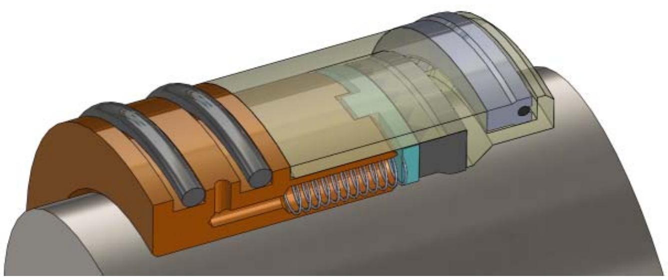
Chapter D9

Preventing skew induced wear of non-ACS seals



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Individual chapters of the Kalsi Seals Handbook™ are periodically updated. To determine if a newer revision of this chapter exists, please visit https://www.kalsi.com/seal-handbook/.

1. Introduction

In abrasive service applications with little or no differential pressure, or low level reversing pressures, all Kalsi-brand rotary seals except ACS[1] Seals, KLS[2] Seals, and packing require axial spring loading to inhibit skew induced wear. Initial radial compression, and thermal expansion during service, cause circumferential compression of the seal. In the absence of differential pressure or spring loading, this circumferential compression promotes local buckling, which skews a portion of the seal relative to the direction of rotation (Figure 1). Because of the skew, environmental abrasives are swept into the dynamic sealing interface, and the seal experiences premature abrasive wear (Figure 2).

Springs should be used in compensation pistons that do not use ACS Seals™

A mud motor pressure compensation piston is an example of an application where low levels of reversing pressure occur. The lubricant pressure is nominally the same as the drilling fluid pressure, but can be either higher or lower due to piston friction. The ACS Seal™ was developed for such applications, but is not available in all sizes and temperature ranges. Because skew induced wear can be a very aggressive wear mechanism, axial spring loading is highly recommended for all non-ACS Seals that are used in pressure compensation pistons for abrasive service applications.

2. What is skew induced wear?

Seal skewing is due to the combined effects of compression and thermal expansion (see Figure 1). The initial radial compression of the seal also imparts a degree of circumferential compression. Thermal expansion of the seal during operation imparts additional circumferential compression. In a manner similar to the buckling of long slender columns, the seal can become locally skewed as a result of the circumferential compression, in the absence of differential pressure. As little as 15 psi (103 kPa) of lubricant pressure will prevent such skewing at 162°F (72.2°C), but establishing such differential pressure is not practical in all applications.

If skewing is present, rotation can cause environmental abrasives to impinge upon and abrade the dynamic sealing lip, and consequently abrade the shaft. If not destroyed by other wear mechanisms, the seal wear pattern caused by skew is distinctive because it is non-parallel to the flat ends of the seal (Figure 2). The tell-tale skewed wear pattern is

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1 ACS is a trademark of Kalsi Engineering, Inc. and is an acronym for "Axially Constrained Seal™".

2 KLS Is a registered trademark of Kalsi Engineering, and is an acronym for "Kalsi Lip Seal".

usually destroyed by other wear mechanisms, because the skew induced wear relaxes some of the circumferential compression.

Seal skewing does not necessarily occur every time a non-axially constrained Kalsi Seal™ is used in a balanced pressure condition, but it will occur often enough to significantly reduce reliability and correspondingly raise equipment maintenance costs. Skew induced wear is a ***very*** aggressive wear mechanism when it does occur. In order to assure consistent performance with any type of rotary shaft seal that is exposed to abrasives, skew induced wear must be addressed.[3]

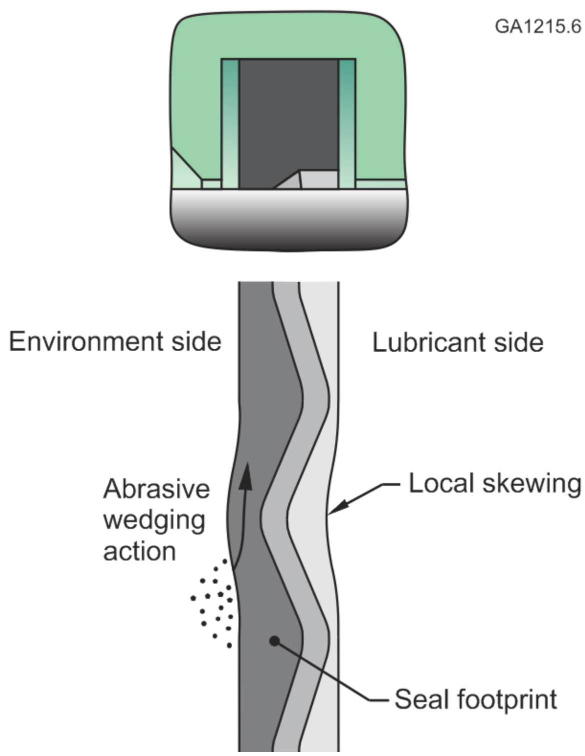

Figure 1

Skew induced wear mechanism

Non-axially constrained elastomeric rotary seals can skew in low differential pressure conditions due to compression and thermal expansion. Rotation causes abrasive fluid to impinge at the skewed location, which can drive abrasives under the seal and cause accelerated wear.

3 For example, in our experience, conventional lip-type axle bearing seals on commercial vehicles can easily have a five to one variation in life depending on how squarely they are seated against their respective shoulders.

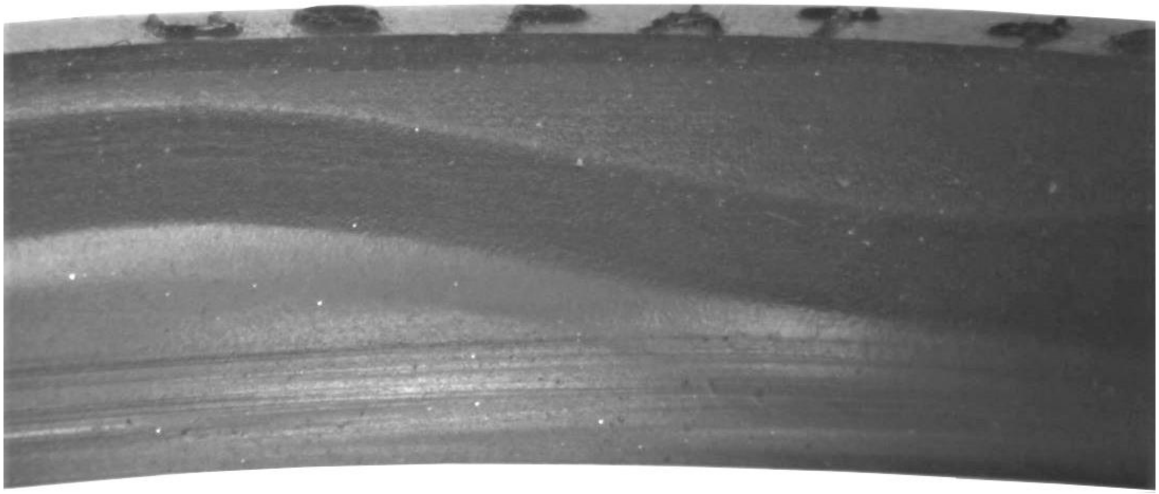

Figure 2

Skew induced wear example

The wear track on this rotary seal is not parallel to the circular exclusion edge, because a local portion of the seal was skewed during operation.

3. Using differential pressure to prevent skew induced wear

Our laboratory tests with 2.75” (69.85 mm) seals and low-pressure extrusion gap clearance indicates that differential pressure acting from the lubricant side of the seal can be used to prevent skew induced wear. A lubricant over-pressure of 15 psi (103 kPa) is enough to prevent skew induced wear at 162°F (72.2°C), and an over-pressure of at least 22 psi (152 kPa) is enough to prevent skew induced wear at 375°F (190.6°C). The differential pressure, acting from the lubricant side of the seal, prevents skew induced wear by holding the rotary seal flat against the environment side groove wall.

Our tests were typically performed with abrasive drilling fluid vented to atmospheric pressure. In some applications, the process fluid is pressurized, and the pressure fluctuates rapidly. For example, in oil well surface equipment applications such as rotating control devices, the drilling fluid pressure fluctuates rapidly as a result of mud pump design.

In such cases, the lubricant over-pressure has to be great enough to maintain a positive pressure differential even when the drilling fluid pressure fluctuations reach maximum values. Since adjustments to the lubricant pressure may lag behind increases in drilling fluid pressure, the lubricant over-pressure value needs to be set high enough to prevent lag related pressure reversals. (Pressure reversals also cause shuttling related seal wear.) Published literature[4] on rotary control device (RCD) sealing indicates that a 200 to 300 psi (1.38 to 2.07 MPa) lubricant over-pressure is adequate with computer-controlled pressurization systems that use a pinch valve to control the pressure of a circulating lubricant.

When using spring loaded piston amplifier type lubricant reservoirs in oil well surface equipment, a lubricant over-pressure in the range of about 160 to 240 psi (1.10 to 1.65 MPa) is enough to deal with mud pump related pressure spikes and piston related lubricant pressure lag. This is just the over-pressure value that was arbitrarily selected; no attempt was made to tease out minimum acceptable over-pressure values.

4. Using axial spring loading to prevent skew induced wear

In applications where typical elastomeric Kalsi Seals® are used with little or no differential pressure, or low levels of reversing pressure, axial spring loading should be used to prevent skew induced wear if an ACS is not available in the correct size and temperature range. Differential pressure conditions can be difficult to predict in some types of equipment. If you are unsure whether axial spring loading is needed, leave room in your design, just in case you find that it is required.

Barrier seals are also desirable in reversing pressure situations

In addition to spring loading, the use of an outboard barrier seal (Chapter D10) is also desirable in seal implementations that are subject to reversing pressure. By providing a clean lubricated environment, a barrier seal can help to prevent Kalsi Seal abrasion damage resulting from pressure induced seal distortion and axial shuttling motion within the groove.

In applications where the reversing pressure is due to slow pressure compensation piston response, the situation can be improved by substituting a fast responding diaphragm.

Various types of springs can be used

A typical spring loaded rotary seal implementation is shown in Figure 3. While the figure illustrates the use of wave springs to preload the seal, the use of other types of springs is possible. For example, expired US patents 1,089,789 and 3,015,505 teach and illustrate the use of a circle of coil type compression springs in pockets to axially preload rotary shaft seals. Figures 4 and 5 show how wave and coil springs can be implemented in pressure compensation pistons. As shown in Figures 3 to 5, a removable groove wall

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4 See US Patents 6,554,016, 6,749,172, 7,004,444, 7,007,913, 7,836,946, and 7,934,545.

permits installation of the seal, a backup washer, and the springs. The backup washer distributes the spring force uniformly around the circumference of the seal. The springs and backup washer are located on the clean lubricant side of the seal, so that they cannot become jammed from contaminants within the environment.

Limiting seal shuttling in reversing pressure conditions

In some applications, such as compensation pistons[5], the differential pressure acting from the abrasive environment side may occasionally exceed the lubricant pressure. In such transient reversing pressure situations, the seal may shuttle axially in the groove, and as a result may temporarily become skewed with respect to the direction of shaft rotation. Although undesirable, axial shuttling causes less harm with a spring loaded seal because any skewing that occurs involves cocking on the flat plane defined by the backup washer, instead of local seal buckling. Ideally, the compensation system should have minimal hysteresis, so it does not cause differential pressure acting from the environment side of the seal that is greater than the spring force applied to the seal.

Runout

The spring force recommendations provided in this chapter were tested with 0.010” (0.25 mm) runout on a 2.75” (69.85 mm) shaft with at 400 rpm using 0.335” (8.51 mm) cross section -11 HNBR seals and a 17-4 PH condition 1025 seal carrier. The radial extrusion gap clearance was 0.020” (0.~~,~~51mm). Scuffing of the environment side groove wall occurred due to runout related radial sliding motion of the inner portion of the seal relative to the groove wall. This scuffing was noted even with relatively low spring loads. This confirms that seal life and reliability increase as runout decreases.

A surface finish of 32 μin (0.20 μm) AA or less is recommended for the inner two thirds of the environment side groove wall to prevent anchoring of seal material, which can accelerate seal abrasion. The outer third of the groove wall can be rougher to inhibit circumferential slippage of the seal within the groove.

The cumulative effect of environment side groove wall abrasion can be minimized by defining the environment side groove wall with a replaceable ring. The replaceable groove wall should be a relatively tight fit with the groove bore to help prevent clearance variations at the extrusion gap, which may accelerate rotary seal abrasion. Replaceable environment side groove walls should be keyed or pinned to prevent rotation.

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5 In oil well downhole drilling equipment, reverse pressure transients can occur from things like compensation system hysteresis, swab pressure resulting from dropping the tool deeper in the hole, pressure drop that occurs along the length of the tool, or wellbore hole collapse. For a description of swab and surge pressure, see U.S. Patent 6,220,087. For information on wellbore hole collapse, see Bourgoyne, et. al., **Applied Drilling Engineering**,1991 and Bradley, et. al, **Petroleum Engineering Handbook**, 1992 (Society of Petroleum Engineers, Richardson, TX).

Another way to minimize environment side groove wall abrasion is to employ a shaft guided seal carrier that follows the dynamic motion of the shaft. This approach provides a number of benefits, including:

- Reducing the impact of rotary seal compression set by minimizing compression variations,
- Reducing shaft and rotary seal wear that occurs when abrasives trapped in the extrusion gap are crushed and displaced by shaft lateral motion, and
- Reducing nibbling type damage related to spring-induced seal extrusion (Such damage is more prevalent with FEPM dynamic sealing lips).

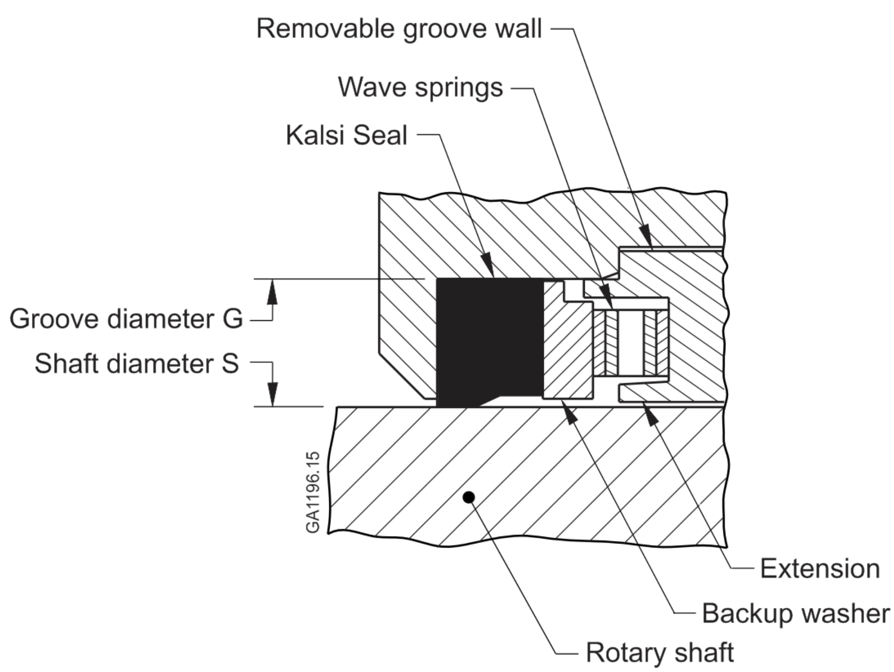

Figure 3

A wave spring and backup washer can be used to constrain Kalsi Seals perpendicular to the axis of shaft rotation to enhance abrasive exclusion in applications having little or no differential pressure. In this figure, the removable groove wall has extensions to limit shuttling of the rotary seal during transient reverse pressure conditions. The extensions can be built into either the backup washer or the removable groove wall. Although the removable groove wall is illustrated on the lubricant side in this figure, as an alternative, the environment side groove wall could be removable provided that it is well guided to maintain concentricity. Conventional retaining rings are a convenient way to secure removable groove walls.

Ideally, a spring loaded seal implementation will limit the magnitude of any axial shuttling motion to only the amount of clearance that is needed to accommodate the maximum anticipated thermally expanded width of the seal. This anticipated width should include some margin for error in temperature estimate. In Figure 5, the shuttling motion is limited by the amount of clearance between the backup ring and the removable groove wall that is defined by the journal bearing portion of the seal carrier. In Figures 3 and 4, the shuttling motion is limited by over travel extensions. If the anticipated transient reverse pressures are quite low, only one such extension is needed, preferably at the OD. If transient reverse pressures can potentially be high, the use of two extensions would be preferred (as shown in Figure 3) to prevent excessive pressure induced deformation of the backup washer and seal cross sections.

The over-travel extensions can be designed to project from the backup washer or from the removable groove wall, and should be designed so that the springs cannot hang up on them. In Figure 3, this has been accomplished by tapering the inner extension, and by causing the outer extension to telescope over a portion of the backup washer.

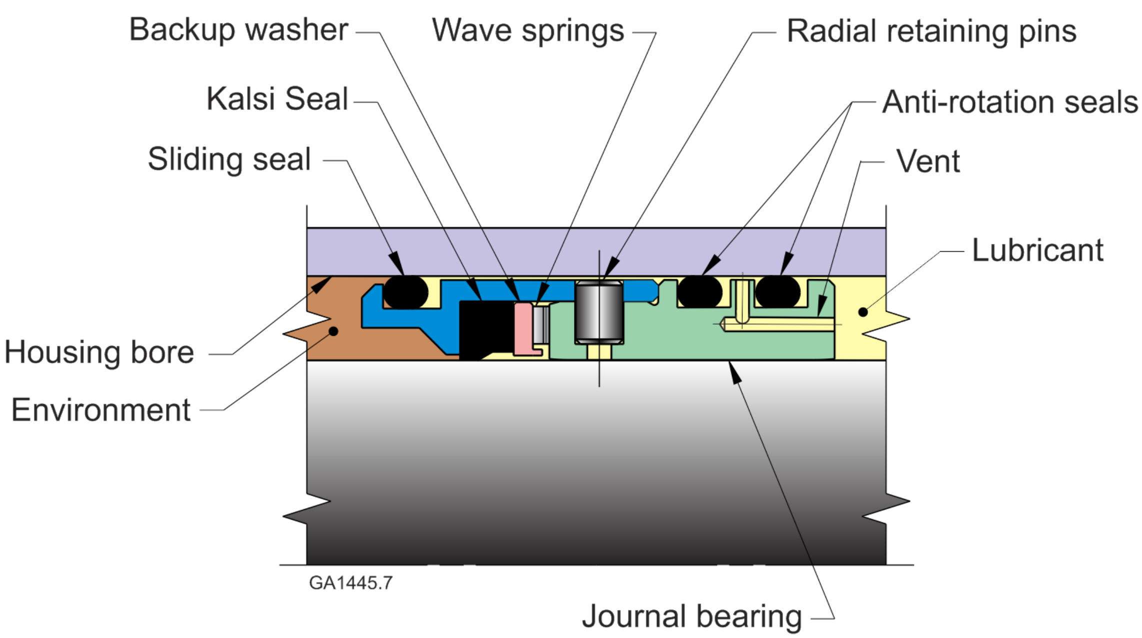

Figure 4

A mud motor compensation piston with a wave spring loaded Kalsi Seal

In this oilfield mud motor pressure compensation piston, the Kalsi Seal is axially preloaded with a wave spring to prevent skew induced wear. Three radial retaining pins in match-reamed holes hold the assembly together. The pins are retained by the housing bore. In Kalsi Engineering's seal research mud motor, this type of compensation piston design allowed the Kalsi Seal to perform much better as a mud seal.

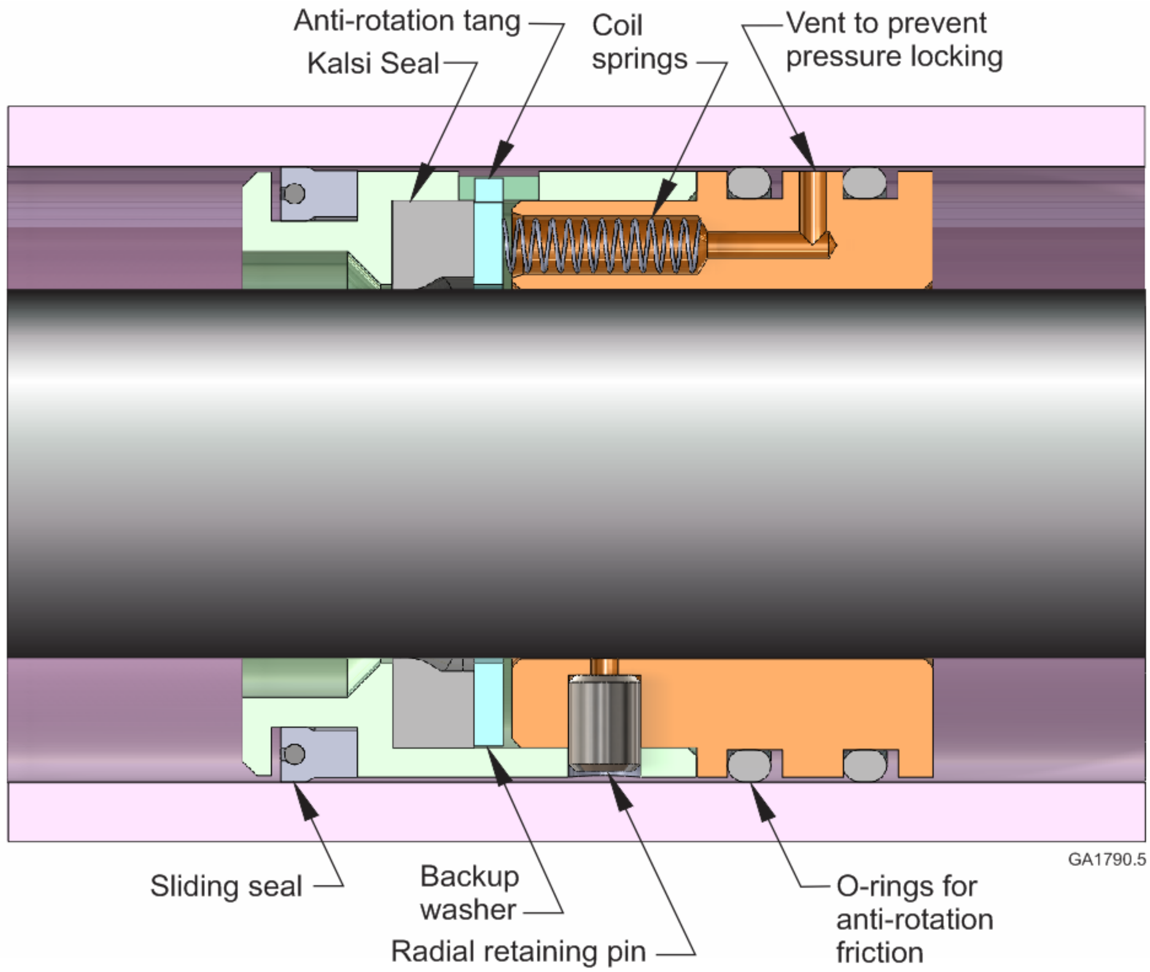

Figure 5

A mud motor compensation piston with a coil spring loaded Kalsi Seal

In this oilfield mud motor pressure compensation piston, the Kalsi 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 with an anti-rotation tang to prevent rotation that could bind the projecting ends of the springs. If desired, the anti-rotation tang could be axially oriented instead of radially oriented. The surface of the backup washer that contacts the Kalsi Seal can be grit blasted to inhibit circumferential slippage of the Kalsi Seal. Designs that incorporate fewer than 3 coil springs should be avoided because they promote tipping-induced skew of the rotary seal in reverse pressure conditions. In larger diameters, this piston arrangement becomes more challenging to implement, because it becomes more difficult to obtain a tight piloting fit between the seal carrier components.

How much spring force?

The amount of applied spring force varies with the width of the seal, which varies due to tolerances and thermal expansion. Estimated seal widths in various temperature and tolerance conditions are provided in Appendix 2. If the Kalsi Seal you intend to use is not represented in Appendix 2, contact Kalsi Engineering for assistance.

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To make our spring force recommendations applicable to a wide variety of seal sizes, we provide the recommendations in pounds per square inch (psi) of equivalent pressure, rather than in pounds. To calculate equivalent pressure, Ep in pounds per square inch, divide the spring force Sf in pounds by the circular area of the seal gland, use the following equation:

Equation 1, Equivalent pressure:

$$Ep = Sf / ((G/2)^2 xPI - (S/2)^2 xPI)$$

See Figure 3 for the variables G and S. Our general-purpose spring force recommendation for 80 to 90 durometer Shore A[6], non-dual durometer[7] seals is:

- At least 15 psi 103 kPa) equivalent pressure at the 162°F (72.2°C) seal width.
- At least 22 psi (152 kPa) equivalent pressure at the 375°F (190.6°C) seal width.
- No more than 45 psi (310 kPa) equivalent pressure at the MMC seal width at the maximum anticipated seal temperature[8]. (80 and 90 durometer seals with Enhanced Lubrication™ waves have been successfully tested at 300°F (148.9°C) with 45 psi (310 kPa) equivalent pressure[9], but they have not been tested at higher temperatures with spring loading.)

For maximum conservatism, apply recommendations in the first two bullet items at LMC seal dimensions and a lower bound estimate of thermal expansion, and apply the recommendation in the third bullet item at the upper bound estimate of thermal expansion[10]. This degree of conservatism is not always possible with high temperature capable seals, because it severely narrows the spring choice and may preclude the use of off the shelf coil springs. One can certainly rationalize the use of nominal seal dimensions instead of LMC dimensions in regard to the first two bullet items, because the mold design and the molding process makes LMC seal dimensions unlikely.

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6 While spring loading is believed to be practical with softer seal materials in some wave patterns, test data is not available. Spring force will probably have to be reduced in proportion to the reduced modulus of the softer seal material.

7 Few Dual Durometer Seals™ have been tested in spring loaded conditions.

8 Composite High Temperature Kalsi Seals™ have been successfully tested at 390°F (198.9°C) against drilling fluid with a combination of 45 psi (310 kPa) equivalent spring load and 45 psi lubricant pressure, assuming a 13E-5 in/in/°F coefficient of thermal expansion.

9 The 300°F (148.9°C) spring force calculations were based on MMC seal dimensions and a 13E-5 in/in/°F coefficient of thermal expansion.

10 We use 13E-5 in/in/°F as an estimate for the upper bound of linear thermal expansion for HNBR~~, and FEPM~~ seals.

The spring rate target can be calculated from the difference between the seal widths referenced by the first two bullet items, the difference in the targeted loads at those two widths, and the number of springs.

Understanding causes of seal width variation

The spring(s) must be designed to accommodate seal width variations while maintaining a useful level of skew resisting force. Seal width variations are related to a number of variables, including seal diameter[11], temperature, tolerances, mounting clearances, and shaft deflection. Temperature is often the most significant variable, because the coefficient of thermal expansion of elastomers is much higher than metals.

Typical published coefficients of linear thermal expansion for various elastomers are shown in Table 1. Volumetric thermal expansion is three times greater than linear thermal expansion. We typically use a linear coefficient range of thermal expansion of 13E-5 in/in/°F when calculating HNBR seal width for spring loading purposes.

Table 1
Typically quoted values for
linear thermal expansion of elastomers

Elastomer	Linear expansion, in/in/°F x 10^{-5}
NBR & HNBR	6.2 to 13
EPDM	8.9
FKM	8.3 to 15.0

The rotary seal is hotter than the environment during rotation because of interfacial lubricant shear. If the environment circulates through the shaft to draw heat away, the seal can be conservatively estimated to be about 50°F (27.8°C) hotter than the environment. For more complex installations with less efficient heat transfer, thermal finite element analysis can be used to predict seal temperature by using estimated rotary seal torque to determine the seal generated heat input for the analysis.

11 Seal diameter influences circumferential compression, which affects installed axial width of the seal. Circumferential compression is greater with smaller diameter seals, so the installed width of smaller diameter seals is greater than the installed width of larger diameter seals having the same cross sectional dimensions; see Appendix 2.

Avoid over confinement of the seal by the spring washer

The most serious spring implementation mistake is seal over confinement caused by failure to anticipate the maximum thermally expanded width of the rotary seal. This over confinement can cause increased interfacial contact pressure, and can cause the seal body to flatten against the shaft (Figure 6), impeding hydrodynamic lubrication, causing increased frictional torque and heat, and potentially causing premature seal failure. When severe over confinement occurs, the typical tell-tale signature is wear across the entire width of the seal (i.e. wear on surfaces that should not be contacting the shaft[12]) and/or extrusion damage. The extrusion damage may appear at one or both ends of the seal[13].

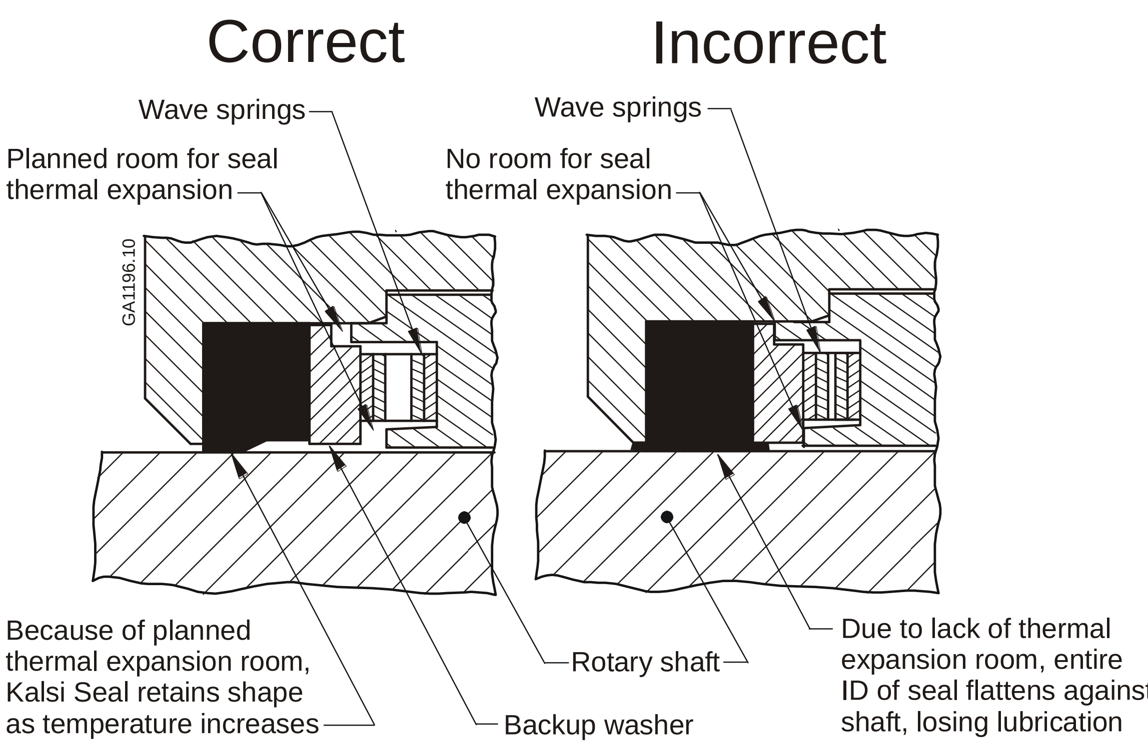

Figure 6

If the spring and backup washer arrangement is not designed to accommodate the thermal expansion and tolerances of the rotary seal, the thermal expansion will cause the seal to flatten against the shaft and lose lubrication.

Excessive spring force can over flatten and damage the seal

A related mistake is to apply too much spring force. The combined effects of high axial spring force and seal thermal expansion cause the seal body to expand toward the

12 Such a wear signature may also indicate high reverse pressure acting across the rotary seal.

13 Extrusion damage on the lubricant end of the seal may also indicate high reverse pressure acting across the seal.

shaft. Excessive spring force, especially in combination with elevated temperature, can cause the seal body and the hydrodynamic edge of the dynamic sealing lip to flatten completely against the shaft. This also can impede hydrodynamic lubrication, and cause premature seal failure[14].

Because axial spring load causes radial expansion of the seal body, it also increases effective installed radial interference. If desired, credit for this radial expansion can be taken when designing the groove diameter G.

For best results, minimize reversing pressures

Spring loading improves abrasive wear resistance of the seal in applications with low transient reverse pressure conditions. This is true even if the spring induced equivalent pressure is less than the reverse pressure transient, although the rotary seal does shuttle axially. Such pressure responsive shuttling is undesirable because environmental media can pack between the seal and the environment side groove wall and tilt the seal, and because the seal may traverse worn or contaminated portions of the shaft. For best performance, minimize the reverse pressure potential by providing rapid pressure balancing of the lubricant to the environment.

5. Backup washer design

The backup washer should be guided by a non-binding slip fit with the groove bore G_{min}, and typically it should have enough clearance with the shaft to prevent contact under any combination of tolerance, backup washer misalignment, and shaft deflection.

Worst case backup washer to shaft clearance can be determined by subtracting the following from the minimum concentric radial clearance:

- The maximum concentric groove diameter G_{max} to washer radial clearance,
- The maximum radial manufacturing eccentricity of the backup washer,
- The maximum shaft to seal carrier lateral offset due to mounting clearances, and
- The maximum lateral shaft deflection from the highest anticipated side load.

When designing the axial thickness of the backup washer, the spring force, the number of coil springs or wave spring waves, and the magnitude of reverse pressure must be considered so that the backup washer does not deflect significantly between spring

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14 In a rotary test of -8 FEPM seals with springs calculated to provide about 81.5 psi (562 kPa) equivalent pressure at 350°F (176.7°C), the seal bodies were completely flattened against the shaft at a bulk lubricant temperature of 375°F (190.6°C), but were not completely flattened against the shaft at a bulk lubricant temperature of 160°F (71.1°C).

contact points (Figure 7). Flimsy backup washers are unsuitable because they deflect significantly under reverse pressure, which imparts a wavy shape to the exclusion edge of the seal that promotes seal abrasion. The backup washer should be a continuous ring to maximize rigidity. We used a 0.1" (2.54 mm) thick backup washer for our tests. This backup washer thickness had a predicted deflection of 0.0027" (0.07 mm) with 61 psi (421 kPa) reverse pressure using the load case shown in Figure 7, and 0.0026" (0.04 mm) using FEA. Less deflection is better than more deflection, in terms of abrasive exclusion.

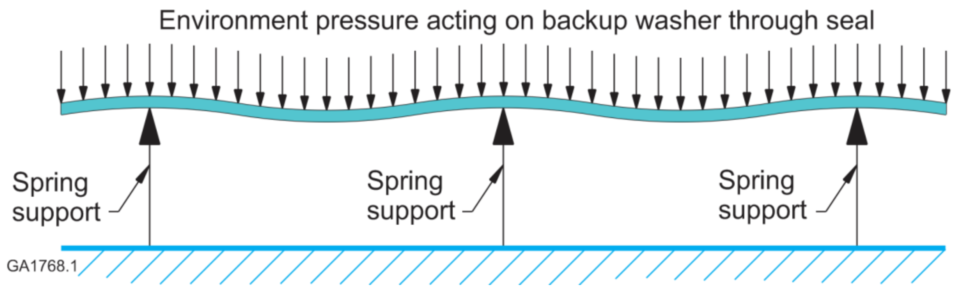

Figure 7

To prevent skew induced seal wear, the thickness of the backup washer must be sized so that very little deflection occurs when the seal is exposed to reverse pressure acting from the environment side. A quick and dirty way to calculate is to use the closed form solution for a continuous beam that has two equal spans and uniform loading. One source for such a formula is **Machinery's Handbook**.

If one or more over travel projections are used, they should be sized to allow travel up to the desired maximum groove width.

Designing the backup washer to inhibit seal slippage

Spring loading helps to inhibit circumferential seal slippage by increasing the frictional force between the seal and the environment side groove wall. If desired, the backup washer can be keyed to the housing (Figures 8, 9 and 10) and the backup washer surface facing the seal can be roughened to further inhibit circumferential seal slippage. One appropriate surface roughening method is the grit blasting process described in chapter D5, provided that the cylindrical surfaces of the backup washer are masked.

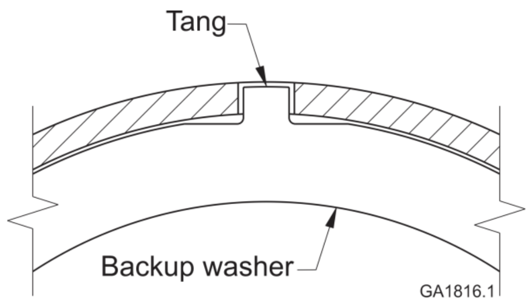

Figure 8

To inhibit circumferential seal slippage, the face of the backup washer can be roughened, and the outer periphery of the backup washer can form an anti-rotation tang that engages a longitudinal slot in the seal carrier.

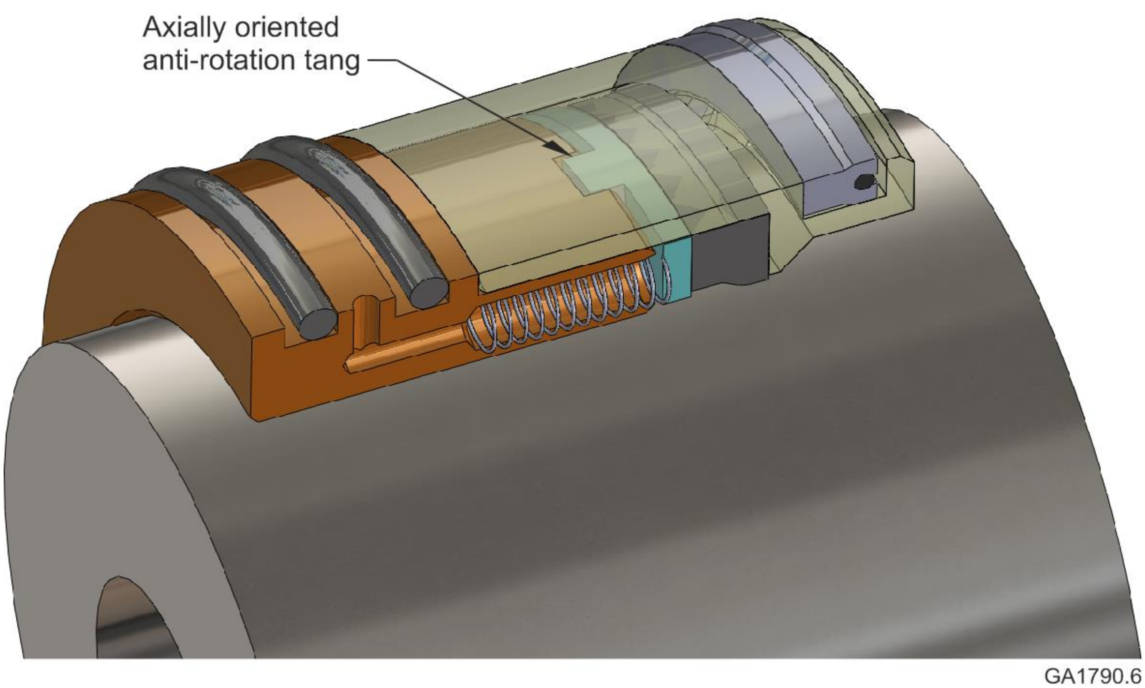

Figure 9

In this cutaway view of a pressure compensation piston, the spring loaded backup washer incorporates an axially oriented anti-rotation tang. The surface of the backup washer that contacts the Kalsi Seal can be grit blasted (See Chapter D5) to inhibit circumferential slippage of the seal.

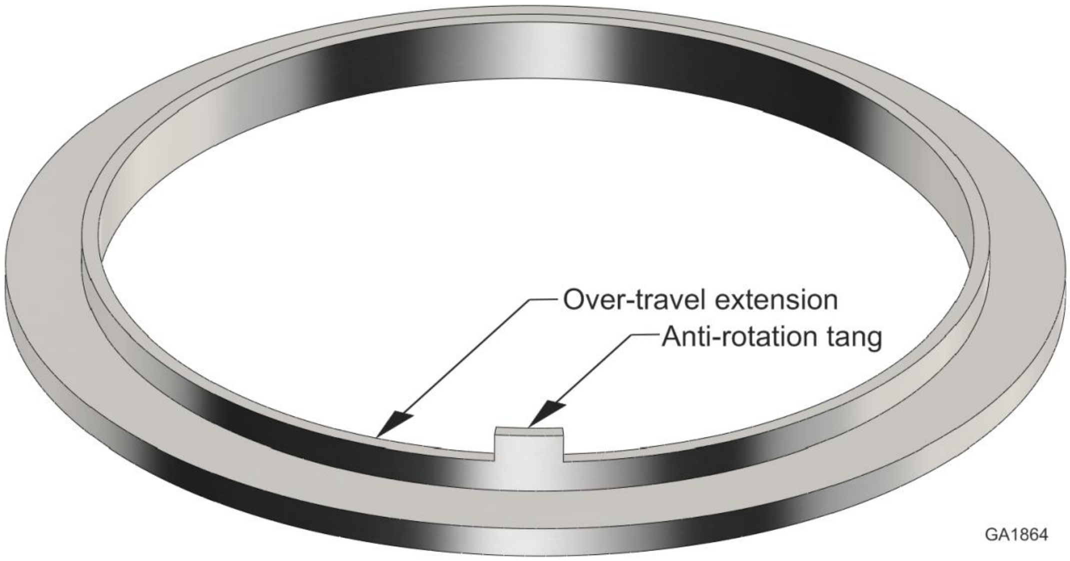

Figure 10

An over-travel extension is sometimes used with backup washers that are used with wave springs. As this figure illustrates, the anti-rotation tang can project from the over-travel extension.

6. Wave spring design

Kalsi Engineering does not manufacture or sell springs of any type, but provides free coil spring design software on our website. Distributors of off-the-shelf springs usually provide extensive spring performance data and material selection guidance in their literature. Custom spring manufacturers typically have staff experts who can provide comprehensive spring design services, often at no cost. Be sure to select a spring material that is compatible with anticipated operating temperature.

If you are designing your own wave springs, software is recommended because of the number of design iterations that are typically required for optimization. Such software is offered by some wave spring manufacturers as a courtesy. Spring material properties are available from a variety of sources, including the Spring Manufacturer’s Institute, Wheeling, Illinois. Spring design formulas and guidelines are also available from a variety of sources.

Wave spring load predictions are considered to be relatively accurate up to 75 to 80% of full deflection if the mean diameter to radial width aspect ratio is greater than 8. Wave springs produce greater force during compression, and less force during unloading, due to friction from movement against mating surfaces (including other wave springs).

Design the spring outer diameter (OD) to be bore guided by a non-binding slip fit with groove diameter G_{min}. Wave spring diameter increases with spring compression, and the spring will cease to function if it is constrained from expanding by the groove. If the spring is constrained from expanding by the groove, it will not allow the seal to thermally expand freely, and may result in rapid seal destruction. This is a critical consideration, especially with non-gap type wave springs.

When designing the spring fit within the groove diameter G_{min}, if the spring approaches full compression in your design (which is not recommended), the maximum spring OD can simply be assumed to be the MMC[15] OD of the spring blank prior to forming the waves. If the spring does not approach full compression in your design, the maximum effective spring diameter should be calculated based on maximum spring compression. The minimum spring OD can be calculated based upon predicted minimum spring compression, or can be simply assumed to be the LMC OD of the spring in the relaxed state. The spring blank diameter increases as a function of the number and height of waves. By increasing the number of waves, the load capacity goes up for a given material thickness, but the deflection range goes down.

If an over travel projection is used on the backup washer, the spring should have a non-binding slip fit with it. If an over travel projection is not used, the spring ID should have enough clearance with the shaft to prevent contact under any combination of tolerance, spring lateral offset, and shaft deflection. When designing the fit with the shaft, the minimum spring ID can be calculated based upon predicted minimum spring compression, or can simply be assumed to be the MMC ID of the spring in the relaxed state. Worst case shaft to spring clearance can be determined by subtracting the following from the minimum concentric radial ID clearance between the spring and the shaft:

- The maximum concentric groove bore to spring OD radial clearance,
- The maximum spring OD to ID radial manufacturing eccentricity,
- The maximum shaft to seal carrier lateral offset due to mounting clearances, and
- The maximum lateral shaft deflection from the highest anticipated side load.

At deflections above 75 to 80%, the actual spring load progressively increases above the load predicted by calculation. The preferred way to implement wave springs is to compress them by less than 75 to 80% at the maximum predicted rotary seal width, and use an over travel limiter to prevent excess seal shuttling[16] (see Figures 3 and 4). Because of the deflection limiter, a spring can be used which has relatively little force change over

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15 MMC stands for “Maximum Material Condition”, and LMC stands for “Least Material Condition”; for more information see ASME Y14.5M – 1994: **Dimensioning and Tolerancing**.

16 Another possible way to implement wave springs is to design them to be compressed by approximately 75 to 80% at the maximum predicted rotary seal width. The remaining deflection serves as a buffer to prevent over-confinement if temperatures go higher than anticipated.

the required stroke. For example, the spring in one of our tests had a calculated spring force of about 27.7 psi (191 kPa) equivalent pressure at the room temperature seal width, and 32.0 psi (221 kPa) at the predicted 350°F (176.7°C) seal width.

A surface finish of 32 μin (0.80 μm) AA or less is recommended for the mating load bearing surfaces of the spring and backup washer to minimize sliding friction. A spring load tolerance of ±20 to 25% is generally achievable, but should not be specified at deflections greater than 75% or less than 25%. The spring should be lubricated during load testing to minimize friction.

Spring designs that incorporate fewer than 3 waves should be avoided because they promote tipping-induced rotary seal skew in reverse pressure conditions. Spring wave height should be as uniform as possible in order to minimize reverse pressure tilting. For the sake of uniformity, continuous circle type wave springs must be formed using dedicated dies, rather than by hand (i.e. bending over bars).

For small diameter seals, a relatively short spring wave pitch and thin spring stock must be used, and it may be necessary to use several springs in parallel to achieve the required load and deflection capability.

Kalsi Engineering wave spring experience has been limited to 17-7 PH condition CH 900 precipitation hardening stainless steel. This material was selected on the basis of its elevated temperature capacity and strength. It is typically available in (but not limited to) thicknesses of 0.005”, 0.008”, 0.010”, 0.012”, 0.025”, 0.031”, 0.035” and 0.042” (0.13, 0.20, 0.25, 0.30, 0.64, 0.79, 0.89 and 1.07 mm) and has a minimum ultimate tensile strength of 240,000 psi (1,655 MPa) and a minimum 0.2% yield strength of 230,000 psi (1,586 MPa). As a general rule, the allowable stress for static and low cycle wave spring loading can be regarded as 100% of tensile strength at 75% deflection (80% of tensile strength for austenitic steels and non-ferrous alloys) provided that pre-setting is used during manufacture to create favorable residual stresses[17]. Consult your spring manufacturer for material and stress recommendations for specific service conditions and manufacturing procedures.

Stagger gaps if you must use gap type wave springs

Avoid gap type wave springs if possible because they are much more flexible at the location of the gap. Under transient reverse pressure conditions, the rotary seal will tilt toward the gap, causing skew induced seal wear. If gap type wave springs cannot be

[17] Adapted from published literature, and a discussion with a local wave spring manufacturer regarding 300°F (148.9°C) service.

avoided, two springs should be used in parallel with the gaps oriented 180° apart (Figure 11) to minimize seal tipping in reverse pressure conditions.

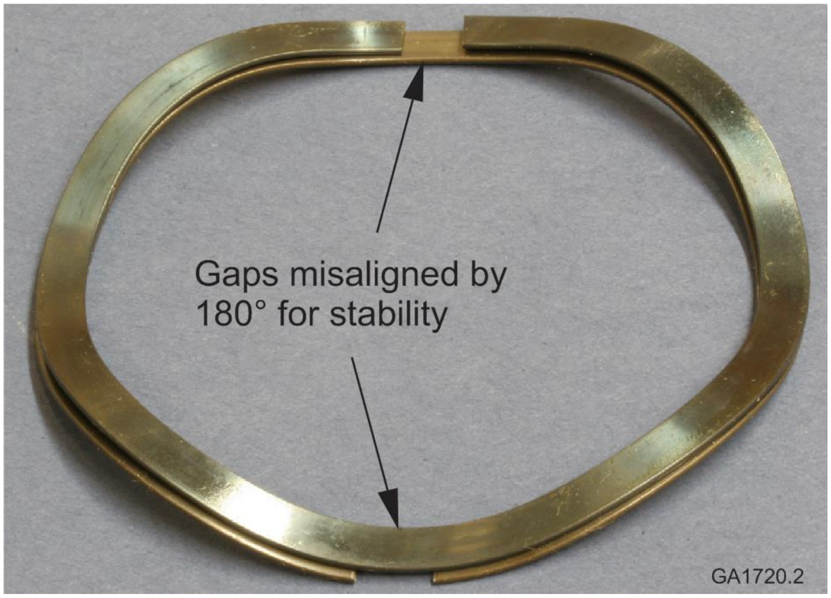

Figure 11

If gap type wave springs must be used to axially load a Kalsi Seal, two springs should be used in parallel with the gaps oriented 180° apart (as shown here). This helps to minimize seal tipping in conditions where the environment pressure is temporarily higher than the lubricant pressure.