston OD on the lubricant side of the sliding seal so that any contact between the piston OD and the housing bore occurs in the lubricated zone. This helps to minimize abrasive crushing in the extrusion gap that can damage the housing bore and can potentially impair axial motion of the piston. Housing bore damage from the crushing of abrasives in the sliding seal extrusion gap can compromise the sealing function of the sliding seal.

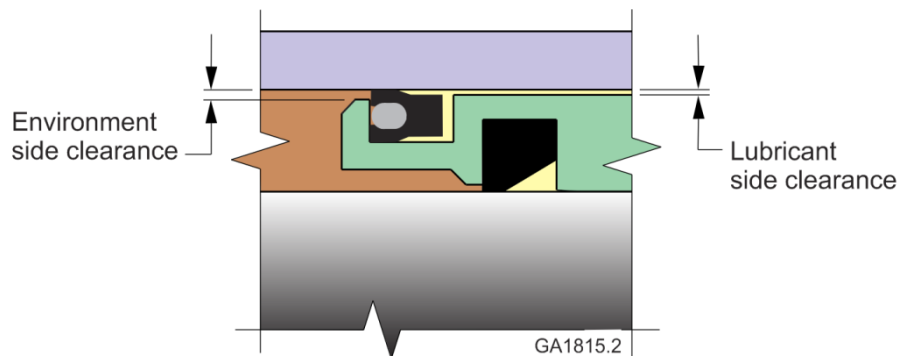


Figure 8
Environment side extrusion gap clearance

To help to avoid crushing environmental abrasives in the environment side extrusion gap clearance (which can damage the housing bore), keep the extrusion gap clearance larger than the lubricant side piston to housing clearance.

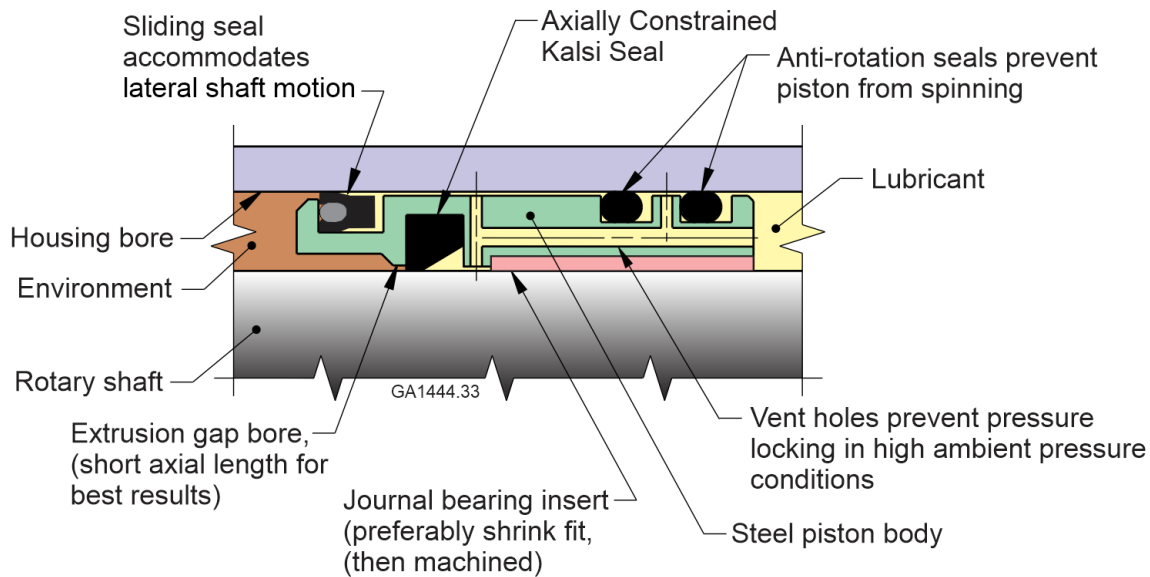
5. Construction materials

Compensation pistons can be a bearing bronze type material, or steel. Pistons may require corrosion resistant coatings or stainless steel construction, depending on the contemplated environment. Pistons manufactured from bearing bronze type materials will suffer increased seal groove wear in the event of circumferential seal slippage, may suffer accelerated wear of the bore that defines the extrusion gap if exposed to environmental abrasives, and may have differential thermal expansion issues in large diameters. In other words, a bronze-based piston expands more than the mating steel shaft at elevated temperatures, and contracts more than the mating steel shaft in cold weather conditions. This means that a bronze-based piston doesn't fit the shaft as well as a steel-based piston.

Steel-based pistons should typically be lined or coated with some type of suitable bearing material or treated to produce a compatible bearing surface. If a coating extends to the rotary seal extrusion gap, avoid coatings that leave ragged edges when machined, because such edges can damage the Kalsi Seal. Potentially suitable linings include slip fit-type DU-type bushings and shrink fit inserts (Figure 9) made from a suitable bearing material, such as bearing bronze. Shrink fit bearing inserts are preferred over slip fit bushings because they can be designed to have less total journal bearing clearance with respect to the shaft, and therefore better isolate the rotary seal from the harmful effects of shaft runout and deflection.

The mating housing bore

If exposed to a corrosive, abrasive environment, the housing bore that receives the compensation piston should be manufactured from, or coated with, a corrosion and abrasion resistant material that is compatible with impact loading and with the environment. The surface finish must be smooth enough to establish a sealed relationship with the sliding seal of the compensation piston.

**Figure 9****A steel piston with a shrink fit bearing insert**

In this compensation piston, the journal bearing fit is defined by a bearing insert that is a shrink fit with the steel piston body. Preferably, for the sake of maximum concentricity, the piston is finish-machined after the shrink fit occurs. Design the shrink fit to accommodate differential thermal contraction between the steel body and the bearing liner at the lowest anticipated operating temperature. If desired, the bearing insert can also be retained to the steel body by a radial pin, to provide additional security against circumferential slippage in extremely cold startup conditions.

6. Miscellaneous

The compensation piston should typically be located reasonably close to the primary radial bearings of the tool to isolate it from as much shaft deflection as possible. In oilfield mud motors, the shaft diameter should typically be kept as large as possible to minimize load-induced deflection, shaft fatigue, and shaft breakage. Using journal bearings to locate the shaft allows a much larger and stiffer shaft, compared to when rolling element radial bearings are used.

In mud motors, the barrier seal in the barrier compensation piston and the high differential pressure seal (the fixed location Kalsi Seal in Figure 1) are likely to be of different geometries, and possibly different materials, and are likely to employ different groove widths even though they have the same shaft diameter. Such seals should be carefully segregated so that they are not inadvertently installed in the wrong locations.

If desired, a high viscosity lubricant such as one having an ISO 1000 viscosity grade can be used to provide greater journal bearing load capacity¹⁰. Verify that the lubricant is

¹⁰ Increasing lubricant viscosity will increase the hydrodynamic pumping related leak rate of the Kalsi Seal, so make sure your reservoir size is compatible with anticipated seal leakage.

compatible with seal and piston materials; some lubricants have EP additives that attack copper and silver-based metals at temperatures above 210°F (98.9°C).

7. Lubricant filling in systems with compensation pistons

Position the piston to allow room for lubricant thermal expansion

Lubricants have a much higher coefficient of thermal expansion than steel. At initial fill, the piston must be positioned at an intermediate stroke location to avoid thermal binding. Failure to allow room for thermal expansion of the total lubricant volume within the tool can destroy seals and hardware (Figure 10). In addition to providing space for lubricant thermal expansion, also allow for the estimated inaccuracy in initial fill volume/piston position. For more information on this subject, see Chapter D13.

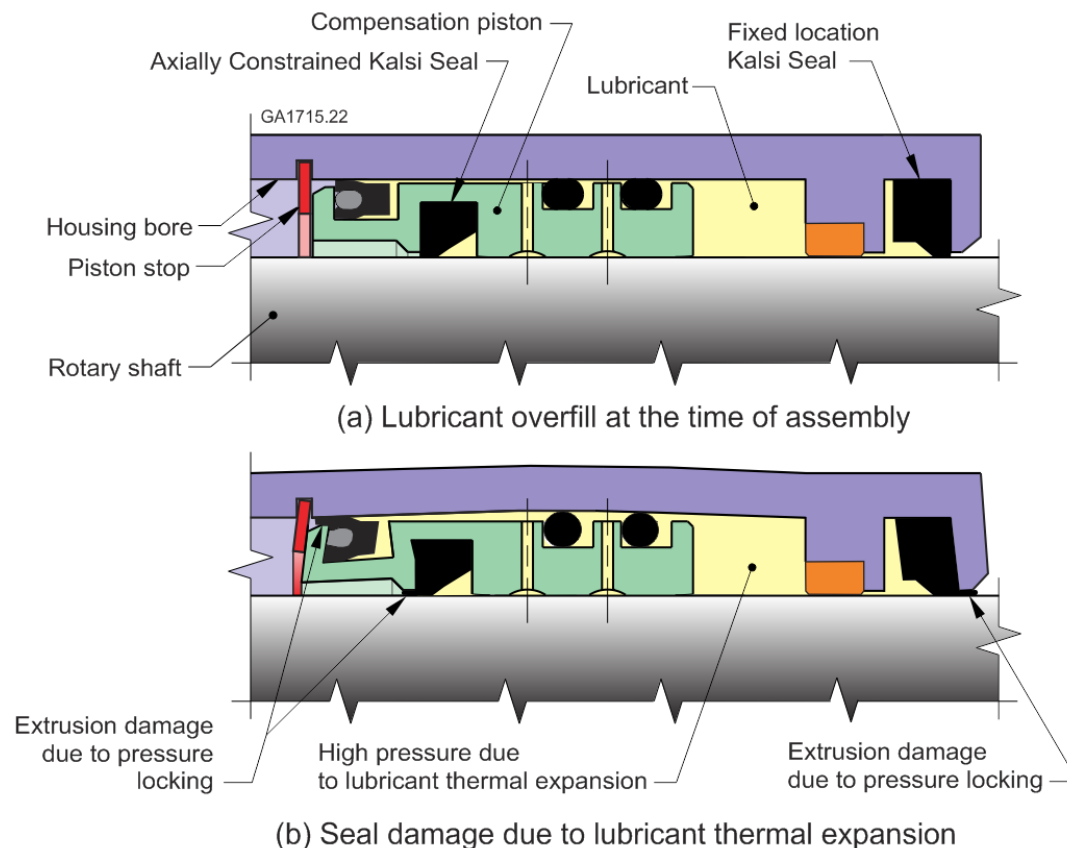


Figure 10

Allow enough room for lubricant thermal expansion

Overfilling the lubricant reservoir is a critical mistake that causes severe seal damage. In the upper image of this schematic, the compensation piston is filled to the 100% full position. As a result, the piston cannot move to accommodate thermal expansion of the lubricant. During operation, lubricant thermal expansion causes extremely high pressures that damage the seals, and can even permanently deform the metal components, as illustrated in the lower image.

Avoid trapping large amounts of air in the lubricant supply

In oilfield mud motors (and other equipment used in high ambient pressure environments) one must avoid trapping large amounts of air in the system, because the high lubricant pressure will collapse the air pockets, potentially causing the piston to bottom out at the empty position. If this happens, the entire ambient downhole pressure acts across the rotary seals. This causes rapid seal destruction, and (because of extreme downhole ambient pressures) may even damage (yield) the metal components.

Air entrapment must also be avoided in applications with vertically oriented shafts, because the air will rise to the top of the assembly. The resulting air pocket may starve the upper Kalsi Seal for lubricant, leading to premature failure. If vacuum filling techniques are not used, systems with lubricant hoses may require purging to eliminate air from the hoses.

Barrier compensation piston initial fill & stroke length

Figure 1 shows an example of a barrier compensation piston implemented in a mud motor sealed bearing assembly to protect the high-pressure seal from abrasives. As with pressure compensation pistons, a barrier compensation piston must be filled initially to an intermediate stroke position so that stroke is available to accommodate lubricant thermal expansion.

In addition to lubricant thermal expansion, the barrier compensation piston may have to accommodate a differential leakage rate between the high-pressure seal and the barrier seal. One basis for determining barrier seal stroke length would be to take into account the known leakage data scatter from various tests at a given temperature (Handbook Section C).

Alternately, select a fixed location Kalsi Seal that has a greater hydrodynamic pumping rate than the barrier seal, and provide a means to vent excess barrier lubricant to the environment.

8. Spring loading the pressure compensation piston

Because of the friction of the sliding and anti-rotation seals, a pressure compensation piston does not always balance the lubricant pressure exactly to the environment pressure. During periods of lubricant thermal expansion, the lubricant pressure tends to be higher than the environment pressure. As the lubricant is slowly consumed by the hydrodynamic pumping related leakage of the Kalsi Seals®, the lubricant pressure drops to a value that is lower than the environment pressure. When the pressure difference is enough to overcome seal friction, the pressure compensation piston moves axially and lubricant pressure becomes more equalized to the environment. During the intervals when

the lubricant pressure is equal to or less than the environment pressure, a non-axially constrained Kalsi Seal is prone to the skew and shuttling related wear mechanisms described in Chapter D9.

One way to combat skewing and shuttling related wear mechanisms is to spring load the pressure compensation piston to overcome sliding seal friction and impart extra pressure to the lubricant¹¹, as shown in Figure 11. While shown with a non-axially constrained Kalsi Seal, the spring-loaded arrangement is also beneficial to Axially Constrained Kalsi SealsTM because the spring can reduce or eliminate the occasions where the environment pressure is higher than the lubricant pressure. This technique should only be used in applications where the environment remains liquid enough to accommodate spring movement.

Ideally, the spring force needs to be strong enough to overcome the piston friction, and maintain a differential pressure that is sufficient to prevent seal skewing, even in the depleted position of the piston stroke. This requires a differential pressure of at least 15 psi (103 kPa) at 162°F (72.2°C), and at least 22 psi (15 kPa) at 375°F (190.6°C) based on testing with 2.75” ID seals.

Kalsi Engineering provides compression spring design software to assist customers who are designing spring loaded lubricant reservoirs. The spring material must be compatible with the environmental chemicals and temperature. When implementing a coil spring, be aware that the diameter of the spring increases as the spring is compressed. The spring must be designed so that it will not bind in the bore when fully compressed, in the maximum material condition. Spring loading of the piston may not be possible in larger diameter equipment because the spring mean diameter to spring wire diameter index may become impractical. This “spring index” should normally be in the range of 4 to 16. The preferred range¹² is 5 to 12.

Venting excess thermally expanded lubricant to prevent overfill

If desired, the housing can incorporate a longitudinal venting slot to prevent lubricant overfilling and associated lubricant thermal expansion related pressure locking. When the sliding seal travels onto the slot, the slot vents lubricant past the sliding seal and into the environment. The end of the slot must be chamfered to ease the sliding seal back into the unslotted portion of the housing bore, and the spring must be strong enough to cause the

¹¹ For additional information on the use of spring-loaded pressure compensation pistons in oilfield downhole tools, see expired U.S. Patent 4,372,400.

¹² The spring index range of 5 to 12 is from the “**Mechanical Spring Design Guide**”, by Green and Mather (Rockwell International, 1973).

sliding seal to reengage with the unslotted portion of the housing bore under all anticipated operating conditions.

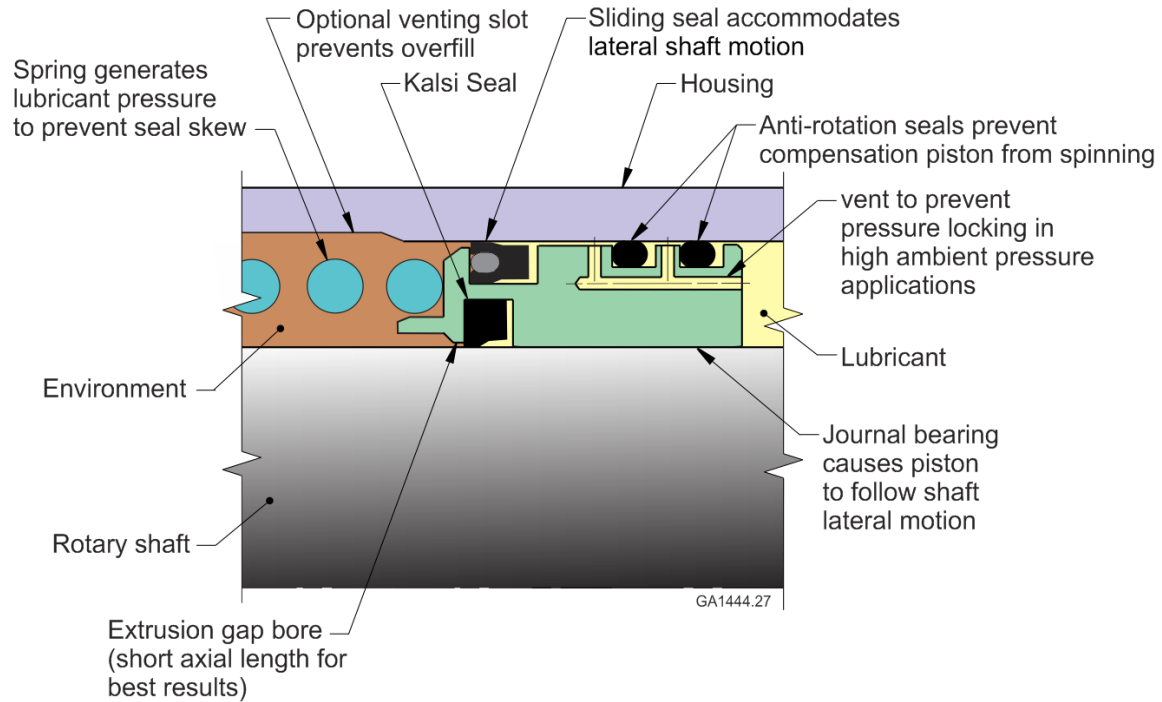


Figure 11

Spring loading the pressure compensation piston

In some applications, a spring can be used to impart pressure to the lubricant, to address skew, shuttling and reverse pressure related rotary seal wear mechanisms.

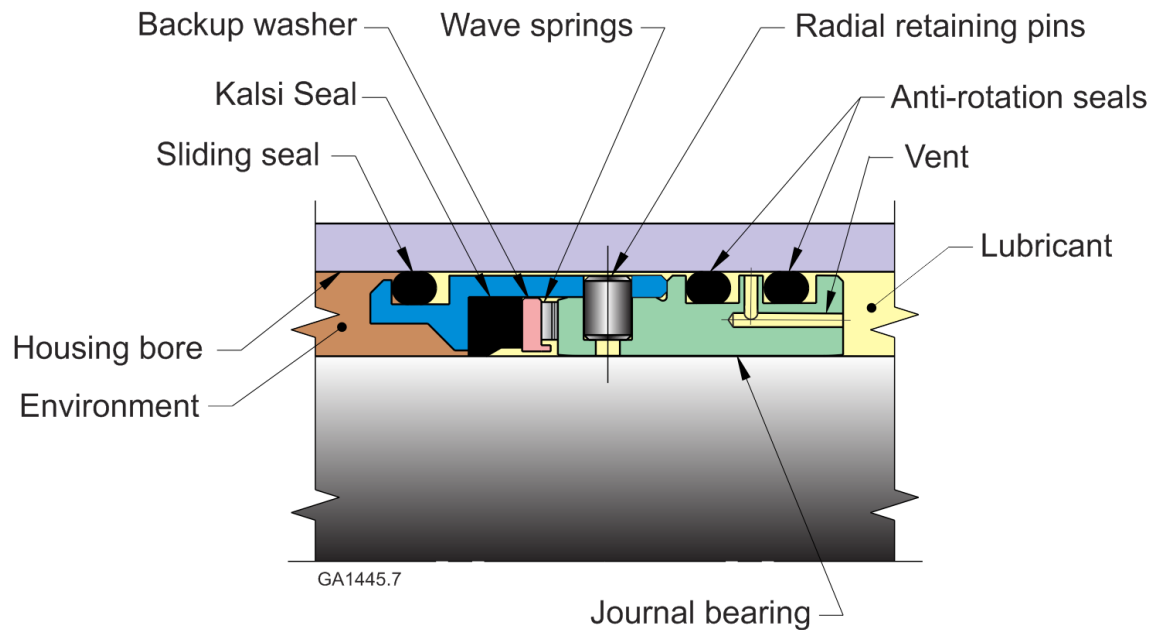
9. Spring loading a Kalsi Seal in a compensation piston

The compensation pistons shown in Figures 12 and 13 fit in the same space as the piston of Figure 2. Instead of using an Axially Constrained Kalsi Seal, the pistons in Figures 12 and 13 employ a backup washer and springs to axially preload a Kalsi Seal to inhibit skew induced abrasive wear. This arrangement can be used in cases where an Axially Constrained Seal™ is not available in the correct size or temperature range. For more information on spring loading Kalsi Seals, see Chapter D9.

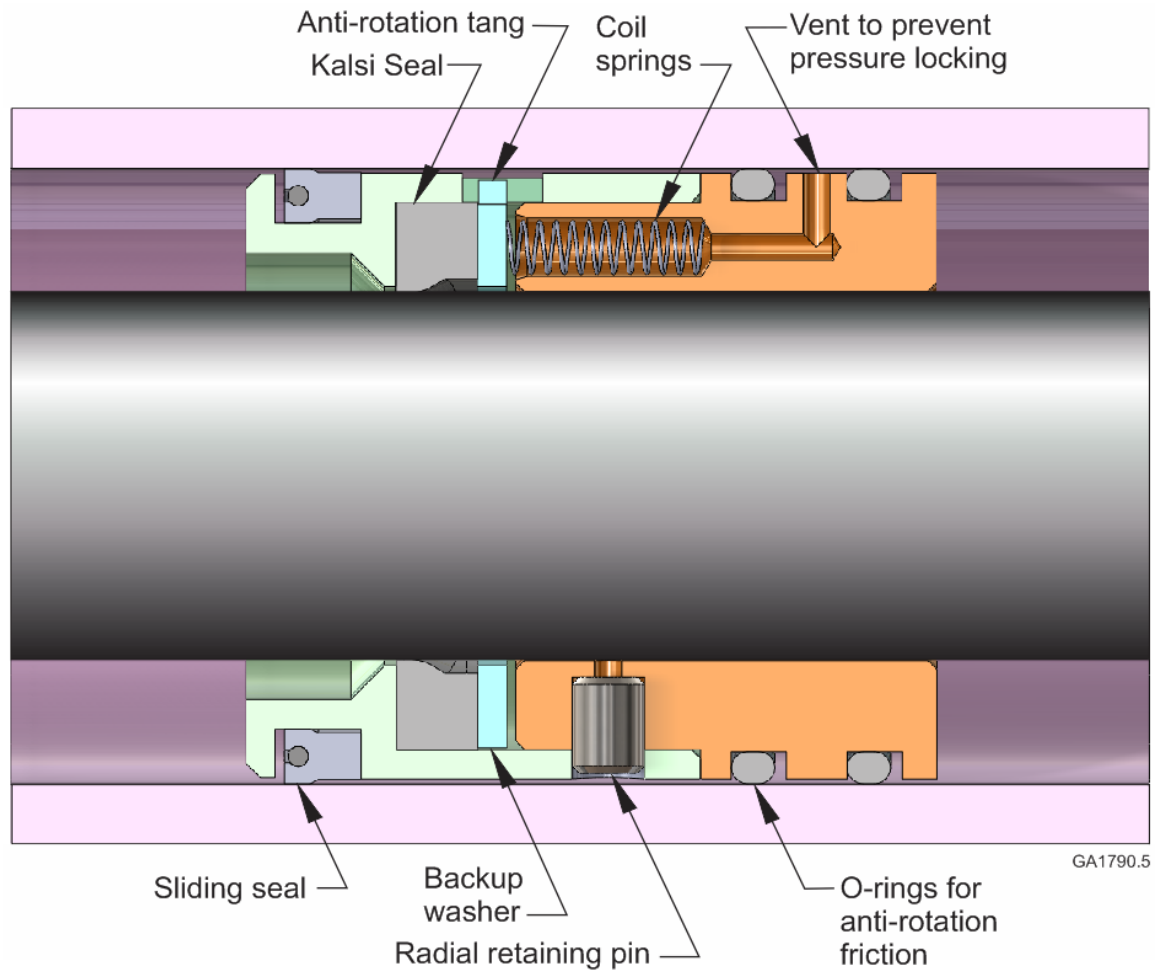
The compensation pistons in Figures 12 and 13 have front and rear members that have a close sliding fit with each other, and are locked together by three equally spaced radial retaining pins captured by the housing bore. (The holes for the radial retaining pins are match drilled and reamed.) The pistons can be quickly disassembled to replace the Kalsi Seal by removing the pins.

When the environment pressure exceeds the lubricant pressure, the resulting hydraulic force can compress the springs and cause axial travel of the Kalsi Seal. As shown in Figure 12, the backup washer (or the removable groove wall formed by the journal bearing) may incorporate one or more projections to limit travel so that the distance between the environment side groove wall and the backup washer cannot exceed the groove width needed to accommodate the thermally expanded seal width. Detailed information on designing spring loaded Kalsi Seal implementations is provided in Chapter D9. Information on the width of some types of Kalsi Seals at elevated temperature is provided in Appendix 2.

As with the compensation pistons described earlier, the sliding seal of Figures 12 and 13 establishes a sealed relationship with the housing bore, and the anti-rotation seals provide extra friction to prevent piston spinning. To prevent pressure locking, the holes for the radial retaining pins, the piloting fit between the piston members, and the cross-drilled vent, defeat the sealing function of the anti-rotation seals.

**Figure 12****A compensation piston with a wave spring loaded Kalsi Seal**

In this oilfield mud motor pressure compensation piston, the Kalsi-brand rotary seal is axially preloaded with a wave spring to prevent skew induced wear. Three radial retaining pins in match-reamed holes are used to hold the assembly together. In an R&D mud motor that Kalsi Engineering built to obtain firsthand field experience, the journal bearing portion of this seal carrier was bearing bronze.

**Figure 13****A compensation piston with a coil spring loaded Kalsi Seal**

In this oilfield mud motor pressure compensation piston, the Kalsi-brand rotary seal is axially preloaded with a circle of coil springs to prevent skew induced wear. Three radial retaining pins in match-reamed holes are used to hold the assembly together. The backup washer is keyed to the piston to prevent rotation that could bind the projecting ends of the springs. If desired, the anti-rotation tangs could be axially oriented, instead of radially oriented. The face of the backup washer can be grit blasted to inhibit circumferential slippage of the rotary seal.

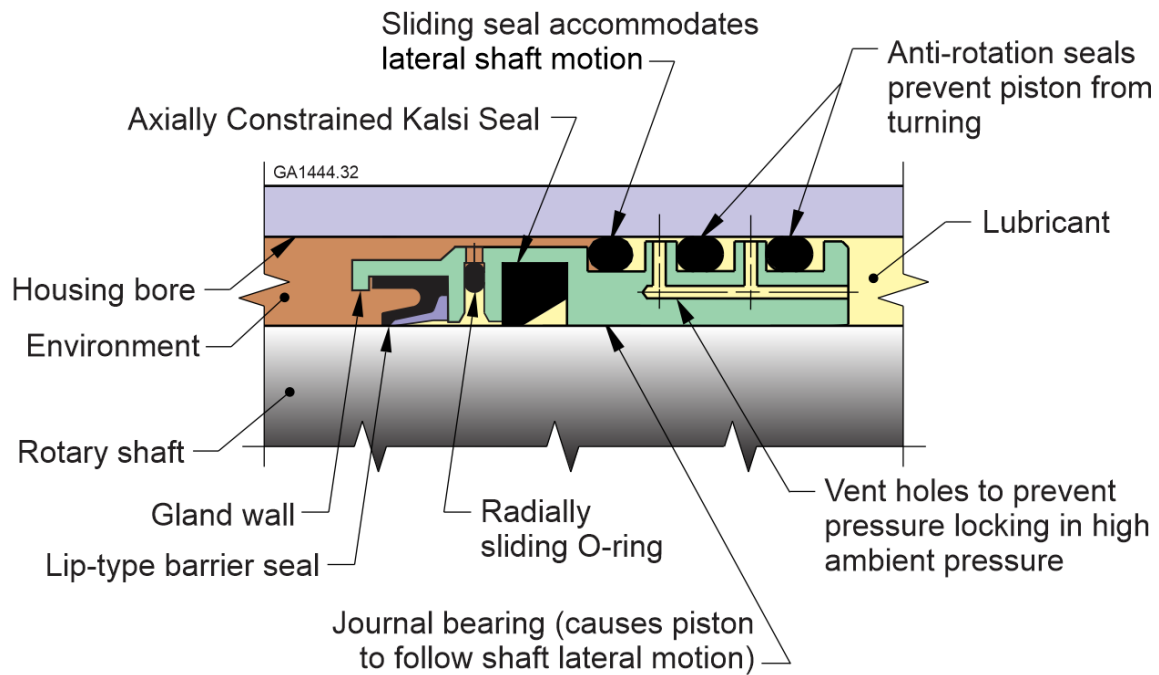
10. Incorporating a barrier seal in a compensation piston

Figure 14 Shows that a lip-type barrier seal can be mounted outboard of the Kalsi Seal, to provide a degree of redundancy. The arrangement that is shown is configured for a high ambient pressure environment. The barrier lubricant between the Kalsi Seal and the barrier seal is pressure compensated to the environment by a radially acting O-ring which is axially compressed in a deep groove.

The pressure of the environment pushes the O-ring radially inward, compressing any entrained air within the barrier lubricant, and equalizing the pressure of the barrier lubricant to that of the environment. This arrangement is directed at preventing the entrapment of atmospheric pressure between the rotary seals, in applications where the piston will be exposed to high ambient pressure, so that the rotary seals are not exposed to high differential pressure.

The mouth of the deep groove is chamfered to facilitate installation of the radially acting O-ring. The lip-type barrier seal is illustrated as a V-spring-loaded elastomer seal with a reinforced PTFE heel. The elastomer portion is available in HNBR or FEPM. The FEPM option seems appealing from a chemical resistance standpoint, especially for oilfield downhole applications such as mud motors and rotary steerable tools.

The gland wall for the barrier seal is radially short for three reasons. First, it eases installation of the seal into the groove. Second, it helps to minimize the risk of installation-related damage to the spring of the barrier seal. Third, it prevents the dynamic sealing lip of the barrier seal from sealing with respect to the gland wall. This helps to ensure that the barrier seal will vent any significant lubricant pressure buildup caused by the hydrodynamic pumping related leakage of the Kalsi-brand rotary shaft seal.

**Figure 14****Incorporating a barrier seal in a compensation piston**

The radial stroke of the radially sliding O-ring helps to compensate for an incomplete lubricant fill in the region between the Kalsi-brand rotary seal and the lip-type barrier seal. This compensation prevents pressure locking in high ambient pressure environments. Smaller O-ring cross-sections have more radial stroke, which may be an important consideration when radial space is limited.

11. Installing sliding seals into compensation piston grooves

The sliding seals of a compensation piston are often quite stiff and difficult to install by hand. The recommended way to install such seals is as follows. First, warm the seal up to soften and thermally expand it. Second, start the seal into its groove on one side of the piston. Third, slip an O-ring under the opposite side of the seal, and then pull on the resulting two loops of the O-ring, using the O-ring as a tool to work the sliding seal into its groove (Figure 15). Finally, pull the O-ring free, and give the sliding seal enough time to gradually retract to its original diameter.



Figure 15

An O-ring can be used as a tool to install a piston sliding seal

12. Redundant compensation pistons

If you elect to use two compensation pistons to achieve seal redundancy in a tool such as an oilfield mud motor, have a suitable plan to eliminate air and provide lubricant between the pistons. One approach is to incorporate an air vent port into the outboard piston, as shown in Figure 16. After installing the inboard piston, fill the tool with oil using a procedure that substantially eliminates air within the tool and moves the inboard piston to the intended position. Next, orient the axis of the tool vertically, and pour lubricant on top of the inboard piston, introducing a volume calculated to achieve the desired position of the outboard piston. After that, with the piston vent port open, install the outboard piston and push it down until oil comes out of the vent port. Finally, plug the vent port.

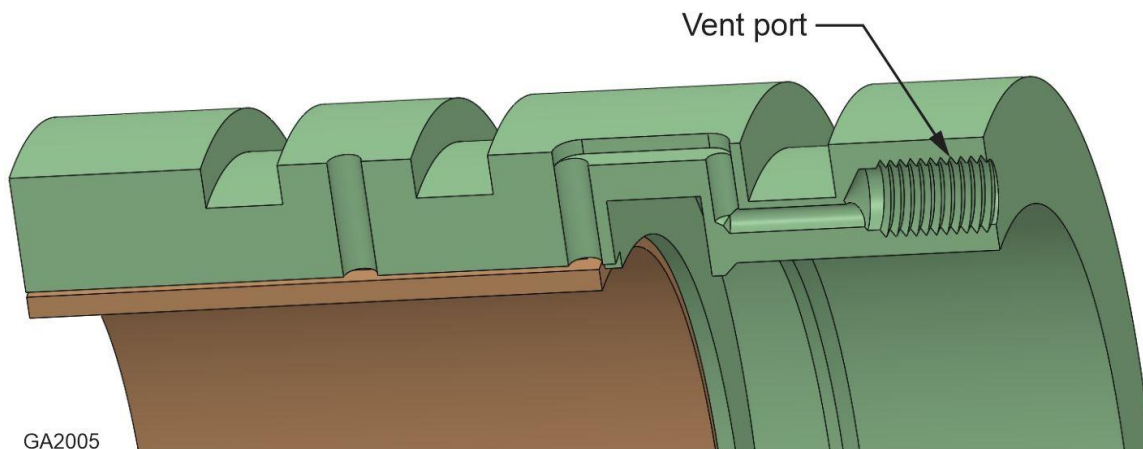


Figure 16

Plan for eliminating air between redundant pistons

The vent port allows air to escape as the outboard piston is installed in the tool.

13. Troubleshooting

Why would the rotary seal in a piston show signs of high reverse pressure?

High reverse pressure across a typical elastomer-type Kalsi Seal may cause extrusion damage to the lubricant end of the seal body and may flatten the seal enough to cause wear to portions of the seal that should not be in contact with the shaft. Some of the factors that might cause a high reverse pressure signature on the rotary seal of a shaft guided compensation piston of a mud motor (or similar equipment) are:

- Piston sticking due to factors such as sticky drawer effect, mechanical binding, or pressure locking of the sliding seals.
- Pressure locking between pairs of rotary seals due to entrapment of atmospheric pressure at the time of assembly, causing one rotary seal to have signs of high reverse pressure, and the other to have signs of high forward pressure.
- The piston bottoming out in the empty position.

What might make a piston bottom out in the empty position?

Some of the factors that might cause a shafted guided compensation piston of a mud motor (or similar equipment) to bottom out in the empty position are:

- Loss of lubricant due to failure of a rotary shaft seal.
- Leakage past the sliding seal on the outside of the piston, due to factors such as insufficient compression or damage to the mating housing bore.
- A leaking fill plug.
- A significant volume of air within the tool that is compressed when exposed to system pressure.
- Insufficient initial lubricant fill.
- A range of shaft deflection that exceeds the compression of a sliding seal or a rotary shaft seal, resulting in premature loss of lubricant volume.
- Insufficient reservoir size that is prematurely depleted due to the normal hydrodynamic pumping action of the rotary shaft seals.