ADVANCED GAS SEALING SYSTEMS FOR HYDROFLUORIC ACID ALKYLATION AND OTHER EXTREMELY HAZARDOUS DUTIES

by

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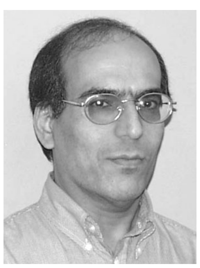

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ABSTRACT

Advanced gas sealing technology has been developed and successfully implemented into field operation for the purpose of exceeding environmental emission regulations, improving safety, and increasing mean time between repair (MTBR). Gas lubricated nonpressurized dual seals represent a significant advancement in the sealing of hazardous processes like hydrofluoric acid alkylation. The use of a low-pressure buffer gas allows the inside diameter (ID) spiral grooved outboard seal to be monitored during normal noncontacting operation and during upset conditions while achieving zero emissions at the pump. The arrangement represents a unique combination of contacting and noncontacting sealing technology that provides end users with unworn and like-new outboard sealing surfaces at the moment of primary seal failure. Noncontacting outward pumping spiral groove technology, which adapts to conventionally seal process upsets, represents a seal face design that is significantly more versatile than inward pumping noncontacting seals. End users have applied this technology throughout the world to increase safety and MTBR. Among other benefits are reduced costs of alkylation process system ownership as well as reduced potential for environmental impact. This paper provides a detailed description of seal face development, including the required adaptation from noncontacting to face contacting operation. Laboratory testing, seven years of field operation, and substantial end user benefits are also addressed.

INTRODUCTION

Thirty years ago spiral groove seal face technology was incorporated into mechanical seal designs for high performance gas compressors. This inventive technology originated from the science of thrust bearings. The success of noncontacting spiral groove seals has allowed the compressor industry to introduce their equipment into more challenging services. In 1992, the sealing industry was introduced to innovative gas seal technology for process pumps. This advancement in sealing technology revolutionized the industry while meeting stringent emission laws.

These tough state and local emission regulation laws have led to optimized single mechanical seals for low specific gravity products, which have successfully resulted in emission levels well below 1000 ppm. Although these single full-face contacting arrangements have produced positive results for many applications, their use is limited when sealing extremely toxic, corrosive, and volatile applications such as those found in hydrofluoric acid (HF) units.

Mechanical seals used in HF processes are flushed with light hydrocarbon liquid (such as isobutane) for cooling the seal faces. These seals also have to handle abrasive particles (ferric fluoride) that result from the corrosive nature of the HF acid on the stainless steel pumps and support piping. In addition, these seals must be able to handle fluctuating operating conditions, which can occur during startup, process upsets, and seasonal temperature changes.

To make a substantial improvement in emissions control and safety requirements, a tandem seal was developed in 1994 that incorporates the science of spiral groove technology. This tandem seal consists of a robust unpressurized dual seal typically running in an isobutane system barrier. In addition, low-pressure nitrogen buffer fluid on the outboard gas seal is used to meet near zero emission requirements and enhance safety. This seal uses a specially designed outward pumping spiral groove array to run noncontacting and ensure a "new until needed" containment seal. The noncontacting seal directs any isobutane emissions to the refinery's vapor recovery system, effectively achieving emission requirements as well as significantly enhancing safety. This innovative seal design was first developed in 1994 for a major refinery in the Midwest. Since its development, refineries on a global basis have used this tandem seal arrangement to improve emissions control, safety, and equipment reliability.

HYDROFLUORIC ACID

The term "hydrogen fluoride" or "HF" refers to anhydrous (water-free) hydrogen fluoride. "Hydrofluoric acid" or "aqueous hydrofluoric acid" refers to the solutions of HF in water. For the purpose of this paper, the HF shall mean both anhydrous fluoride and hydrofluoric acid.

HF is a pure, colorless liquid. When encountered in a refinery, it will be a brownish color due to contamination. HF has the same density as water. At atmospheric pressure, it boils at approximately 68°F and freezes at minus 117°F. HF is very soluble in water and, to a limited extent, in hydrocarbons and chlorinated solvents. HF rapidly vaporizes at atmospheric pressure and temperatures above 68°F, releasing a cloud of white or yellowish steam-like vapor, which has a distinct sour taste and irritating odor. HF or HF mixtures are extremely corrosive and will attack all materials containing silica such as glass, porcelain enamelware, asbestos, and certain cast-irons. It will also attack slag in poorly welded carbon steel as well as wood, rubber, and most plastics, except Teflon® and Kel-F®. Neoprene and polyvinyl chloride (PVC) are relatively resistant to the acid.

Note: See APPENDIX A for the effects of HF on the human body.

SAFETY PRACTICES WHEN DEALING WITH HF

Each refinery has developed their own safety practices for protecting their employees. The items listed here are some of the general/universal safety practices followed by all refineries using HF.

Among the more universal protocols: always minimize your exposure to HF acid and wear the appropriate protective equipment at all times. Material and equipment must be neutralized before transferring it from the unit. All tools that have been used on equipment containing HF must be neutralized. All protective clothing that could possibly have been contaminated must not be worn outside the HF unit. If HF acid or vapors become evident while an employee is in the HF unit, the area should be evacuated.

OTHER HAZARDOUS FLUIDS

Ethylene oxide is an odorless gas at room temperature and normal pressure, but is a liquid with an ether-like odor at 50°F. It is completely miscible with water, alcohol, acetone, benzene, ether, and most organic solvents. At room temperature, it is an extremely flammable and reactive gas. Its vapors are inflammable and explosive. It is highly reactive and potentially explosive when heated or in the presence of alkali metal hydroxides and highly active catalytic surfaces. Incomplete combustion releases carbon monoxide. It has been used in flame-retardants and to accelerate the aging and seasoning of tobacco leaves.

Propylene oxide is a volatile, flammable liquid that is soluble in water, alcohol, and ether. It is highly dangerous when exposed to heat or flame. It has a violent reaction with hydrogen chloride, chlorosulfonic acid, hydrogen fluoride, and oleum. It should not be stored in the presence of acids, bases, chlorides of iron, aluminum, and tin, or peroxide of iron and aluminum; any of these may cause violent polymerization. When exposed to flame, propylene oxide will burn with a hot flame and may explode if confined. Polymerization may occur due to high temperatures or contamination with alkalis, aqueous acids, amines, and acidic alcohols.

Note: See APPENDIX A for additional information on the hazards of ethylene and propylene oxide.

The hazard in processing both ethylene oxide and propylene oxide is the presence of an oxygen molecule. Both chemicals are hazardous to human health, but the overriding hazard is ignition and explosion. The presence of an oxygen molecule makes both these chemicals extremely hazardous to process. Controlling leakage, even a minimal amount, is paramount.

EVALUATION OF SYSTEM CRITERIA

The system discussed in this paper is that of a refinery located in the Midwest that went through extensive research on new seal technologies eight years ago. The refinery utilized single seals to control emissions, but the seals were unpredictable and inadequate in some cases. The refinery criteria included a life expectancy of three years from the seals and zero volatile organic compound (VOC) emissions during startup, normal, and standby operation as measured using Environmental Protection Agency (EPA) Method 21. Also requested was computer generated theoretical analysis on seal performance under specified conditions as well as verification testing of the seal under normal operating conditions and upset failure modes. The analysis and testing of the seal were to evaluate durability, safety, wear rates, and seal life.

The seal design selection criteria were safety and reliability. Primary considerations were the highly corrosive characteristics of the HF and the possibility of a "stick-slip" situation when vaporization and dry running occur in services with high flashing indexes. Metal bellows seals were rejected because of the very thin bellows convolution thickness of approximately 0.178 mm (.007 inch), and the "stick-slip" phenomenon exciting the resonant vibration mode of the all metal construction.

The incorporation of advanced noncontacting dry running spiral groove technology provides a highly reliable containment seal in conjunction with an industry proven balanced wet running primary seal design developed for safety and reliability for HF alkylation units.

SEAL ARRANGEMENT

Two seal arrangements were investigated: a wet/dry running contacting tandem and a wet/dry noncontacting tandem.

A dry running contacting secondary seal arrangement eliminates the need for the wet lubrication system. Figure 1 illustrates a typical tandem arrangement utilizing a low emission seal inboard and a secondary containment seal outboard. The secondary containment seal functions as a safety seal in the event of an inboard seal failure and restricts process emissions to the atmosphere (Wasser, 1993). The secondary containment seal has been specially designed with a light spring load, to keep heat generation to a minimum. This design also uses a specially designed silicon carbide mating ring to help transfer heat away from the sealing interface. The dry running seal runs dry in the normal operating mode, but is capable of handling full process pressure. Containment pressure rating under inboard seal failure conditions is dependent upon the process fluid properties. It is recommended to incorporate an alarm or shutoff switch within the piping to the vapor recovery system to warn of an inboard seal failure.

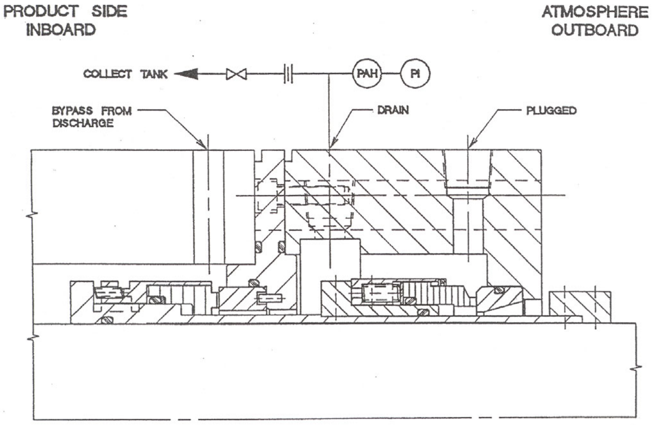

Figure 1. Dry Running Contacting Secondary Containment Seal Arrangement.

Although dry running, contacting secondary containment seal arrangements have proven to provide low emissions sealing for over 10 years, the arrangement did not meet the strict safety requirements for sealing HF.

The second seal arrangement considered was a wet/dry tandem with an outboard bushing in a cartridge format, coupled with a self-contained inboard restriction bushing assembly (Figure 2). The throat bushing isolates the primary seal from the process fluid and serves to maintain a liquid lubricating environment for the seal faces. This is achieved by controlling the pressure and flow of the flush media (propane/isobutane) in the inboard cavity. A hydropadded seal in the inboard seal position will control the leakage of the flush liquid into the outboard dry seal cavity. The flush leakage is sealed by a special dry running noncontacting seal that is designed to compress moisture-free nitrogen gas from the inside diameter to the outside diameter into the outboard cavity. The mixture of the nitrogen with the vaporized flush leakage is then vented to the header/flare. Nitrogen is supplied outboard of the dry tandem seal and the safety bushing restricts its flow.

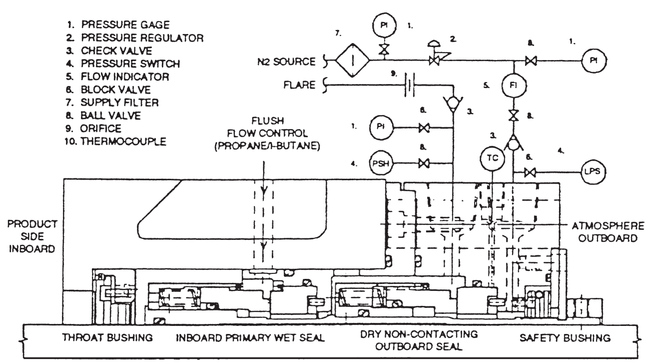

Figure 2. Wet/Dry Running Gas Seal Tandem Seal Arrangement.

The buffer area between the primary and dry tandem seal does not require any external liquid lubrication source. The dry running seal faces incorporate a series of partial spiral grooves, which are designed to operate on a thin film of gas. The grooves are configured to pump from inside diameter (ID) to the outer diameter (OD) of the seal faces. The nitrogen quench supply ensures an inert spiral groove film, of which an extremely small amount (~0.01 scfm) will move to the vapor recovery system.

NONCONTACTING GAS SEAL TECHNOLOGY

Satisfactory life for any mechanical seal depends on the ability of the design and the materials of construction to minimize the effects of contact friction. Without the proper design and material considerations, a seal will destroy itself due to the thermomechanical effects of contact friction. For gas seals, the noncontacting design eliminates the contact friction, allowing the seal to be used where energy levels are too high to run dry running contacting seals.

There are many face profiles used to generate the noncontacting feature, but unidirectional spiral grooves have proven to be effective for over 30 years. The gas seal faces ride on a gas film generated by spiral grooves, as shown in Figure 3. This spiral groove pattern is a series of logarithmic spiral or evenly spaced spirals recessed into the harder face ring. The sealing dam is the area from the outer diameter of the spiral groove to the outer diameter of the face of the opposing ring.

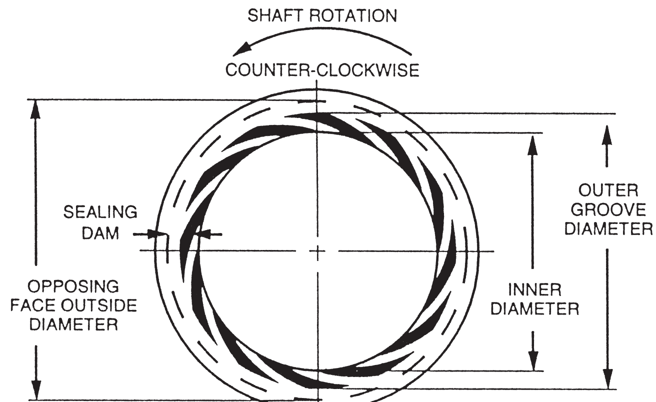

Figure 3. ID Spiral Groove Face Profile.

Spiral groove seals operate using the principles of fluid mechanics. As the seal rotates, gas flows into the spiral grooves through a viscous shearing action and is compressed. At the sealing dam, gas is expanded. The combined film pressure results in an opening force greater than the closing force that separates the faces a few hundred micro inches. During pump shutdown, hydrostatic forces along with the spring load act to close the faces. Seal balance and the design of the spiral groove prevent damage to the faces at startup and shutdown, before separation, when the faces contact (Wasser, et al., 1994).

The location of the spiral grooves on the mating ring is determined by the direction of the pressure differential. A seal designed to handle pressure at the inside diameter of the primary ring normally has the spiral grooves on the ID of the mating ring. The pattern shown in Figure 3 is for a counterclockwise shaft rotation and ID pressure of a stationary mating ring.

There are many advantages to incorporating spiral grooves into a seal design/arrangement.

- Since the faces are separated, frictional heat is eliminated, thus eliminating the need for a cooled external liquid support system.
- The noncontacting performance also significantly extends the useable life of the outboard seal. Sacrificial face wear associated with contact design is nonexistent for the gas seal.
- Unlike a liquid double seal system, the dry tandem seal system's barrier fluid levels do not have to be checked on a regular basis to ensure proper levels are maintained.
- The dry buffer fluid also eliminates any contamination or solubility problems experienced using a pressurized double liquid seal arrangement.
- In the event of an inboard seal failure, the noncontacting seal faces will be like-new and will close off quickly creating a tighter seal of the process to the atmosphere.

By design, the spiral grooves only extend partially across the seal face width providing an ungrooved dam that, when pressurized hydraulically, overcomes the hydrodynamic lift of the grooves and essentially functions as a wet contact seal. In the event of an inboard seal failure, the outboard seal is designed to close, thus creating the vapor recovery system as the path of least resistance. One major advantage of the noncontacting feature is when there is an inboard failure, the outboard seal faces are like-new and will seal off quickly. This is in contrast to dry running contacting seal designs that may be relying on faces that could have two years of wear on them.

SUPPORT SYSTEM

Vaporous leakage past the inboard seal under normal mode is completely isolated from escaping to the atmosphere by the nitrogen quenched seal; therefore all emissions are directed to the vapor recovery system through the gland vent. Inclusion of an orifice (item 9 in Figure 2) in this vent line allows restriction of the more significant levels of leakage such that a pressure rise will result. Location of a pressure switch between the seal and the orifice provides detection of a pressure rise associated with primary seal failure. A check valve (item 3 in Figure 2) must be positioned upstream of the orifice on the vapor recovery system header line to block flow of the flare line pressure to the dry tandem seal cavity. The nitrogen quench between the secondary seal and the segmental bushing must be a constant source to maintain a nitrogen blanket on the seal face ID. Nitrogen leakage of the gas seal will be entering the vapor recovery system header line through a vent in the gland plate. Nitrogen leakage across the segmental bushing will be released to the atmosphere, thus preventing any atmosphere from entering the vapor recovery system. The segmental bushing also serves as an additional containment seal in the unlikely event that both the primary and secondary seals fail.

MATERIALS OF CONSTRUCTION

The inboard sealing faces of the tandem seal are silicon carbide versus silicon carbide. They provide excellent corrosion resistance and wear with the hydropadded feature. The dry running noncontacting outboard seal faces comprise a corrosion resistance carbon (rotating face/primary ring) versus silicon carbide (stationary face/mating ring). This combination provides the excellent wear characteristics in the event the mode of operation becomes a contacting one. Special litharge cured fluorocarbon elastomers have been chosen for the secondary sealing elements based on their proven performance on existing HF acid applications. The metal components are made of Monel®, which provides excellent corrosion resistance and mechanical properties.

ANALYSIS OF THE INBOARD AND OUTBOARD SEAL FACES

A proprietary PC-based finite element analysis tool was used to computer analyze the seal faces. The software uses an iterative technique to find the convergent steady-state solution to the equations involving axisymmetric flow, heat transfer, and structural deformations with continuously updated fluid properties. A special finite element analysis (FEA) computer model handles these interactions among fluid, structure, and heat flow at the interface.

In the computer model, interfacial fluid pressures and leakages are determined by the seal gap and the fluid properties, each of which are functions of pressure, temperature, and fluid mixture. The fluid pressures have two components: hydrostatic and hydrodynamic. Hydrostatic pressures are calculated according to Reynold's Equation for Poiseuille flow. Hydrodynamic pressure relationships for contacting seals are empirically derived and are based on extensive testing for over eight years. Noncontacting seals' hydrodynamic pressures are based on numerical solutions of theoretical equations. A surface contact model is also included to account for contact pressure variations as a function of the seal face gap. All the above combine to counteract the closing force and affect the seal face temperature and distortion. These interactions together determine the steady-state seal gap.

Some fundamental assumptions used in the calculations are coefficient of friction (based on testing), surface roughness, and initial amount of seal face out of flatness. Uniform values, which are consistent with experience, are used for all the calculations in order to have comparable results. Other factors influencing results are pressure and temperature boundary conditions and the interaction with neighboring seals such as with dual seals. Figure 4 illustrates the heat transfer boundary location assumptions. In this regard, the calculated temperature from the inboard seal model is used as the temperature input to the outboard seal model.

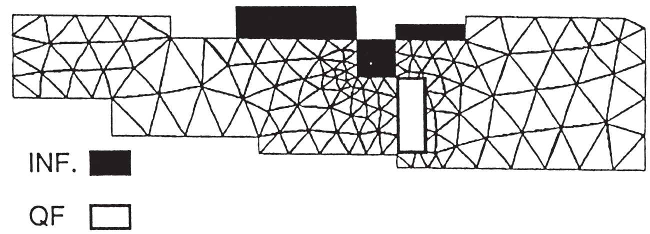

Figure 4. Assumed Heat Transfer Boundary Locations for the Wet, Inboard Seal.

Results of the computer analysis include predictions of performance; among the more important permanents are seal leakage rates, seal face temperatures, film thickness, hydrostatic, hydrodynamic, and contacting pressures, seal distortion, and contact patterns with and without seal face wear-in considered.

Figure 5 illustrates the pressure boundary location for the noncontacting gas seal faces, while Figure 6 illustrates the heat transfer boundary assumptions. Figure 7 illustrates the predicted distortion of the noncontacting sealing faces due to a combination of pressure and thermal effects.

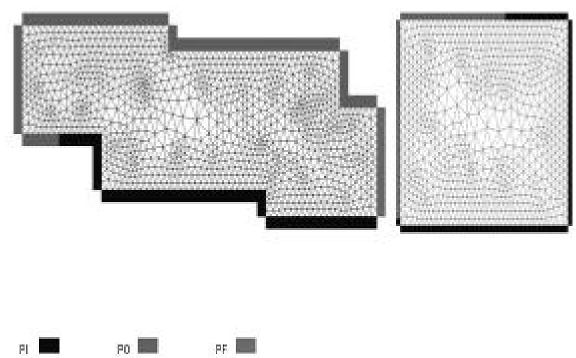

Figure 5. Pressure Boundary Locations for the Noncontacting Gas Seal Faces.

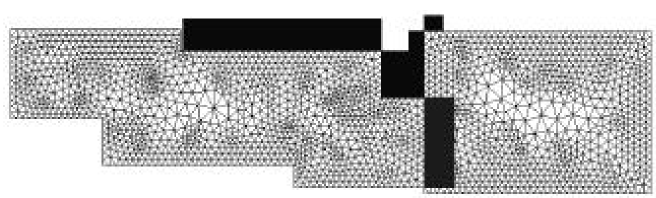

Figure 6. Heat Transfer Boundary Location Assumptions for the Noncontacting Gas Seal.

LAB TESTING

The specific purpose for the refinery's requirements on testing was to provide factual proof of the seal's ability to perform reliably in actual environments. The test rig included all auxiliary hardware as required to simulate seal flush and buffer systems. The seal flush

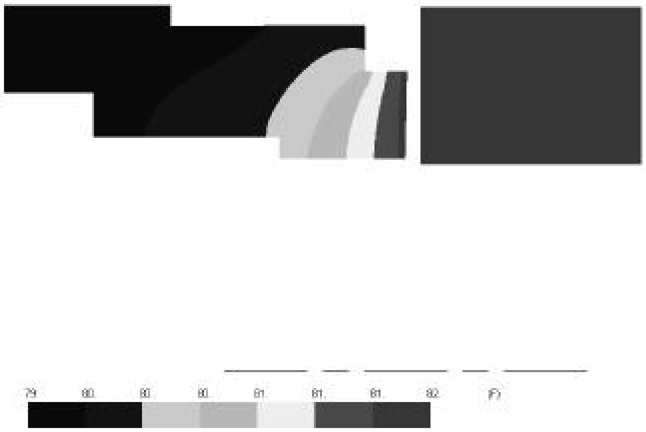

Figure 7. Predicted Distortion of the Noncontacting Gas Seal Faces Due to Pressure and Temperature.

system was developed through continuous partnering with the refinery's Project Engineers. The lab testing discussed in this paper was preformed in 1994 and 1998.

The tests consisted of the following:

- Propane was used as the flush media.
- The test duration was a minimum of 100 hours, but included five cyclic tests, for flushing excursions into vapor and back to liquid (flash and recovery). Three of the cyclic test modes maintained pressure at a constant while cyclic temperatures were induced. The remaining two tests maintained constant temperatures while pressures were cycled. This mode induced flashing as well as pressure surges.
- The seal was tested as a complete assembly.
- The dry running noncontacting seal was subjected to a four-hour failure mode.

The following parameters were recorded: primary seal chamber temperature and pressure, barrier fluid cavity temperature and pressure, VOC emissions, and pressure differential across the noncontacting secondary containment seal.

The first part of the test consisted of a 100-hour dynamic test, which was held statically for 10 minutes after the first 50 hours and four hours at the end of the test. The second part of the test was the induced failure mode condition in which the outboard dry running seal becomes flooded with propane to simulate an inboard seal failure. The second part of this test involved fluctuating the pressures and temperatures for a total of three operating conditions. The first condition was the maximum temperature and pressure, the second was the mean or normal operating conditions, and the third was to the minimum temperature and pressure conditions. These three conditions produced values that could be used as set points for different instrumentation alarms. This second part of the test also indicated that the seal would not catastrophically fail and that there would be adequate time to safely repair the pump or replace the cartridge seal if needed.

Operating Conditions for the 100-Hour Test

- Buffer fluid: Nitrogen at 0.02 to 0.03 scfm at ambient temperature, 55°F
- Process fluid: Propane at 280 psig at 110°F at 6 gpm (first 50 hours) and 4 gpm (second 50 hours)
- Speed: 1750 rpm
- Special: Back pressure (simulation of vapor recovery system pressure)
  75 hours - 2 to 5 psig
  25 hours - 5 psi increments to 25 psig

Reference Figure 8 for rig test.

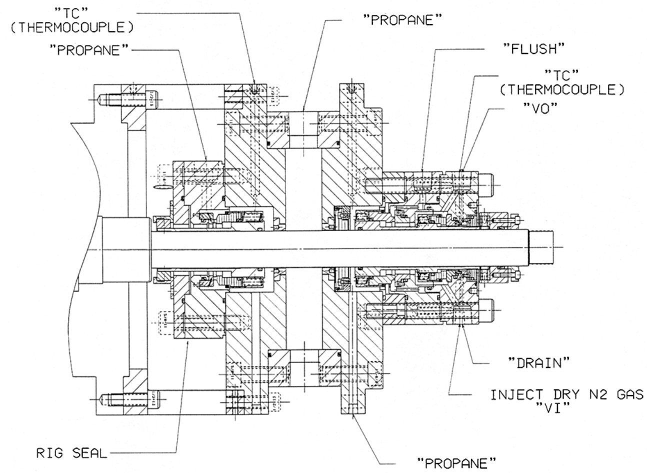

Figure 8. Test Rig Setup.

The leakage rate for the noncontacting gas seal can be determined by subtracting the background reading (position #4) from position #1 in Figure 9. The distance between position #4 and position #1 was approximately three feet. The background readings in this test were much higher than actual leakage from the inboard seal. This is due to propane vapors being present in the test cell. These vapors are the result of setting up the test and flushing of the system before test startup. The gas seal leakage was actually the same as background value at startup. It was assumed the tandem seal leakage was achieving a zero emission because the value did not increase over the length of the test.

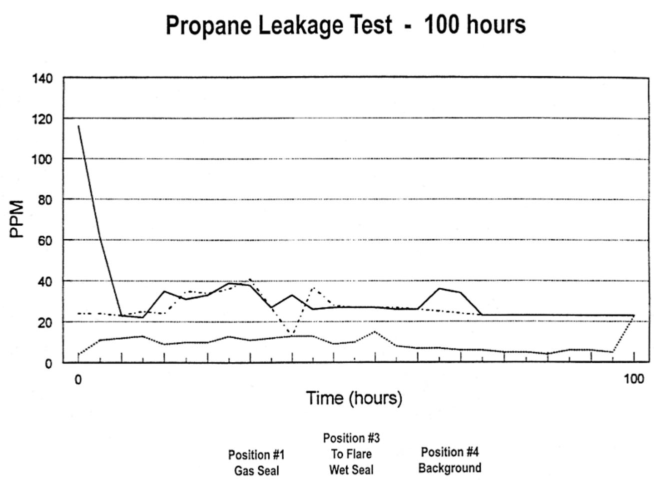

Figure 9. Leakage Rate for the Noncontacting Gas Seal.

The noncontacting gas seal was calculated to have an average temperature of approximately 88°F. Figure 10 does reflect the test value to be approximately 20°F higher than expected for this ambient temperature. This temperature increase was thought to be the result of heat soak from the inboard seal. The fluctuations that are shown can be contributed to propane vapor leakage from the inboard seal that may have migrated toward the secondary containment gas seal. This includes any condensate that may be present in the propane and/or nitrogen supply pressure fluctuations, both of which have been witnessed to have negligible impact on seal performance. The temperatures obtained in this test were used to set the maximum/minimum temperature settings on the control panel alarm.

The running torque that is plotted is the value for the primary seal, rig seal, and outboard bushing. The secondary seal torque during operation will be negligible due to its noncontacting feature. So, torque shown can be divided in half (approximate) for actual seal torque values.

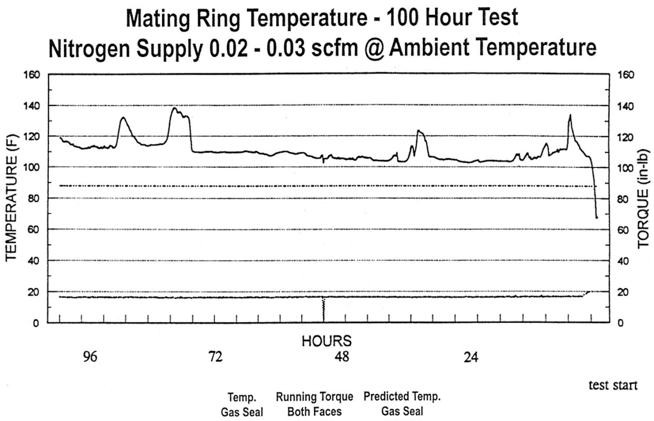

Figure 10. Gas Seal Mating Ring Temperature During Testing.

Operating Conditions for Gas Seal Induced Failure Mode

- Buffer fluid: Propane (to simulate inboard seal failure) at 2 gpm
- Process fluid: Propane
- Speed: 1750 rpm
- Operating temperature and pressure: Hour 1 - 380 psig at 230°F Vapor conditions were encountered at the start of the test due to fluctuations in system pressure and temperature stabilization.
  Hour 2-3 - 280 psig at 175°F
  Hour 4 - 210 psig at 110°F

The gas seal failure mode test provides data for limitations on the seal or where set points for instrumentation can be made (Figure 11). The sealing integrity was maintained even at elevated temperatures, above 400°F for short periods. The seal successfully sealed at the specified conditions stated above. Figure 12 plots the temperature of the gas seal during a loss of nitrogen pressure or flow supply. Ambient temperature at the start of the test was 60°F. Figure 13 plots the temperature rise of the gas seal with an increase in the nitrogen supply pressure. Ambient temperature was 60°F. Figure 14 plots the rise in temperature of the secondary seal's mating ring while being tested at the pump original equipment manufacturer's (OEM) test facility. During this test, the outboard seal chamber was flooded with city water (~70°F). The gas seal continued to pump a small amount of nitrogen across the faces, against the higher-pressure water, and the temperature rise was only 1 to 2 degrees. During testing, a site gauge was installed in the vapor recovery vent and air bubbles were witnessed. This indicated the gas seal was pumping a small amount of gas into the outboard seal cavity.

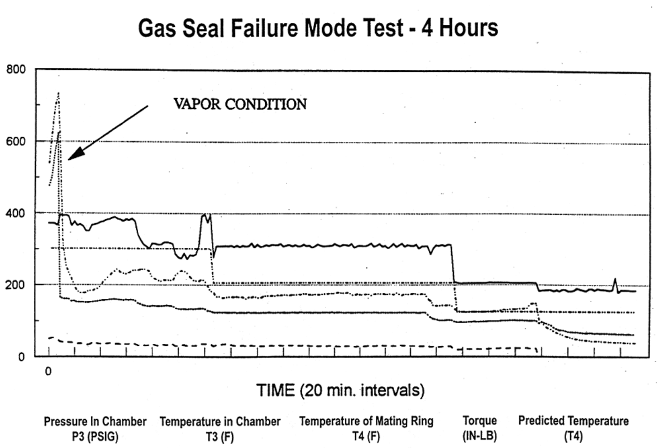

Figure 11. Failure Mode Test Results.

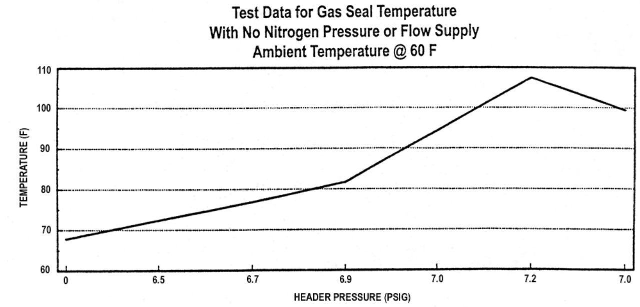

Figure 12. Gas Seal Temperature During a Loss of Nitrogen Supply.

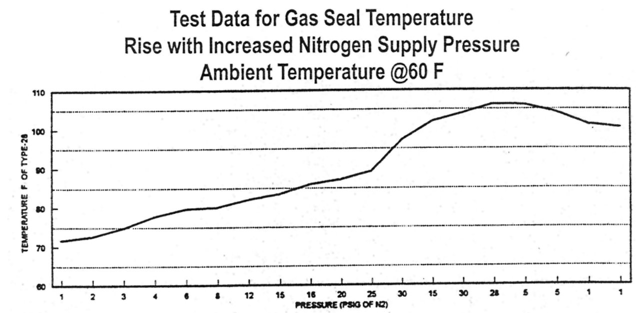

Figure 13. Gas Seal Temperature with an Increase in Nitrogen Supply Pressure.

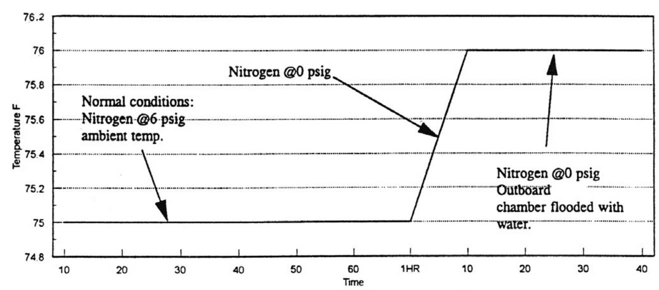

Figure 14. Gas Seal Performance During Pump OEM Testing.

These three possible situations graphed in Figures 11, 12, 13, and 14 indicate the stability of the secondary seal during very realistic operating upset conditions. These values were used to set the ambient and alarm temperatures on the seals installed on the HF units.

In the scenario of an inboard dynamic failure it was recommended that a 0.125 inch orifice be positioned upstream to achieve a pressure buildup before leakage exits to the vapor recovery system. Figure 15 indicates what quantity of gas (vapor) may be expected.

In the event the nitrogen is not supplied to the secondary seal, the seal will draw the atmosphere through the bushing as its buffer. The life of the seal at this condition cannot be quantitatively predicted because of the introduction of contaminates such as moisture or dust. In this event, the failure set point would be temperature.

As mentioned earlier, this testing was completed in 1994 for seals that were installed into service in late 1994. This same testing was also completed in 1998 for seals that were being installed in Europe in late 1998. That customer did not require the same level of documentation, but the seal preformed as predicted since the proprietary software files were modified as a result of the original testing.

INSTALLATION PREPARATION AND STARTUP

The first seals were successfully installed in late 1994 in a major Midwest refinery. A total of seven pumps, five different shaft sizes,

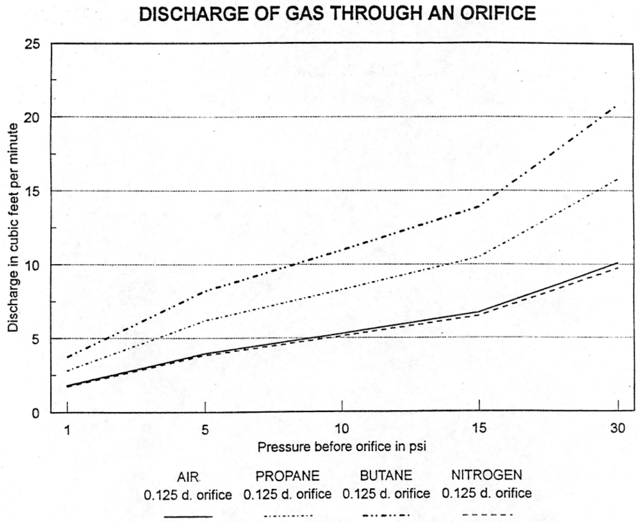

Figure 15. Discharge of Gas Through an Orifice.

was fitted with the seal arrangement. After hearing of the success at their competitor, another major refinery located in the Midwest fitted their two HF units with the tandem gas seal arrangement.

Prior to installing the gas seals, both locations utilized single liquid lubricated seals. Attempts were made to use both perfluoroelastomers and polytetrafluoroethylene (PTFE) secondary seals and carbon primary rings without much success. Seals were lasting nine months to a year, and failures were sudden and not predictable. Single seals were not meeting emission requirements and were costly and time consuming to repair or replace. When processing with HF, a plant has only a very short time to repair an emission problem.

Both customers' main drivers for selecting the wet/dry dual arrangement were safety, predictable performance, failure recognition, and real time control and condition monitoring. The gas seal permitted temperature monitoring as discussed earlier in the paper. This allowed the customer to track and record the temperature of the outboard seal faces ensuring the seal was performing as designed. Maximum temperature limits were set and alarms would activate if temperature exceeded the stated range. This allowed the refinery to more closely monitor seal performance and schedule maintenance according to their safety procedures. On the process side, the refineries recognized iron oxide particles collecting around the seals would be a problem along with the possibility of fluctuating supply pressures of the barrier system. Accumulators were installed inline on the nitrogen supply to provide a consistent supply pressure in the event of a loss of supply or fluctuating pressures. Low-pressure alarms were also incorporated in the nitrogen support system to warn of a drop or loss of barrier supply. Both provided additional safety features to the sealing arrangement.

Pump preparation was essential for ensuring long seal life, be it a new installation or repair. Shutdown, draining, and decontamination procedures for the pump and seal were documented and reviewed before each installation. These HF seals have been installed on end suction, overhung, and double-ended pumps. Pumps, such as the ones fitted with the wet/gas seals, which have been in service for a prolonged period, need to be checked for wear, grooving, and pitting. Any parts with excessive wear were replaced. The alignment of the pumps and subsequently the alignment of the mechanical seal assemblies are critical to the successful long-term performance of any seal. These seals were installed on pumps that fell within the runout and radial bearing fits of .001 inch per inch of shaft diameter, and endplay tolerances for thrust bearings range from .001 inch to .005 inch. Impeller adjustment had been checked on existing pumps due to the abrasive service these pumps can experience during operation, hence the clearances can widen due to erosion. After the pumps have been cleaned and brought back to manufacturing operating specifications, seal installation can proceed.

Pump startup procedures must be followed making sure that proper venting/flooding of seal chamber/priming of pump occurs. This is extremely important due to the vapor pressures of the HF; this will ensure that the inboard seal faces are properly lubricated. The nitrogen supply to the buffer chamber must be on at startup, this will allow an inert, clean buffer gas to the spiral grooves. Field and test runs have indicated that the seal will continue to run without the nitrogen supply, but understanding of atmospheric air into ones flare/header should be intrinsic. The vent piping (refer to Figure 2) initially did not have a 6 inch drip level with drain, but after experiencing high temperatures on the noncontacting outboard containment seal in the field, it was realized that residual fluid could migrate back from the flare header since several pumps share flares.

SEAL PERFORMANCE

The initial operating life of the seal cartridge was approximately nine months to a year. Problems that occurred included large amounts of iron scale buildup in and around the seal causing excessive amounts of abrasion on seal parts. The buildup of solids at the seal faces may restrict the flush media from reaching the faces, thus starving the seal of lubrication and cooling. The result would be excessive heat generation and wear resulting in decreased seal life. Solids buildup may also reduce the flexibility of the seal resulting in the inboard seal not being able to ride on a fluid film or not allowing the seal to completely close during a static mode, which results in leakage into the outboard barrier chamber. Modifications were made to increase flush porting size and Monel® seal components' clearances to allow for abrasives not to collect around the seal. An additional problem was found due to the Y strainer being clogged, starving the seals of flush. The piping downstream of the filter was changed to Monel® to prevent any contamination from the lines (downstream of the filter) from entering the flush port on the gland plates. The failures are noted to point out that the seal reliability during failure modes worked properly, as tested, sealing off, and set points were reached to indicate various types of problems. This allowed for a safe shutdown of the pump to occur over a four-hour period, and in some cases the seals recovered from an upset, settling back to normal temperature levels. This allowed the seals to remain installed, minimizing shutdown of that particular piece of equipment.

Several seals to date have been operating for over four years' continuous run, while others have been in operation for over several years due to their predicable performance under normal and upset operating modes. The seals installed in Europe have exceeded four years of service without a single seal failure since being installed in late 1998. Both installations have experienced pump-upset conditions, which tested the secondary containment gas seal's ability to handle the process fluid and pressure. In all cases the seals performed as predicted and contained the hazardous process fluid. One common failure has been due to iron scale buildup in the flush line and the vapor recovery line. Increasing the diameter of the flush line and the vent tap and piping eliminated this problem. On a few of the installations, the outboard seals experienced "icing" problems from the primary seal leakage (isobutane flush) flashing within the buffer chamber. Warming the vapor recovery lines solved this problem.

CONCLUSION

The seal arrangement described has been applied by virtually all the leading gasoline refineries around the world. The reliability of a full cross section unpressurized dual cartridge coupled with an outward pumping, noncontacting secondary seal in a new-when-needed role represents an insightful and elegant design, which many operators refer to as "the safety seal available, worldwide." The application of the seal arrangement described does not automatically ensure safety. Systems and instrumentation, as well as drive motor control logic and corporate safety practices must not be overlooked. Leakage of extremely hazardous or explosive fluids is a historical and chronic problem for refiners and chemical processors. This sealing system does indeed address these problems with unique technologies resulting in much greater operating confidence.

APPENDIX A

HF, in any concentration, as a liquid or a vapor, has two harmful effects on body tissues. First, it causes burns due to destruction of tissue, as do other strong acids such as sulfuric acid. Secondly, fluoride ions penetrate below the surface of the skin and continue to attack and destroy the tissue and bone until they are neutralized by medical treatment. If not treated promptly, the fluoride ions cause deep-seated, hard-to-heal ulcers.

The effect of HF on the skin depends on the concentration of acid. In strong concentrations, it causes immediate and serious burns on contact. Dilute solutions <50 percent also produce serious burns, but symptoms may not develop until more than one hour up to one day after exposure. This is referred to as a *delayed action burn*.

Inhalation of concentrated HF vapors may produce severe pulmonary irritation, which may be followed by delayed pulmonary congestion. Significant HF exposure can also deplete the body of calcium, which can result in death. Prolonged or repeated breathing of vapors can also result in liver, kidney, and lung damage and possibly death.

Ethylene oxide is classified as a carcinogen linked to peritoneal cancer and leukemia, and also has mutagenic and reproductive effects. Inhalation causes nausea, vomiting, neurological disorders, and even death. Traces of gas in gloves or clothing may cause burns. Inhalation may result in pulmonary edema. Excessive exposure may cause irritation of lungs and central nervous system depression.

Propylene oxide is a colorless liquid with an ether-like odor that is used mainly as a chemical intermediate in the production of polyurethane polyols, which are used to make polyurethane foams, coatings, and adhesives. It is used in the manufacture of propylene glycol, which is used in fiberglass-reinforced plastics, foods, cosmetics, pharmaceuticals, cigarette tobacco, packaging materials, dyes, and hydraulic fluids. Propylene oxide is classified as a fluid, which may reasonably be anticipated to be a carcinogen.

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