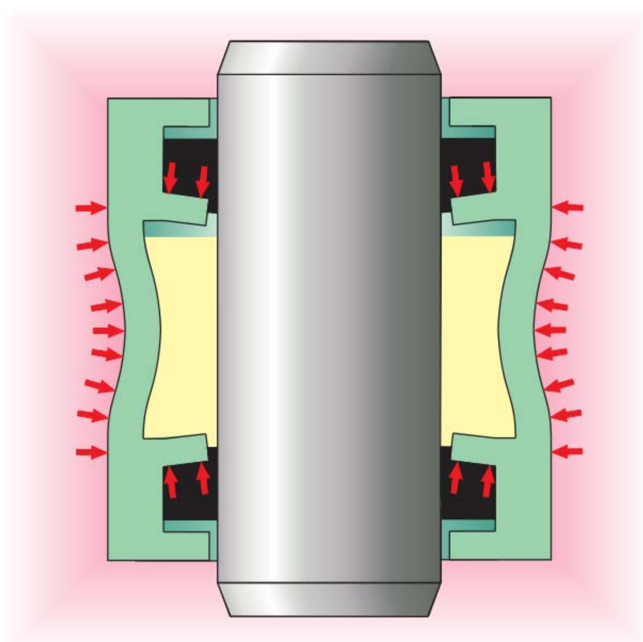
Chapter D13

Preventing seal pressure locking



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1. Thermal expansion can damage seals, bearings, and housings

Introduction to thermal expansion

All liquids, gases and solids expand when heated. This effect is called "thermal expansion", and it is proportional to the size of the item and to the temperature change increment.

Thermal expansion is quantified as a linear coefficient or a volumetric coefficient. The linear thermal expansion coefficient α is the fractional change of the length of a solid item for a temperature rise of one degree. The volumetric thermal expansion coefficient is the fractional change of the volume of a liquid, gas or solid for a temperature rise of 1°. For solids, the volumetric thermal expansion coefficient β is three times the linear thermal expansion coefficient α. The thermal expansion coefficients of various materials are typically published by their manufacturers, by manufacturing associations[1] and in engineering handbooks, along with other material properties.

Thermal expansion must be considered in many aspects of the design of rotating machinery. Some components expand faster than others, due to differences in operating temperature, or the use of materials which have differing coefficients of thermal expansion. For example, if a long shaft runs significantly hotter than the bearing housing, then putting the opposing bearing thrust shoulders at opposite ends of the shaft would be unwise, because the differential thermal expansion between the shaft and the housing is likely to overload the thrust bearings. Another example, the differential thermal expansion between the shaft and the housing should be considered when determining the diametric fit of bearing races, to avoid overloading the bearings.

Calculating linear thermal expansion

The formula for determining temperature-induced linear dimensional change of a solid item such as a shaft is provided as Equation 1.

Equation 1, Linear thermal expansion:

$$\text{Dimensional Change} = \text{Original Dimension} \times \alpha \times \Delta t$$

Where:

Δt = change in temperature

α = linear coefficient of thermal expansion

For example, if a 20-inch-long carbon steel shaft, with a linear coefficient of thermal expansion of 0.000007 inches per inch per °F, were increased in temperature by 200°F**,**

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1 The Copper Development Association provides comprehensive material properties for copper and its various alloys.

then the shaft length would increase by 0.028”. If the housing had the same length and linear coefficient of thermal expansion as the shaft, but only increased in temperature by 150°F, then its length would increase by 0.021”. Thus, the linear differential thermal expansion between the housing and the shaft would be 0.028”-0.021” = 0.007”. Clearly, if thrust capable bearings were located at the opposite ends of the shaft, then the bearing(s) at one end of the shaft would have to allow relative axial motion (e.g. slippage) to occur, in order to accommodate the differential thermal expansion. Consult your bearing manufacturer’s technical guide for additional information on this subject, and for specific bearing implementation guidelines.

Calculating volumetric thermal expansion

The formula for determining temperature-induced volumetric change is provided as Equation 2.

Equation 2, Temperature-induced volumetric change:

$$\text{Volumetric Change} = \text{Original Volume} \times \beta \times \Delta t$$

Where:

Δt = change in temperature

β = volumetric coefficient of thermal expansion

For example: If a machine contains 100 ml of lubricant that has a volumetric coefficient of thermal expansion of 0.0005/°F, then an increase in temperature from 70°F to 300°F (Δt = 230°F) would cause the lubricant volume to increase by 11.5 ml.

Lubricant thermal expansion can cause seal and hardware damage

Lubricants have a much higher coefficient of thermal expansion than the metals that are used in machine construction. For an example of the coefficient of thermal expansion of metal, see Table 1-5.0 in Appendix 1 of the 1989 Section III, Division 1 ASME Boiler and Pressure Vessel Code, which lists the mean linear coefficient of thermal expansion of carbon steel to be 0.00000687 inches per inch per °F for the temperature range of 70°F to 300°F. This equates to a volumetric coefficient of thermal expansion of 0.0000206/°F. According to one lubricant manufacturer, a typical volumetric coefficient of thermal expansion for their liquid type lubricants is 0.0005/°F. With these examples, the lubricant has a volumetric coefficient of thermal expansion that is more than 24 times greater than that of the steel.

Since oil is largely incompressible, the machinery must be able to accommodate the thermal expansion of the lubricant. Figure 1 illustrates the seal and housing damage that can occur due to high pressure resulting from confined lubricant thermal expansion.

Estimating the pressure rise of a trapped fluid due to thermal expansion

A simplified formula for estimating the temperature-induced pressure increase of a trapped fluid is provided as Equation 3. This formula assumes that the container is rigid and does not change in volume due to temperature and pressure increases.

Equation 3, Estimating temperature-induced pressure change:

$$\Delta P = K_m \times \beta \times \Delta t$$

Where:

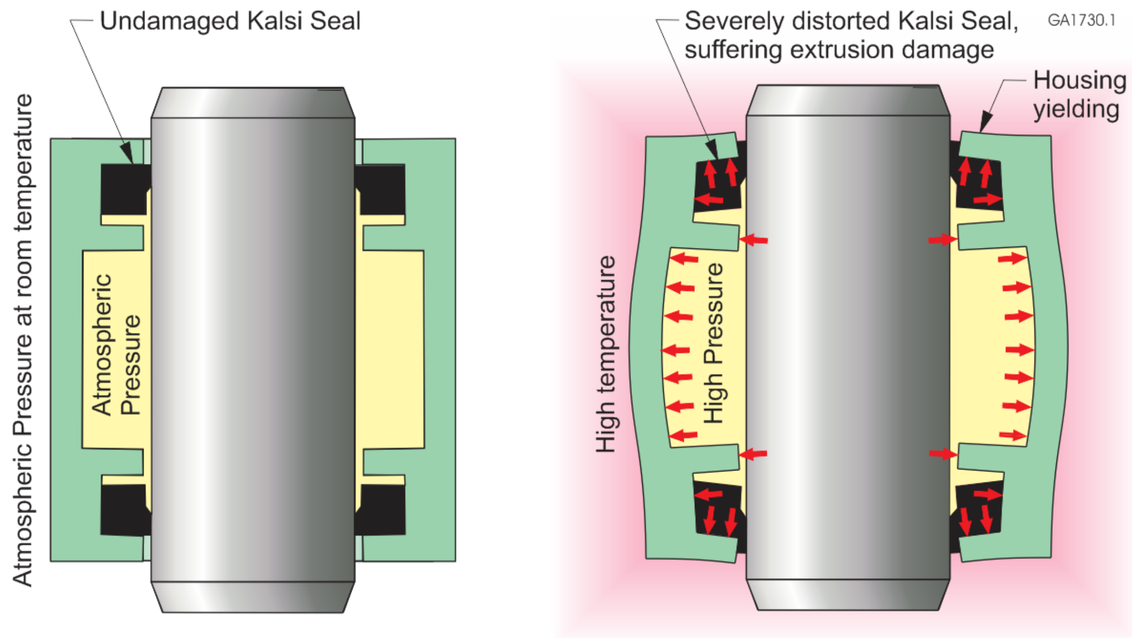
ΔP = change in pressure

K_m = secant bulk modulus

Δt = change in temperature

β = volumetric coefficient of thermal expansion

For example, at pressures above 3,000 psi the published volumetric coefficient of thermal expansion of one popular synthetic hydrocarbon lubricant is 0.0004/°F, and the bulk modulus is 220,000 psi when the entrained air is 10% or less. Using these values, the calculated increase in pressure for a 100°F increase in temperature is 8,800 psi. As a rule of thumb, the calculated value using this simplified formula can be decreased by 10% to account for changes to the volume of the reservoir, giving an adjusted pressure of 7,920 psi. Using this formula, one can appreciate that even a moderate temperature increase of a trapped fluid causes pressure that is greater than most rotary shaft seals are capable of withstanding.



(a) **Complete oil fill, atmospheric pressure trapped between seals at time of assembly**

(b) **Seals, and possibly even the housing, are severely distorted when unit is subsequently exposed to high temperature**

Figure 1

Schematic of pressure locking due to lubricant thermal expansion

This lubricant-filled housing does not have a pressure compensation arrangement to balance the pressure of the lubricant to the pressure of the ambient environment. The lubricant is trapped within the housing at atmospheric pressure during assembly, as shown at *"a"*. When the unit is subsequently exposed to, or generates, elevated temperature, the lubricant expands faster than the metal housing. The rotary seals and the housing are distorted and damaged by the resulting high differential pressure, as shown at *"b"*.

2. High ambient downhole pressure can damage seals and housings

Introduction to oilfield downhole ambient pressure

Downhole oil well equipment is subject to immense ambient hydrostatic pressure, due to the depth of the well and the weight of the fluid column within the well, and occasionally due to pump flow blockages. Ambient pressure in the many thousands of psi is common at typical well depths. Although well fluid temperature changes with depth, and the weight of a given volume of fluid changes with temperature because of thermal expansion, the approximate pressure at any given depth can be estimated by multiplying the nominal fluid specific weight by the depth of the well in feet.

For example: With a drilling fluid weight of 74.8 pounds per cubic foot, (10 lb/gallon) and a well depth of 10,000 feet, the ambient fluid pressure at 10,000 feet from the fluid column alone, discounting pump pressure, is 748,000 pounds per square foot (5,194.4 psi).

Mud weight is typically given in lb/gallon, and well depth is typically feet. Equation 4, which ignores downhole temperature- related[2] fluid density changes, is often used for estimating well pressure at a given depth.

Equation 4, Estimating well pressure:

$$P_{DEPTH} = 0.052 \times F_{WEIGHT} \times D_{WELL} + P_{SURFACE}$$

Where:

D_{WELL}	=	Depth of the fluid column within a well, in feet
F_{WEIGHT}	=	Fluid specific weight, in lb/gallon
P_{DEPTH}	=	Well pressure at a given depth, in psi
$P_{SURFACE}$	=	Well pressure at the surface, in psi

In Equation 4, the factor 0.052 is the rounded off result of the conversion factor 7.4805195 for converting pounds per US gallon to pounds per cubic foot, multiplied by the conversion factor 0.0069444 for converting pounds per square foot to pounds per square inch.

As shown in Figure 2, if the lubricant within the oilfield downhole equipment is not pressure compensated to the high ambient downhole pressure, then the rotary seals are exposed to high differential pressure acting from the environment side of the seals. This causes severe distortion and damage to the seals, and dramatically shortens rotary seal life. Transverse radial cross sections of seals that have been damaged from exposure to high ambient pressure type pressure locking are illustrated in Figure 3. It is critically important that the lubricant pressure within a downhole drilling tool (such as a mud motor or rotary steerable tool) be balanced to the ambient downhole pressure.

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2 As a rule of thumb, static well temperature typically increases by 1 to 2°F per 100 feet of depth. Temperatures are lower with circulation, and well temperature varies by geographic location (Source: "**Petroleum Engineering Handbook**", third printing, Society of Petroleum Engineers, 1992).

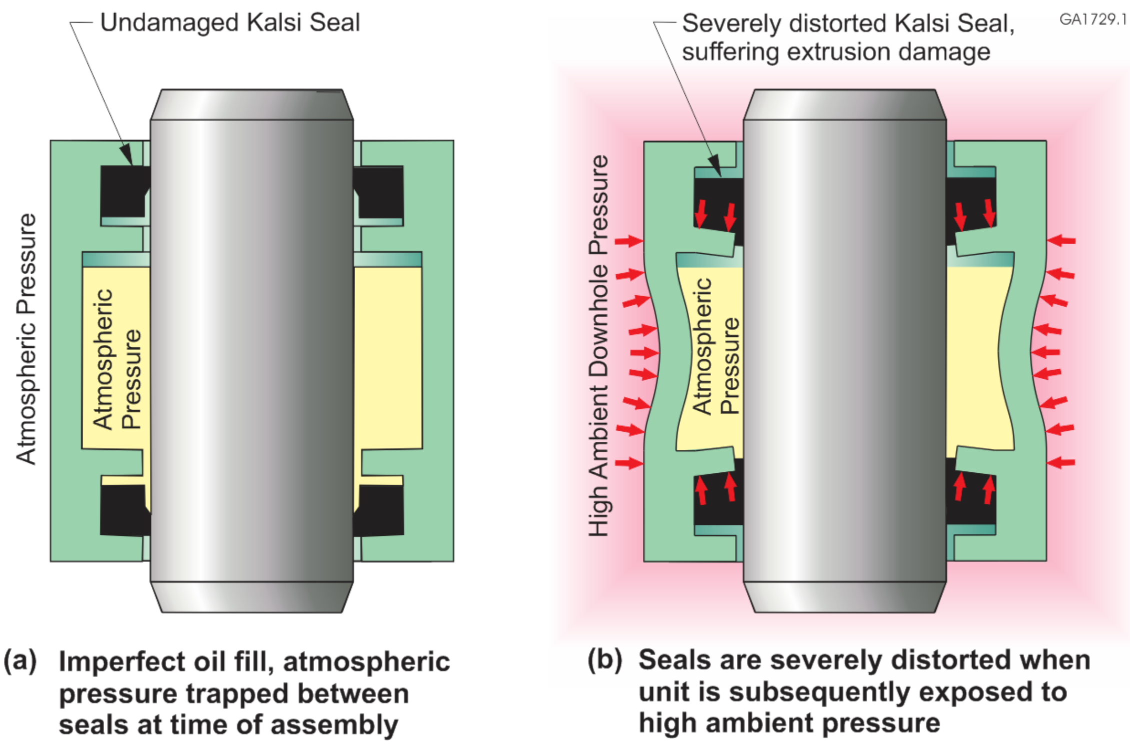

Figure 2

Schematic of hydraulic pressure locking due to lubricant under-fill

The lubricant-filled, sealed housing illustrated here does not have a pressure compensation arrangement to balance the pressure of the lubricant to the pressure of the ambient environment. The lubricant is trapped within the assembly at atmospheric pressure during assembly, as shown at *"a"*. When the unit is subsequently submerged in the high ambient pressure oilfield downhole environment, the lubricant remains at atmospheric pressure. As a result, the rotary seals and housing are distorted and damaged by the resulting high differential pressure acting across them, as shown at *"b"*.

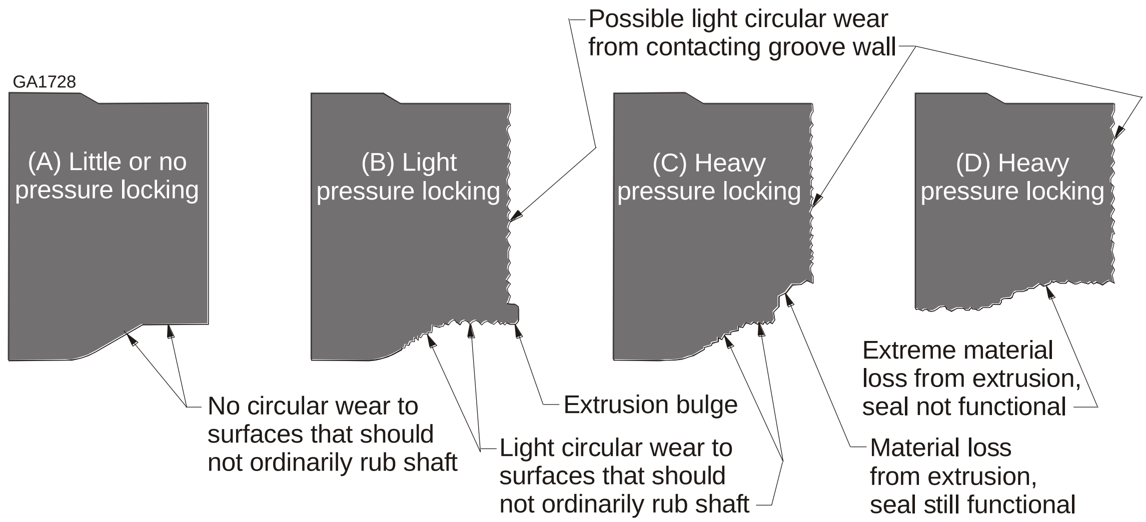

Figure 3

Signs of pressure locking on used Kalsi Seals® related to high ambient pressure

This figure illustrates the typical pressure locking damage that occurs to Kalsi-brand rotary seals when they are used in a high ambient downhole pressure environment with equipment that lacks lubricant pressure compensation. The seals were exposed to high differential pressure acting from the left. Due to the abrasive nature of the downhole environment, the dynamic surface may exhibit a very wavy band of third body wear. Such wear bands are circular during operation, when the seal is severely distorted. When the seal relaxes after disassembly, the wear band becomes wavy.

Seal material compressibility in high ambient pressure conditions

Although elastomer materials are resilient, allowing Kalsi Seals to be installed “in compression” (with radial squeeze), in strict engineering terms, elastomer materials are relatively incompressible. This means that volume remains substantially the same as squeeze-induced seal deformation occurs. Resiliency is not the same as compressibility.

Seal volume also remains relatively constant in high ambient pressure conditions. For example, at 72°F (22.2°C), the volume of a Kalsi Seal™ made from -11 HNBR material is estimated to decrease by only about three percent at 20,000 psi (137.9 MPa).

Elastomer compressibility increases as temperature increases, because the bulk modulus decreases. For example, with one common sealing material, the bulk modulus drops by about 45% between 72°F (22.2°C) and 350°F (176.7°C). This sounds significant, but is not, because the increase in compressibility is more than offset by elastomer thermal expansion. Seal compressibility is irrelevant in downhole drilling applications because higher ambient pressure is also accompanied by higher ambient temperature.

3. Preventing pressure locking in completely sealed systems

Using a pressure compensation piston to prevent pressure locking

In order to prevent seal and hardware damage from pressure locking in completely sealed systems such as oilfield downhole tools, and to accommodate the hydrodynamic pumping related leakage of the Kalsi Seals, a moveable member such as a piston or a diaphragm[3] must be used. Figure 4 shows a typical axially movable compensation piston[4]. This piston moves downward to balance the lubricant pressure to the environment pressure P1, and to accommodate the hydrodynamic pumping related leakage of the rotary seals. The piston also moves upward to accommodate lubricant thermal expansion. Due to piston sliding seal friction, the lubricant pressure can be slightly higher or lower than pressure P1. If the piston does not bottom out in either direction of travel, the lubricant pressure will be approximately balanced to that of the environment pressure P1.

In order to prevent thermal expansion-based pressure locking, the lubricant filling process must leave the piston at an intermediate axial position that allows room for enough thermal expansion induced piston movement. One way to achieve this intermediate position is to fill the reservoir until piston motion is arrested by a temporarily installed motion stop, and then remove the stop. Another way to achieve the intermediate position is to fill the reservoir with a known quantity of oil. To avoid thermal expansion problems, the reservoir can also be filled completely, and then the piston can be moved back to an intermediate position, expelling a predetermined oil volume.

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3 For examples of diaphragms being used to pressure compensate lubricant within an oilfield drilling tool, see U.S. Patents 3,007,750, 3,721,306, 4,199,856, 4,335,791, 4,372,400, 4,727,942, and 7,673,705

4 Mud motor lubricant pressure compensating pistons were first described in the now-expired 1973 U.S. Patent 3,730,284.

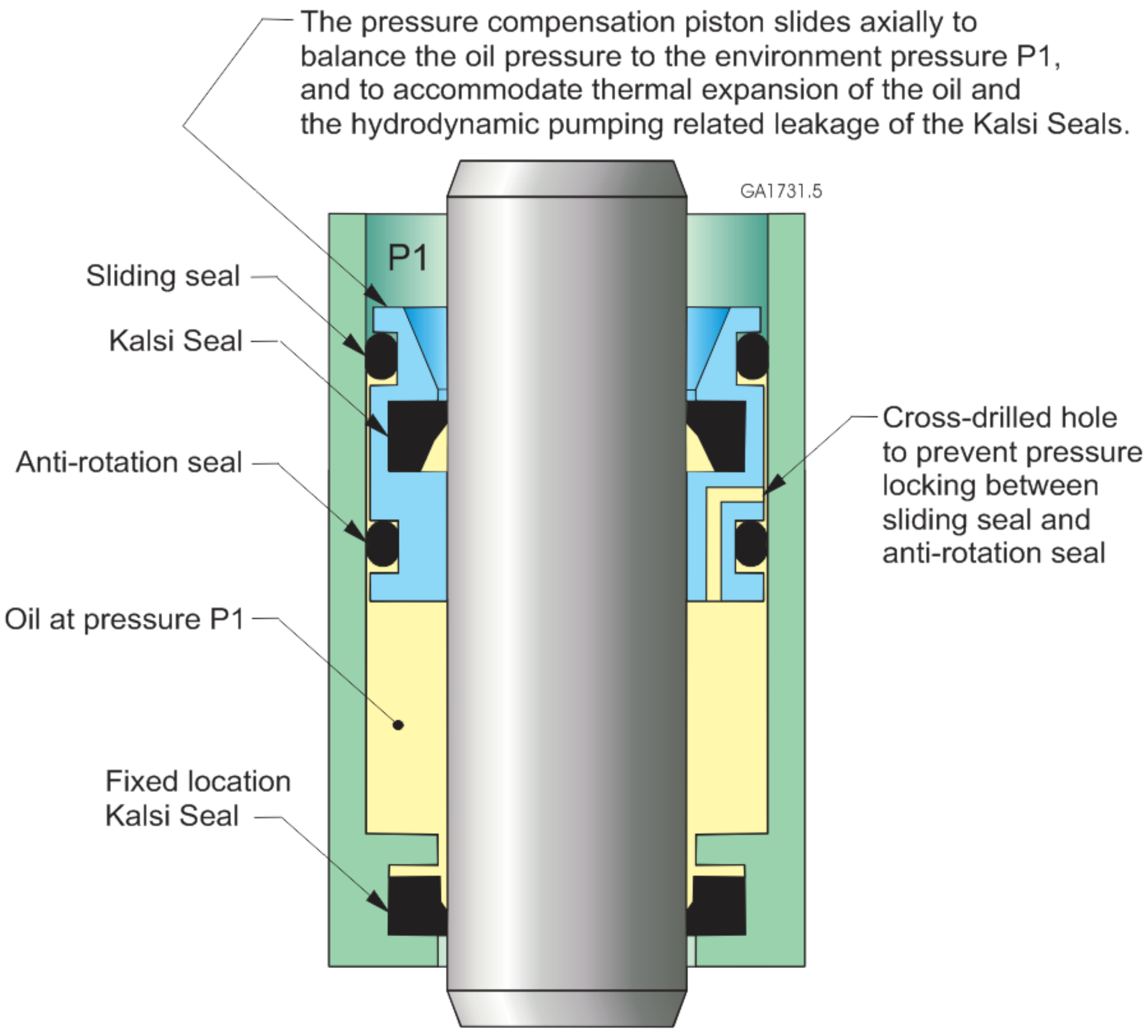

Figure 4

Schematic of a pressure compensation piston for preventing pressure locking

A pressure compensation piston can be used to balance the oil pressure to the environment, to accommodate thermal expansion of the oil and to accommodate the hydrodynamic pumping related leakage of the Kalsi-brand rotary seals. Multiple sliding seals are used to generate enough friction to prevent the piston from rotating with the shaft. A cross-drilled hole prevents pressure locking between the sliding seal and the anti-rotation seal.

Employing a diaphragm to prevent pressure locking

As shown in the schematic of Figure 5, a flexible elastomeric diaphragm can be used to prevent pressure locking. The diaphragm flexes inward to transmit the environment pressure to the lubricant, and flexes outward to accommodate lubricant thermal expansion. The diaphragm also serves as a lubricant reservoir to accommodate the hydrodynamic pumping related leakage of the Kalsi Seals.

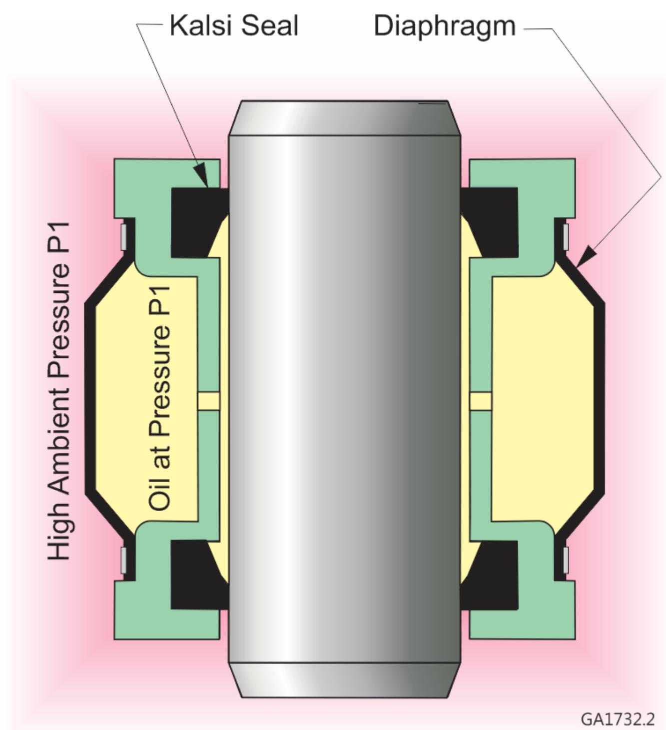

Figure 5

Schematic of a diaphragm for preventing pressure locking

A flexible elastomeric diaphragm can be used to prevent pressure locking, and to serve as a lubricant reservoir to accommodate the rotary hydrodynamic pumping related leakage of the Kalsi-brand rotary seals. The high ambient pressure is transmitted to the seal lubricant via the diaphragm. Many different diaphragm designs are possible, including convoluted designs.

The diaphragm shown in the schematic is an example of a very simple form of annular diaphragm[5] 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

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[5] For an example of an annular diaphragm in an oilfield downhole drilling tool, see expired U.S. Patent 4,462,469, "**Fluid motor and telemetry system**".

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 handbook.

One benefit of a diaphragm over a piston is that a diaphragm is relatively friction free, and extremely fast acting, compared to an axially sliding compensation piston. This means that there is ordinarily little or no differential pressure acting over the Kalsi Seals.

Because diaphragms will rupture if inadvertently exposed to high differential pressure, applications should be carefully evaluated to ensure that no unusual circumstances can occur that would expose the diaphragm to high differential pressure. Diaphragms can be configured to automatically vent the pressure that results from lubricant thermal expansion; see expired U.S. patent 4,727,942.

Because diaphragms expose significant area to the environment, the diaphragm material should be selected for compatibility with the environment.

Using a syringe to extract a known volume of lubricant

In equipment with diaphragm-based pressure compensation, or with an easily movable pressure compensation piston, a known volume of oil can be extracted with a syringe (Figure 6) to prevent thermal expansion related pressure locking.

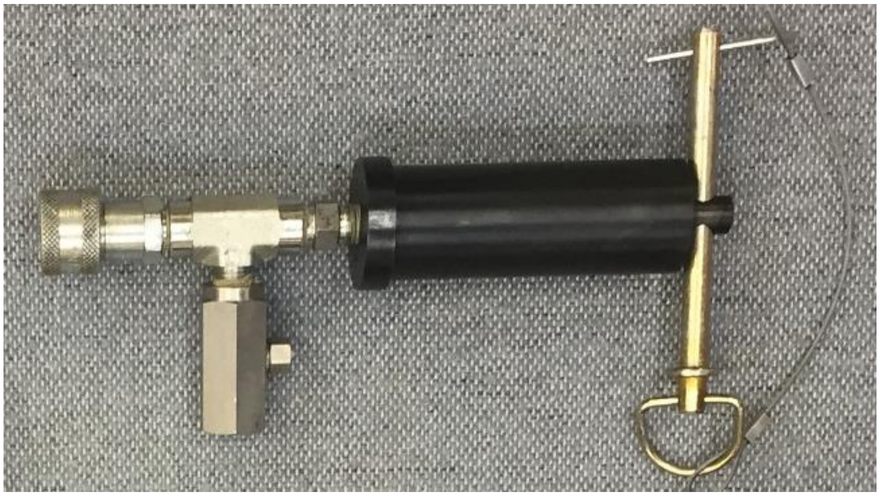

Figure 6

Syringe used to extract a known amount of oil from a lubricant reservoir

This syringe was designed to remove a known amount of oil from a piston-based lubricant reservoir to prevent thermal expansion related pressure locking. Because the quick connect provides a sealed connection with the equipment housing, atmospheric pressure moves the pressure compensation piston axially within the lubricant reservoir as the syringe extracts oil from the lubricant reservoir.

The oil extraction syringe can be custom designed to extract the correct volume of oil with a single stroke. During lubricant extraction, the syringe should have a sealed connection with the equipment housing. With a sealed connection, atmospheric pressure causes the pressure compensation piston or diaphragm to move as oil is extracted by the syringe. This prevents air from entering the equipment as oil is extracted by the syringe.

4. Preventing pressure locking of piston sliding seals

The piston in Figure 4 has a journal-bearing relationship with the rotary shaft, enabling it to follow any shaft runout and deflection that may be present. This arrangement minimizes compression changes to the rotary seal if the shaft moves laterally. For example, in mud motors, the shaft moves laterally due to drilling forces and nutation of the rotor of the Moineau[6] type positive displacement motor.

As shown in Figures 4 and 7, shaft guided compensation pistons employ more than one sliding sealing element between the piston and the surrounding housing. This provides friction to prevent the piston from spinning due to seal and bearing torque. In oilfield downhole tools, the piston also has to resist the rotational acceleration and deceleration of the drillstring.

Pressure locking (Figure 7) can occur between any pair of seals subjected to a high ambient pressure environment, including the sliding seals of a compensation piston. If the sliding seals become pressure locked, then friction increases significantly. More differential pressure is required to overcome the friction and make the piston slide, which exposes the piston-mounted Kalsi Seal to higher, potentially damaging, reversing differential pressures.

As shown in Figure 4, to prevent pressure locking in high ambient pressure environments, only the environment side sliding seal is permitted to establish a sealed relationship with the housing. The anti-rotation seal is prevented from sealing by providing pressure communication to the lubricant supply. In Figure 4, this communication is accomplished by a cross-drilled hole.

6 For those with historical interests, see Moineau's U.S. patent 2,028,407.

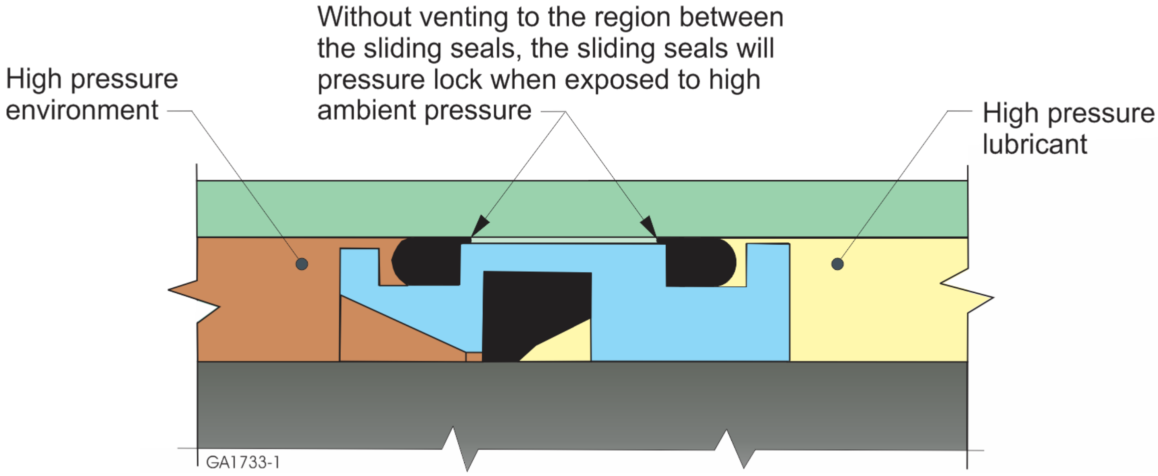

Figure 7

Pressure locking of piston sliding seals in high ambient pressure conditions

In high ambient pressure service (such as an oilfield mud motor), failure to vent to the region between the sliding seals will result in pressure locking due to trapped atmospheric pressure between the seals. This makes the piston difficult to move, which causes abnormally high differential pressure across the rotary seal.

5. To avoid pressure-locking barrier seals, see Chapter D10

The preceding material focuses primarily on the concept that the bulk lubricant volume of a downhole tool needs to be pressure compensated to the high-pressure ambient environment. As illustrated by Figure 7, the potential for pressure locking needs to be evaluated for every pair of seals that are exposed to high ambient pressure. As shown by Figure 8, barrier seal arrangements are not immune to pressure locking. For a detailed look at barrier seal implementation, see Chapter D10.

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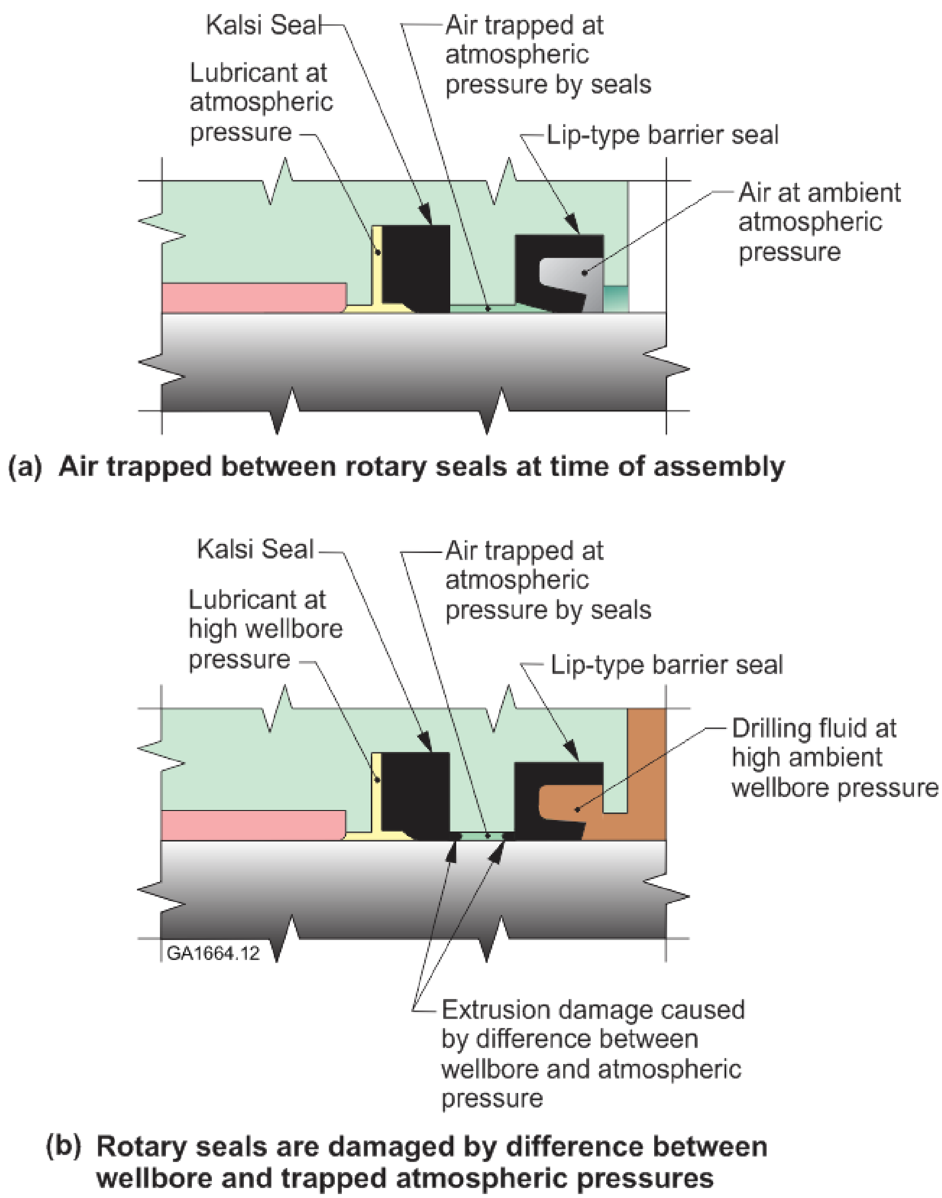

Figure 8

Barrier seals can also experience pressure locking-related damage

As this figure illustrates, a barrier seal can create a pressure locking situation by trapping atmospheric pressure at the time of assembly. For tips on avoiding pressure locking when using barrier seals, see Chapter D10.