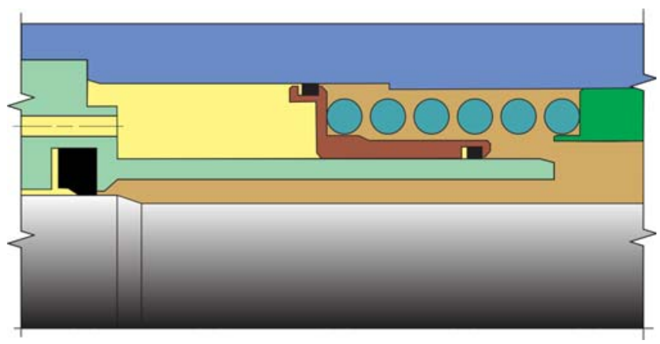
Chapter D11

Reservoirs for pressurized seal lubricant



Revision 8 October 12, 2020

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1. This chapter is applicable to seals that require pressurized lubricant

With a few exceptions,[1] Kalsi Seals® designed for high differential pressure service are configured to retain pressurized lubricant. Because of this, a pair of Kalsi Seals are typically used to retain a pressurized process fluid, as shown by the Figure 1 schematic. The lubricant between the rotary seals is pressurized to a value that is equal to, or greater than, process fluid pressure. The right-hand seal partitions the lubricant from the process fluid, and the left-hand seal retains the pressurized lubricant.

The purpose of having the lubricant pressure greater than the process fluid pressure is to properly orient the partitioning seal to promote seal abrasion resistance.[2] If lubricant over-pressure cannot be used, and the lubricant pressure is merely balanced to the process fluid pressure, the partitioning seal must be axially spring loaded (Chapter D9), or must be an Axially Constrained Seal™ (Chapter C4).

A suitably sized lubricant reservoir is used to pressurize the lubricant and accommodate lubricant thermal expansion and the hydrodynamic pumping related rotary seal leakage. Various types of lubricant supplies are described in this chapter. Some are external (as implied by Figure 1), and some are integrated into the equipment.

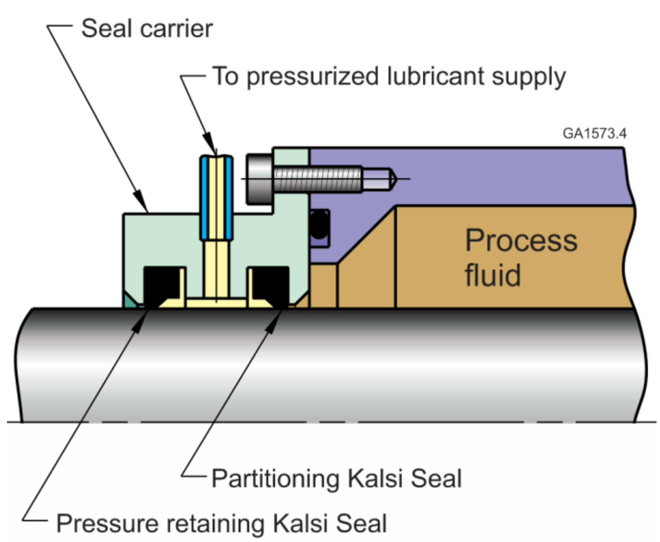

Figure 1

A pair of seals, fed by a lubricant supply, are typically used to retain process fluid

1 The KLS® high pressure lip seal and the BDRP Seal™ are designed for sealing fluids having a pressure that is greater than the lubricant pressure. These seals allow the use of seal lubricant that is gravity fed from a lubricant reservoir (such as a bearing housing) that is vented to atmospheric pressure. When designing remote gravity fed lubricant reservoirs for such seals, any connecting piping must be free of restrictions (such as small valves) that can prevent gravity-induced lubricant flow.

2 Kalsi Seals do not rely on the lubricant over-pressure for interfacial lubrication. The lubricant over-pressure is used solely to properly orient the partitioning seal.

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2. Integral piston type lubricant reservoirs

The lubricant reservoir can sometimes be incorporated integrally to the equipment being sealed. For example, in the oilfield mud motor sealed bearing assembly of Figure 2, the reservoir is provided by the stroke of the axially movable annular pressure compensation piston, which also pressure balances the lubricant to the drilling fluid that is located in the bore of the drillstring. Design considerations for such pressure compensation pistons are included in Chapter D14. Any piston will have to be designed with enough guide length to avoid the "sticky drawer" effect described in Chapter D21.

As another example of an integral lubricant reservoir, the reservoir of the coring swivel[3] of Figure 3 is provided by the stroke of the spring-loaded compensation piston. The spring amplifies the lubricant pressure slightly above the high-pressure drilling fluid that circulates through the swivel. This pressure amplification increases the abrasion resistance of the upper rotary seal by holding it against the environment side groove wall.

Figure 4 shows how to spring load a compensation piston using a compression spring that is located on the lubricant side of the piston.

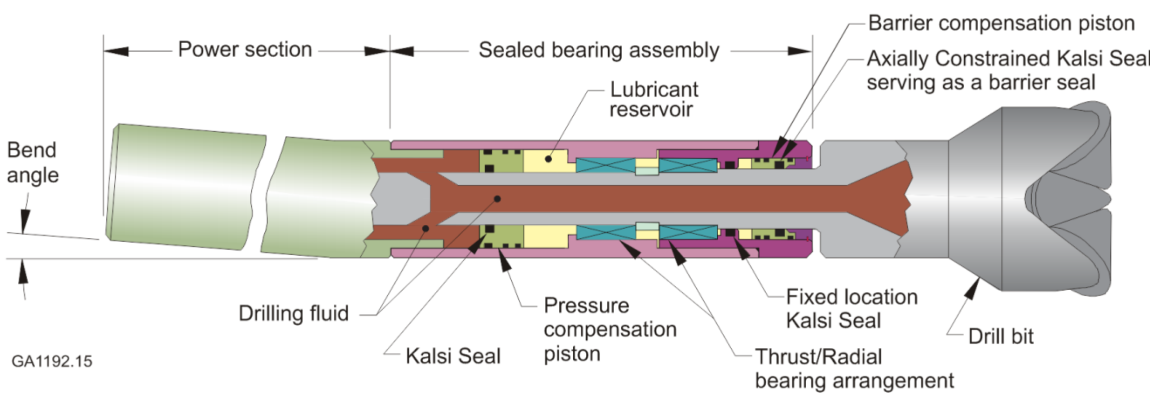

Figure 2

Schematic of an oilfield downhole drilling mud motor sealed bearing assembly

In a mud motor sealed bearing assembly, the pressure of the lubricant within the tool is typically balanced to the drilling fluid pressure within the drillstring by an axially movable shaft guided pressure compensation piston. The stroke of the piston defines the lubricant reservoir. As the lubricant is consumed by the hydrodynamic pumping related leakage of the Kalsi Seals, the piston moves axially. Likewise, the stroke of the barrier compensation piston defines a lubricant reservoir for the barrier Kalsi Seal™. The barrier lubricant provides the critical fixed location Kalsi Seal with a clean lubricated environment, which prolongs sealing life.

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3 To learn more about the development and initial field use of the coring swivel, see the article *"Hybrid Rigs Developed for Continuous Coring Exploration"*, **Oil & Gas Journal**, April 27, 1992.

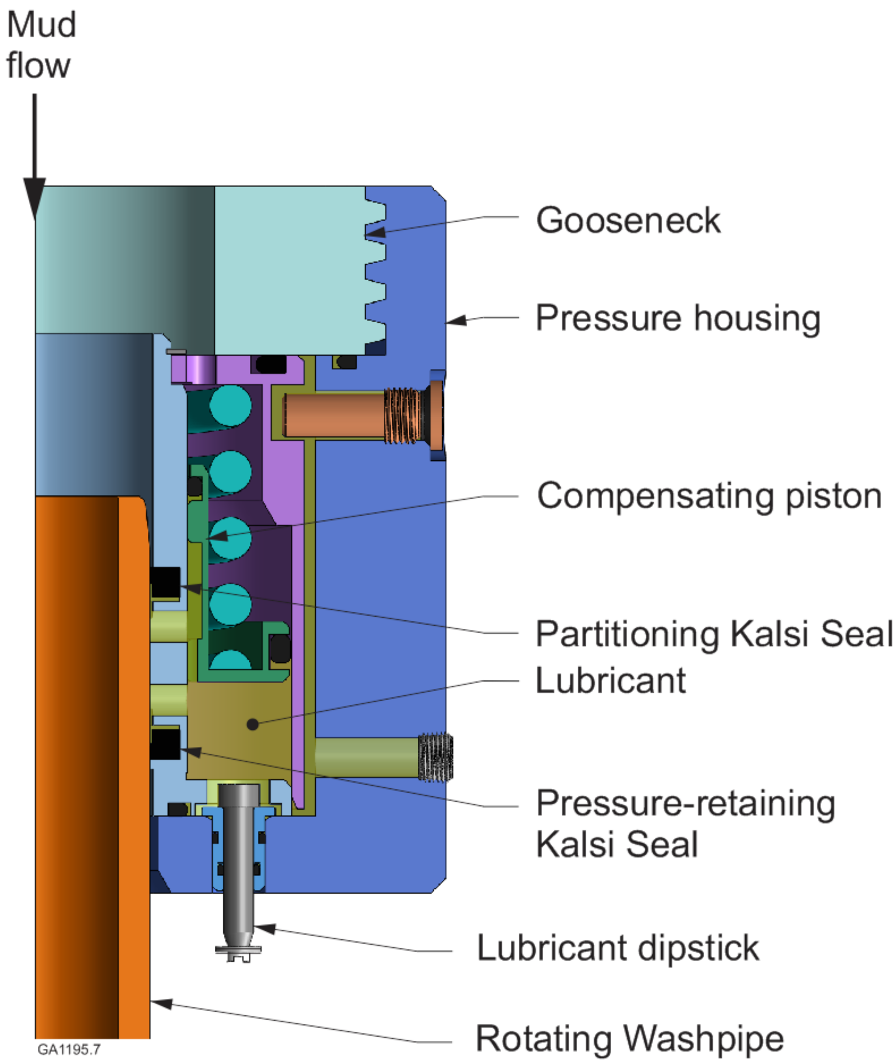

Figure 3

High pressure, high speed coring swivel sealing arrangement

In this oilfield coring swivel seal assembly, the lubricant reservoir is defined by the stroke of a pressure compensation piston within a housing. A spring acting on the piston amplifies the lubricant pressure above that of the drilling fluid pressure to improve the abrasion resistance of the upper Kalsi Seal. The spring telescopes over the piston to conserve space. The seal carrier is designed to self-align on the washpipe at the time of assembly, before the pressurized mud is introduced. During rotary operation of the swivel, the washpipe functions as a typical hydraulically force-balanced, articulating washpipe. During assembly, the washpipe is temporarily clamped rigidly to the hollow rotary shaft it is mounted to, causing the axis of the washpipe to temporarily be parallel to the axis of the hollow rotary shaft. This temporary clamping step drags the seal carrier into nominal alignment with the axis of the hollow rotary shaft. During operation, the drilling fluid pressure holds the seal carrier in its aligned position.

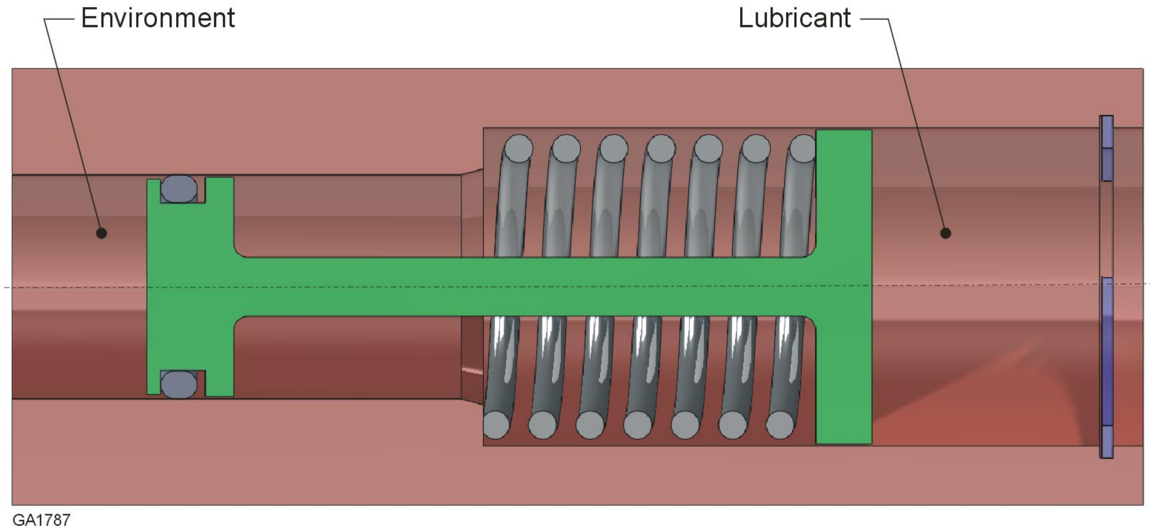

Figure 4

Spring loaded compensation piston with the spring in the lubricant

This figure shows a way to spring load a pressure compensation piston using a compression spring that is located within the seal lubricant. The spring amplifies the lubricant pressure to a value that is greater than the environment pressure. The stroke of the piston within the housing defines the lubricant reservoir. This arrangement protects the spring from the environment.

3. Accommodating lubricant thermal expansion

Lubricants have a much higher rate of thermal expansion than metals and are substantially incompressible. Because of these factors, a reservoir that is filled completely at room temperature will generate extreme lubricant pressure at elevated temperature. This pressure buildup can damage seals and hardware (Figure 5). Unless specially designed to prevent buildup of excessive lubricant pressure, a lubricant reservoir must initially be filled to an intermediate position, so that any thermal expansion of the lubricant can be accommodated by piston movement (Figure 5).

The initial fill position is calculated to accommodate the maximum anticipated temperature of the machine, using the coefficient of lubricant thermal expansion that is provided by the lubricant manufacturer. The calculation for the initial fill position must account for the total lubricant volume of the system, rather than just the volume of the lubricant reservoir. The fill position calculation should also provide for the estimated range of fill position inaccuracy that will occur during assembly.

Mechanics should be given periodic retraining to maintain their awareness of the pitfalls of overfilling. When practical, use a fill position gauge during filling to help assure fill position consistency. Alternatively, one can fill to the completely full position, and then use a syringe to extract a known volume of lubricant. For simplicity in use, the syringe should be designed to extract the correct lubricant volume. If an automatic refill system is used, ensure that it cannot fill the reservoir to the completely full position.[4]

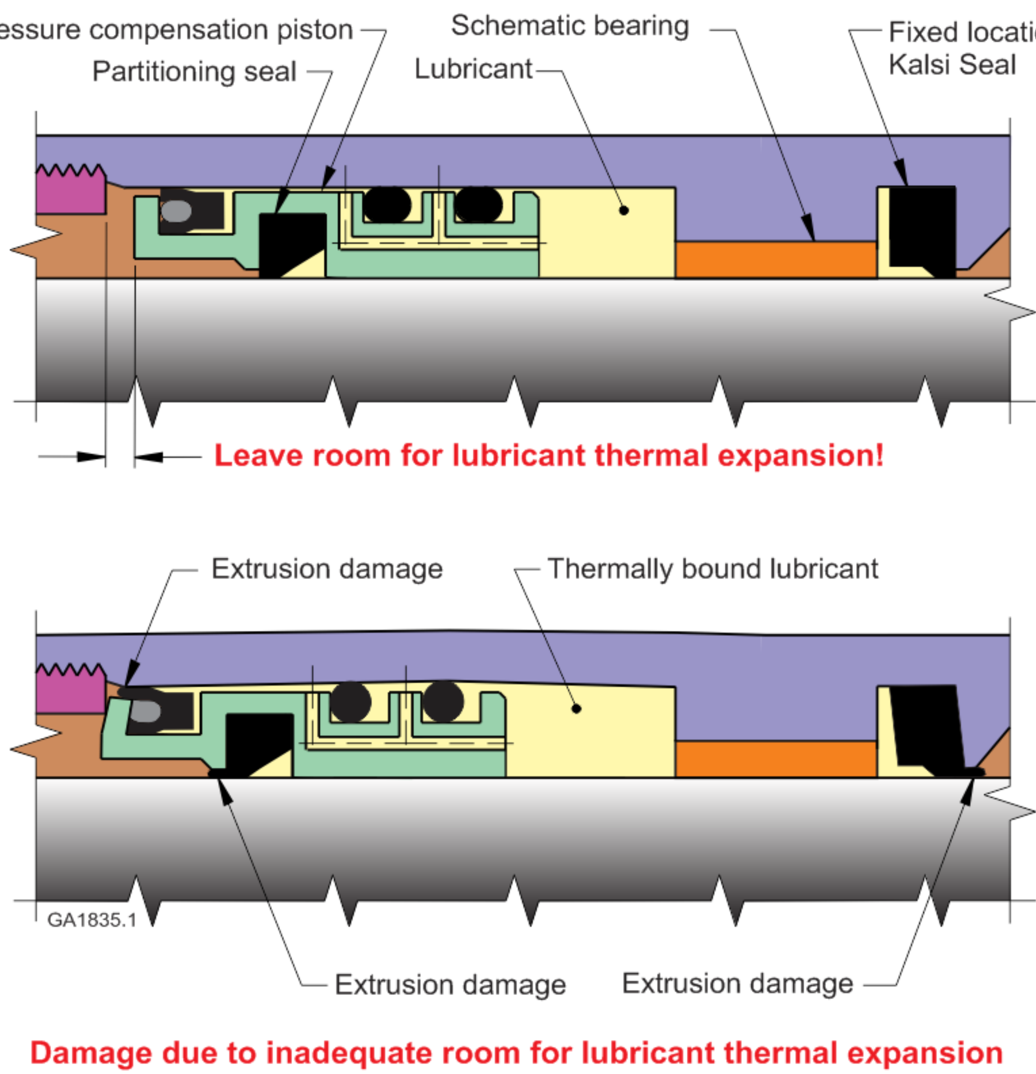

Figure 5

Provide room for lubricant thermal expansion

Re-train frequently to make sure your mechanics know to leave room for thermal expansion of the lubricant. (In this arrangement, the reservoir serves both Kalsi Seals). Failure to provide room for lubricant thermal expansion can create enormous pressure that destroys the rotary seals and may even yield the metal hardware.

4 With spring loaded piston-based reservoirs, such as Figure 4, this can be accomplished with a refill pressure selected to match the lubricant pressure generated by the spring at the desired fill position. With this method, the refill pressure isn't high enough to stroke the piston to the completely full position. During the refilling process, the lubricant preferably enters the reservoir through a check valve.

In some types of equipment, a vent seal can be incorporated into the pressure compensation, as shown in Figure 6. This seal vents lubricant thermal expansion into the environment if the piston is bottomed out in the full position. With this type of arrangement, the reservoir can be filled to the completely full position, without needing to allow extra piston stroke to accommodate lubricant thermal expansion. Venting should occur at a pressure that is greater than the differential pressure required to move the piston, or the venting rate should be much slower than the reservoir filling rate, so that the piston moves during filling operations. In either case, the full position can be detected by a rise in lubricant pressure.

In equipment with extreme axial acceleration, it is conceivable that the acceleration could produce enough lubricant pressure to cause the venting action to occur. With repetitive acceleration (for example, extreme vibration), it is conceivable that repetitive venting could empty the reservoir.

The venting seal should be separate from the sliding seals of the piston, so that it does not have to slide over contaminated surfaces during axial motion of the piston.

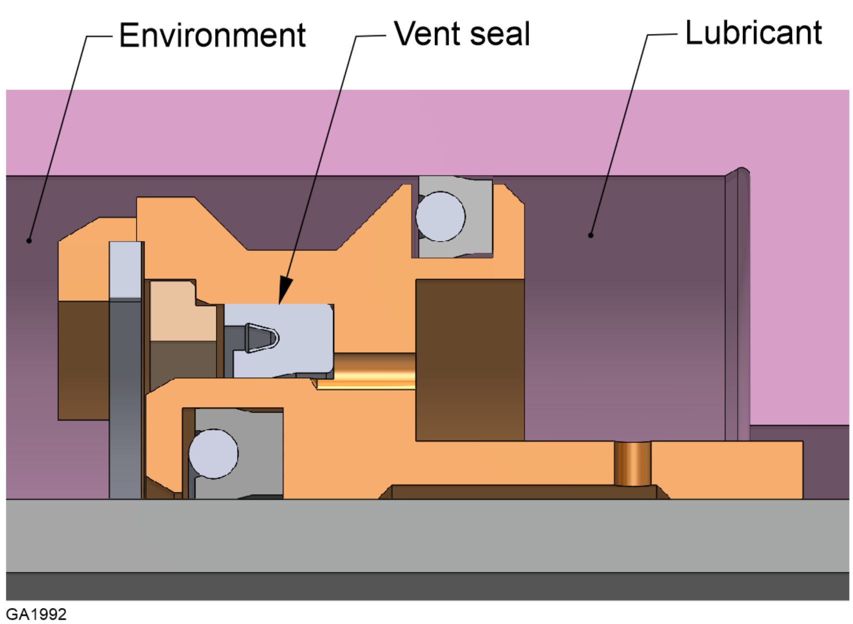

Figure 6

Another way to accommodate lubricant thermal expansion

In some types of equipment, a vent seal can be incorporated into the pressure compensation piston to accommodate lubricant thermal expansion. If the piston is bottomed out in the completely full position, then extra lubricant volume from thermal expansion vents past the vent seal and into the environment.

4. Differential area piston amplifier type lubricant reservoirs

As described elsewhere in this chapter, when a Kalsi Seal is being used to partition a process fluid from a lubricant, the lubricant pressure should ideally be greater than the process fluid pressure to minimize abrasive wear of the seal. This lubricant over-pressure can be achieved with custom designed piston type lubricant reservoirs[5] that use differential hydraulic area to amplify the lubricant pressure above the process fluid pressure, as shown in Figures 7, 8, and 13. Also see IADC/SPE paper no. 59107.

The piston rod subtracts area from the lubricant side of the piston, causing the lubricant side of the piston to have less hydraulic area than the process fluid side of the piston. The hydraulic amplification of the lubricant pressure is equal to the ratio of the two areas. The axial position of the piston rod provides a useful indication of lubricant level and can be instrumented for remote monitoring and control purposes if desired. Such systems are adaptable to automatic refill using a limit switch and a remotely located pump that draws from a vented tank.

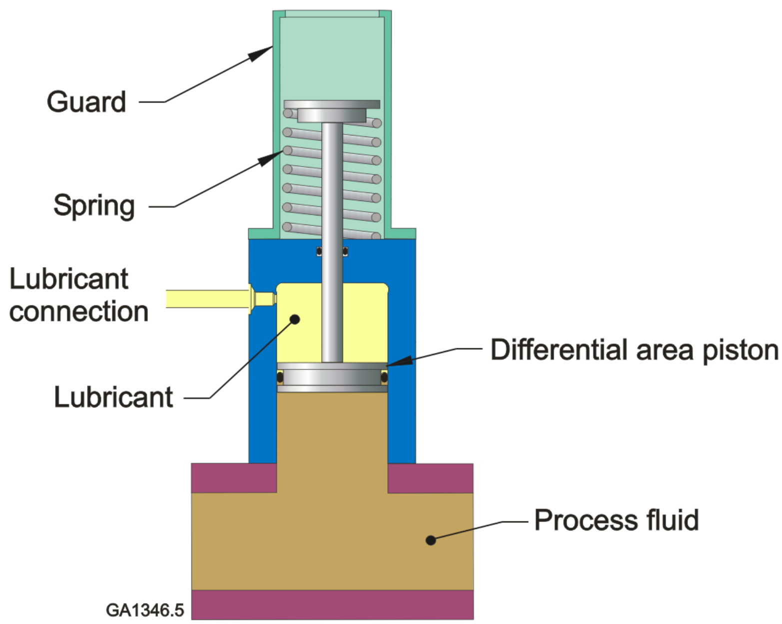

Figure 7

Lubricant reservoir with hydraulic amplification and supplemental spring force

In this piston type lubricant reservoir, the lubricant pressure is maintained above the process fluid pressure by hydraulic amplification and spring force. The spring helps to avoid detrimental

5 For an early description of such lubricant reservoirs, see Figure 177 of T. C. Thomsen's 1920 book "**The Practice of Lubrication**" (McGraw-Hill Book Company, Inc.)

"pressure lag"; in other words, it helps the lubricant pressure to "stay ahead" of any rapidly increasing process fluid pressure. To minimize pressure lag, the end of the reservoir opens directly into the process fluid conduit.

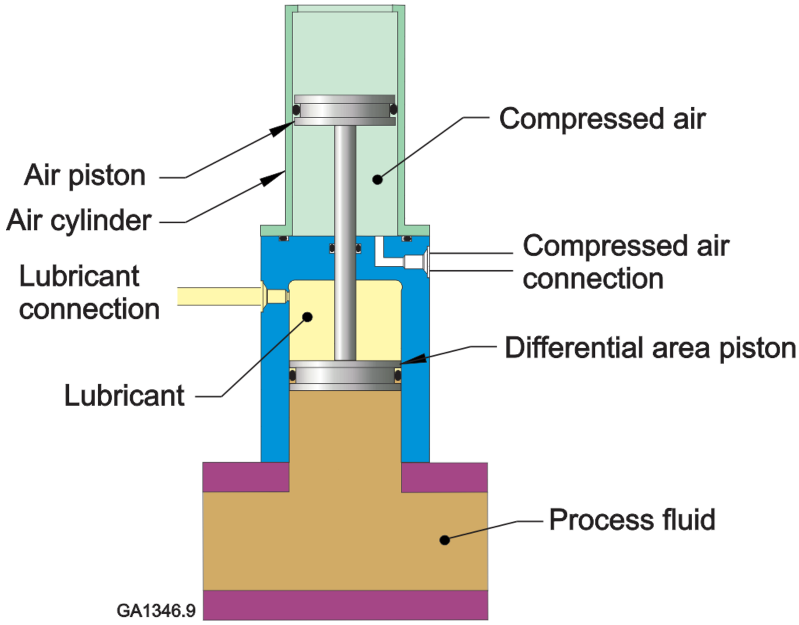

Figure 8

Lubricant reservoir with hydraulic and pneumatic amplification

In this piston type lubricant reservoir, the lubricant pressure is maintained above the process fluid pressure by hydraulic amplification and pneumatic force. The regulated air pressure applies constant axial force to the piston, regardless of the position of the piston.

Applying supplemental force to prevent pressure lag

In cases where the pressure of the process fluid increases rapidly, the lubricant pressure takes time to reach the rotary seals. This lubricant "pressure lag" is due to the time required to overcome factors such as sliding seal friction, piston inertia, entrained air in the lubricant, pressure induced expansion of lubricant hoses, and restricted passageways. For example, the lubricant has to flow in order to compress entrained air and expand lubricant hoses, and that flow takes time. This lubricant pressure lag can result in the very undesirable situation where the lubricant pressure at the rotary seals is less than the process fluid pressure. Such pressure reversals can distort the rotary seal that faces the process fluid, causing seal skewing and shuttling that seriously impairs environmental exclusion.

To help to address pressure lag, one can apply a supplemental force to the piston, in addition to the hydraulic amplification. This supplemental force gives the seal lubricant pressure a "head start", allowing it to "stay ahead" of the increasing process fluid pressure. In Figure 7 the supplemental force is applied with a coil spring, and in Figure 8 the

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supplemental force is applied with a pneumatic piston. When a coil spring is used, changes in spring force due to piston motion must be considered to ensure that lubricant pressure remains greater than process fluid pressure. Ideally, the spring force needs to be strong enough to overcome the piston friction, and maintain enough differential pressure to prevent seal skewing, even in the depleted position of the piston stroke. Kalsi Engineering can provide compression spring design software to customers who are designing spring loaded lubricant reservoirs. When designing a spring-loaded arrangement, ensure that the design accommodates the increase in spring diameter that occurs as the spring is compressed.

If a dependable supply of regulated compressed air is available, a pneumatic piston is preferred over a coil spring, because the supplemental force remains constant over the entire piston stroke. If desired, the air cylinder can contain a spring to serve as a backup lubricant over-pressure method in the event of air pressure loss.

Rod and piston seal breakout and sliding friction should be considered when designing the hydraulic amplification and the supplemental applied force. Rod and piston seal friction can be minimized through material selection and reduced compression, and an appropriate surface finish on the mating parts. We recommend that the surface roughness of the rod and the reservoir bore be no more than 16 micro-inches (0.40 μm) AA.

Avoid restricted process fluid connections

In Figures 7, 8, and 13 the piston is exposed directly to the process fluid through a large opening, and the pressure of the process fluid cannot increase without instantly pushing on the piston. Compare that desirable situation to Figure 9, where the process fluid is communicated to the piston by a slender hose. A significant amount of time is required for the process fluid to travel through the hose and compress the trapped air located on the process fluid side of the piston. As a result, an undesirable pressure reversal may occur at the partitioning rotary seal (see Figure 1). When the process fluid pressure drops, the compressed air expands and expels the process fluid.

In Figure 10, the process fluid connection is illustrated as a hose. The reservoir is oriented to keep process fluid in the hose and against the piston. An air vent is provided to allow air to escape when the process fluid side of the piston is initially filled. The Figure 10 arrangement has much less potential for lubricant pressure lag, compared to the arrangement of Figure 9. Even so, a certain amount of time is required for process fluid to flow through the restricted process fluid hose, and compress any entrained air within the lubricant and process fluid, expand the hoses, etc.

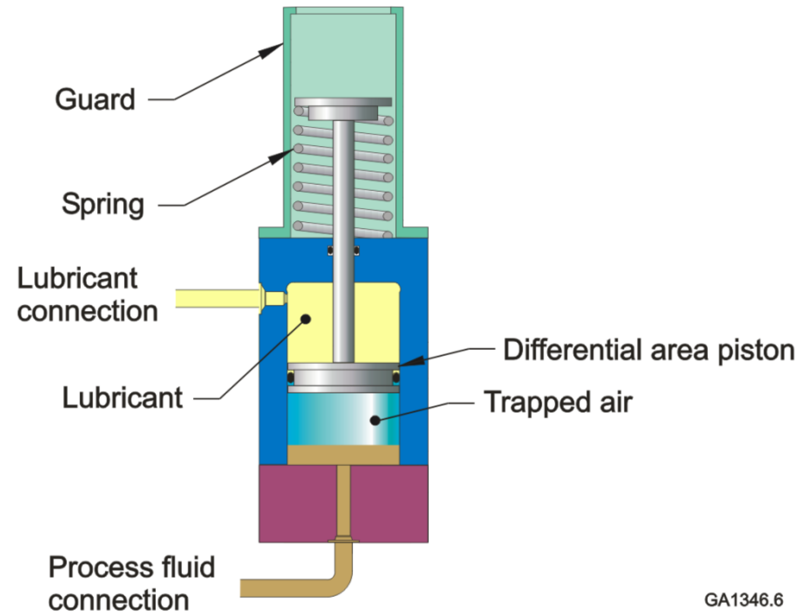

Figure 9

Restricted process fluid connection causes undesirable pressure lag

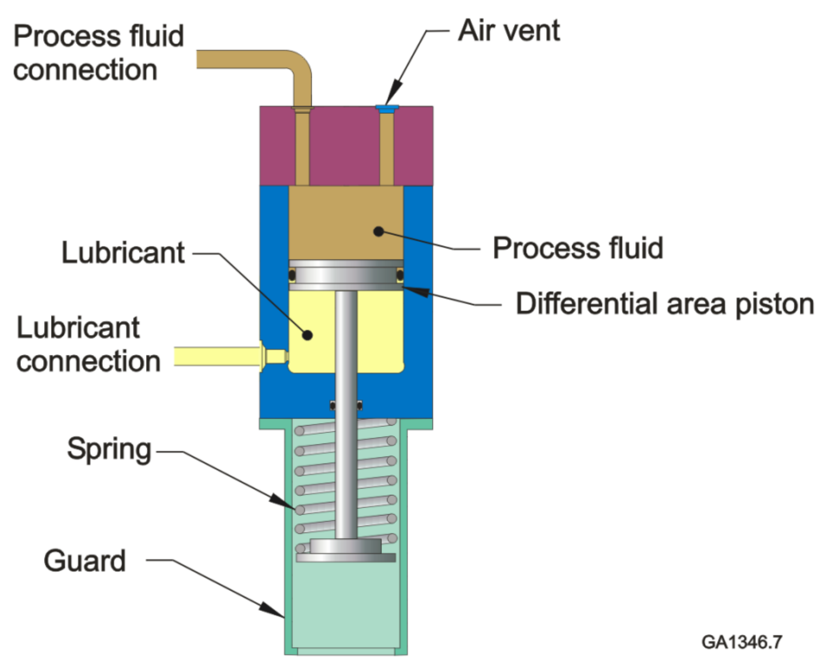

Figure 10

In this orientation, gravity helps to keep process fluid against the piston

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The restricted process fluid hose is also undesirable for another reason: Hoses can become clogged with the process fluid if the fluid contains solids or has gelling properties. For all the reasons described above, it is preferred that the differential area piston be directly exposed to the process fluid by a large, rigid port, as shown in Figures 7, 8, and 13.

Eliminating air from the reservoir during lubricant fill

In order to minimize pressure lag, the lubricant reservoir can be "vacuum filled." Hook the lubricant reservoir to a vacuum pump and evacuate the air, then turn off the pump and monitor the vacuum for a few minutes to be sure there are no leaks. Finally, introduce the lubricant into the evacuated reservoir. Alternatively, a bleed port can be provided to allow air to escape as lubricant enters the reservoir.

The integrity of the lubricant fill can be determined by applying pressure to the process fluid side of the piston and observing the resulting piston motion. If there is little or no air on the lubricant side of the piston, the piston will move very little when pressure is applied to the process fluid side of the piston.

If desired, the lubricant can be degassed before being introduced into the lubricant reservoir, but this requires a custom lubricant degassing apparatus. Kalsi Engineering has drawings for a small lubricant degasser. Degassing is typically only considered for low pressure vertical shaft rotary sealing applications, to prevent the formation of a detrimental air pocket under the top rotary seal.

As is true with nearly all types of lubricant reservoirs, do not completely fill an amplifier piston type lubricant reservoir with lubricant. Leave enough free travel to accommodate lubricant thermal expansion. Clearly mark the unit so that the mechanic understands the initial fill position of the piston. Also provide a piston position mark that indicates when the unit should be refilled. On the spring type arrangements, the piston position can be directly observed by putting a slot in the spring guard. Figure 12 shows one convenient way to compress the spring at the time of assembly.

Assembling a piston amplifier type lubricant reservoir

In the differential area piston amplifier type of lubricant reservoirs shown in Figures 7-10, the reservoir is illustrated as having one-piece construction. In some designs, the portion of the reservoir that defines the necessarily tight rod seal extrusion gap bore is retained to the reservoir cylinder with bolts. In such designs, the bolts should be tightened with the piston in the full stroke empty position, so that the piston rod locates the rod seal extrusion gap bore laterally. This helps to ensure freedom of piston motion throughout the entire piston stroke.

Reducing overfill risk in low pressure piston amplifiers

Spring loaded piston type amplifier lubricant reservoirs are suitable for both low and high process fluid pressures. In low pressure applications, the risk of overfilling the lubricant can be addressed by providing a longitudinal undercut which is exposed by the rod seal as the piston reaches the full-stroke position, as shown by Figures 11 and 13. The undercut allows excess lubricant to leak out of the reservoir. This arrangement requires that the pressure amplification spring has enough force to return the piston to a sealed position in all anticipated operating temperatures. (Seals are stiffer at lower temperatures, and more force may be required to move the piston.)

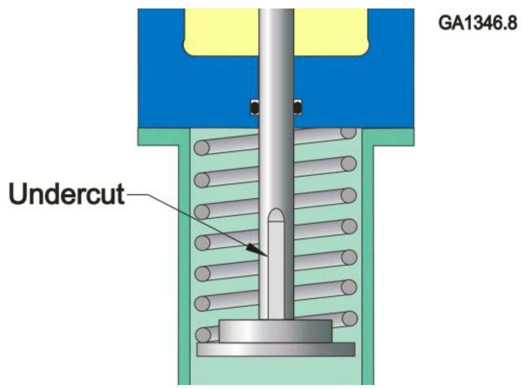

Figure 11

Undercut prevents lubricant overfill on low pressure amplifiers

This figure shows how an undercut can be used to prevent overfilling of the lubricant reservoir. The spring must provide enough force to move the piston back into full sealing engagement with the rod seal in all operating temperatures. The corners of the undercut need to be smooth radii or chamfers, and the tapered ramp at the end of the undercut needs to be smooth. New undercut designs should be tested to ensure damage free operation and reliable resealing. This arrangement is not recommended for high pressure applications, because the rod seal is prone to extrusion damage when encountering the undercut.

Differential area piston amplifier type lubricant reservoirs

Differential area piston amplifier type lubricant reservoirs require one seal to be exposed to high differential pressure — typically the pressure between the lubricant and the atmosphere. In the rod type units shown in this chapter, that seal is the rod seal. The rod seal is typically relatively small in diameter, so the frictional contribution of the seal is relatively small despite the high differential pressure. Additionally, the small diameter makes it easier to maintain the small extrusion gap necessary for sealing the high differential pressure. In an annular arrangement, the friction of the atmosphere-exposed piston seal would be high due to its large diameter. This would create significant hysteresis, and significantly impair the ability of the piston to provide appropriate lubricant pressure. Failure to provide the appropriate lubricant pressure can cause destructive pressure reversals across the partitioning rotary seal. Additionally, in an annular piston, it is more difficult to provide the atmosphere-exposed piston seal with the small extrusion gap that is needed for reliable sealing of high differential pressure.

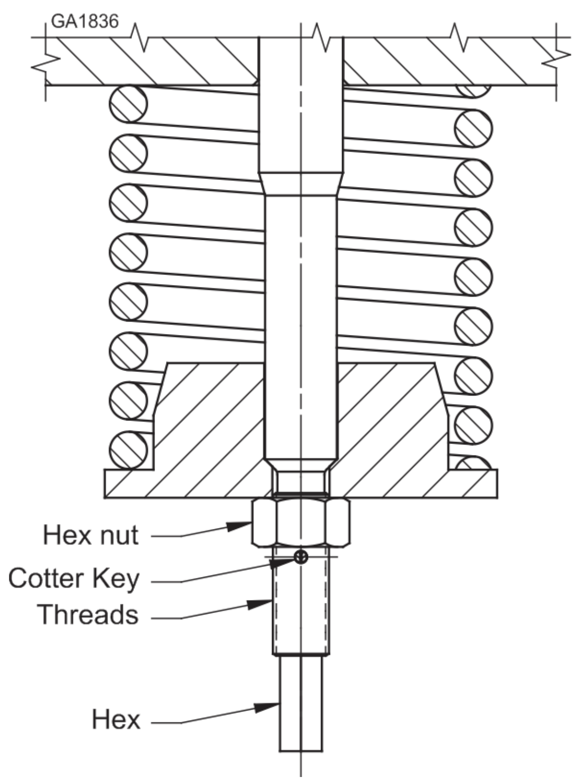

Figure 12

Safely compressing and retaining the spring

When the lubricant reservoir is empty, the spring can be safely compressed and uncompressed with the hex nut. When the unit is assembled, the nut is captured by the cotter key. A hex drive is required to prevent piston rotation while turning the hex nut. This figure shows the hex drive located on the piston rod. A better place for the hex drive would be inside the reservoir, on the piston itself, because a hex drive there can be bigger, and less likely to strip. **DO NOT ATTEMPT TO DECOMPRESS THE SPRING WHILE THE RESERVOIR CONTAINS LUBRICANT, BECAUSE THE SPRING WILL STILL BE COMPRESSED WHEN THE HEX NUT DISENGAGES FROM THE PISTON ROD.** If the hex drive is located on the lubricant side of the piston, the reservoir has to be emptied before the hex drive can be accessed, so there is less chance of human error.

Minimizing downtime

In some applications, such as offshore oil rigs, it is critical from a cost standpoint to minimize maintenance time. In such cases, a spare lubricant reservoir and rotary seal assembly should be kept on hand so that a quick exchange can occur if a lubricant reservoir becomes contaminated due to rotary seal failure. After the unit is back up and running, the contaminated lubricant reservoir and seal assembly can be disassembled and cleaned at the mechanic's leisure. Critical sealing surfaces of the lubricant reservoir should be provided with adequate corrosion resistance and appropriate surface finishes for the intended service. In oilfield service, we have successfully used chrome for the sealing surfaces of the piston rod and the reservoir bore.

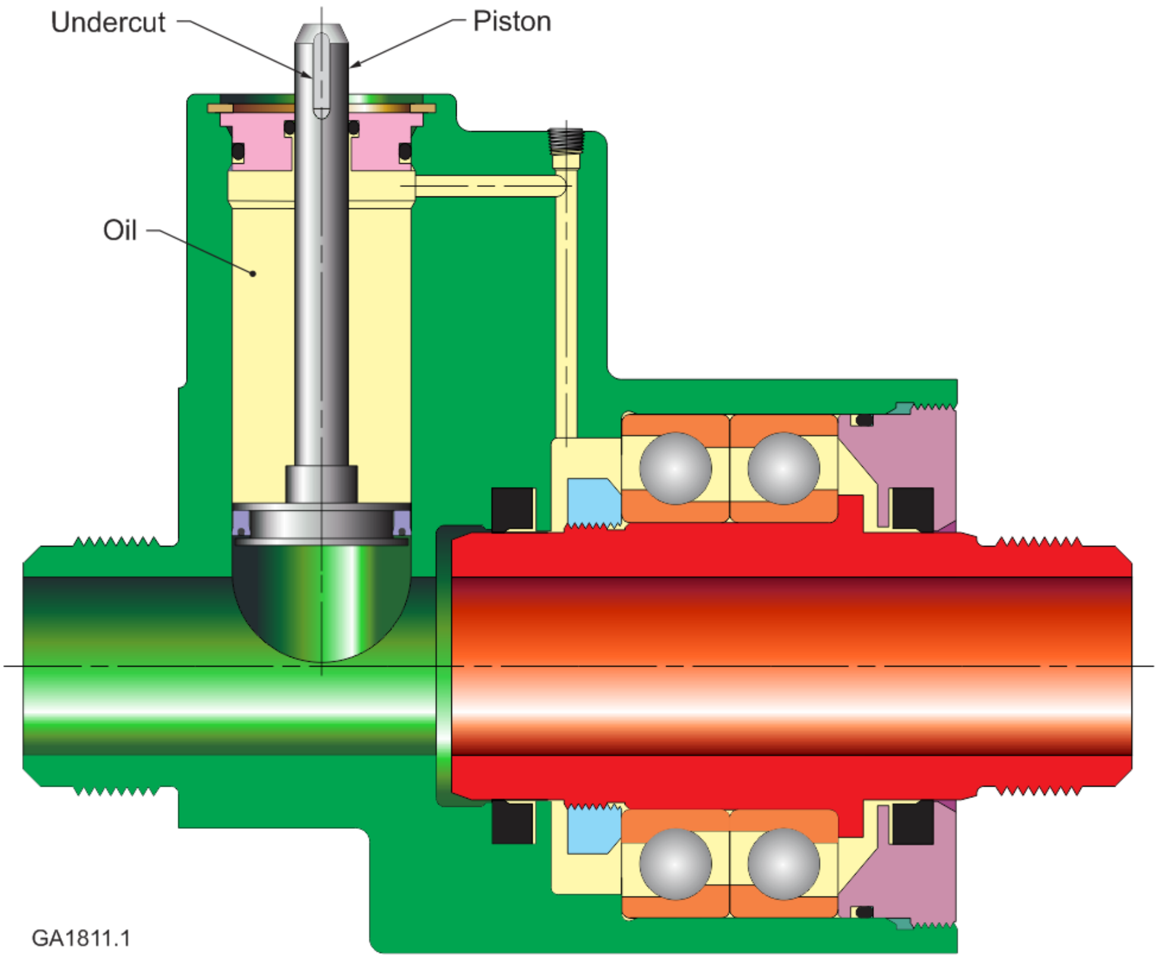

Figure 13

Integrated piston amplifier type lubricant reservoir

This figure shows a piston amplifier type of lubricant reservoir that is integrated into a swivel. As soon as the pressure in the process fluid increases, it pushes on the piston, which amplifies the oil to a pressure greater than the process fluid. For maximum concentricity, the right-hand seal carrier pilots on the housing bore that locates the outer bearing races. Preferably, the piston should be spring loaded to prevent pressure lag.

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5. General comments on rod and piston seals

O-ring energized Rod and piston seals

Figure 5 shows the use of O-ring energized seals as piston seals. Unlike O-rings, O-ring energized seals are immune to spiral failure in reciprocating applications. Call your seal supplier for availability of O-ring energized seals before specifying them. Not all cataloged sizes, styles, and materials are readily available. When O-ring energized seals are used as rod or piston seals, reduced compression may be desirable to minimize sliding friction. In surface equipment where elevated temperature isn't an issue, we have had excellent results with O-ring energized lip seals having a 70 Shore A FKM O-ring energizer, and a low friction carboxolated nitrile shell.[6]

Table 1 shows dimensions and tolerances that were provided to Kalsi Engineering by one major manufacturer of O-ring energized lip seals in 1990. This information is provided for reference only, as the best available information we have on this type of seal. Please contact your lip seal manufacturer or supplier to obtain their current dimensions and tolerances, rather than relying on Table 1.

Table 1
O-ring energized seal dimensions from one manufacturer

Size & type	Radial lip dimension	Lip tolerance
1/4" Regular	0.287"	±0.012"
5/16" Regular	0.355"	±0.016"
3/8" Regular	0.423"	±0.016"
3/8" Deep	0.437"	±0.016"

Extreme pressure rod seals

When Piston amplifier type lubricant reservoirs are used in extreme pressure situations, the pressure capacity of commercially available rod seals may be exceeded. In such situations, consider licensing our floating backup ring (Chapter D17) for use with the rod seals, to maximize rod seal pressure capability.

Piston seal extrusion gaps

Figure 14 is an enlargement of the piston seal region of Figure 13, to show extrusion gap details. The extrusion gaps on the lubricant and process fluid sides of the piston seal differ in both clearance and length. The extrusion gap clearance is larger on the process fluid side,

[6] We tested an annular piston with 3.500" AND 5.700" sealing diameters. This piston is shown in Figure 6. After allowing the assembly sit over the weekend, the pressure required to initiate piston movement was less than 20 psi.

so that any contact between the piston and the surrounding housing occurs in a lubricated zone. The extrusion gap clearance on the process fluid side is axially short, to minimize entrapment of abrasive particles.

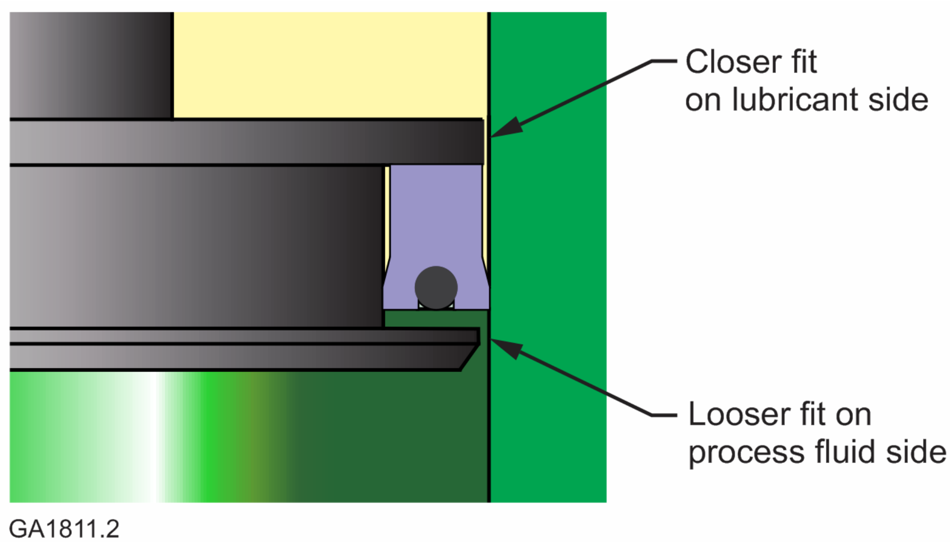

Figure 14

Extrusion gap detail, piston seal of lubricant reservoir

This is an enlargement of Figure 13, but is applicable to any piston based lubricant reservoir. A closer fit is provided on the lubricant side of the piston seal, for piston guidance. The extrusion gap clearance on the process fluid side is larger, so that any contact between the piston and the surrounding housing occurs in a lubricated zone. The extrusion gap clearance on the process fluid is axially short, to minimize entrapment of abrasive particles between the piston and the surrounding housing.

6. Annular lubricant reservoirs

Figure 15[7] is a schematic illustrating an annular spring-loaded piston that surrounds a rotary shaft and produces lubricant overpressure. The partitioning seal is located at a fixed axial location in a laterally floating seal carrier, instead of being mounted in an axially moving piston. The spring is telescoped over part of the piston, to conserve equipment length while maximizing lubricant volume.

The radial extrusion gap fit on the process fluid side on the piston seals is relatively large and axially short, to minimize abrasive particle entrapment, and associated damage to the guide rod and the bore of the housing.

7 Also see Figure 3.

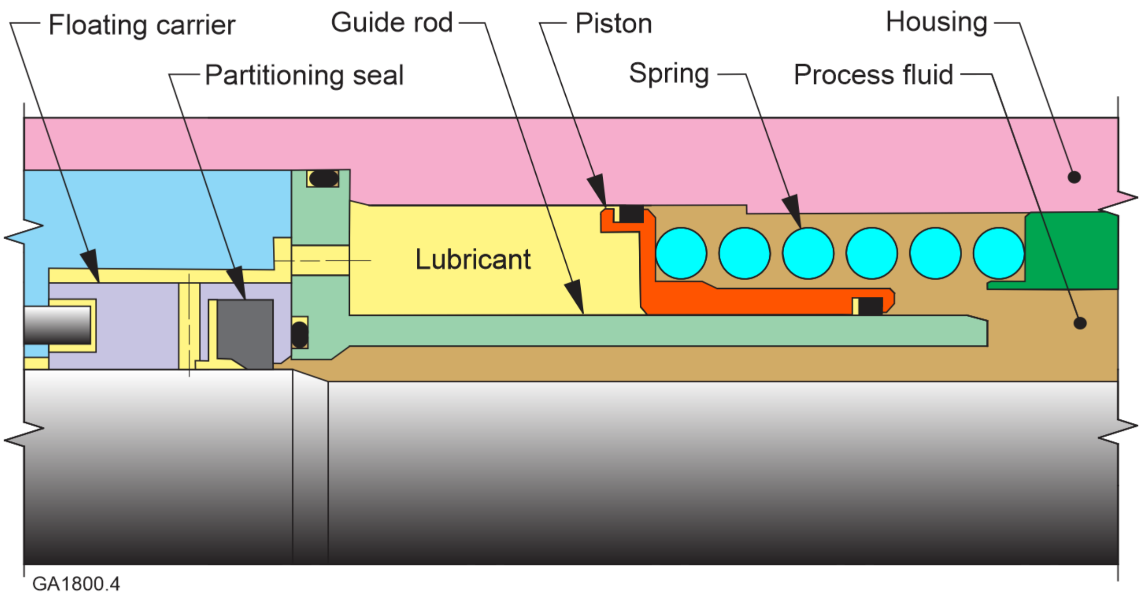

Figure 15

Schematic of a spring telescoped over annular piston

It is sometimes desirable to mount a lubricant reservoir around a rotary shaft, without having the inner seal of the piston in sealing engagement with the rotary shaft. This allows the partitioning rotary seal to be mounted in a fixed axial location, rather than being mounted in the piston. As shown here, the spring can telescope over a portion of the piston to conserve length.

In some cases, it will be desirable to provide a piloting relationship between the housing and the right-hand end of the guide rod. However the annular piston arrangement is designed, it is imperative to provide for the free flow of process fluid into the spring area, so that the pressure of the lubricant increases as fast as the pressure of the process fluid in the region adjacent to the partitioning rotary seal. If the process fluid cannot enter the spring area rapidly, the partitioning seal will be exposed to a destructive situation, where the pressure of the process fluid is temporarily greater than the pressure of the seal lubricant.

In Figure 15, note that the installation chamfer for the partitioning seal is placed close to the partitioning seal. This helps to protect the partitioning seal during installation. Even if the stepped down shaft surface is damaged, that surface cannot damage the partitioning seal during installation because it has radial clearance with the seal during installation.

The coil spring is piloted at both ends, to prevent its catching on the end of the piston. The piloting surfaces engage the inside diameter of the spring and are sized to clear the minimum inside coil diameter, considering tolerances. Piloting to the outer diameter of the spring is less desirable, because the spring expands diametrically as it compresses.

Have a plan to safely compress and decompress the spring during assembly and disassembly.

Figure 16 shows a reservoir where, instead of spring-loading the piston to produce lubricant overpressure, the rotary seal is axially spring-loaded to prevent skew-induced wear. Information on axial spring loading of seals is provided in Chapter D9.

The annular piston arrangements shown in Figures 15 and 16 have an advantage over arrangements where the rotary seal is mounted in a rotary shaft-guided annular piston (Chapter D14), because the torque of the rotary seal does not act on the piston. This means that the piston doesn't need multiple sliding seals to generate friction to resist spinning with the rotary shaft. Less piston seal friction means that the pressure of the lubricant tracks the pressure of the process fluid more accurately.

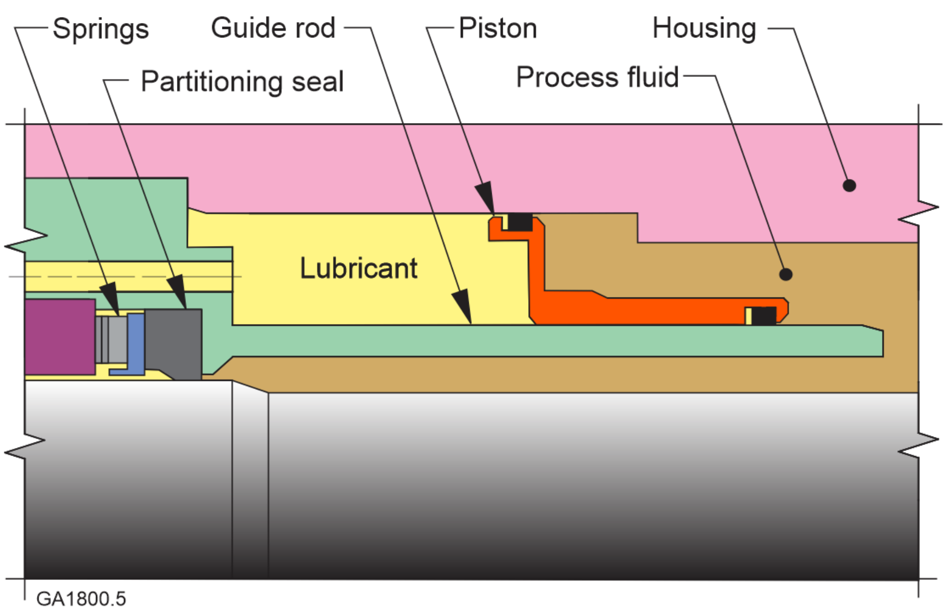

Figure 16

Schematic of a spring-loaded seal with an annular piston

In this schematic, the rotary seal is spring-loaded, rather than the annular piston. The piston balances the pressure of the lubricant to the pressure of the process fluid. The axial spring load holds the seal in a circular configuration to prevent the skew-induced wear that is described in Chapter D9. This arrangement is superior to the rotary shaft guided annular pistons described in Chapter D14 because the rotary seal is not mounted in the piston. As a result, the rotary seal does not reciprocate with the piston, and the piston does not have to incorporate extra friction-producing sliding seals to resist the torque of the rotary seal.

7. Troubleshooting rotary seal issues related to piston type reservoirs

Introduction

Rotary seal issues related to piston-type lubricators can often be traced to one of the following issues:

- Lubricant overfill
- Empty reservoir
- Stuck piston
- Pressure lag

Lubricant overfill – piston bottomed out in the full position

Lubricants are relatively incompressible and have a higher coefficient of thermal expansion than metal. Lubricant overfill (failure to provide room for lubricant thermal expansion) can cause both the partitioning seal and the pressure retaining seal to experience high pressure extrusion damage (Figure 17) to the circular exclusion edge of the dynamic lip. The damage is caused by the extremely high system pressure that can occur as a result of thermal expansion of the lubricant (Figure 5). Unless the piston incorporates a vent seal, train personnel to leave room for lubricant thermal expansion!

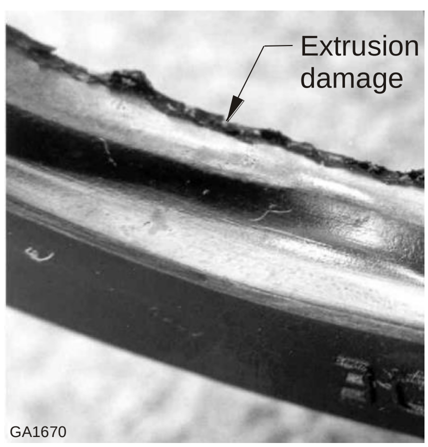

Figure 17

Example of high-pressure extrusion damage

This photo shows high pressure extrusion damage to the circular exclusion edge of a Kalsi-brand rotary seal. Such damage is typically progressive in nature and can work its way across the entire dynamic lip. In extreme cases, nearly all of a local portion of the seal can be consumed, as more and more material is fed into the extrusion gap and destroyed.

Empty reservoir – piston bottomed out in the empty position

In a high pressure two-seal arrangement like Figure 1, an empty reservoir exposes the partitioning rotary seal to high “reverse pressure”; i.e. high differential pressure acting across the seal from the environment/process fluid side.

The distinctive seal damage that occurs when partitioning rotary seals are exposed to high reverse pressure is shown in Figure 18. Seals exposed to less reverse pressure may experience a high rate of third body abrasive wear due to pressure-induced seal distortion. This is sometimes accompanied by visual evidence of a wider than normal contact zone resulting from the seal distortion.

There are various potential causes of an empty reservoir, such as forgetting to fill or refill the reservoir, having so much air in the equipment that the piston bottoms out in the empty position before pressurizing the lubricant, leakage past the piston seals (or static seals) due to a variety of potential causes, or the normal consumption of lubricant due to the hydrodynamic pumping related leakage of the Kalsi-brand rotary shaft seal.

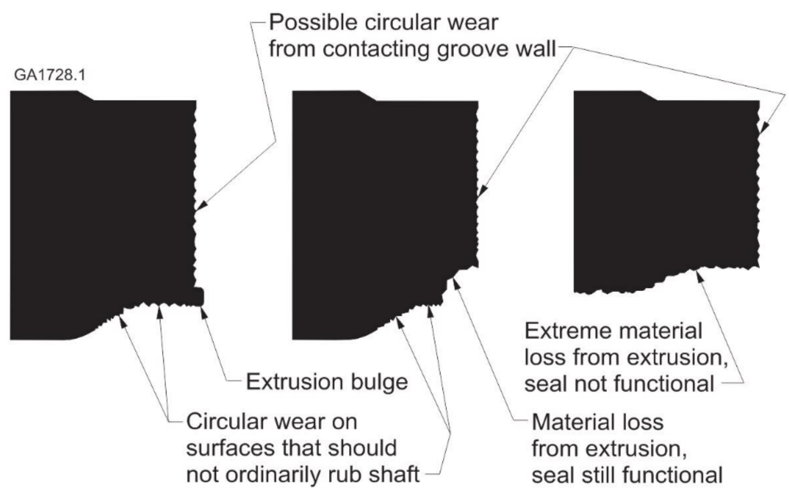

Figure 18

Examples of partitioning rotary seals exposed to high “reverse pressure”

This image depicts partitioning rotary seals that have received varying degrees of damage due to high differential pressure acting from the environment side of the seal. The degree of illustrated damage increases in severity from left to right.

Stuck Piston

In a high pressure two seal arrangement like Figure 1, a stuck pressure compensation piston may cause reverse pressure damage (Figure 18), high pressure extrusion damage (Figure 17), or both types of damage may occur at different times. If the piston sticks and lubricant thermal expansion occurs, the result can be high pressure damage (Figure 17) to both the partitioning seal and the high-pressure seal. Eventually, as hydrodynamic pumping related leakage of the rotary seals occurs, the lubricant pressure drops, and the result can be high “reverse pressure” across the partitioning seal (Figure 18). If the whole apparatus is immersed in a high-pressure environment, the leakage-related lubricant pressure drop can also cause high reverse pressure across the pressure-retaining seal. One example of a piece of equipment that is immersed in a high-pressure environment is an oilfield mud motor sealed bearing assembly used in directional drilling.

There are various potential causes of stuck pistons, such as sticky drawer effect (Chapter D21), excessively tight piloting, pressure locking of the sliding seals (Chapter D13), attempting to have a tight guiding fit at both the ID and OD of an annular piston (Chapter D14), and equipment deformation due to mechanical or pressure loading[8]. For another example, excessive eccentricity can cause binding of both annular pistons and the rod and piston arrangement of differential area-based piston amplifier type reservoirs (Section 4 of this chapter).

Combinations of conditions can also exist, such as binding an annular piston between the housing bore and the guide rod as a result of equipment deformation, combined with a tight fit with both the guide rod and housing bore. Annular pistons should be shaft-guided[9], providing a looser fit with the housing to accommodate equipment deformation that can cause lateral movement between the piston and the housing bore.

Pressure lag

During periods of increasing process fluid pressure, the pressure of the seal lubricant may lag behind the process fluid pressure, resulting in reverse pressure across the partitioning rotary seal. Depending on factors such as the type of partitioning seal, the way the seal is implemented, and the amount of reverse pressure, third body abrasion of the partitioning seal may occur due to reverse pressure induced seal distortion and axial displacement.

One cause of pressure lag is sliding seal friction. In some types of equipment, this can be addressed by axially spring loading the piston and reducing the initial compression of the piston seals (and in amplifier type reservoirs, the rod seal). Pressure lag related to sliding seal friction can also be minimized by using more favorable ratios between piston area and

8 For example, mud motor sealed bearing assemblies (Figure 2) use annular pressure compensation pistons that are located between the housing and the upper end of the shaft. The housing flexes due to drilling side load, and the upper end of the shaft flexes due to nutation-related rotor side load. Additionally, the upper end of the shaft has lateral motion due to bearing clearance and drilling side load.

9 For the same axial guide length, a rod guided annular piston is also less prone to sticky drawer effect, compared to a bore guided annular piston.

the circumference of the sliding seals. With more area, it takes less differential pressure to create the force necessary to overcome the breakout friction of the sliding seals.

Tight lubricant passage clearance can sometimes cause pressure lag, because it takes time for pressure changes to pass through tight clearance, particularly with higher viscosity lubricants. For examples of tight passage clearance producing detrimental pressure lag at the partitioning seal, and suggested solutions, see Figure 22.

With external piston-based lubricators, the pressure changes may have to travel through a lubricant hose or a tube to reach the partitioning seal. For example, see Figure 7. With such external reservoir arrangements, entrained air within the lubricant may cause pressure lag because it takes time for lubricant to flow through the hose or tube to compress entrained air between the rotary shaft seals, and within the hose or tube. The pressure lag effect of entrained air can be minimized by degassing the lubricant, using a larger tube or hose, selecting a rotary shaft seal that is compatible with a lower viscosity lubricant, and (within reason, considering other factors) minimizing the lubricant volume between the partitioning seal and the pressure retaining seal.

Referring to Figure 1, care must be taken to eliminate air between the pressure retaining rotary seal and the partitioning rotary seal when filling the assembly with lubricant. Several methods are possible, such as removing the air with a vacuum pump prior to introducing the oil, or by providing a strategically positioned bleed port that allows air to escape as the lubricant is introduced between the rotary seals.

With external piston-based lubricators, the pressure changes may have to travel through a process fluid hose or a tube to reach the pressure compensation piston. Such systems should be avoided, if possible, especially in cases were the process fluid tends to solidify or pack up. If such systems cannot be avoided, consider using gravity to keep the hose or tube full of process fluid (Figure 10) and using a relatively large diameter process fluid hose. In some cases, it may be possible to use a diaphragm to separate the process fluid from a clean, low viscosity, degassed fluid that transfers pressure to the piston.

Extreme vibration

In oilfield downhole drilling, the downhole equipment is subject to extreme levels of reversing axial G-force from drilling related forces and drill string elasticity. In view of high G-forces, it seems prudent to minimize the weight of compensation pistons in downhole drilling equipment, so that vibration related changes to lubricant pressure are minimized.

8. Gas-Charged Lubricant Reservoirs (Air over Oil)

In surface equipment, compressed gas-charged lubricant reservoirs can be utilized to provide a lubricant pressure that is higher than the process fluid pressure, as shown by Figures 19 and 20. The lubricant reservoir of Figure 19 uses a fixed charge of pressurized gas, and the lubricant reservoir of Figure 20 is connected to a regulated pressure source.

When designing gas-charged lubricant reservoirs that use a fixed charge of gas, the pressure effect of changes in ambient temperature and changes in lubricant level must be taken into consideration to ensure that the lubricant pressure remains higher than that of the process fluid pressure.

Figure 19

Gas charged lubricant reservoir

In some applications, nitrogen-charged lubricant reservoirs can be employed to keep the lubricant pressure greater than the process fluid pressure. The resulting pressure differential across the partitioning rotary seal provides improved resistance to third-body abrasion.

The Figure 19 lubricant reservoir has a simple sight glass for monitoring lubricant level. A pressure relief device such as a relief valve or rupture disk should be used to prevent overcharging and pressure damage to the lubricant reservoir or the sight glass. The Figure 20 lubricant reservoir incorporates a float type liquid level transducer which allows remote monitoring. Kalsi Engineering uses this type of lubricant reservoir on its rotary seal test fixtures, so computers can monitor and control the tests. Note the float snorkel, which allows the float to remain buoyant even when very high gas charge pressure is used.

Depending on how the lubricant reservoir is filled, it may be necessary to bleed the lubricant connection line to eliminate entrapped air. (In vertical shaft applications, such trapped air may collect near the upper rotary seal, potentially causing lubricant starvation.)

Commercial air/oil reservoirs, which are designed to use shop air pressure to produce hydraulic pressure, can also be used for pressurizing the seal lubricant. Precautions should be taken to avoid excessive moisture in the compressed air supply line.

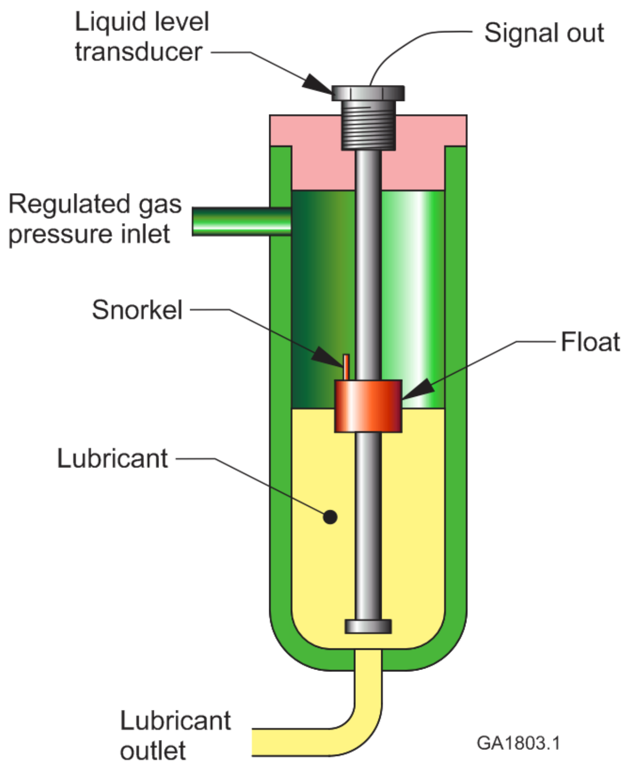

Figure 20

Regulated gas charged lubricant reservoir with transducer

Kalsi Engineering's rotary seal test fixtures use regulated nitrogen pressure to pressurize the seal lubricant. The float type liquid level transducer allows the lubricant level to be monitored by the data acquisition computer. The float snorkel allows the float to remain buoyant even when very high gas pressure is present.

9. Pump generated lubricant pressure

Computer control of pump generated lubricant pressure

One reliable method for producing and controlling lubricant pressure is to use a fixed displacement pump to circulate the lubricant through a computer controlled pinch valve that creates backpressure[10]. The pressures of the lubricant and the process fluid are sensed with pressure transducers, and the computer controls the orifice size of the pinch valve to maintain a lubricant pressure that is a desired amount greater than the pressure of the process fluid. The pressure capacity of such systems is limited to a few thousand psi by the pressure capacity of commercially available pinch valves.

If the process fluid is oilfield drilling fluid, the pressure transducer can be protected with a conventional diaphragm-type gauge protector. The drilling fluid passage to the transducer or diaphragm should be as large in diameter as practical, and as short as possible, to inhibit clogging in cases where drilling fluids with solidifying tendencies are encountered.

The circulating system can be adapted to cool or warm the seal lubricant, as required. (Warming of the lubricant can be desirable in arctic applications, to minimize rotary seal startup hydrodynamic pumping related leakage.) Startup leakage can also be minimized by temporarily applying greater lubricant pressure until the system warms up.

With computer controlled lubricant over-pressure systems, the machine designer must decide whether to circulate and pressurize a dedicated seal lubricant, or to utilize an existing system fluid (such as hydraulic fluid or bearing lubricant) as the seal lubricant. The use of a dedicated seal lubricant, isolated from other system lubricants, has the following advantages:

- The viscosity of the seal lubricant can be tailored to the seal, which can be used to minimize the hydrodynamic pumping related leakage of Enhanced Lubrication Seals™.
- The system fluid, and the related mechanical components, do not become contaminated if only one of the rotary seals fails that defines the seal lubricant chamber.
- Pressurization of the bearing cavity, which can result in some loss of bearing control due to pressure expansion of the bearing housing, is avoided.
- Pumping of the bearing lubricant, which may be too viscous for effective pumping in some temperature conditions, is avoided.

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[10] This method is protected by third-party patents in at least one type of equipment.

- The pressure response may be faster, because less fluid volume is involved, compared to pressurizing the bearing lubricant.

Manual control of pump generated lubricant pressure

In addition to the computer-controlled arrangement described previously in this chapter, lubricant pressure can be controlled manually with a back-pressure valve or by controlling pump speed. This method is more suited to relatively low process fluid pressures and requires that the lubricant pressure be established before the process fluid is pressurized.

10. Dealing with rapid process fluid pressure fluctuations

Using lubricant over-pressure to deal with pressure fluctuations

In some applications, the process fluid pressure fluctuates rapidly. For example, in oil well surface equipment applications such as rotating control devices (RCDs), the drilling fluid pressure fluctuates rapidly as a result of mud pump design.

In rotating control devices, the lubricant pressure is maintained at a value that exceeds the greatest drilling fluid pressure fluctuations, in order to prevent pressure reversals across the rotary seal that partitions the drilling fluid from the lubricant. Published literature on computer-controlled over-pressure systems for RCDs mention 300 psi (2.07 MPa) lubricant over-pressure as being satisfactory.

In addition to dealing with rapid pressure fluctuations, computer controlled lubricant over-pressure systems must deal with the rise in drilling fluid pressure when the pumps are turned on, and the drop in drilling fluid pressure when the pumps are turned off. During the periods of rising pressure, we *estimate* that the lubricant overpressure should always be at least 15 psi greater than the instantaneous drilling fluid pressure. During steady state conditions, we recommend that the lubricant overpressure be at least 100 psi[11] greater than the instantaneous drilling fluid pressure. During periods of dropping pressure, we recommend that the lubricant over-pressure be no more than 1,500 psi greater than the instantaneous drilling fluid pressure. These estimates and recommendations assume a -11 HNBR mud-to-lubricant partitioning seal facing a 0.010” radial extrusion gap clearance.

In our experience with smaller oilfield surface equipment using spring loaded piston amplifier type lubricant reservoirs of the type shown in Figures 7 and 8, a lubricant over-pressure in the range of about 160 to 240 psi (1.10 to 1.65 MPa), discounting piston friction,

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[11] This recommendation is based on testing performed with a mud-exposed -11 HNBR seal, 0.010" radial extrusion gap clearance, and 100 to 300 psi lubricant over-pressure. Additional testing would be required to determine the compatibility of a 0.010" radial extrusion gap with lubricant over-pressure less than 100 psi.

was enough to deal with mud pump related pressure spikes and piston related lubricant pressure lag.

Fast acting diaphragms

Another way to deal with rapid process fluid pressure fluctuations is with a fast acting, low hysteresis diaphragm which instantly applies the process fluid pressure to the seal lubricant. The diaphragm shown in the schematic of Figure 21 is an example of a very simple form of diaphragm that is often produced by a mandrel wrapping process. Many other diaphragm shapes are possible, such as shapes with molded convolutions that allow the diaphragm to “roll” as the lubricant volume changes. Commercially available rolling diaphragms can be spring loaded, and some designs provide for a relatively long stroke. Diaphragm design is a specialized field that is beyond the scope of this chapter.

Non-spring-loaded diaphragms are only recommended for use with Kalsi-brand partitioning seals that do not require differential pressure acting from the lubricant side; i.e. Axially Constrained Seals™ (Chapter C4) or axially spring-loaded seals (Chapter D9).

Because diaphragms expose a large area to the process fluid and the lubricant, the elastomer used to fabricate the diaphragm must be carefully selected for chemical compatibility. Diaphragms are not suitable for applications that may place high differential pressure across the diaphragm, acting from the lubricant side, in unusual operating conditions. Diaphragms can be configured to automatically vent excess lubricant volume resulting from lubricant thermal expansion; see expired U.S. patent 4,727,942.

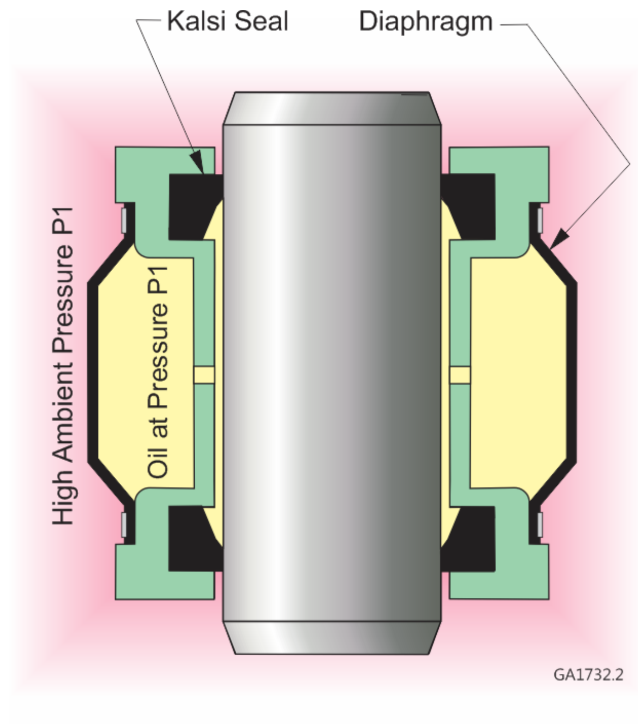

Figure 21

Schematic of a diaphragm for balancing lubricant pressure to ambient pressure

A flexible elastomeric diaphragm can be used to balance the lubricant pressure to the ambient pressure, and to serve as a lubricant reservoir to accommodate the hydrodynamic pumping related leakage of the Kalsi Seals. The ambient pressure is transmitted to the seal lubricant via the diaphragm. Many different diaphragm shapes are possible.

Filling diaphragm and bladder type lubricant reservoirs

One possible way to fill a diaphragm or bladder type lubricant reservoir, without risk of overfilling, is to provide an inlet and an outlet, and fill via the inlet until excess lubricant runs out of the outlet.

Lubricant side seal carrier to shaft clearance influences pressure lag

The clearance between the shaft and the seal carrier, and the length of that clearance, influences the time it takes for a lubricant reservoir pressure change to reach the rotary seal (Figure 22). Design your equipment to minimize pressure lag.

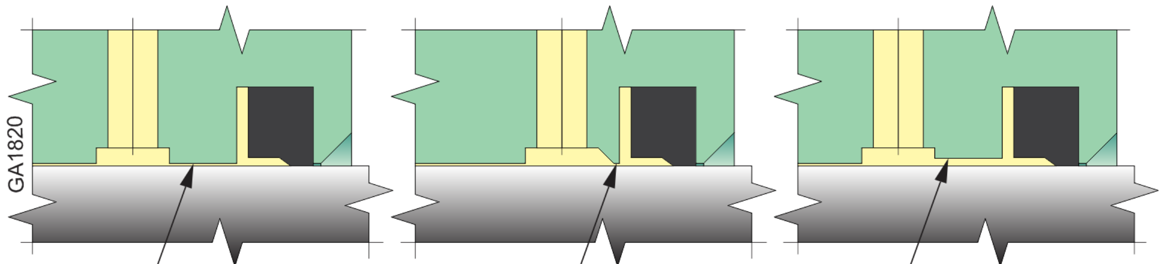

Figure 22

Seal carrier to shaft clearance influences pressure lag

In this figure, the lubricant pressure is communicated from the lubricant reservoir to the rotary seal through the radial hole. The lubricant side clearance between the seal carrier and the shaft, and the length of that clearance zone, influences lubricant pressure lag. Long, tight clearances cause demonstrable pressure lag. If the clearance has to be tight to support the seal in the event of inadvertent high reverse pressure, keep the length of the tight clearance axially short, as shown in the middle view.

11. Lubricant reservoir material selection

Materials used to construct lubricant reservoirs should consider all applicable service requirements, such as but not limited to:

- Corrosion resistance.
- Resistance to the stress and fatigue of the application.
- Any special low or high temperature material requirements.
- Galling resistance of sliding components.

Elastomers selected for use in the lubricant reservoir should consider all applicable service requirements, such as but not limited to:

- Compatibility with the seal lubricant, environment chemicals, and operating temperature range.
- Compatibility with the anticipated differential pressure.
- Sliding seal material friction characteristics.

12. Reservoir volume and strength

The lubricant reservoir should be sized to accommodate the hydrodynamic pumping related leak rate of the Kalsi-brand rotary seals over the desired time interval between refills. Volume calculations should account for the anticipated increase in hydrodynamic pumping related leak rate due to cold weather conditions. In some cases, lower viscosity lubricants, including arctic grade lubricants, may be a desirable way of reducing the hydrodynamic pumping related leak rate if prolonged cold weather operation is anticipated.

Designers should consider testing equipment, including lubricant reservoirs, in anticipated real-world temperature conditions, to confirm performance. Reservoirs used in lubricant overpressure systems should typically be designed to the same pressure rating as the device to be sealed, so that they can be used to supply the pressurized seal lubricant during proof testing.

13. Types of lubricant reservoirs typical to oilfield downhole tools

This chapter can be confusing to designers of downhole oilfield equipment, because it describes several reservoir types that are only appropriate to surface equipment. Downhole lubricant reservoirs substantially equalize the pressure of the seal lubricant to the pressure of the ambient environment, while also accommodating differential thermal expansion of the lubricant, volumetric compression of the lubricant, and the hydrodynamic pumping related leakage of the Kalsi Seal brand rotary shaft seals. Downhole reservoir types are typically either piston-based or diaphragm/bladder[12] based. Because a typical oilfield tool contains a central passage to conduct the drilling fluid, downhole reservoirs are typically annular in nature. The lubricant reservoirs in mud motors, for example, are typically based on annular, shaft guided pressure compensation pistons.[13] Certain types of oilfield downhole equipment, such as roller reamers,[14] use piston-type lubricant reservoirs that are non-annular in nature.

14. Constant displacement lubricators

In some types of surface equipment, such as oilfield RCDs, a constant displacement pump flushes lubricant past one or more lip seals that face the process fluid, while a separate rotary seal retains the lubricant pressure. The constant displacement of lubricant into the sealed area causes the lubricant pressure to exceed the process fluid sufficiently to cause the lip seals to vent lubricant into the process fluid. Although Kalsi Engineering does not

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12 Diaphragms have been used in oilfield downhole equipment for many years. For an example of a downhole tool design that includes a diaphragm, see expired U.S. Patent 4,462,469, which shows an annular diaphragm providing pressure balancing between the bore of an oilfield drilling tool and the fluid within the tool.

13 Mud motor lubricant reservoirs based on pressure compensation pistons were first described in expired U.S. Patent 3,730,284. For an example of a mud motor with a spring-loaded compensation piston, see expired U.S. Patent 4,372,400. For examples of sealed bearing mud motors that incorporate a barrier compensation piston, see U.S. Patents 5,150,972, 5,248,204, and 5,664,891.

14 For an example of a sealed roller reamer with a lubricant reservoir based on a spring-loaded piston, see expired U.S. Patent 4,542,797.

sell lip seals for this type of service, a Kalsi-brand rotary shaft seal is an excellent choice as the pressure retaining seal. Contact us for assistance in selecting the most appropriate seal design.