DOUBLE GAS SEAL WITH COPLANAR COAXIAL
RAYLEIGH PAD FACES FOR PUMP SEALING APPLICATIONS

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ABSTRACT

There has been a rapid growth in the application of noncontacting double seals with a gas buffer fluid to pump sealing. The advantages of such a sealing arrangement are:

- There is no liquid buffer fluid to maintain or dispose of.

- Gas buffer fluid causes a minimum of contamination to the process.

- There are no emissions, so no monitoring may be required.

- Friction power is only a fraction of that associated with liquid seals.

- Because there is no contact, seal life can be several times that of a liquid/contacting seal.

One of the limitations of a conventional double seal arrangement, be it gas or liquid, is that there must be either considerable axial length or considerable diametral space available. There are many pump applications that could benefit from a double gas buffered sealing system where space is severely limited. Thus, there is a need for a small size double gas sealing device.

The double gas seal with coplanar coaxial Rayleigh pad faces has two gas sealing faces concentrically located on the same plane. Buffer gas is supplied to an annular groove between the two sealing faces. The outside gas seal leaks to process and the inside gas seal leaks to atmosphere. Both faces have Rayleigh step pad lifting surfaces as well as a sealing dam similar to a conventional gas seal. The process fluid is to the outside of the outer seal face so as to minimize process fluid contamination of the gas seal face.

With the configuration described, the double gas seal with coplanar coaxial faces can be made into a small package that fits entirely into a seal gland, which itself has minimum dimensions. Thus this seal will fit to most conventional small bore ANSI pumps, and this makes it possible to apply double gas sealing technology to many existing process pumps.

In this paper, the theory of the double gas seal with coplanar coaxial faces is described. Considerable design work was required to find useable shapes and balance moments on the seal so that the inside and outside sealing gaps stay open under variable process and gas supply pressures. Optimization of the Rayleigh step pad bearings was performed. A complete simulation model was developed. Deflection results and predicted performance (leakage) are presented. Extensive testing has been conducted, and the paper presents predicted and measured leakage as a function of process and gas pressure. Loss-of-gas testing has also been performed, and these results are presented. Several field tests have been initiated, and these are described. Observation so far has been that the seal performs as expected, so the double gas seal with coplanar coaxial faces can be used to provide double gas sealing solutions for many pumps having limited available space.

INTRODUCTION

There has been a rapid growth in the application of noncontacting double seals using a gas buffer fluid to seal pumps. The advantages of such a sealing arrangement are:

• There is no liquid buffer fluid to maintain or dispose of.

• Gas buffer fluid causes a minimum of contamination to the process.

• There are no emissions, so no monitoring may be required.

• Friction power is only a fraction of that associated with liquid seals.

• Because there is no contact, seal life can be several times that of a liquid/contacting seal.

Several different approaches have been taken for the design of such sealing systems. One of the first such systems is described in detail by Wasser, et al. (1994). In this conventional double seal arrangement (Figure 1), there are distinctly two gas seals axially separated into inboard and outboard seals where both seals are pressurized by gas on the outside. The inner of these seals faces the sealed product on the inside diameter and the outer seal faces atmosphere at its inner diameter. Both of these spiral groove gas seals have their own primary and mating ring assemblies.
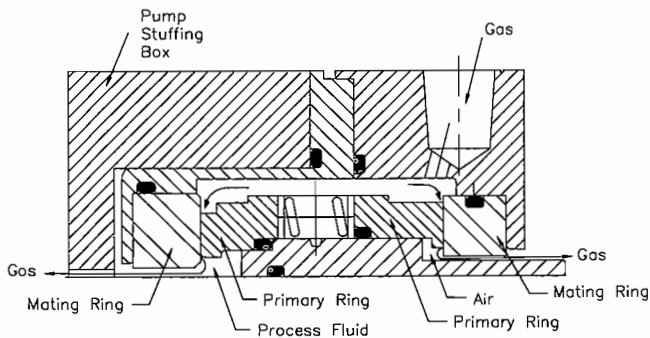

Figure 1. Traditional Double Gas Seal Arrangement.

This same basic arrangement has been used by others to achieve a double seal using a gas buffer fluid. Young and Huebner (1998) show an arrangement similar to Figure 1, but use wavy faces to achieve liftoff rather than spiral grooves.

To achieve a smaller package, O'Brien and Wasser (1997) introduced a new design where both seals share a common rotor with lifting features being placed on both sides of the rotor. This leads to a design that is much more axially compact, and this design can be made to fit many existing ANSI pumps having small stuffing box bores and small axial space.

Figure 2 shows the concept of interest here. There are two distinct coplanar seal faces, one coaxially outside the other. Both the primary faces and the mating faces are made common to the mating ring and the primary ring. Thus, both the primary ring and the mating ring each have two faces. Gas pressure is applied between the two faces. With this configuration one has achieved the smallest axial space possible, and all parts of the seal may be positioned outside the stuffing box itself. The obvious advantage of this seal is that it can fit a pump with any size stuffing box, as long as it can fit the axial and diametral space available outside the stuffing box. Thus, this seal would be expected to have advantages in fitting more small space pumps than some other designs.

The disadvantages of the configuration in Figure 2 are that two faces, and related parts and material thicknesses, must be made to fit between the bolt circle and the shaft, as opposed to just one face in the previous designs. Also, the arrangement in Figure 2, while it may function like a double seal, does not have the redundancy of parts as is found in conventional double seal arrangements.
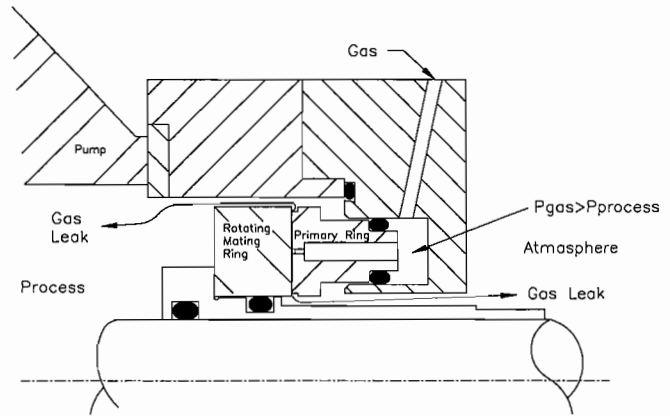

Figure 2. Coplanar Coaxial Double Gas Seal Concept.

Nevertheless, given the apparent advantages for this configuration, a development program was initiated to develop a seal based on the concept of Figure 2. The details of this development, the final product, and test results are described in the sections to follow.

DESIGN OBJECTIVES

• Design is to utilize smallest axial space and radial space so as to fit the maximum number of pumps

• Leakage to be consistent with competitive designs (2 scfh range at 100 psi)

• Minimum film thickness to be large enough to assure no contact in normal circumstances

• Seal is to have ability to operate over wide range of gas pressures and product pressures while performing satisfactorily

• Speed to be 1800 rpm or 3600 rpm, with possible operation at zero speed

• Pressure to 300 psig

• Temperature to 450°F

• Ability to seal any product compatible with gas discharge to product including abrasive, sticky, and highly corrosive fluids

• Ability to seal product in the event of loss-of-gas supply

• Ability to recover to normal operation after loss-of-gas with clean nonsticky process liquids

COPLANAR COAXIAL DOUBLE RAYLEIGH GAS SEAL CONCEPT

Method of Operation

Figure 2 shows the general concept of operation. Gas at higher than process pressure enters the space between the inner and outer seal faces. This gas then leaks to atmosphere across the inner face and it leaks to the product across the outer face. Since gas is constantly leaking across the outer seal face into the product, the product cannot leak out across these same faces.

Figure 3 shows more detail about the concept. To assure reliable operation in gas, one generally must use lifting features on the faces. So in effect, one now has two gas seals on the two faces. The inner seal is an outside pressurized seal and the outer seal is an inside pressurized seal. The effect of the lifting faces is to cause the gas pressure to distribute itself in such a way that the film has as much stiffness as possible. The essential characteristic of film stiffness is that it keeps the faces from touching in the event of upsets, overloads, and out of flatness, and it causes the film thickness to reliably establish itself at a repeatable and known level.
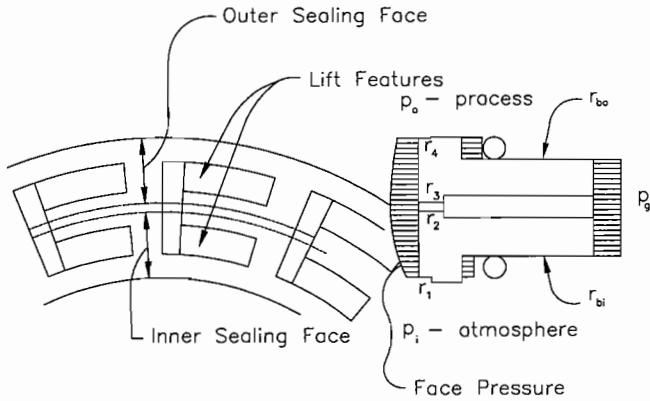
Figure 3. Coplanar Coaxial Double Gas Seal Hydrostatic/Hydrodynamic Face Features and Force Equilibrium.

Figure 4. Inside Pressurized Rayleigh Pad Parameters.

Figure 3 shows how the gas pressure acts on the primary ring. Acting to the left, the gas pressure acts between the two balance diameters, r_{bo} and r_{bi} as shown. The process pressure acts between r_{bo} and r_4, and the atmospheric pressure acts between r_{bi} and r_1. On the outer seal face, one will have a distribution of pressure that depends on the details of the seal grooving, but in general the shape of the distribution will be as shown where the pressure drops between the gas pressure at r_3 to the process pressure at r_4. The pressure may actually increase over some portion of this face if the grooves are used to pump outward. Looking at the way the loading pressures are acting and the pressures on the face, this outer seal is much the same as a *single face inside pressurized gas seal*. As long as the balance ratio is properly selected so as to apply the right load to the faces, the seal will respond favorably and maintain a useful film thickness.

All these same arguments can be made for the inside seal as well, so the inner seal acts like a *single face outside pressurized gas seal*. So, the seal becomes a combination of the two seals, each of which can be separately analyzed. The interaction between the two occurs when there is a deflection of the faces. As the pressure on one face changes due to deflection, this changes the load on the second face as well.

Rayleigh Step Pads

The seal concept shown in Figure 2 can be made to operate with no features on the face, but there would be some contact of the faces and it would be difficult to control heat and wear. To make a gas seal operate successfully, one must choose a face geometry so that the seal interface develops an increased load support as the faces are moved closer to each other. This is called stiffness of the film and in fact is normally a negative number, in that the force increases as the gap between the faces decreases.

There are many different ways to obtain stiffness of the faces in a gas seal. One can use various spiral groove profiles, tapering waves, radial tapering, and of course Rayleigh pads. For the current development, Rayleigh step pads were chosen because they are known to work well and the technology is in the public domain. Figure 3 shows the unidirectional pattern chosen. While one can design a bidirectional Rayleigh pad, the unidirectional design offers greater stiffness.

Figure 4 shows the parameters to be considered when designing Rayleigh pads. These are summarized below:

Δr – Face width
n – Number of pads
r_d – Fractional width of the sealing dam
r_p – Fractional width of the pad
t_f – Fraction length of the feed groove
t_p – Fraction length of the pad
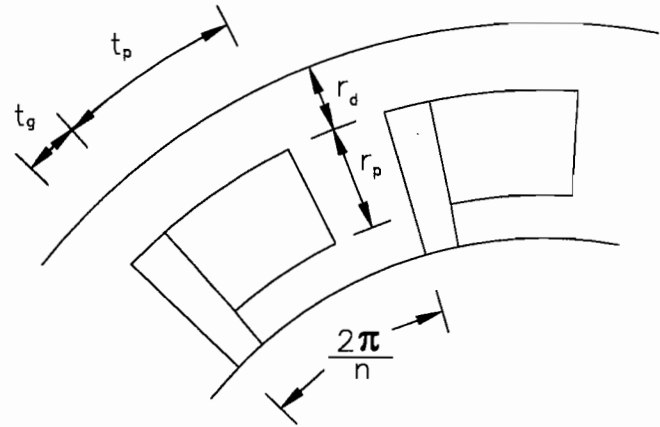h_f – Feed groove depth
h_p – Pad depth

Design Optimization

Additional seal design parameters that must be considered are:

B_i – Inside seal balance ratio
B_o – Outside seal balance ratio

Performance criteria are:

h_{min} – Minimum film thickness
k – Film stiffness
Q_i – Inside leakage
Q_o – Outside leakage

The Rayleigh step pad parameters, balance ratios, and the minimum film thickness values must be chosen. The larger optimization problem of design must be considered. Several of the parameter values to be chosen are based on compromise and tradeoff. Yet other parameters can be chosen based on optimization. Now all the parameters and the basis for decisions are discussed.

The first choice to be made is the value of the nominal minimum film thickness. For these analyses, this value has been chosen at about 80 μin. This number represents a practical minimum value considering expected out of flatness of the parts, and it represents a reasonable compromise for leakage. Larger values are desirable because they provide a larger margin for unwanted but naturally occurring waves and radial taper (that cause touching), but leakage increases rapidly with film thickness. Smaller values are desirable because leakage is smaller, but these are more likely to lead to contact of the faces. Thus, minimum film thickness is normally a compromise.

The second major choice is face width. The wider the face, the higher will be the stiffness and the lower will be the leakage. However, as can be seen from Figure 2, for this type of design there is a practical upper limit for face width that is set by the distance between the bolt circle and the shaft diameter. One must have enough radial space for two face widths plus a thickness of the housing, plus O- rings, plus minimum thickness of metal, plus an allowance for radial motion. For the 1-7/8-inch shaft size seal used as the prototype design here, these factors limit the face width (for this size) to about 0.2 inch.

Next, one must consider the balance ratio for each of the two seals. For practical reasons, they are assumed to be about the same value. For gas seals of the Rayleigh pad type, a workable balance ratio is about 0.8. At this value, one will find that minimum film thickness will increase somewhat at higher operating pressures. This has the advantage that, as pressure increases, distortions increase, and therefore one needs to allow for more distortion. Making the balance ratio higher will reduce the amount of increase in minimum film thickness. Making the balance ratio lower will increase the minimum film thickness at a given operating point, so the value of *B* is set based on the lowest acceptable film thickness at the lowest operating speed.

Finally, the remaining parameters have to do with the geometry of the Rayleigh pads themselves. The criterion for choosing these other parameters is usually stiffness. One can readily show that at a given minimum film thickness there are an optimum number of Rayleigh pads that give maximum stiffness. This argument also applies to the pad depth and the feed groove depth. One must not make the dam width too small or leakage will be too large. There is also an optimum length of the pad considering stiffness. Thus, the proportions of the pads used in the example below are an approximate optimum. One should note that the values are not critical, on the other hand. The seal will, for example, operate fine if one uses 14, 16, or 18 pads, but not eight and so on. Pad depth is important. Experimental results and calculations, as well, show that pads must not be allowed to become too shallow. It is better to error on making the pads deeper rather than shallow.

Computational Tools

Before one can pursue the issue of detailed design or design optimization, one must have computation tools that predict seal performance. In particular, given an arbitrary face profile and face features, one must predict the details of the pressure distribution and the minimum film thickness for a given load (balance ratio). Methods for predicting pressure distributions for thin films using compressible Reynolds equations have been well known for some time, and Szeri (1980) and Gross, et al. (1980), summarize such theory. The specific methods used here are described by Lebeck (1991). For the double coaxial coplanar seal shown in Figure 3, a special formulation was made whereby one could solve simultaneously for the minimum film thickness at load equilibrium and the pressures on both faces. While the pressure distributions of each of the faces are independent of each other (given a constant film thickness), the load support depends on both. So, solution is easier if one solves for the pressures on both faces at the same time. This formulation is referred to as FSGASCC (1998). This algorithm made the computation of minimum film thickness much easier, particularly later in the development where complete equilibrium solutions with face deflection included are essential.

Consider, for example, the following case:

$$RPM = 3600$$
$$p_o = 114.7 \text{ psia}$$
$$p_g = 134.7 \text{ psia}$$
$$p_i = 14.7 \text{ psia}$$
$$\mu = 2.7 \cdot 10^{-9} \text{ psi·s}$$
$$r_1 = 1.075 \text{ in}$$
$$r_2 = 1.275 \text{ in}$$
$$r_3 = 1.300 \text{ in}$$
$$r_4 = 1.500 \text{ in}$$
$$B_i = 0.790$$
$$B_o = 0.815$$

Values for the pads themselves are:

$$n = 14$$
$$r_d = 0.3$$
$$r_p = 0.5$$
$$t_f = 0.15$$
$$t_p = 0.6$$
$$h_f = 300 \text{ μin}$$
$$h_p = 200 \text{ μin}$$

Figure 5 shows the pressure prediction using FSGASCC (1998) computer program. Note that the gas pressures on both faces are shown and the difference between the inside and outside pressure distribution. For this particular computation (where the faces have been made parallel), the resulting performance parameters are:

$$h_{min} = 87 \text{ μin}$$
$$Q_o = 0.018 \text{ scfm}$$
$$Q_i = -0.043 \text{ scfm}$$
$$P = 6 \text{ B/h}$$
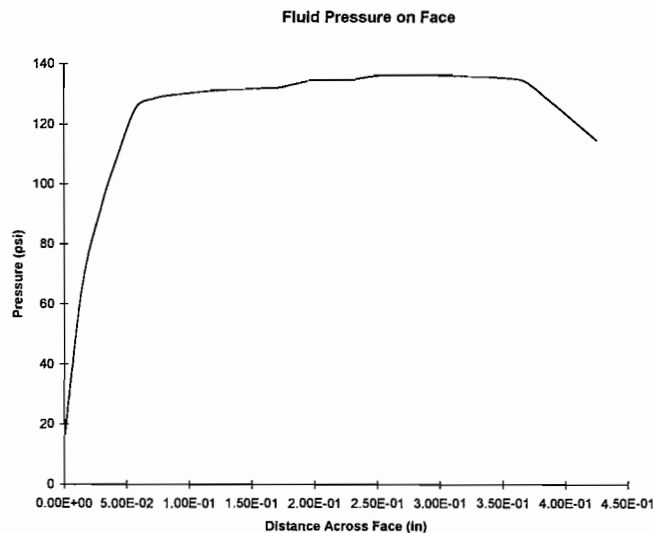$$k = -422000 \text{ lb/in}$$

Fluid Pressure on Face

Figure 5. Typical Pressure Distributions from Computational Model.

Thus the computational tool described was used to conduct the trade off and optimization studies described so as to arrive at the understanding and choices made.

STRUCTURAL DESIGN

Structural Design Issues

So far, only the design of the faces has been considered. To make the faces operate as predicted above, one must consider a number of other features related to the structural and related geometrical assembly. Some of the issues are:

• The faces must stay relatively flat and parallel (minimum radial taper), in spite of changes in the gas and process pressures.

• Axial friction must be low, so that the applied load stays relatively constant.

• The design must be relatively insensitive to temperature changes.

• Features must be incorporated to handle the occasion where gas supply is lost.

• Both sets of rings must be driven and sealed to the housing/shaft assembly.

• Spring loads must be applied.

Compliant Face Concept—First Structural Design

There are many different structural configurations that can be used to carry the faces. The first approach taken for this development was the compliant face design shown in Figure 6. Here the carbon has a small axial dimension, which makes it relatively flexible in bending. Thus, the cross section of the carbon rotates about its centroid relatively easily. Now the two faces each create forces F_o and F_i, and both seal faces have inherent stiffness k_o and k_i. The rubber at the right side of the carbon seal acts only to transmit pressure to the seal ring and does not affect the stiffness of the carbon part. One can set up a deflection analysis of the carbon considering all the different pressures acting on the face, the gas pressure, and the sealed pressure. The seal faces and film thickness are included in the model. Then one can find the equilibrium operating deflection of the carbon. This deflection can be represented in terms of a radial taper. The difference between the minimum film thickness for the inner and outer seal is the criterion for satisfactory solution.
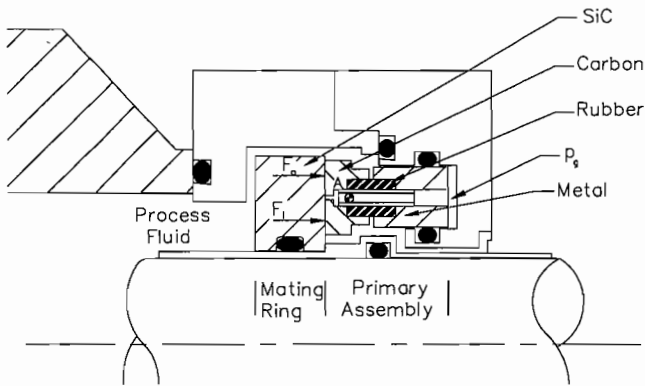
Figure 6. Compliant Face Concept—First Structural Design.

Using the interface model described above to predict gas pressure distribution coupled with an FEA to predict deflection, plus an iterative technique, the radial taper of this compliant design could be predicted. Using iterative procedures, proportions of the rubber and the carbon of the primary ring were found that allowed the faces to stay relatively parallel over the entire range of operation. Clearly one reason such operation is possible is that both faces have individual stiffness, which tends to make the film thickness somewhat the same on both faces, assuming the imbalance is not too large.

One feature of this design is that the gas pressure force is transmitted between the metal part of the primary and the carbon portion of the primary through the rubber itself. The rubber acts somewhat like a fluid in transmitting this pressure between the two elements of the primary. Since the area of the rubber is relatively large, this means that the rubber transmits relatively large forces. This behavior ultimately led to the rejection of this design approach. Given that the carbon primary is relatively flexible, any slight variation in the pressure transmitted by the rubber causes the carbon to become wavy to the point that the seal will drag at certain spots.

Since it is commonly assumed that rubber transmits pressures almost hydrostatically like a liquid, the source of this problem is not so evident. However, careful analysis shows that the slightest tangential variation in Youngs modulus of the rubber will produce waves (and dragging). Further, even small variations (0.001 inch) in the thickness of the rubber or the flatness of the mating surfaces will also cause waves that in turn lead to contact.

The final alternative considered was to mold the rubber in place between the carbon and the metal. This would assure that there would be no nonuniformity in the contact pressure caused by the imprecision of the parts. However, even this procedure could not guarantee the tangential uniformity of elastic properties.

By using great care in making surfaces flat, the seal design was made to operate without touching, but it was decided that such measures would be too expensive in production, and that even these could not guarantee uniformity over time. Thus, the first structural design was abandoned.

Monolithic Primary—Second Structural Design

Figure 7 shows the second structural design concept. Here the primary ring becomes much longer and therefore becomes relatively much stiffer. In this type of design, one relies on stiffness to maintain alignment, whereas the compliant design relies on compliance and the film stiffness of the two faces to achieve a radially parallel face. Of greatest concern with this design concept is the fact that the outer seal is an inside pressurized seal. With inside pressure, a seal can become thermally unstable from contact and heat generation at the face. Thermal coning causes a pressure direction divergent film thickness (touching at the I.D. of the faces). This in turn causes the contact to become worse, rather than

reducing contact load. Thus, inside pressurized gas seals have some risk of thermal instability if they touch. Outside pressurized gas seals are of course thermally stable.
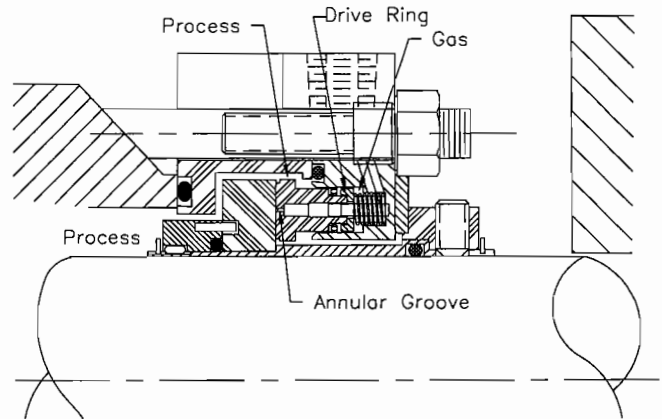

Figure 7. Monolithic Primary—Second Structural Design.

For the design concept of Figure 7, careful analysis was made using the model described above. Even though the outside face becomes divergent under contact load conditions, the inside face becomes more convergent and in fact carries the load. Thus, because the two seal faces are part of the same structural assembly, the potential problem described for inside pressurized seal does not occur. The coplanar coaxial seal is thermally stable.

The greatest challenge to produce a working version of the concept in Figure 7 was to adjust the shape of the cross section of the carbon so that the radial taper is small enough under all the various operating conditions. The adjustment of the cross section consisted of adding the increase in thickness of the part just to the right of the face and adding the deep groove at the right hand end. The effect of the groove is shown in Figure 8 where, under the condition of having outside pressure and gas pressure acting, the seal face deflects radially inward. The right hand end does not deflect inward as much because it is not subjected to radial differential pressure. The groove reduces the coupling between the left and right ends, so that the left end can deflect radially while not severely tapering at the same time.

PRESSURE AND CONCENTRATED LOAD CAUSED DEFLECTION
DPSEAL9 - SHORTER SOLID

Figure 8. Seal Deflection With Gas and Process Pressures.

Using the seal model described, numerous iterations were made on the details of the shape of the carbon until a reasonable compromise was found where the radial taper would stay within

reason over the range of operating conditions. Table 1 shows a few of the results for the design in Figure 7. One has relatively small changes in the radial taper, ϕ.

Table 1. Performance Predictions for Structural Design 2.

RPM	p_i	p_g	p_o	h_o	h_i	ϕ	Q_o	Q_i
	psia	psia	psia	μin	μin	μrad	scfm	scfm
3600	14.7	134.7	114.7	90	84	11	0.020	−0.041
3600	14.7	134.7	14.7	87	88	−1	0.059	−0.047
3600	14.7	334.7	314.7	128	118	20	0.119	−0.64

One will note in Figure 7 the use of PTFE spring loaded u-cup type of seals rather than O-rings. Testing showed repeatedly that while a seal would start running at predicted leakages, over time the leakage would become smaller and smaller. This problem was traced to a combination of O-ring friction and shaft expansion. In the test machine used here, as the shaft warms up, there is a small relative motion trying to close the seal. O-ring friction was so high that gas leak rates would become a fraction of those levels at the start of a test because the faces become overloaded. The minimum film thickness was being reduced to less than one-half the starting value. Replacing the O-rings with the PTFE u-cup seals eliminated this problem.

The carbon is driven by a special drive ring shown in Figure 7. The drive ring engages lugs on the carbon, and it engages drive pins pressed into the housing. The deep groove in the carbon makes a direct drive impossible in this design.

If one loses gas pressure so that gas pressure becomes less than product pressure, then the u-cup seal will leak so that the right hand side of the seal becomes pressurized by the product all the way down to the lower balance diameter. The balance ratio on the inside seal becomes the same as under gas operating conditions and the outside seal performs no sealing function. Loss-of-gas tests showed that the seal would then operate with a thick film in water (due to the face features) and leak of the order of 30 ml/min. This was considered to be too large, so an alternative design was developed.

Loss-of-Gas/Overbalanced—Third Structural Design

Figure 9 shows the third structural design. This design is very much like the second design, except for the addition of an O-ring at the outer balance diameter of the primary and allowing the mating ring to move to the right. Under a loss-of-gas scenario, because of the location of the O-ring at the I.D. of the mating ring, the mating ring balance diameter becomes smaller than the inner balance diameter on the primary ring, so the mating ring moves to the right pushing the primary ring with it until the primary bottoms. The extra O-ring in Figure 9 keeps the fluid from leaking past the u-cup to the gas passage and then out across the inside seal. The seal operates in this position as an overbalanced seal (B>1) and seals off the product. Leakage is now much lower than with design 2 above, but due to the overbalance, heat generation is high. Thus, three support protrusions were added to the support between the carbon and the housing so that as the carbon is forced to the right, the carbon develops waves, which in turn provide additional lubrication to the faces. Thus, under these conditions, the seal leaks about 3 ml/min.

LABORATORY TEST RESULTS

Testing was performed for all three structural designs as a part of the design development process. During testing, both inside and outside leakage were measured. Based on temperature rise, observations about touching were made. After each test, observations about film thickness and contact were made based on
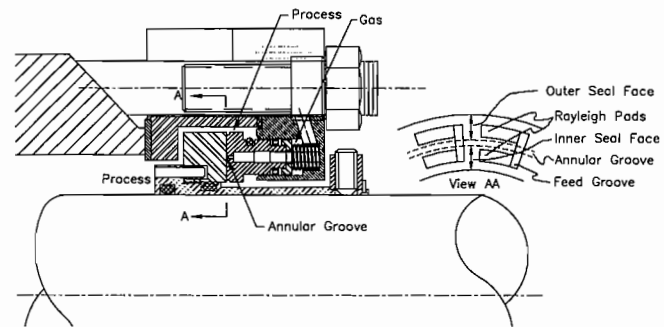

Figure 9. Loss-of-Gas/Overbalanced—Third Structural Design.

an inspection for contact marks. The most important measurement was the leakage. After some experience, using inside and outside leakage alone, one could anticipate the conditions of operation of the seal.

Some of the final test results for design 3 above are reported in Table 2. One should carefully note the relationship between predicted leakage and measured leakage. One will see that the leakage trends are predicted. For example, cases 1, 2, and 5 show that leakage decreases as speed decreases indicating the film thickness decreases as predicted. At higher gas pressures, cases 6 and 7 show significantly increased leakage as predicted. Careful observation of case 3 suggests that when product pressure is low, the seal may be converging more than predicted (indicated by the inside leakage becoming lower than predicted). So, some adjustment of the cross section might still be made. Agreement is better for cases where product pressure remains higher.

Table 2. Performance Predictions and Measurements for Structural Design 3 ($p_i = 12.5$ psia).

Test	p_o psia	p_g psia	RPM	Predicted				Measured	
				h_{min} μin	ϕ μrad	Q_i scfh	Q_o scfh	Q_i scfh	Q_o scfh
1	82.5	112.5	3600	79	71	1.6	2.0	1.3	3.0
2	82.5	112.5	1800	66	68	0.9	1.3	1.9	1.4
3	26.5	112.5	1800	63	63	0.8	2.2	0.2	3.5
4	52.5	112.5	3600	75	61	1.3	2.7	1.7	4.0
5	82.5	112.5	0	36	63	0.2	0.4	0.6	0.9
6	262.5	312.5	3600	102	199	26.6	33.5	41	39
7	262.5	332.5	1800	87	179	19.6	34.1	25	55
8	12.5	112.5	0	44	70	0.3	1.3	1	6

There are several reasons why the numbers in Table 2 do not compare better. The prediction of radial taper is not exact, even though it is close. There is always some axial friction force acting on the seal, tending to either close it or open it, thus reducing or increasing observed leakage. Initial radial tapers may easily vary as much as 20 μradian to 40 μradian (one light band equates to 20 μradians in this design). So, from test to test there will be some inconsistency of results as parts are relapped and have a different initial radial taper. Of course the seal design itself must be tolerant to these factors. So, interpreted in this way, the results of Table 2 show that the leakage in all cases remains significant, indicating that there is an adequate running clearance. Post test inspection for drag marks verifies this observation.

FIELD TEST RESULTS

The coplanar coaxial gas seal has been installed in several field applications.

Chemical plant—Mixed chemical fluids in transfer pump
Mark II, Group 2, 1-7/8 shaft
Temperature: 250°F
Suction pressure: 2.6 psig
Process head pressure: to 100 psig
Nitrogen barrier gas pressure: 80 psig
Speed: 1750

Chemical transferring takes place in planned processing sequences. Due to the nature of the transferring cycle, conventional wet seals were running dry and sometimes had to be replaced after one batch. While the gas seal worked for a number of cycles over a two month period, apparently the product hardened at some point such that when the pump was restarted, there was a serious radial misalignment causing the silicon carbide mating ring (Figure 9) to hit the inside of the housing and break out the drive pin hole. The seal was rebuilt and put back into service with the caveat that it can withstand only limited radial misalignment (0.050 inch max). At the same time, gas supply pressure was increased to 125 psi because it appeared there were occasions when the stuffing box pressure may have exceeded the 80 psi previously used.

Chemical plant—Corrosive resins in transfer pump
Mark III, Group 1, 1-3/8 shaft
Temperature: 250°F
Suction pressure: 2.6 psig
Process head pressure: to 100 psig
Nitrogen barrier gas pressure: 80 psig
Speed: 3600

This is a similar application to above, but the operation is such that after the batch is completed, both pump discharge and pump inlet are closed off and the pump continues to operate essentially dry. As a consequence, the pressure in the pump chamber gradually approaches the gas supply pressure so that the seal then operates with no differential pressure across the outside seal, and operates in its loss-of-gas mode where the mating ring is pushed to the right (Figure 9) and overloads the faces. This leads to some wear and deterioration of the gas sealing faces, so this seal has been modified to capture the mating ring so that there is less load on the faces, and the seal can operate at the condition where process pressure is equal to gas supply pressure over extended periods of time. This application points out an important issue in applying double gas sealing technology. If one is sealing to a closed (and relatively small) volume, be it a blocked off pump or a small volume closed process, the gas leakage from the seal will pressurize the system so that eventually the gas seal no longer operates with a pressure differential.

Chemical plant—Vinyl pyrrolidone bottoms slurry
Dean Brothers PB 232 Pump, 1-3/4
Temperature: 368°F
Suction pressure: 13 psig
Discharge pressure: 43 psig
Nitrogen pressure: 40 psig
Speed: 1750

Conventional seal life for this application was about one month. The coaxial coplanar seal was installed in March 1998, and has been running without failure since that time.

Pulp and paper mill—68% black liquor—abrasive
Worthington D1011 recirculating pump, 1-7/8
Temperature: 200°F max
Suction pressure: ?
Discharge pressure: 40 psig
Nitrogen pressure: 60 psig
Speed: 1800

Here conventional double seals were lasting about two months. A coaxial coplanar gas seal was installed and ran successfully for three months. At that time the pump was shut down for two weeks.

Apparently the fluid solidified around the seal so that when the pump was restarted, the mating ring drive pin was sheared and this partially broke the mating ring. The seal was repaired and placed back in service, with the caveat that some care must be taken during shut down to flush the seal chamber out with water and to leave the gas supply on so that the chance of sticking will be minimized.

LIFE AND LIMITS OF APPLICATION

Field experience above plus additional laboratory measurement has led to several additional findings.

● In applications where a pump is run in a closed off mode where gas pressure can accumulate on the process side, causes design 3 to contact and wear (equal gas and process pressures) because of its loss-of- gas design feature. Design 2 is used in such situations where it has been shown under testing that the seal operates with no contact when gas supply pressure equals process pressure.

● Some laboratory tests have been made where the ingress of sealed liquid across the faces has been monitored. It is observed that when the seal is pressurized but not rotating, there is some ingress across a portion of the outer sealing dam. Thus, when sealing sticky or highly viscous fluids, this fluid layer must be sheared on restart. This creates a high torque load on the seal parts, and design must address this issue. If the sticky or viscous fluid does not harden and can flow, the seal will restart rotation and resume normal operation. For example, experiments show that seal faces completely wetted in oil will restart and resume normal operation even though there will remain a minute oil film on the faces.

● Based on observations to date with *continuous operation*, expected life is high. One would expect seal failure to occur by chemical attack or material degradation, and this could be years for some benign environments. For applications with *start/stop* operation, because some amount of fluid enters the seal interface during stoppage, life may be limited because:

• The fluid hardens on the interfaces thus preventing proper fluid film load support (touching), or

• Sticking together may cause a mechanical failure on restart if the sticking torque is too high.

On the other hand, if the fluid is simply viscous and does not harden, then the seal could recover normal operation on restart for any number of cycles. *Polymerizing fluids* in conjunction with start/stop are particularly difficult because during stopping, some small amount of fluid enters the outer part of the interface. When the seal is restarted, this liquid may polymerize. This will cause an irreversible buildup of hard material on the face and eventual seal failure.

CONCLUSIONS

A coaxial coplanar double gas seal design has been developed, laboratory tested, and field tested. The idea of using two coaxial coplanar active gas seal faces coupled to the same mating ring and primary ring works well. Leakage is consistent and predictable. Laboratory testing confirms noncontact operation. Field testing has shown reliable operation of the seal under normal operating conditions, but points out several issues related to abnormal operation that can cause gas seal failure. These observations suggest that one must carefully evaluate the application before applying this gas sealing technology.

NOMENCLATURE

B_i	=	Effective balance ratio for the inside seal
B_o	=	Effective balance ratio for the outside seal
h_f	=	Feed groove depth
h_p	=	Pad depth

h_{min}	=	Minimum film thickness
k	=	Axial film stiffness
n	=	Number of face features
p_o	=	Outside or process pressure
p_g	=	Gas supply pressure
p_i	=	Inside or atmospheric pressure
Q_i	=	Volumetric leakage to the inside
Q_o	=	Volumetric leakage to the outside
r_1, r_2, r_3, r_4	=	Radii of the outer and inner faces
r_d	=	Dam width to face width fraction
r_p	=	Pad width to face width fraction
r_{bo}	=	Balance diameter for the outer seal
r_{bi}	=	Balance diameter for the inner seal
rpm	=	Rotational speed
t_f	=	Feed groove circumferential length to length of one period fraction
t_p	=	Pad circumferential length to length of one period fraction
μ	=	Viscosity of the gas

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