HIGH PRESSURE NONCONTACTING MECHANICAL SEAL FOR API APPLICATIONS

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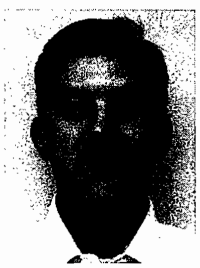

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ABSTRACT

Since the introduction of noncontacting, gas lubricated, dual pressurized mechanical seals for process pumps over five years ago, they have primarily been applied to ANSI pumps and other applications where the barrier pressure was below 300 psig. This pressure limit was not due to a limitation in material strength or face pattern technology as noncontacting seals have been applied in compressors at pressures well above this. The challenge came in fitting a dual pressurized cartridge seal into the limited size seal cavity available in most process pumps. The radial and axial dimensions of a pump seal chamber limit the size and shape of the seal faces as well as the supporting hardware needed to make a convenient cartridge seal design.

A pressure responsive primary sealing ring geometry has been developed to allow a dual pressurized sealing arrangement to routinely handle pressures up to 600 psig. This pressure capability enables the dual pressurized gas seal to be utilized in applications generally found in the API market where contacting liquid lubricated seals are the commonly used options. Discussed herein are the design, theoretical results obtained through the use of finite element analysis modelling and test results. Also discussed are the results of a modified API 682 qualification test and details of field installations.

INTRODUCTION

The noncontacting gas pressurized dual mechanical seal arrangement has been embraced by industry as a viable solution to many pump sealing applications. This type of seal has offered industry another option when facing hazardous emissions restrictions. It has also contributed to savings in maintenance and energy consumption and increased reliability and mean time between planned maintenance (MTBPM). On applications where single mechanical seals cannot be used, previous solutions utilized pressurized and nonpressurized dual mechanical seals and sealless pumps. In some cases, these solutions have inherent shortcomings that the dual gas seals address.

Dual pressurized and nonpressurized liquid lubricated seals both require the selection of a compatible barrier or buffer fluid as well as a liquid support system where pressures and levels need to be maintained. Seal flush rates have to be properly chosen and applied as the seals operate in a contacting mode resulting in heat generation, wear, and energy consumption from face torque. As liquid seals rely on the liquid being pumped to lubricate and cool the seal faces, liquids that are pumped in conditions near their vapor point can cause problems for the seals. The liquid tends to vaporize across most of the seal face, resulting in excessive heat and wear of the seal. Nonpressurized seal arrangements have utilized a noncontacting seal as the outboard seal to obtain the advantages that this seal offers. However, the inboard seal is typically still a liquid lubricated seal susceptible to the problems listed above. Sealless pumps often require extensive monitoring equipment and are limited to certain fluid types based on viscosity and solids content. Numerous previous papers have discussed the advantages and operation of various gas seal designs that have made them a popular alternative. (Wasser, et al., 1994; Adams, et al., 1995; Young, et al., 1996; O'Brien and Wasser, 1997).

While dual gas seals have been used with success along with contacting, liquid lubricated seals, the current available designs are directed toward the ANSI market and are limited to barrier pressures below 300 psi. This leaves contacting seals as the available option for higher pressure applications. Such high pressure applications generally exist in the API and refinery markets as well as in chemical plants, where federal emissions legislation has had an important impact. The limitations of the existing gas seal designs are generally due to the design and shape of the seal faces. Excessive distortions of the seal faces due to pressure, result in a decrease of the gas film, thus causing face contact. A traditional method of handling higher pressure by using larger cross section parts is restricted due to limited available space in the seal chamber. A patented pressure responsive seal face geometry has been devised to overcome the pressure limitation and make the option of using noncontacting dual pressurized seals, with its numerous benefits, available to a larger market.

GAS SEAL TECHNOLOGY

Inherent in understanding the benefit of this advancement as well as the use of gas seals in general, it is important to have an understanding of how the gas seal works. The geometric design concept is similar to contacting wet seals with the addition of special seal face features that enable the gas seal to operate noncontacting on a film of gas, thus eliminating liquid support systems. To achieve face separation while running, gas seals create pressure across the seal face using gas. Special face features are used to accomplish this. One face in a mating pair of faces is typically lapped flat and smooth, while the other face incorporates the lift generating design (Wasser, 1993). This design can be of various shapes, including T-slots, V-slots, and wavy faces. The seals discussed here will use spiral grooves.

These special grooves are used to create hydrodynamic lift. The gas seal faces ride on a film generated by the spiral grooves while the shaft is rotating. These grooves are recessed into one of the mating face pairs, usually the harder material as shown in Figure 1. A sealing dam, or ungrooved area of the face helps to restrict gas leakage and form a seal while static and during reverse pressure incidents. As the seal rotates, gas flows into the spiral groove and is compressed. At the sealing dam, it expands. The combined film pressure results in an opening force that balances the closing force when the faces have reached the operating film thickness and the faces separate (Wasser, et al., 1994). The pressure profile created is shown in Figure 2. This separation greatly reduces face temperature and reduces horsepower consumption compared with a conventional contacting seal. Figure 3 shows the theoretical prediction of the interface pressure generated by the spiral grooves.

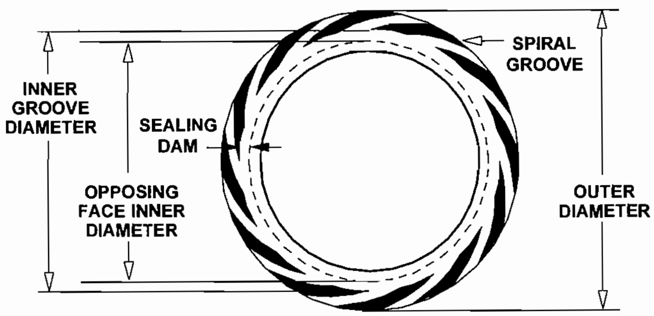

Figure 1. Typical Spiral Groove Sealing Surface.

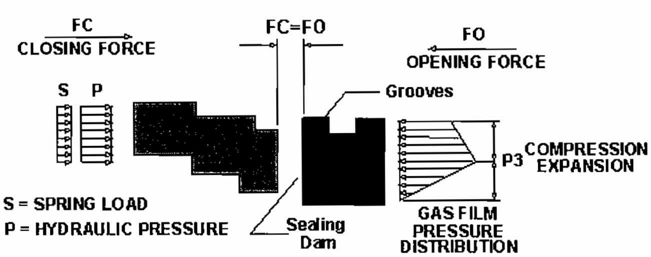

Figure 2. Pressure Profile of a Noncontacting Gas Seal.

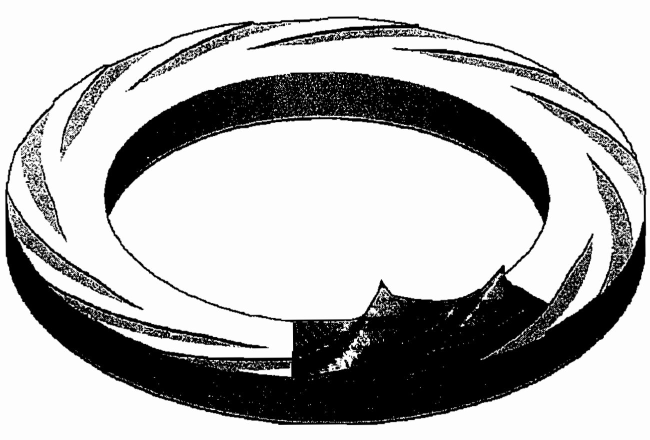

Figure 3. Theoretical Prediction of Pressure Generated by Spiral Grooves.

Seals incorporating such face features can be utilized as single seals when operating in a gas such as with a fan or blower, or they can be used as a backup seal in a nonpressurized dual arrangement. The seal design as discussed here is incorporated into a back-to-back dual pressurized cartridge seal arrangement. The barrier area is pressurized dead ended with an inert gas at a pressure about 50 psi above pump stuffing box pressure. As the seal operates, a small amount of this gas is moved across the seal faces into the product and out to the atmosphere. The barrier support system is simplified versus the requirements of a wet dual seal whose typical arrangement is shown in Figure 4. All that is required is a clean, dry inert gas source that can be regulated. Pressure and flow switches, flowmeters, and accumulators can be added as desired (O'Brien and Wasser, 1997).

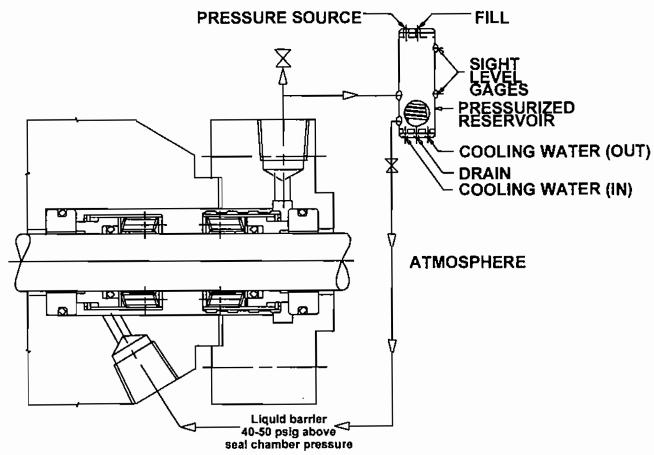

Figure 4. Typical Dual Pressurized Liquid Lubricated Seal Arrangement.

DESIGN

Dual pressurized gas seal designs for pumps generally have used a primary ring shape that is similar to wet seal designs. The shape is dictated by the limited radial space in pump seal chambers and the properties of the long thin cylindrical shapes to handle pressure with little distortion. Space is limited not only due to the shape of the pump, but by the desire to package the seal in a cartridge arrangement to make installation easier. A cartridge design includes hardware to hold all the seal parts together and to create a barrier chamber that can be pressurized. Pressure exerted on the seal faces by the barrier pressure typically causes a seal face to distort around its centroid so that the outside diameter of the face deflects axially toward the mating face more than the inside diameter. Contacting wet seals can tolerate more of this distortion as they have a counteracting distortion due to temperature rise caused by the contact. In contacting wet seals, the balancing of these two distortions allows the seal faces to run parallel to each other, which is the desired mode. Noncontacting gas seals experience the same distortions from pressure; however, since the mating faces are not touching during operation, there is very little heat buildup to provide compensating distortion. Therefore, gas seal designs have used different methods in their design to counteract this distortion and allow the faces to run noncontacting.

Once barrier pressures reach levels around 300 psi, the pressure distortions are too great to overcome with current design methods. This distortion results in inadequate sealing while static, as well as excessive leakage while dynamic. Pressure distortions also cause seal face contact, resulting in heat generation and wear. Other traditional methods of compensating for higher pressures include hammerhead designs, which stiffen the seal face to resist distortion. Such designs proved successful for a limited pressure range; however, its benefit cannot be fully recognized again due to radial space constraints. As mentioned before, gas seal designs have been used at very high pressures on compressors and turbines. The solution there was to use a large radial cross section with a wide face width, as shown in Figure 5. The radial and axial space available in ANSI and API pumps does not allow the use of faces of this design. Other methods used for high pressure gas seals include using stiffer face materials. This leads to using hard face materials such as silicon carbide and tungsten carbide. These are run against a mating surface that is of a similar material. A major disadvantage of the hard face versus hard face arrangement was its inability to handle upset conditions. Any face contact generates excessive amounts of heat and destroys the faces in a short period of time. Using a softer material, such as carbon, for one of the faces allows the seal more flexibility in handling incidental contact from startup and shutdowns as well as system upsets that may cause temporary reverse pressurization.

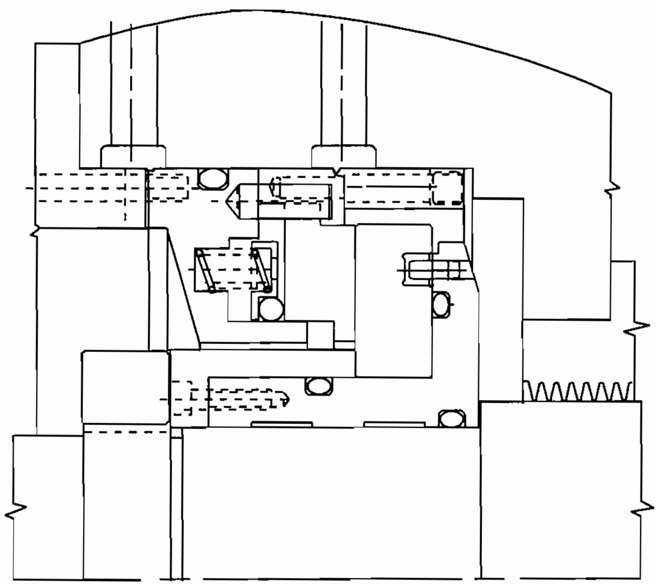

Figure 5. Large Cross Section Gas Compressor Seal.

The desired design for a seal to handle higher pressures above 300 psi in pump applications would maintain a similar design to existing gas seals for large bore pumps, illustrated in Figure 6. It would be in a cartridge arrangement with an inboard and outboard pair of seal faces and pressurized with an inert gas. Two sets of faces in a dual pressurized arrangement made for an inherently safer seal and allowed for operation with zero product emissions to the atmosphere. The rotating faces would be of a carbon material and the stationary faces would be of a harder material.

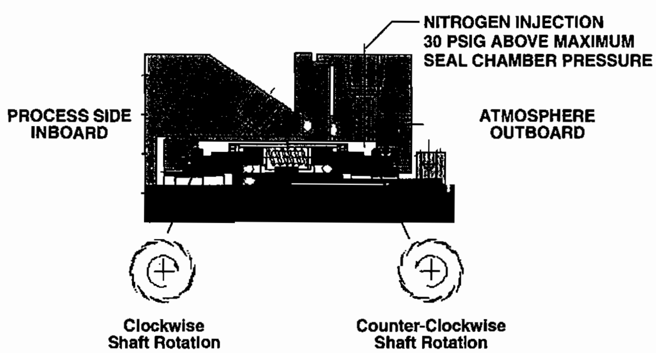

Figure 6. Noncontacting Dual Pressurized Gas Seal for ANSI Large Bore Pumps.

The method devised for the carbon faces to be able to handle the higher pressures was to incorporate a saddle shape cutout on the outer diameter of the primary ring. This general shape is shown in Figure 7. Typically shaped seal primary rings distort under pressure around the part's centroid. This causes the part's outer diameter to collapse toward the mating face, causing the seal face gap to become wider at the inner diameter of the seal face interface and to narrow at the outer diameter. A computer generated prediction of this distortion is shown in Figure 8. This distortion is not conducive to sealing capability because the effects of the spiral grooves pumping the gas across the face are diminished, possibly causing face contact and wear. The saddle shape cutout essentially enables the distortion of the front and back sections of the seal ring to be controlled, allowing the front section to remain nearly parallel to the mating face, while the back section is distorted as normal from pressure (Figure 9). This saddle shape cutout creating a thinner center radial section makes the sealing ring more compliant, in that distortion of the back portion of the ring does not cause the front section to also distort in the same direction. The front section essentially distorts in the opposite direction, allowing the faces to come into the desired orientation for efficient face separation. Additional distortion of the front section results from pressure buildup in the seal interface that is caused by the pressure buildup in the spiral grooves. Therefore, the location of the saddle shape cutout as well as its width and depth, along with the location of the peak pressure created by the spiral grooves, will have varying effects on the performance of the seal. The angle of both the mating and primary ring faces distort with increasing pressure. The deflection in each ring occurs simultaneously to permit the faces to retain a desired relative angle at the high pressures.

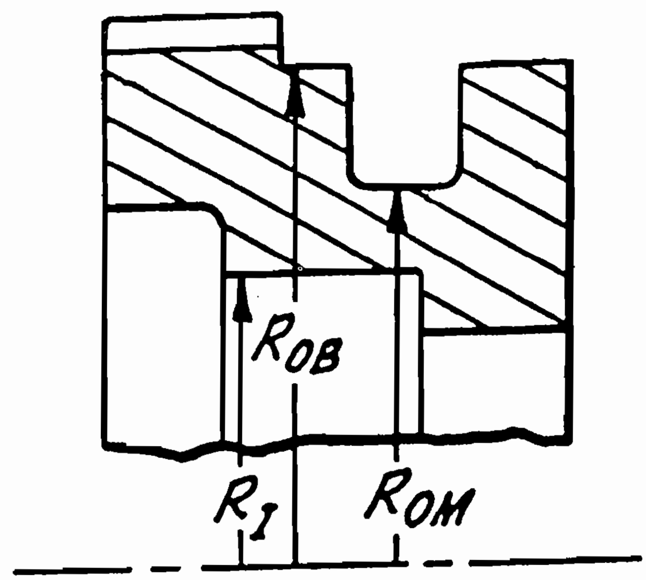

Figure 7. Saddleback Shaped Primary Ring.

The saddle cutout location relationships can be optimized to provide controlled barrier consumption throughout an operating pressure range as well as to reduce torque due to the high pressures at startup. More specifically, there are certain ratios in part thickness that are optimal. These ratios are defined as the radial thickness of the back section in relation to the radial thickness of the thinner center section of the seal face. If the middle section is too thick relative to the back, then only a minimal degree of flexibility in the middle section will make the primary ring insufficiently responsive to the pressure differential. If it is made too thin, the result is a loss of structural support between the back section and the seal face section, resulting in an increase in the radial tensile stress occurring on the middle section. This may result in the fracture of the ring, especially if the seal is subjected to numerous pressurization cycles. A metal filled carbon grade with a high modulus of elasticity and tensile strength was selected for the primary rings. This provides an advantage of being able to

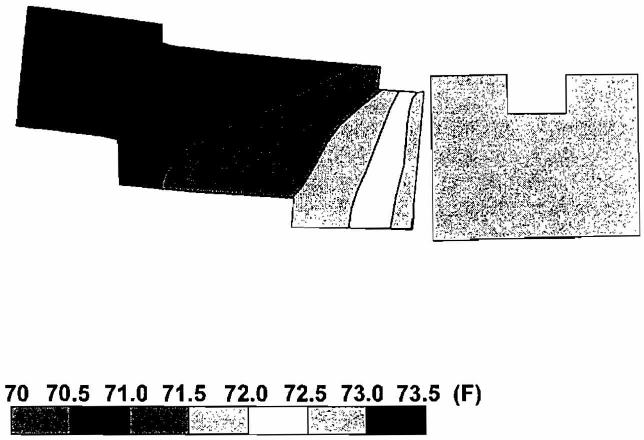

Figure 8. Predicted Face Distortions on Lower Pressure Gas Seal.

Figure 9. Predicted Face Distortions on Saddleback Shaped Primary Rings.

use a soft face material for one of the faces. This is important in providing long seal life and reliable operation. In case of face contact during upset conditions or reverse rotation, the carbon material running against a harder face will be much more tolerant than two hard faces running together creating heat and causing damage. The strength of this carbon material, however, is high enough to handle the stresses at the high pressures, especially in the thin area below the saddle shape cutout.

In the seal design, the saddle shape described above is employed in two primary rings that are positioned back-to-back and held in place with a metal retainer and snap rings. The retainer is attached to a sleeve that rides over the pump shaft. The sleeve is driven by a collar that contains set screws that are snugged to the pump shaft. The mating faces are made of a hard material such as tungsten carbide. These stationary faces have a spiral groove pattern machined into them. These faces are held into metal gland plates that form the cartridge and the barrier chamber. The secondary seals are O-rings. The placement of the O-rings in the set of inboard faces is positioned in relation to each other so that, in the case of reverse pressure or when the pressure in the pump stuffing box exceeds the pressure of the barrier gas, the seal faces will remain closed and restrain any process fluid from escaping to the atmosphere. This arrangement is shown in Figure 10. Due to the noncontacting feature of the seal faces while in operation, very little heat is generated by the seal during operation, in contrast to contacting seals. Therefore, the temperature limits of this type of seal are restricted only by the capability of the elastomers that are used as secondary seals. This seal design is similar in arrangement to the gas seals used in large bore stuffing boxes in ANSI services. Pump stuffing box chambers in API pumps are typically large enough to accommodate the cross section of this seal.

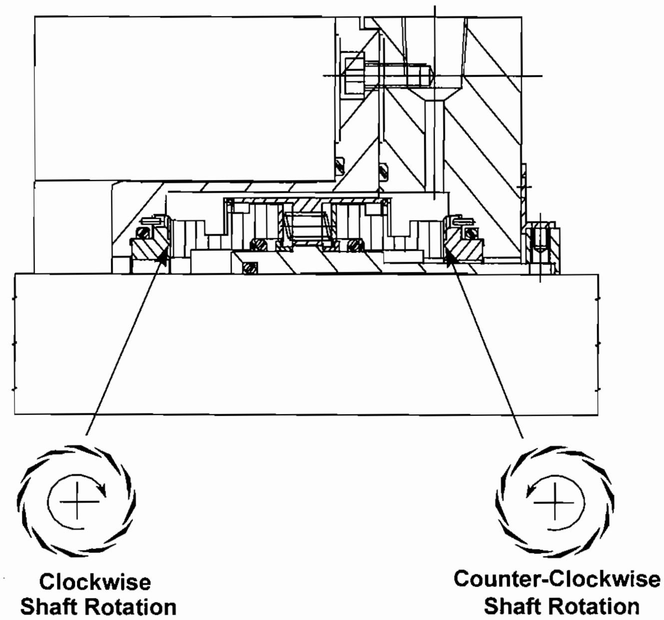

Figure 10. High Pressure Dual Gas Seal for API Pumps.

The dual pressurized seal requires a barrier gas supplied at a pressure higher than what is in the pump stuffing box. Any available source of inert gas may be used including nitrogen, air, and steam. This gas can be provided from an available plant header, compressor, or bottled gas. In some applications, a nitrogen bottle may be exhausted fairly rapidly, so this source is typically used as a backup to the other systems. As many plant headers are below 300 psi, an air pressure amplifier can be used to boost the supply to the required pressure. These are variable pressure, variable flow devices powered by the same air that they amplify. A barrier support system panel can be fabricated into a convenient package to regulate and control this barrier gas, as shown in Figure 11. Gas is brought to the panel and filtered through a coalescing element to remove moisture and large particles. The gas can then be regulated to the required pressure and its flow can be read from a flowmeter. A low pressure alarm can be incorporated to warn of loss of barrier pressure. A high flow switch can also be added to indicate and warn of seal problems. The gas is then dead ended into the seal cartridge.

Although the seal as described and tested has been used in a dual pressurized seal arrangement, the same concept of the pressure responsive primary ring could be used in numerous other arrangements. A single seal arrangement could be used when sealing a gas in a fan or blower. In a tandem arrangement where the inboard seal is seeing high pressures, it could be used as the outboard standby seal. Its high pressure capabilities would be able to handle this high pressure, in the case of an inboard failure, for a period of time until an operator could shut down the unit. The concept could also be incorporated into contacting wet seal designs to help reduce face pressures, heat generation, and wear that tend to shorten the life of these types of seals.

ANALYSIS

To aid in the optimization of the design, CSTEDY℠, a proprietary finite element analysis (FEA) computer program was

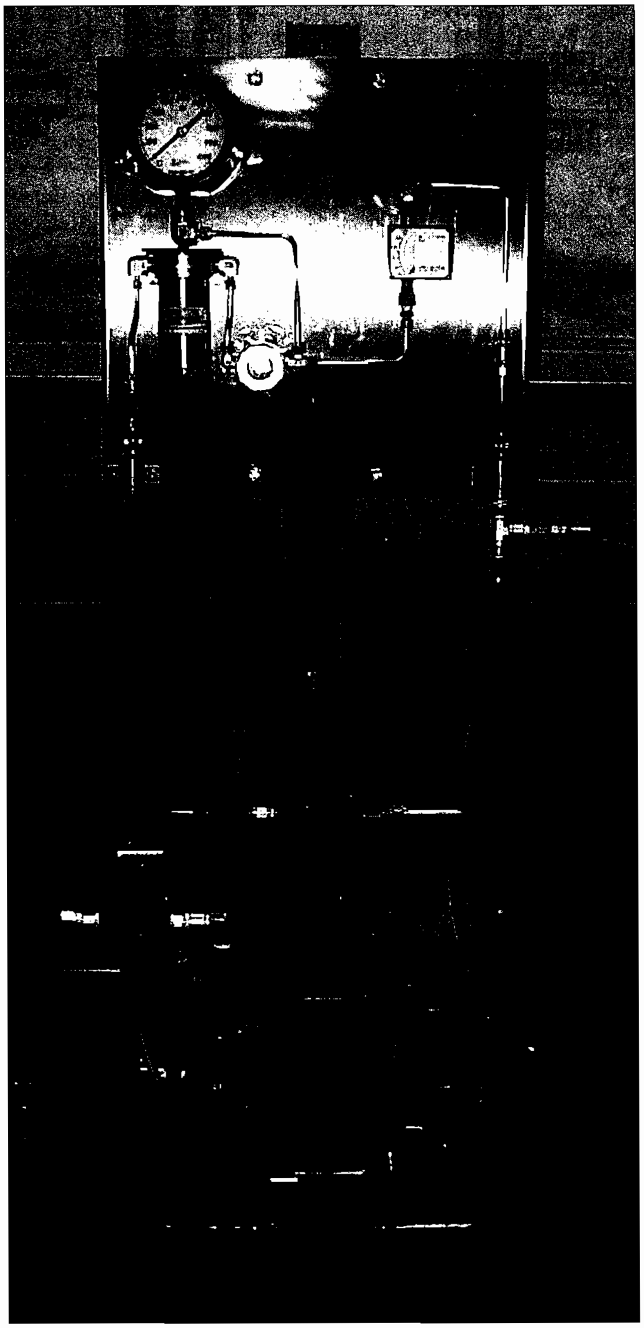

Figure 11. Gas Barrier Supply Panel for Seal with Amplifier.

utilized. This program takes into account the effects of pressure, temperature, shaft speed, materials of construction, sealed fluid, and spiral groove geometry. The program predicts radial face profile, film thickness, gas consumption, stress, and distortions due to temperature and pressure. For the analysis, the barrier gas was nitrogen and the process was water, in order to correspond with what was used on the test stand. Through iterations in design, the program was used to optimize the initial face profile of the seal faces and location and size of the saddle shape cutout. Results from these iterations were used to strike a balance between seal face separation and barrier gas consumption. Most of the barrier gas used is released to the atmosphere because of the large pressure differential between the barrier pressure and the atmosphere. Only a small amount enters the process. On the process side, there is typically a 50 psi differential between the barrier pressure and the process pressure. However, it is advantageous to minimize barrier consumption so as not to overwhelm the system used to supply the seal. Testing of the seal was performed to correlate results with the program predictions in order to verify assumptions and to enable new sizes to be designed with confidence without having testing done for each one. Test results enabled the establishment of a minimum film thickness necessary, as predicted on the computer model, to ensure proper separation of the seal faces and avoid contact during operation. Face distortions and barrier gas consumption vary with pressure and speed, so each computer model was run through the full operating range of the seal operation to ensure consistent performance. The current standard operating ranges of the seal are barrier pressures from 300 psi to 600 psi, at pump speeds of 1800 rpm and 3600 rpm. As stated before, temperature limitations are dictated by the O-ring material used.

TESTING

Seal Design Testing

The test setup consisted of a motor driven variable speed shaft that passed through a housing. Bolted to the housing was a cartridge arrangement dual pressurized mechanical seal made to fit over a 3.000 inch shaft, consisting of two carbon primary rings incorporating the saddle shape, and two tungsten carbide mating rings with a spiral groove pattern machined into them. A single contacting wet seal inboard of the seal cartridge formed a chamber through which water would be passed from a tank with a circulating pump to simulate a process fluid in a pump. A cap was placed over the end of the shaft with an opening for a flowmeter. This arrangement was used to measure barrier gas consumption from the outboard seal. Barrier gas flow was also measured as it entered the seal cartridge. The difference between these two readings gives the amount of gas that would be going into the process. Typically 80 percent of the barrier consumption was through the outboard seal. This test setup is shown in Figures 12 and 13. Barrier pressure was supplied by an air pressure amplifier and was set at a level around 50 psi above process water pressure. Pressure to the amplifier was supplied by a shop air header at 300 psi and could be boosted to 1000 psi by the amplifier. The amplifiers are sized per the application to limit the number of cycles per minute and ensure reliable operation for several years. Temperatures of the faces were monitored with thermocouples inserted into the seal barrier cavity at close proximity to the sealing interfaces. Process water temperature and pressure were also monitored.

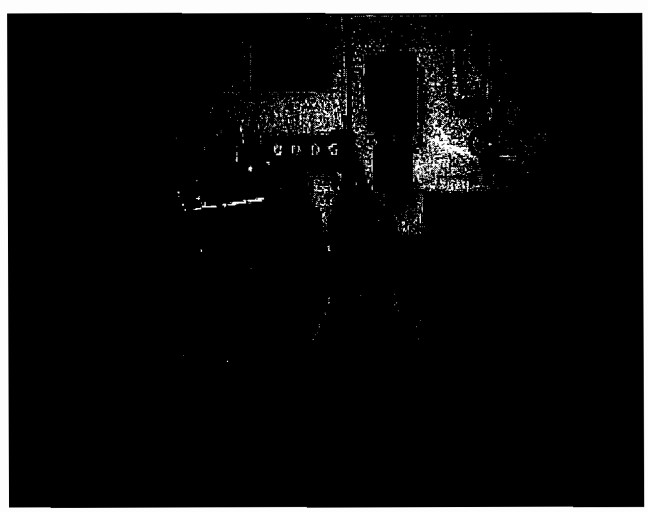

Figure 12. High Pressure Gas Seal Test Rig.

Tests were run in this configuration, varying speeds from 1400 rpm to 3600 rpm, and with barrier pressures ranging from 300 psi to 600 psi. Temperature rise at the seal faces averaged 10°F above

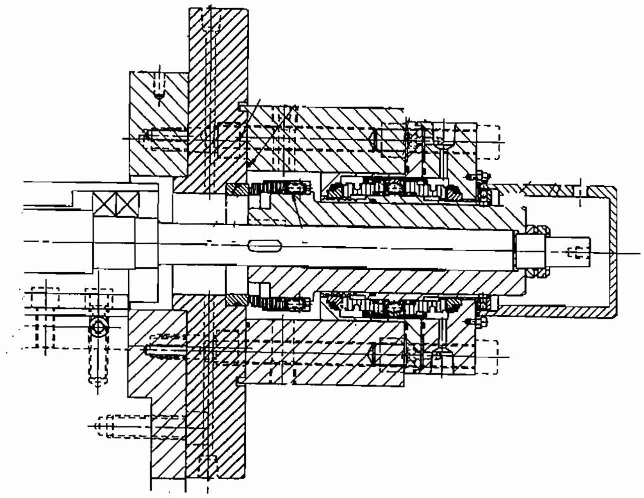

Figure 13. High Pressure Gas Seal Test Rig Arrangement.

the ambient temperature. Barrier consumption varied predictably with changes in pressure and speed. Testing variables included varying initial face profiles on the mating rings and primary rings, in order to minimize barrier consumption and to correlate results with the computer model. Varying shapes and location of the saddle shaped groove were tested. The seals were also run using fluoroelastomer, perfluoroelastomer, and TFE elastomer O-rings. Throughout all the varying conditions, including numerous starts and stops, the seal faces showed no contact. To simulate a loss of barrier gas, the seal was subjected to reverse pressures of 600 psi under static conditions. There was no loss of seal integrity and no release of product to the atmosphere.

The computer model was used to predict performance at higher pressures and showed good results, so the same seal arrangement was then run at selected pressures to investigate the seal's capabilities and ensure a safety factor for the design and the materials. After around 80 hours of dynamic testing, the test results were similar to the predicted model, with the seal faces showing no contact. Temperature rise at the seal faces was minimal.

The geometric characteristics of the saddle shape determine the amount of barrier usage while static and in operation. For seal faces that are shut under static conditions, high barrier pressures result in large face pressures, causing high levels of torque at startup. This can cause slight scuffing of the seal faces as they contact, while the shaft is running slow during startup and shutdown. Lower seal face balance and spring loads can be used to minimize the damage done during these periods. The pressure distortion of the primary ring with the saddle shape cutout can also be utilized to generate some hydrostatic liftoff. This will result in some barrier consumption while the seal is static. Typical static consumption rates in testing were around 1 scfh. Applications of the seal are developed closely with the customer to optimize barrier consumption with respect to the system and what it can handle. Most applications are continuous services or include standby conditions where liquid is flowing through the pump, carrying with it any barrier gas leaked into the pump. Therefore, the gassing up of a pump causing startup difficulties has not been a problem. Optimizing the seal performance is accomplished by controlling this distortion while minimizing barrier consumption. Operations with long continuous runs would have less need for hydrostatic design than an operation with many starts and stops.

Seal Development Testing

Testing was then performed to simulate an operation where the pump was started and stopped once a week. The same test setup was used as described previously using a seal for a 2.500 inch shaft. Process water was set at 510 psig and barrier pressure was 620 psig. The seal was run at 3600 rpm and saw 150 starts and stops, in addition to 350 hours of dynamic running. The seal faces were in excellent condition and showed no signs of face contact or damage, as can be seen in Figure 14. During operation, the temperature near the seal faces measured only a two to three degree rise over the ambient temperature. Total barrier consumption was measured at 22 scfh to 24 scfh, with 3 scfh to 4 scfh of this entering the process. The larger amount of the barrier leakage was to atmosphere because of the larger pressure differential between the barrier pressure and atmosphere, versus the inboard seal differential of the barrier pressure and the process pressure. The number of starts and stops equates to about three years of operation for this given application, with no wear shown on the seal faces.

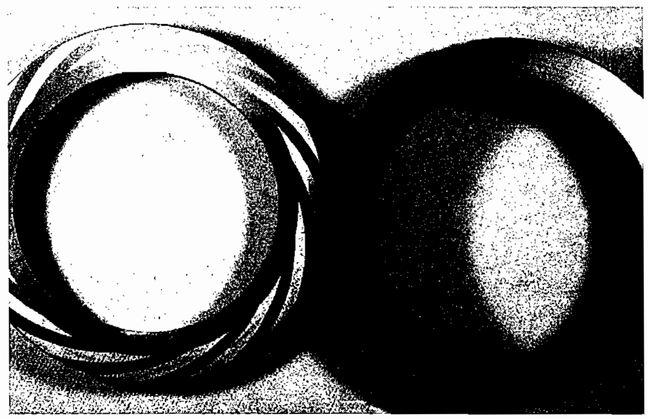

Figure 14. Seal Faces After Start/Stop Testing.

Similar testing was done on a seal for a 3.000 inch shaft. The test setup again was the same. Process water pressure was at 450 psig, while barrier pressure was set at 500 psig. The seal was run at 3600 rpm and experienced 76 starts and stops along with 190 hours of running. After disassembly, the seal faces showed no face contact. Total barrier consumption was measured at 23 scfh to 25 scfh, with only 3 scfh to 4 scfh of this entering the process.

Noncontacting seals are currently not specifically addressed in API 682 (1994), the specification directed toward new pumps purchased for use in the petroleum refinery services. However, future revisions most likely will include specifics on this seal type. As a large number of applications for this type of seal will be in the API market, the seal was tested using a modified API 682 qualification test procedure.

The test was run in a dedicated testing facility capable of handling hazardous process fluids. The seal used fit over a 3.000 inch shaft and was arranged, as in all the previous tests, in a dual pressurized arrangement. The test rig and seal arrangement is shown in Figures 15 and 16. The seal faces used were carbon versus tungsten carbide with fluroelastomer O-rings. A wet contacting seal was again used inboard of the test seal to form a chamber in which to run the process. The process used in these tests was liquid propane, which has a vapor pressure of 152 psig. This was circulated at a rate of 5 gpm through the chamber by means of a circulating pump, at a pressure of 250 psig, and a temperature of 90°F. The barrier gas used was nitrogen pressurized to 300 psi supplied by nitrogen bottles. The shaft speed was 3600 rpm. Nitrogen consumption was monitored along with seal face and process temperatures. Leakage to the atmosphere of the propane was monitored by an organic vapor analyzer that provides emissions data in accordance with EPA Method 21.

As specified for the API 682 seal qualification test, the first dynamic test was run continuously at the above conditions for 100 hours. The seal was then held static for 15 minutes. These results are in Figure 17. The seal was then run through a series of five dynamic cycles in which the pressure on the propane was released so that it was flashed to a gas. This took only a few minutes. Pressure was then reapplied until the propane returned to a liquid state. This process can take up to an hour. This process of flashing and repressurizing the propane was repeated five times. After this cycling, a static test was done again before the seal was disassembled and inspected. These results are shown in Figure 18. This procedure was repeated for barrier pressures of 400 psi, 500 psi, and 600 psi, each time with the propane pressure being 50 psi less than the barrier pressure. For each test, the seal faces showed no contact or damage. There was no leakage of propane to the atmosphere and the seal chamber temperature averaged an increase of 5°F above ambient temperatures. The only deviation from the standard API 682 qualification test was that a procedure of turning off the flush to the seal was skipped as this is a step specific to wet contacting seals.

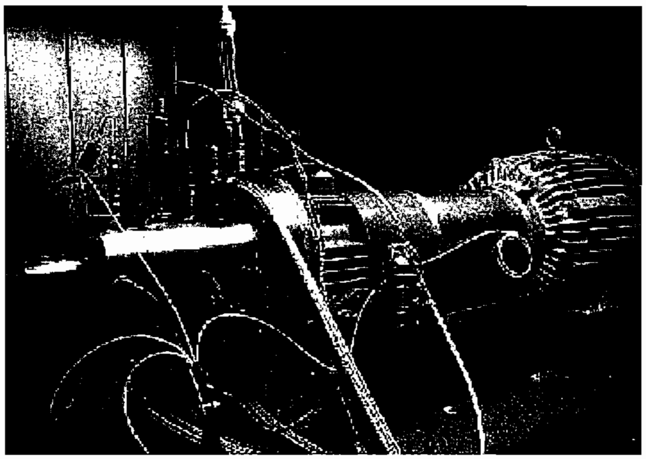

Figure 15. High Pressure Gas Seal Propane Test Rig.

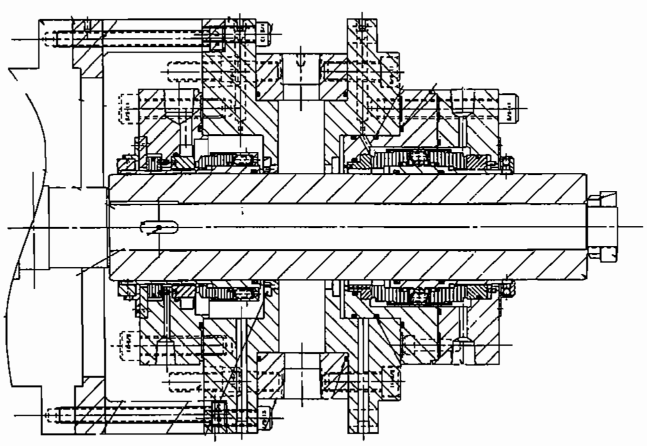

Figure 16. Propane Test Rig Arrangement.

FIELD EXPERIENCE

To date, there are over 25 different applications on which the seal has been successfully installed. Shaft sizes on which they have been installed range from 1.750 inch to 4.750 inch, with barrier pressures ranging from 300 psi to 600 psi. The pump speeds have generally been 1750 rpm and 3600 rpm. The processes have included a wide range of hydrocarbons with temperatures ranging up to 500°F. A partial list of applications is in Table 1.

The first seals were installed at a chemical plant in Texas, in April 1997. They were on a pump with a 2.500 inch shaft handling hydrocarbons at 270°F, 3560 rpm, and a suction pressure of 550 psig. Over time, suction pressures increased, so barrier pressures were increased to 640 psig from 600 psig. Nitrogen pressure to the seals is provided by a plant header system with a

Table 1. Field Installations of High Pressure Gas Seal.

Shaft (in.)	Pump Model	Process Fluid	Suct. Press.	Dis. Press.	Speed	Temp. (F)
1.375	Union Pump VLK - 7 GPI	Light Hydrocarbons	445 psig	482 psig	3550	120
1.875	Bingham CAP 6x8x11.5	Propylene	320 psig	388 psig	3560	127
2.000	Union Pump HHS 1.5x2x10B	Carbamate	240 psig	400 psig	3550	212
2.500	Ingersoll Rand 6 x 16 JH	Benzene	378 psig	627 psig	3570	433
2.625	Goulds 3700 L 7th Ed.	Benzene	240 psig	500 psig	3600	485
2.750	Union Pump 3x4 MOB - 4 Stg	Tertiary Butyl Alcohol	230 psig	629 psig	3550	200
3.000	United WMSND-H 6x11	Propylene-Propylene Oxide	285 psig	985 psig	3600	124
3.031	Worthington T-6 HED 16 DS	Hydrocarbons	144 psig	403 psig	3540	283
3.250	Union Pump HOL 8x10x16	Benzene	378 psig	510 psig	3550	490
3.375	Union Pump HHS 6x6x8-26	Propylene	304 psig	412 psig	1760	123
4.750	Union Pump MRF 8x10 4 Stg	Unstabilized FCC Gasoline	260 psig	359 psig	1750	220

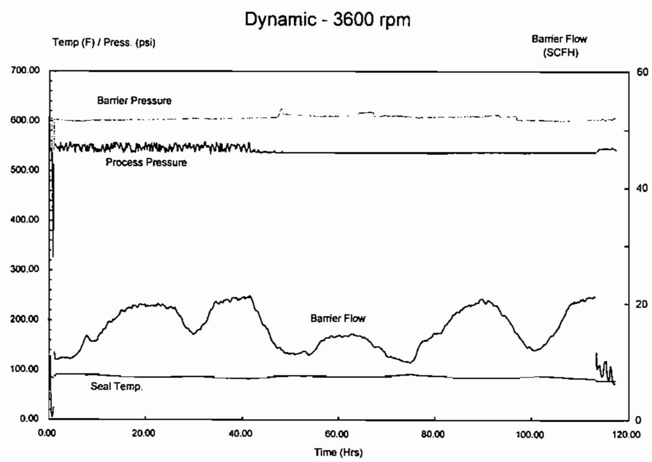

Figure 17. Test Results for Propane Dynamic Test.

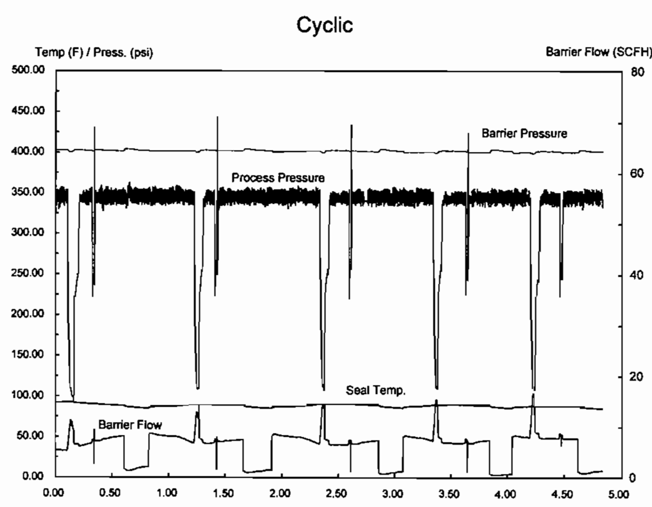

Figure 18. Test Results for Propane Cyclic Test.

booster to raise the pressure. Gas seals were utilized in this application as a solution to the emissions problems they were having with wet contacting seals. The contacting seals were also being run in a process that was close to its vapor pressure, resulting in product vaporizing across the seal faces, resulting in burned up seals and short life. The wet contacting seals were lasting less than six months. Lower pressure noncontacting gas seals were being utilized in the same plant on similar applications and have been running successfully for over three years. With this experience, it was decided to try the high pressure gas seals. Several other seals have since been installed at similar facilities on processes such as benzene, propylene, and tertiary butyl alcohol with similar operating conditions.

CONCLUSION

A new pressure responsive primary sealing ring geometry was developed that increases the operating range of the existing available gas barrier dual pressurized mechanical seals. This advancement now makes available to the API market the advantages of a noncontacting dual mechanical seal. This design achieves successful operation with barrier pressures up to 600 psi at normal pump speeds, without having to use hard face on hard face combinations. The controlled distortion of the primary allows carbon material to be used for the primary ring, eliminating concerns due to incidental contact causing heat generation and fracture.

The design was analyzed using advanced finite element modelling and was verified with numerous laboratory trials, including a modified API 682 qualification test on propane. Long term life was demonstrated with multiple start and stop operations while controlling barrier gas consumption. Success of the design was determined by minimal increase in face temperature and zero wear of the sealing faces. The distortion of the seal faces and resulting face separation is predictable and repeatable, resulting in barrier consumption that can be tailored for application requirements. The distortion under pressure, while static, can result in hydrostatic face separation that reduces torque at startup and minimizes face contact at slow speeds and in upset conditions.

Successful field experience for over a year has resulted in savings in maintenance of seals and barrier support systems, savings in energy consumption, and compliance with emission regulations. The cartridge arrangement allows easy installation and operation in conjunction with a simplified barrier support system.

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