FIRST OPERATING EXPERIENCES WITH THE INTEGRATION OF SPLIT SEALING CONSTRAINTS TO A GAS SEAL CHALLENGE

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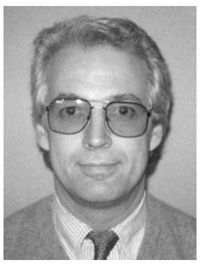

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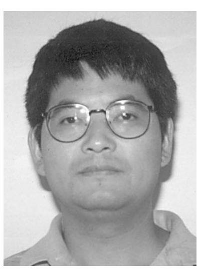

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ABSTRACT

Split mechanical sealing technology has lagged nonsplit seal applications because of the many difficulties introduced by the requirement of assembling separate parts to a high degree of precision in the field. It has at first been applied to equipment that simply cannot be practically disassembled such as stern tubes in boats. Then the migration was made to packing replacement for larger pumps handling mostly water. But recently, many complex chemical, pharmaceutical, and other applications have been successfully addressed with split seals. This paper deals with the design and field application of a completely split mechanical gas lubricated double seal in a challenging situation by overcoming various new hurdles.

INTRODUCTION

The science of mechanical sealing involves the prevention of fluid movement from one area to another. This goal must be accomplished in an environment in which high pressures and high rotating speeds exist and in which rubbing contact between mating seal faces must occur. The survivability of the material is a necessary element, as the containment of fluid under pressure is the primary goal. Originally, packing, woven fibers impregnated with lubricant, were compressed against the shaft in a narrow cylindrical chamber known as the stuffing box. This provided an effective fluid containment but with limited leakage control. The packing can be assembled over the shaft and replaced without disturbing the equipment. For all its practicality this method also has its limitations. It is somewhat maintenance intensive as the proper compression of the packing must be maintained as it wears, its life is limited as there can be significant damage on shaft sleeves, and of course, controlled leakage is no longer acceptable in many applications.

Mechanical seal technology was developed to effectively tackle the problems of high maintenance, wear, and leakage. One major drawback is the requirement that the equipment needs to be disassembled in order to slide the seal over the end of the shaft and onto its proper axial location. On most small pieces of equipment this is not much of a concern, but on large pieces of equipment weighing several tons quite a few compromises of the type "live with the leakage versus rent a crane" have to be made.

One obvious solution to this dilemma was the use of split mechanical seals—obvious to the user, but not so obvious to the designer. Compared to standard seal technology, split mechanical seals multiply the number of possible leak paths. The first hurdle to overcome is to bring two seal ring segments back into alignment to the exacting tolerances required for sealing, in the order of a fraction of a micron. This must be done reliably and without elaborate setups to be successful in the field. The next problem is how to seal the metal halves in the split plane. This becomes particularly difficult at the junction of other sealing elements in the axial direction, such as O-rings.

Many elaborate alignment mechanisms have been devised involving clamps, lapped support surfaces, precision alignment pins, cones of many varieties, and quite a few other sophisticated arrangements. Although many versions were successful for their intended purpose of aligning face segments, most suffered from lack of practicality due to size, complexity, and high manufacturing costs. One method proved to be practical, reliable, and cost effective—that of relying on a resilient axial support for the split face segments. Each segment is evenly supported with uniform mechanical and pressure forces, pushing the faces toward each other, thereby relying on the lapped seal faces to ensure the alignment of the segments. This has been used in a number of designs and has been instrumental in increasing the use of split mechanical seals. However, one limitation of this design approach is that the device is limited in its ability to handle pressure reversal.

In many single mechanical seal applications this is not a critical limitation as pressure reversals are infrequent. When double seals are used, the barrier pressure is set with respect to process pressure and only a process upset would change the relationship. In a great number of cases this is acceptable. There are, however, quite a few single seal applications, such as in reactors, or pumps with negative suction pressure (for example vertical pumps) where this limitation is severe. The problem can be quite restrictive in the domain of double sealing as well. There are cases where process upsets will occur, and the seal has to be able to operate under the adverse pressure conditions due to the critical nature of the application.

A GEOMETRIC PUZZLE

When the pressure is acting on the majority of the outside surface of a seal face made of split segments, it will hold the segments together. When no mechanical clamping is used to avoid distortion and cost problems, this proves to be quite a beneficial arrangement. The segments can be free to align themselves on the sealing plane, and the higher the pressure, the more they are forced together thereby creating an effective seal at the splits.

When the pressure is internal, a point is quickly reached where the resilient members (typically elastomeric members) can no longer hold the halves together. The segments separate, and the seal leaks. Even with clamping arrangements there are typically some significant limitations to the ability to handle internal pressure. Metal clamps used in order to induce the necessary compressive stress required (to preload the rings to counteract the tensile hoop stress generated by internal pressure) can easily damage the faces as the materials are quite brittle. This does not even take into account the face deflection induced from differential thermal expansion.

A number of designs have been used to place the pressure on the outside of the seal faces. Figures 1, 2, and Figure 3 (Sandgren, 1994) represent fairly typical examples for a single split seal.

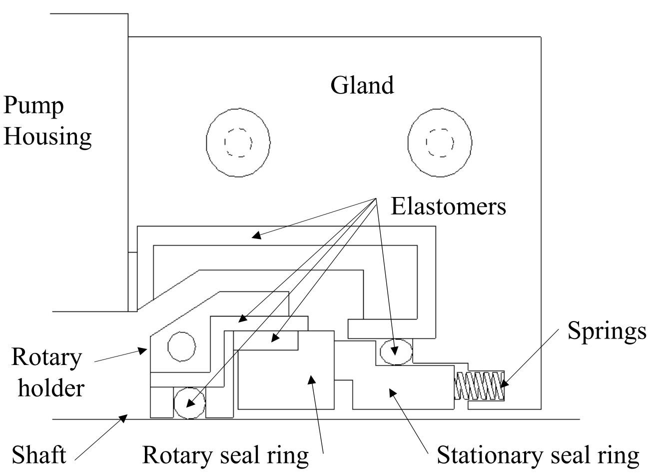

Figure 1. Generic Split Mechanical Seal.

Figure 4 depicts a good example of an arrangement with two seals placing the barrier fluid pressure on the outside of the seal faces, with the process on the inside of the inboard seal. It is essentially two single seals in a back-to-back configuration.

Such a design would experience immediate failure if the barrier fluid pressure were lost. Process would fill the cavity between the two seals and one would have to rely on the outboard seal for backup. Furthermore, the process is on the inside of the inboard faces, and that configuration can be quite a detriment when solids are present in the process.

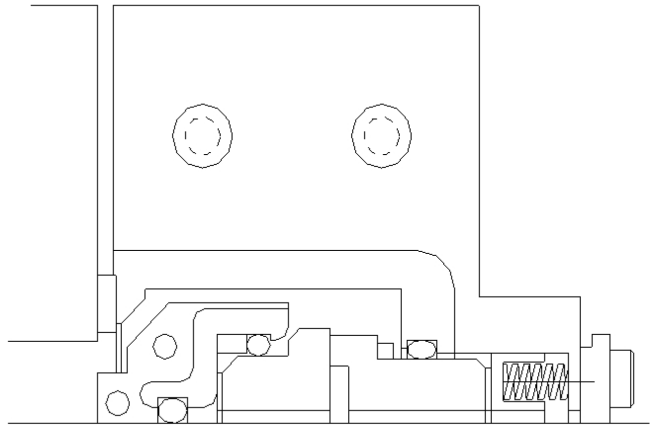

Figure 2. Single Split Seal Outside Pressure (A).

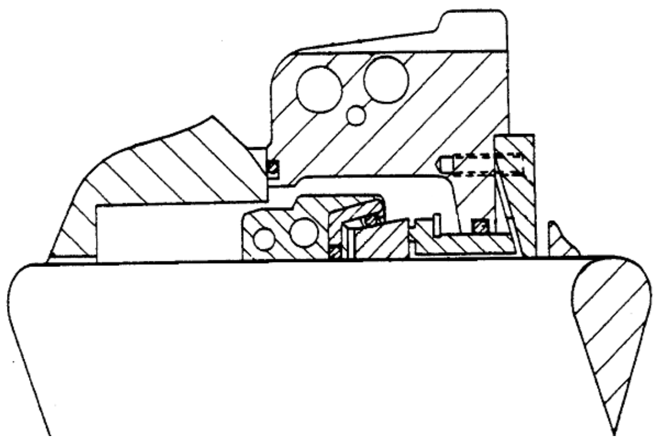

Figure 3. Single Split Seal Outside Pressure (B).

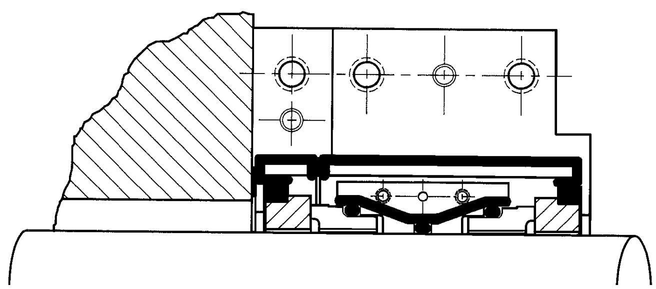

Figure 4. Back-to-Back Dual Split Seal.

So designers have been quite innovative in their arrangements. It appears that the scientific literature is sparse on this topic; however, the patent literature is rich in examples of numerous inventions. Figure 5 depicts a very interesting example of juggling geometry to have product on the outside of the seal faces and, at the same time, being able to maintain a barrier fluid pressure higher than the process pressure (Reagan, 1998). Furthermore, by judicious placement of the sealing point between process and barrier in line radially with the seal face, pressure forces tending to separate the seal face segments are kept to an absolute minimum. This is the case whether the difference between process and barrier pressures is positive or negative. Other than some minor packaging issues of requiring a substantial amount of radial space, it would appear that the problem of adapting split seal technology to pressure reversal had been conquered. However, a careful examination would reveal other perilous deficiencies.

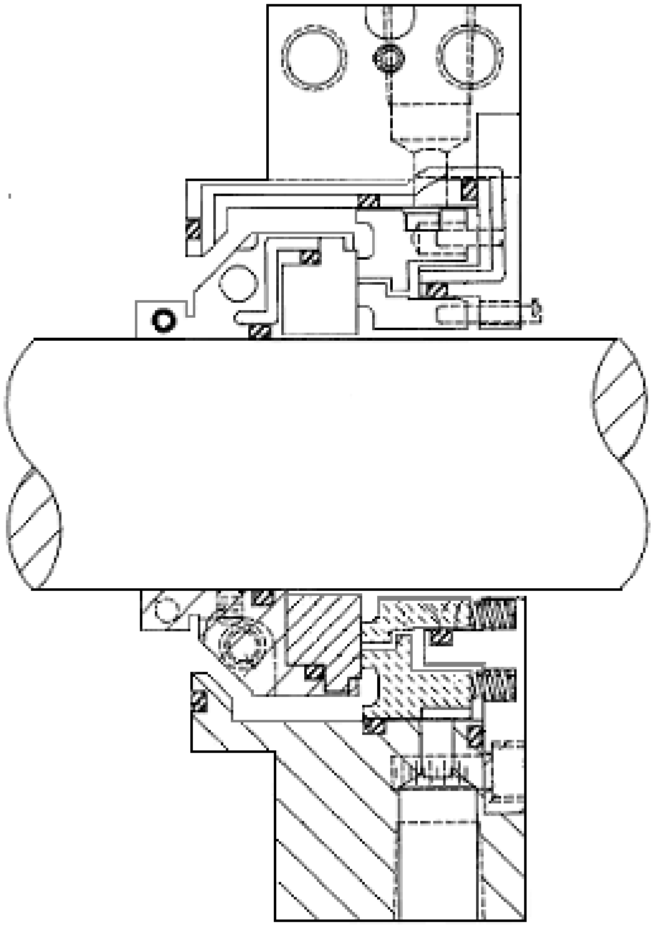

Figure 5. Concentric Dual Split Seal.

Although the problems of maintaining the physical integrity and the seal interface flatness of the seal face segments might be solved in the radial direction, the same could hardly be said in the axial direction. And this is yet another example of the requirements for splitting the seal being contrary to the requirements of an advanced balanced seal design. Indeed, in the axial direction, seal designers have, for quite some time, conquered the geometrical constraints to balance the axial pressure load and thereby to balance the seal's opening and closing forces. In the example above, the only contribution to the closing force on the inboard (outer) seal from the process pressure side is from the spring. As soon as the pressure acting from the seal face to the outer O-ring sealing diameter overcomes the spring force, the seal will open. The seal face segments are kept in contact but the seal faces are axially separated. Conversely the barrier pressure acts over an area that is substantially larger than the seal face area, creating closing forces several times greater than the opening forces, thereby limiting the differential pressure capabilities of the inboard seal. In a typical dual seal application the differential between process and barrier pressure is small, but this arrangement would not be tolerant of either barrier of process pressure loss.

THE PUZZLE

When it works as a split seal, it may not work as a standard seal, and when it works as a nonsplit seal, for example, the concentric coplanar double gas seal presented by Wu, et al. (1999), it may not work as a split seal.

Following is the requirement for the capability of handling pressure reversals: in a design where the faces are resiliently supported to maintain face flatness, find a geometry that will have pressure maintaining the integrity of the face segments radially, and, at the same time, achieve control over the axial loading of the faces, regardless of the levels of difference between process and barrier pressure.

Figure 6 shows a geometry addressing these requirements (Azibert, et al., 2000a, c, d). There are some significant differences in this design from the previous ones we have looked at. In Figure 4, there are two sealing interfaces with four seal rings used. In Figure 5, there are two sealing interfaces with three seal rings used. This geometry goes one step further in that the sealing interfaces are done on common seal rings. There are thus inboard and outboard seal faces, but only two seal rings are used. It is similar to the single seal designs shown in Figures 1 through 3, in that there is a gland, a rotary metal holder, assorted elastomeric seals and hardware, a rotary seal ring, and a stationary seal ring. But with passages within the seal rings, four seal faces are created with a barrier fluid cavity.

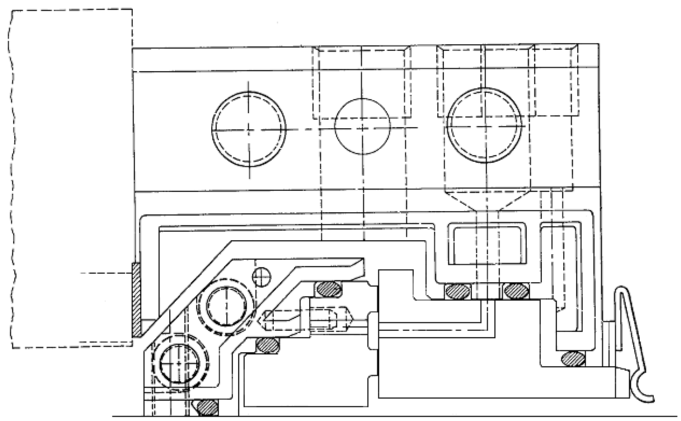

Figure 6. Concentric Common Face Geometry.

With this arrangement process pressure is on the outside of the seal rings, holding seal ring segments together, and barrier pressure is also directed through various passages to the outside of the seal rings. For example, Figure 7 shows axial and radial pressure loadings for the barrier fluid and axial pressure loadings for the process pressure.

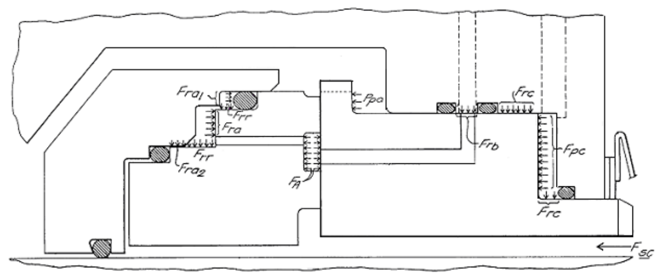

Figure 7. Pressure Acting Forces.

One may note that the axial pressure forces act on the outside of the seal rings in a manner similar to a balanced single mechanical seal. In this particular embodiment the barrier pressure cavity is separate from a closing pressure cavity. Although the geometry is such that the two cavities can be joined and can be relied on to create balanced axial forces for the inner (outboard) seal, this particular example is shown with separate cavities.

With such a novel design, particularly in gas sealing applications, one may wonder about the design philosophy and its operating principles. To answer these questions, it is helpful to examine some design aspects of a conventional mechanical face seal, for example, its balance ratio. The balance ratio of a conventional face seal is an important parameter pertaining to its design, since it is a measure of the seal face load exerted by the sealed fluid pressure. This balance ratio (often referred to only as balance) can be expressed as:

$$B = \frac{A_h}{A_f} \tag{1}$$

where A_f is the sealing face area, and A_h is the back area with sealed fluid pressure. From this expression, immediate conclusions can be drawn, that is, it is simple and it does not change with the change in sealed fluid pressure. Then, looking at the coplanar double gas seal design as depicted in Figure 6, it is apparent that the balance of this new seal design is not going to be as simple. It has two seals, an inboard seal and an outboard seal; and yet these two seals share one seal ring. Nonetheless, following the traditional balance ratio definition of mechanical face seal (Lebeck, 1991), the following expression can be derived as the apparent composite balance ratio of this coplanar double gas seal:

$$B_a = \frac{A_{f,in} + B_{out} \cdot A_{f,out}}{A_f} - \frac{\Delta p_{bp}}{p_b} \cdot \frac{(1 - B_{in}) \cdot A_{f,in}}{A_f} + \frac{\Delta p_{cb}}{p_b} \cdot \frac{B_{in} \cdot A_{f,in} + B_{out} \cdot A_{f,out} + A_{ann}}{A_f} \tag{2}$$

where B_{in} and B_{out} are balance ratios of inboard and outboard seals obtained as if they were two separate seal rings; $A_{f,in}$ and $A_{f,out}$ are sealing face areas of inboard and outboard seals, respectively. A_{ann} is the annular area in the interface separating the inboard and outboard sealing face, and A_f is the total sealing area of both inboard and outboard seals. Δp_{bp} and Δp_{cb} are pressure differentials between barrier pressure, p_b, and process pressure, p, and between closing pressure, p_c, and barrier pressure, p_b, respectively.

For the seal in this field application, substituting all parameters with dimensional data, Equation (2) can be simplified as the following:

$$B_a = 0.817 - 0.382 \cdot \frac{\Delta p_{bp}}{p_b} + 1.082 \cdot \frac{\Delta p_{cb}}{p_b} \tag{3}$$

One apparent difference between Equation (3) and Equation (1) is that the balance in this coplanar double seal is no longer constant with respect to the sealed fluid pressure. In fact, as shown in Figure 8, if the two pressure differentials are kept constant, the balance increases as the sealed fluid pressure rises. The balance ratio also changes when the pressure differentials change as shown in Figure 9: it decreases when the pressure difference between barrier gas and sealed fluid is increased, and it increases when the pressure difference between closing cavity and barrier is increased.

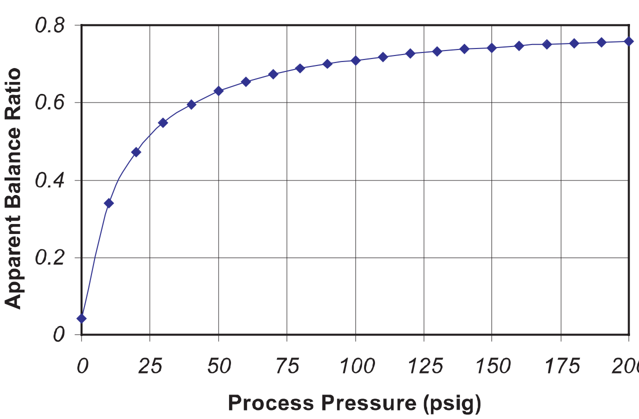

Figure 8. Changes in Apparent Balance Ratio with Respect to Sealed Fluid Pressure When Pressure Differentials Are Kept as Constants.

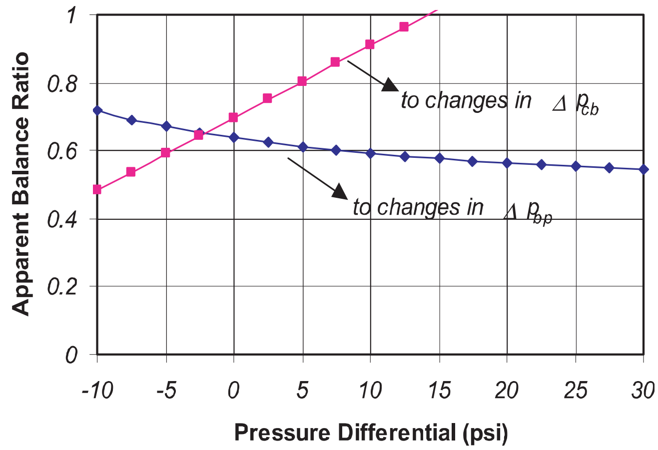

Figure 9. Characteristic Apparent Balance Ratio with Respect to Pressure Differentials for a Given Sealed Fluid Pressure.

This is precisely the design philosophy and its operating principle so far as the axial force balance is concerned. By regulating the two pressure differentials, the contacting load and barrier gas usage rate can be controlled. In ordinary applications, the pressure differential between barrier gas and sealed fluid is typically set to a value no less than 15 psi, so as to ensure no sealed fluid leaks through the interface and to maintain proper seal face distortion. Then, the pressure differential between closing cavity and barrier gas is adjusted for optimum operating performance. Lowering closing pressure and/or raising barrier pressure, the desired zero or near zero contacting load can be achieved, such as in the cases of all gas (vapor) applications. Conversely, increasing closing pressure and/or decreasing barrier pressure, the mating faces can be brought closer so as to decrease the interfacial film thickness and thus reduce barrier gas usage. To put all these into practical applications, it is obvious that an effective pressure control system is of necessity, which is described later in this paper.

Of course, the use of separate seal faces in common seal rings does introduce some significant design challenges. Axi-symmetric face deflection must be controlled to a significantly greater degree than with separate seal faces. The face rotation or coning multiplies the distances between the seal faces not only from the inside diameter to the outside diameter, as one is used to seeing in a single seal, but in this case also from the inside diameter of the inner face to the outside diameter of the outer face, which is a distance over three times as much as one normally has to deal with. In order to maintain the required face profile, the axial support of the rotary seal ring was adjusted to achieve the correct profile. Figures 10 and 11 show the difference in face deflection when the support is moved radially from the outer O-ring to the inner O-ring under the same pressure conditions. Even using one O-ring as support, the placement of the O-ring within the geometry of the part is critical to the overall deflection characteristics as illustrated in Figures 12 and 13 (Azibert, et al., 2000b).

As previously discussed, one further complication was introduced by the desire to make this a gas-lubricated seal with controlled net closing forces. The arrangement allows the regulation of a closing pressure separate from the barrier pressure. This creates significantly more flexibility in tailoring the seal to the operating conditions.

The regulation mechanism can be external to the seal and set with conventional pressure regulating devices. The application, which is detailed later, utilized differential pressure regulators built in the gland of the seal itself (Azibert, 2000.) The reference pressure is the process pressure. In turn the barrier and closing pressure differential can be set by external regulating screws. Figures 14 and 15 show the positioning of the pressure regulators within the seal.

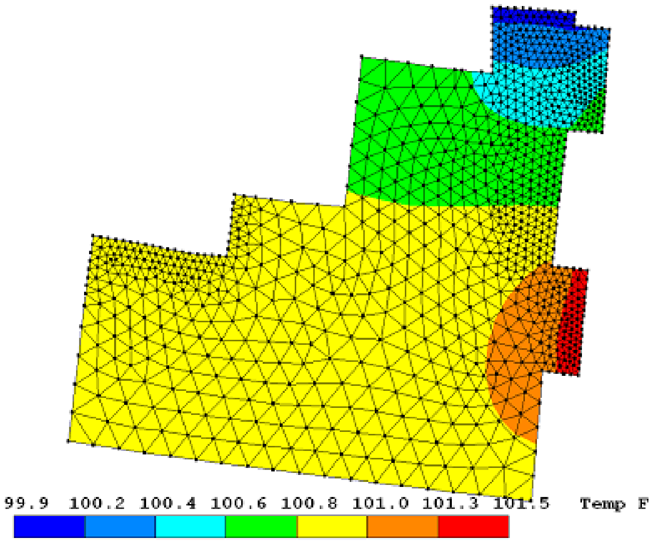

Figure 10. Outer O-Ring Support.

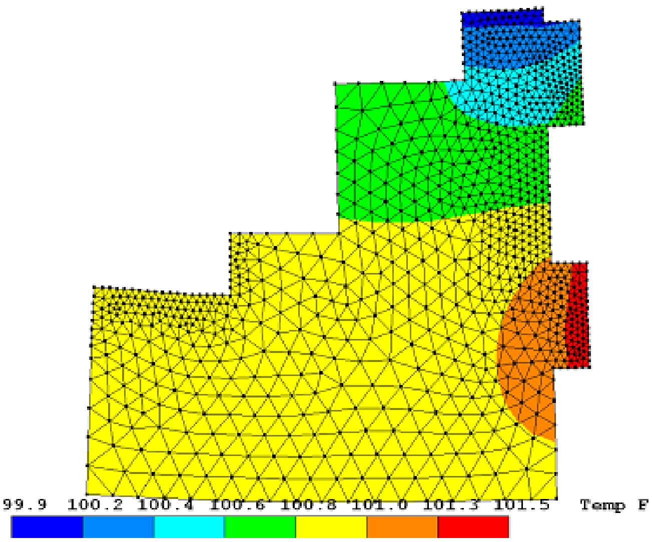

Figure 11. Inner O-Ring Support.

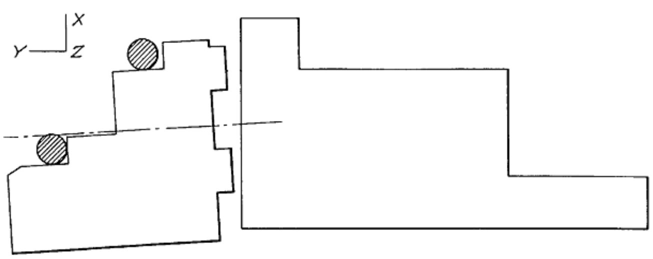

Figure 12. Positive Coning Geometry.

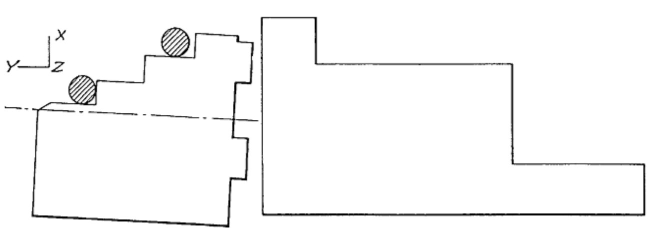

Figure 13. Negative Coning Geometry.

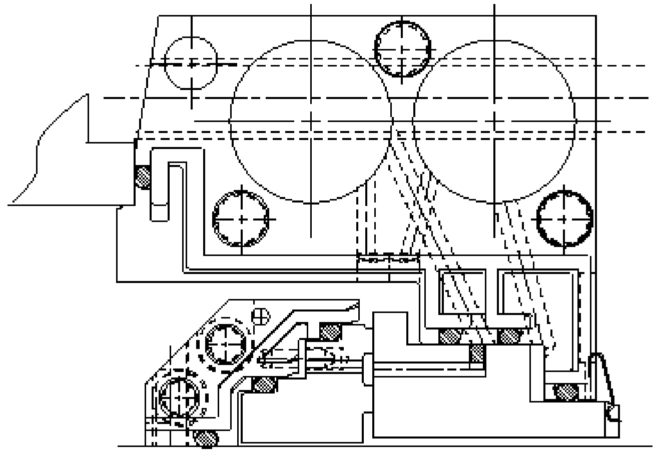

Figure 14. Cross Section View with Dual Regulators.

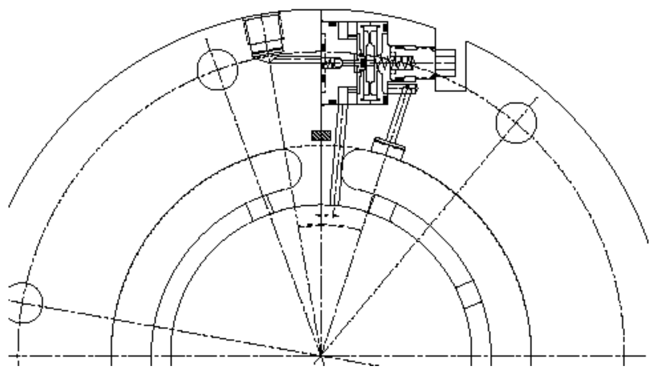

Figure 15. End View of the Differential Pressure Regulator.

APPLICATION OF THE THEORY TO THE FIELD

A critical application of a blower with a lethal gas (carbon monoxide, CO) provided a background for this type of design. This blower had a shaft diameter of 90 mm (3.54 in.) and rotating speed of 1800 rpm. The sealed gas had a pressure of 35 psig (2.4 bar) and a temperature of about 300°F (150°C). A split seal was chosen because the entire plant operation relies on this piece of equipment and it takes two days to replace a solid seal. A barrier gas seal was particularly desired to avoid any potential contamination inside the machine, which, if present, could have resulted in corrosion. The requirements were that sealing integrity had to be maintained even if barrier gas supply was lost. Based on the application requirement and its operating conditions, a series of analyses and calculations were carried out for parameter settings to optimize the performance. It was determined that, for a process pressure of 35 psig, a barrier pressure of 52 psig, and a closing pressure of 42 psig with the consideration of face load imposed by springs would give rise to a near zero contacting condition, which would yield an apparent balance ratio as defined by Equations (2) and (3) to be 0.484. And the final pressure settings should be determined by the total barrier gas consumption rate. This barrier gas consumption rate should be between 5 and 10 scfh (2.36 and 4.72 l/min), which would correspond to an operating film thickness of 150 to 200 µin.

The initial installation went extremely well with the installation completed in less than three hours (Figures 16 and 17.) Note the two pressure gauges for the barrier and closing pressure. The supply goes into one port and is then internally directed to the two regulators. A thermocouple is used to monitor the stationary face temperature. The machine was started without any pressure and all indications were positive.

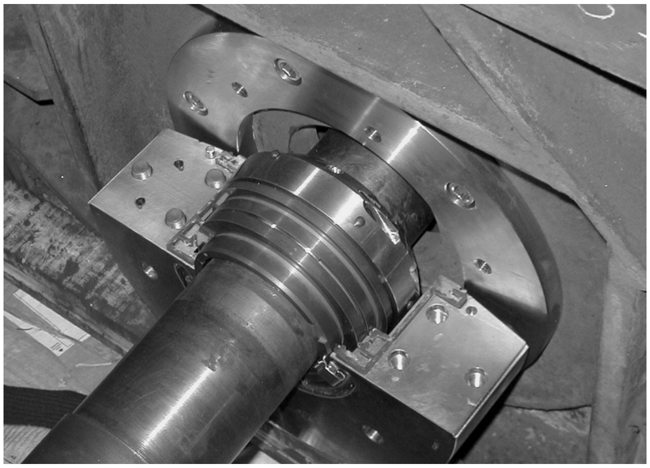

Figure 16. Split Seal Installation.

Figure 17. Installed Split Gas Barrier Seal.

When the blower was pressurized (on a trial run) with nitrogen, the face temperature went up and the seal started to have a gas consumption of 35 l/min. After many attempts at troubleshooting the operation, it was discovered that the housing moved more than 3 mm (.11 inch) relative to the shaft. This amount of axial movement was more than had been designed into the seal. So a second attempt was made with a face mounting plate secured to the baseplate, and a convoluted bellows to compensate for the deflection of the vessel wall (Figures 18 and 19.)

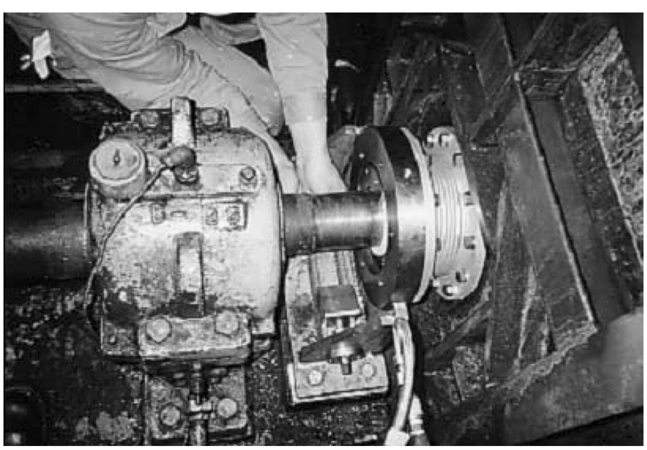

Figure 18. Convoluted Bellows Isolator.

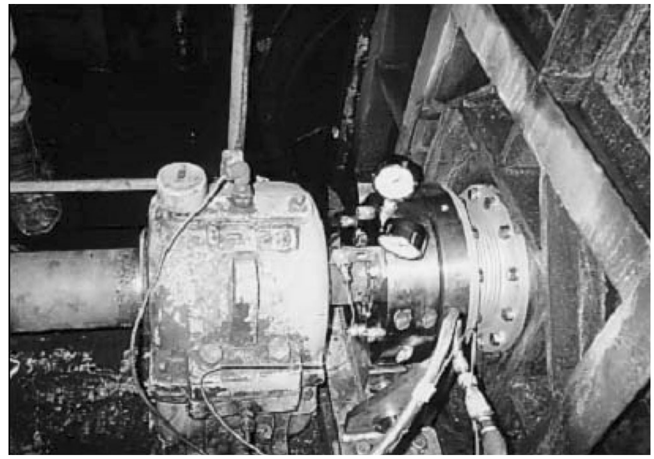

Figure 19. Final Installation.

Very high hopes resided on this investment. Unfortunately the seal behavior remained the same: high face temperature and gas usage when the blower was pressurized. At this point it became quite difficult to understand what was happening. High gas consumption would indicate face separation and, therefore, we would expect minimal temperature rise. Conversely, a high temperature rise would indicate face contact and the gas consumption would be expected to be negligible. After much scrutiny it was decided that parts had to be returned for inspection.

Only one clue became clear after sifting through all the installation and testing procedure. The seal would start to leak after the process pressure increased past 1.8 bar (26 psi). And this was quite reliable. After much discussion and many calculations the culprit was finally identified. The antirotation drive pin protrusion on the rotary metal holder was longer that the hole inside the rotary seal face. An oversight hidden inside a few hundred dimensions (Figure 20.)

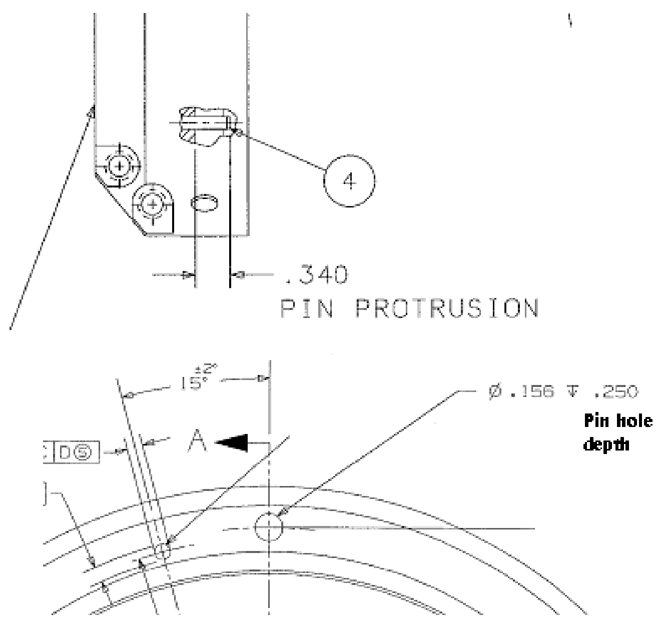

Figure 20. Drive Pin Interference.

To appreciate why such a small detail could cause such a significant difference in performance, one must go back to the design premise that the seal face segments must be uniformly supported in the axial direction. It is important to understand that the split face segments remain in alignment when the axial support is uniform. There is a resilient support, an O-ring, to achieve that goal. If just a machined metal surface had been used, the back of the seal face would take the shape of the support. If the machined surface is lapped, and the back of the seal face is also lapped, then sufficient flatness could be maintained. This was not done, however, because of the manufacturing difficulty associated with this operation and its related cost. It is much simpler to rely on elastomers to provide the even support.

The interrelated requirements of the seal being split and balanced regardless of pressure differential generated a geometry that evolved from the conflicting requirements. One must look at the forces holding the segments together, at the forces holding the faces against each other, as well as the axial support forces acting of the seal face assembly. In the design, the axial reaction forces are finely balanced. The axial support is on the springs from the barrier gas pressure, on the rotary holder from the closing pressure, and on the holder from the process pressure. When the support switched from the spring to the holder by the increase in the process pressure past 1.8 bar (26 psi), the support became the pin, a one point support. At this time the face halves became misaligned, and we experienced high temperature because of localized contact, and high leakage because the face was no longer flat. The rotary face that had been running for a while (the machine was left idling form 500 to 700 rpm for a while with the high barrier gas consumption as no process gas leaked) and it was 0.13 mm (.005 inch) thinner in the area of the drive pin than in the opposite side of the face. This is evidence that the pin was the support.

After the problem was corrected and the seal was installed for the third time, success was achieved. The following is an excerpt of application report on the operating conditions:

16:15 (4.5hrs since start up)
CO gas temperature = 140degC.
Face temperature = 60degC. (Stable last 2 hrs.)
Process Pressure = 2.4bar.
CG = 3.1bar.
BG = 3.50bar
Flow = 4L/min.
No detectable CO gas leakage.

From the final settings of field application, the apparent balance ratio is calculated to be 0.574, which is larger than the predicted value of 0.484 for an optimal operation. This indicates that in the sealing interfaces, there are more dynamic actions than initially predicted. If the 4 l/min (8.5 scfh) barrier gas is consumed all through the sealing faces, then the operating film thickness will be around 200 μin (5 μm). In actuality, taking other leaking paths (of a split seal) into account, the estimated operating film thickness should be in the range of 150 to 200 μin. Nevertheless, it is clear that the seal performance agrees rather well with the design calculations and predictions. At the time of this writing, the seal has been in continuous running for over 3500 hours.

CONCLUSION

Split mechanical seal technology has evolved to the point where the most difficult applications requiring the most technically advanced solutions can be addressed. The requirements placed on design solutions by having parts radially split can be conquered even for gas lubricated seals. One should never forget that simple mistakes can easily compromise the most advanced and highly engineered mechanical designs.

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