DESIGN AND OPERATION OF A NONPRESSURIZED DUAL GAS SEAL IN LIGHT HYDROCARBONS

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ABSTRACT

A major producer of ethylene installed and is successfully operating a nonpressurized dual gas seal in liquid methane service. The least reliable pump in the unit was a single stage overhung centrifugal pump in demethanizer reflux service. The process fluid is marginally over its vapor pressure at a temperature of −140°F (−96°C) and pressure of 502 psig (3461 kPa). The pump was originally installed with a dual nonpressurized mechanical seal using n-propanol as the barrier fluid. There were numerous seal problems on the pump and the mean time between repair (MTBR) averaged four months. Many modifications were made to the sealing system with little or no improvement in the MTBR.

A joint development project was initiated involving plant and seal supplier personnel to investigate the use of dual gas seal technology in this service. A cartridge, nonpressurized series arrangement dual gas seal was designed and thoroughly tested before installation in the plant. Testing included operation with a light hydrocarbon in both the liquid and vapor phases. Incorporating an interstage and external labyrinth with an inert gas sweep satisfied low environmental emissions requirements. A purge and vent control panel system was developed to monitor and establish appropriate gas purge and vent settings for containment and emissions purposes. Installation at the plant included a close clearance throat bushing and removal of the seal flush to encourage vapor accumulation in the seal chamber.

The first seal was installed in the plant on January 1, 1997, and has operated through November 1998 without problems. The methane vapor emissions at the outboard labyrinth are consistently measured at less than 250 parts per million. The seal has dramatically improved the reliability of this critical pump even under upset conditions. This paper details the seal design parameters, laboratory testing, seal installation, and operation of the pump and support system. The application of this technology can assist plants in reducing emissions, increasing pump MTBR, reducing maintenance costs, and increasing equipment reliability.

INTRODUCTION

Equistar Chemicals, a joint venture between Lyondell Chemical Company, Millennium Chemicals, and Occidental Petroleum Corporation, is a major producer of ethylene and ethylene derivatives, with a rated capacity of 1.7 billion pounds of ethylene per year. The subject of this paper is the improvement of mechanical seal performance and reliability on the least reliable pump in the unit: a critical methane recycle process pump. The core problem with this pump is the low vapor margin of the methane in the seal chamber. In a survey of petrochemical plants, approximately 10 percent of all light hydrocarbon pumps have inadequate temperature or pressure differential to provide stable performance of liquid lubricated seals. For this reason, companies like Equistar have identified liquid lubricated seals in poorly lubricated light hydrocarbon service as their major cause for short mean time between repair (MTBR). At the request of Equistar, the Fluid Sealing Division of Flowserve Corporation was invited to partner in the development of a mechanical seal with noncontacting seal face technology for light hydrocarbon service.

PROBLEM DEFINITION

The subject pump is a Pacific SVCN-7 API single stage overhung centrifugal demethanizer reflux pump with a 2.250 inch (57.2 mm) shaft. The normal seal chamber pressure is 502 psig (3461 kPa), the system temperature is −140°F (−96°C), and the shaft speed is 3600 rpm. These operating conditions establish a very close vapor margin in the seal chamber. The vapor pressure at −140°F (−96°C) is 487 psig (3358 kPa). The system has a primary and spare pump with the same piping arrangement and performance. The initial mechanical seals were traditional dual nonpressurized balanced cartridge seals with an n-propanol barrier fluid. Despite the proper design and implementation of the piping and pump system, the low vapor margin at the inboard seal faces was the root cause of the mechanical seal failures.

When the pump is started, the pressure and temperature conditions in the seal chamber, coupled with the seal face generated heat from the inboard seal faces, induce methane vaporization. This not only hinders the desired pumping, but it creates conditions for a high infant mortality rate of the inboard seal faces. Vapor is a poor lubricant for liquid lubricated seal faces and causes excessive face wear and damage. One of the first modifications employed to improve startup was to machine the throat out of the seal chamber and add a five degree taper to the seal chamber bore. This change helped to improve the startup success by increasing the pressure in the seal chamber and providing adequate fluid circulation. Another successful technique was the addition of precooling to the seal chamber and barrier fluid. However, making this step a consistent part of the pump's standard startup procedures was not easily implemented.

If proper precooling were applied or if the column pressure were momentarily high, the pump could start without incident. Unfortunately, the seal chamber, or at least the region around the inboard seal faces, remained as methane vapor. The normally contacting inboard seal faces were actually separated while operating in a full fluid film condition. The faces indicated no evidence of wear after removing the seal from service. Also, there was excessive leakage of methane vapor into the barrier system. The barrier tank flare vent initially had a one-sixteenth inch (1.59 mm) orifice, but the leakage was so high that the barrier pressure would reach 30 psig (207 kPa) and set off system alarms. The vent restriction was changed to a one-eighth inch (3.175 mm) orifice and the barrier pressure remained lower than the alarm trigger of 15 psig (104 kPa). However, the flowrate was so high that the n-propanol barrier fluid was evaporated by the escaping methane vapor. The two gallon supply tank had to be completely filled every day. Through monitoring the barrier system, it was determined that the methane waste was excessive.

The final system change was the addition of a dry ice bath around the barrier tank to maintain a low barrier fluid temperature. Although this may have marginally increased the seal life by lowering the barrier temperature and the temperature of the inboard seal faces, the system maintenance requirements were increased, especially during summer months (the plant is located in Houston, Texas).

Each change to the system conditions was methodically intended to increase the vapor margin at the inboard seal faces and improve seal performance. The effect on MTBR was an improvement from zero to four months. However, maintenance requirements were significantly increased and operator confidence was never satisfied. Furthermore, an inconsistent success rate of four months between repairs is not the most cost effective way to operate a critical process pump.

Another factor influencing the pursuit of alternate seal technology was governmental changes in environmental emissions standards. Although methane emissions from this pump were typically less than the 500 ppm company threshold, the n-propanol barrier fluid was recently added to the Clean Air Act's list of controlled volatile organic compounds (VOCs). A proactive move away from n-propanol would reduce future company compliance efforts.

A viable alternate seal technology that thrives in the same vapor environment detrimental to conventional seals is a noncontacting seal face gas seal. A dual nonpressurized arrangement would provide low vapor leakage across the noncontacting inboard and outboard seal faces. The dual arrangement also provides emergency backup in case of an inboard seal failure. With the addition of a nitrogen purge through the barrier, minimum hydrocarbon emissions can be maintained and the n-propanol barrier fluid could be eliminated. These advantages and performance expectations inspired the development of the subject dual-pressurized gas seal design.

GAS SEAL DESIGN

Noncontacting seal face gas seal technology has been in operation on compressors since the mid 1970s and more recently available on pumps (Adams and Parker, 1994). Both designs rely on a gas, either from the product or from an external source, to facilitate seal face separation. One of the more popular seal face configurations utilizes a pattern of shallow spiral grooves manufactured into one of the seal faces. The particular design used was patented by the seal vendor (Patent Number 5,556,111) and it incorporates a tapered groove depth to the spiral grooves that feed into a circumferential groove at the sealing dam (Figure 1). The shallow grooves provide both hydrostatic and hydrodynamic lift to create seal face separation. With applied pressure, hydrostatic lift is present on the seal faces as pressure acts through the shallow grooves and decays across the sealing dam. Hydrodynamic lift is generated by viscous shear of the gas film by the nongrooved counterface during rotation. The net result is a balance of opening and closing forces at a specific separation of the seal faces.

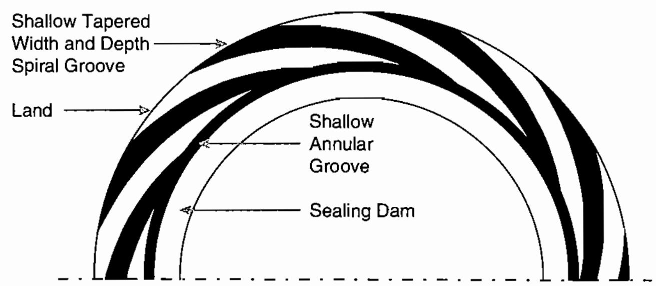

Figure 1. Shallow Groove Terminology.

The hydrocarbon product, if in a liquid state, would flash to a vapor at a specific location on the seal face, as defined by the local vapor pressure. Depending on the radial location of the flashpoint with respect to the spiral groove pattern, the hydrodynamic life generated by the groove pattern is influenced by a combination of hydrocarbon liquid and vapor. The ensuing film thickness would be typical of a liquid, and this larger gap, coupled with the liquid/vapor volume expansion ratio, would simply provide slightly greater leakage than a typical gas seal but with equal stability. Analytical methods modelled from classical narrow groove theory can be used to precisely predict seal face separation and gas consumption as a function of seal geometry, operating conditions, and fluid properties. The modelling becomes more complex if a two phase fluid is evaluated. There is a precise relationship between liquid phase, vapor phase, and flashpoint location on the sealing face. Along with the standard model inputs such as operating conditions, seal geometry, and face pattern details, additional parameters affecting local fluid properties must be included. Some examples are local pressure, local temperature, viscous or churning heat, heat transfer characteristics, and heat soak. Due to the relative infancy of this analytical procedure, testing was required to confirm performance.

One unique characteristic of the subject gas seal application is the state of the process fluid at the inboard seal faces. Experience with conventional liquid lubricated seals has proven that the hydrocarbon is a vapor at the seal faces. This vapor condition is somewhat a function of the seal face generated heat from the contacting seal faces. Noncontacting gas seals typically produce only five percent of the heat of contacting liquid lubricated seals. Therefore, to have a gas seal operate properly in a marginal vapor state, the seal face design must tolerate both liquid and vapor conditions. This is also true for the nonpressurized outboard seal faces that primarily operate in vapor, but could experience full product pressure if the inboard seal was to fail. Another approach is to maintain a constant vapor condition in the seal chamber and use an established gas seal design.

Several laboratory tests were performed under liquid product conditions with two shallow groove designs: one optimized for completely vapor conditions and one for flashing hydrocarbon conditions. The details and conclusions of the testing are discussed later. The basic difference between the liquid and vapor groove designs is the level of dam balance, an increase of approximately 20 percent as dictated by the sealing dam location relative to the balance diameter. Dam balance is a measure of opening forces on the seal face generated by the groove pattern. The liquid groove face has a high dam balance, so that the hydrocarbon flashpoint is closer to the seal face outer diameter. This approach provides lower leakage, but compromises performance if the product is already a gas.

Across the inboard seal faces, the high pressure hydrocarbon drops to the low barrier pressure and carries a predictable amount of process gas into the barrier cavity. To maintain low atmospheric hydrocarbon emissions, a nitrogen purge feature was added to the barrier system to carry away inboard seal leakage to the flare system. The gas available to the outboard seal faces would be a mixture of nitrogen and hydrocarbon. The purge flow was estimated to be set at 10 scfh (4.7 slpm) with a 10 psig (69 kPa) differential pressure. A secondary purge was also included between the outboard seal faces and a labyrinth. This purge would be used to remove minor hydrocarbon emissions from the outboard seal faces or to act as the primary purge in an emergency backup situation. The secondary purge would be set at a lower flowrate such as 2 scfh (0.94 slpm).

After laboratory testing and design analysis, a spare seal was built with a close clearance interstage labyrinth between the inboard and outboard seal faces. The primary purge inlet is located on the outboard side of the labyrinth and the outlet is located on the inboard side. This arrangement creates an additional hydrocarbon flow restriction to keep inboard seal leakage from the outboard seal faces. The seal design nomenclature and purge scheme are shown in Figure 2. A gas supply control panel, as shown in Figure 3, was built to maintain and monitor the primary and secondary purges from a common nitrogen header. As with the conventional liquid seals, the atmospheric emissions goal was less than 500 ppm of methane. Laboratory testing was designed to determine the appropriate purge flowrates to satisfy emissions limits.

The gas seal materials of construction were conventional. The rotors were a premium resin impregnated carbon, the stators were self-sintered silicon carbide, and the metallurgy was 316 stainless steel except for the 20 stainless steel springs and pins. To accommodate the −140°F (−96°C) operating temperatures, spring energized PTFE seals and graphite packing were used in critical locations. The interstage labyrinth wear material was bronze, while the outboard labyrinth was reinforced PTFE.

LABORATORY TESTING

Although the charter methane service was expected to be completely methane vapor, the seal had to tolerate the occasion of liquid service. Therefore, laboratory testing focused on validating the gas seal's performance with a liquid hydrocarbon as the product. The seal vendor's hydrocarbon seal test facility has a Class 1 Division 1 Group D hazardous area rating and was calibrated to operate with propane liquid or vapor. Propane was used instead of methane because of its easier handling and control in a simple test loop. Correlations were made between the propane test results and the expected performance with methane.

The first goal of testing was to confirm that the seal faces separate under operation. Thermocouples were mounted to the silicon carbide stator faces to check for seal face contact and to set

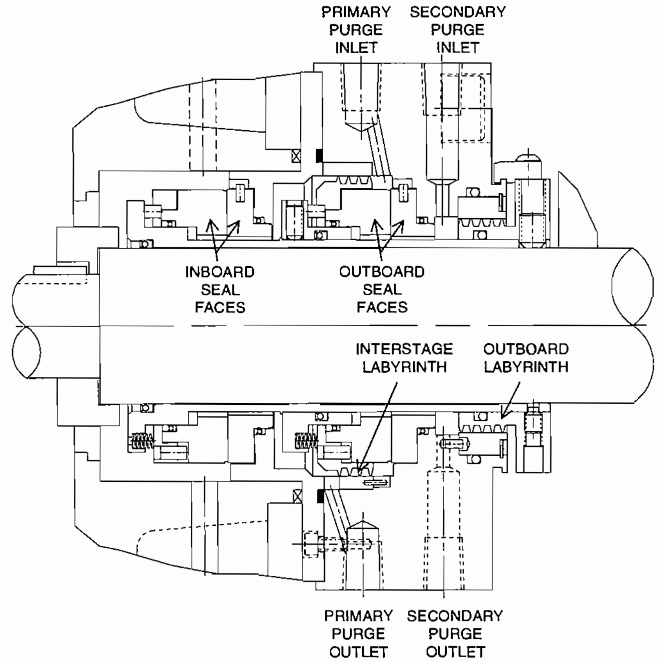

Figure 2. Dual Nonpressurized Gas Seal Design for Light Hydrocarbon Services.

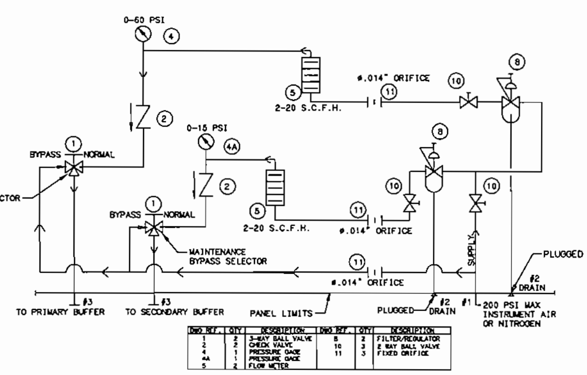

Figure 3. Dual Gas Seal Purge Supply Control Panel Schematic.

a safety control. The automated control system was set to shut down the test if the inboard stator temperature exceeded the seal chamber temperature by 50°F (28°C) or if the outboard stator temperature exceeded the primary purge inlet temperature by 50°F (28°C). If the seal faces are noncontacting, the seal face temperatures should follow the system temperatures. The tester control system and data acquisition were operated from the same computer and included measurements for temperature, pressure, emissions, and motor speed. Emissions monitoring was performed by a single flame ionization detector (FID) sensor with a six channel linear sequencer. The experimental tester arrangement is illustrated in Figure 4.

Altogether, six dynamic tests were performed with a total run time of 6.7 hours. Each test's duration was a function of the volume of propane in the test loop relative to the gas seal leakage. System pressure was maintained at 250 psig (1725 kPa) by pressurizing the bladder of a small accumulator. When the accumulator is empty, the system pressure begins to drop. A safety control was set to stop the test if the system pressure was to come within 20 psig (138 kPa) of the vapor pressure, so that the ensuing volume expansion would not damage the accumulator. Each test ended with a low pressure alarm and automatic system shutdown.

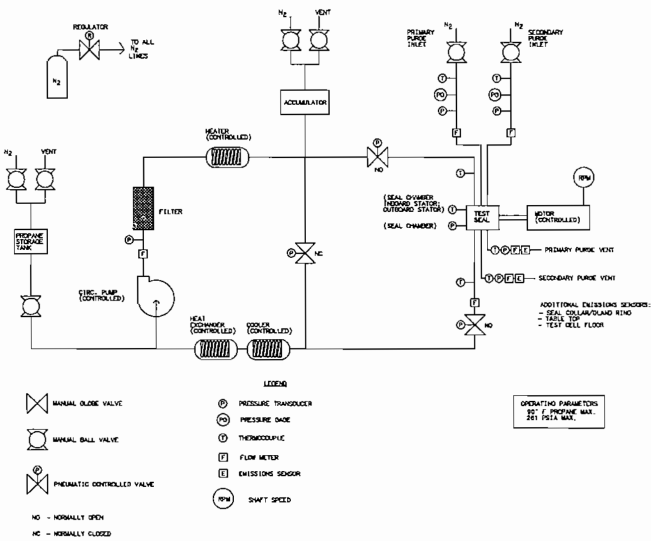

Figure 4. Laboratory Tester and Data Acquisition.

In every test, the seal face temperatures verified full fluid film operation. For example, in the first test, the temperature of the inboard seal faces dropped 20°F (11°C) within 10 seconds after startup. After one minute, the temperature stabilized at 57°F (13.9°C) from an initial temperature of 84°F (28.9°C). This temperature drop not only confirms that the seal faces were not contacting, it also shows that the liquid propane flashes to a vapor as it crosses the seal faces. The temperature drop is attributed to the latent heat of vaporization for propane. The measured temperatures for the first 40 seconds of operation are shown in Figure 5. Upon inspection of the post-test hardware, the seal faces appeared pristine, and a single set of faces was used for the first five tests.

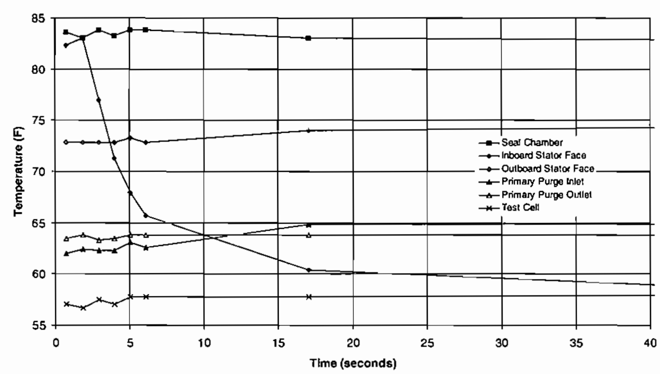

Figure 5. Measured Seal and System Temperatures from Test 1.

The purpose of running several tests was to establish the most effective nitrogen purge flows and arrangements for low emissions. Even though testing was performed on a seal without an interstage labyrinth, the initial primary and secondary purge settings of 10 scfh (4.7 slpm) and 2 scfh (0.94 slpm), respectively, were determined to be adequate in reducing emissions at the rear of the gland to less than 500 ppm. For example, the emissions measurements taken during Test 3 are shown in Figure 6. The constant emissions plateaus represent the sampling time and delay as the six channel sequencer switches channels every 45 seconds for a sampling period of 4.5 minutes. The sharp drop in gland emissions at the 37 minute mark was the result of changing the purge flows. The primary purge was increased from 3 scfh (1.4 slpm) to 10 scfh (4.7 slpm), and the secondary purge was increased from nothing to 2 scfh (0.94 slpm). Increasing the purge flows to the initial target immediately reduced the atmospheric emissions to less than 500 ppm.

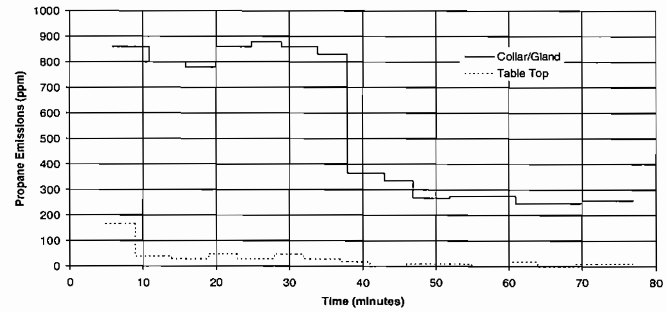

Figure 6. Measured Propane Emissions from Test 3 with Changing Purge Flows.

Another example of emissions measurements is from Test 5 as shown in Figure 7. In this test, the primary and secondary purges were set at 10 scfh (4.7 slpm) and 2 scfh (0.94 slpm), respectively, but the outboard labyrinth was worn from previous tests. During assembly, the outboard labyrinth has a slight interference fit between the teeth and wear material. A close clearance, torturous leakage path is created after the teeth bite into the wear material during operation. If the labyrinth is disassembled and reassembled, the nested teeth disturb and remove the wear material between the teeth, inducing greater clearance and a lower flow restriction. The new labyrinth installed in Test 3 had been reused and reinstalled in Tests 4 and 5. This resulted in the higher gland emissions shown for Test 5 relative to Test 3. For delivery to the customer, a new wear bushing is installed each time the seal is assembled.

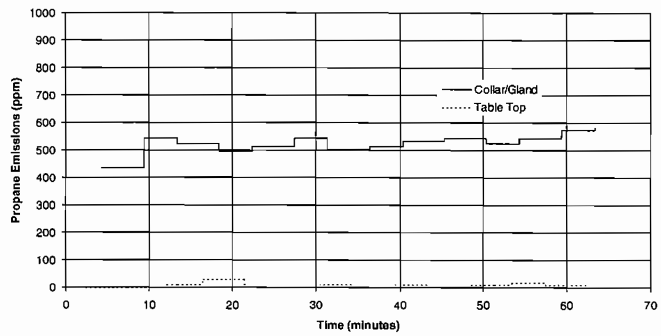

Figure 7. Measured Propane Emissions from Test 5 with a Reused Labyrinth.

The first five tests validated the successful performance of a gas seal face designed for a hydrocarbon vapor but operated in a hydrocarbon liquid. Test 6 used a gas seal face designed for a hydrocarbon liquid and was operated in a hydrocarbon liquid. The result was much lower propane leakage across the full fluid film inboard seal faces. This was due to the increased dam balance and closing forces that decreased the seal face separation. Atmospheric emissions were low even with the worn outboard labyrinth. Further testing of this design was not pursued, due to the proven flexibility of the vapor face design to tolerate a liquid hydrocarbon. Analytically, the larger closing forces in the liquid face design exceed normal target performance conditions when operating in a vapor and is therefore not as tolerant of off-design operation.

Laboratory Testing Conclusions

The liquid propane testing proved that the nonpressurized dual gas seal could provide low emissions levels, even with a face groove pattern designed for vapor conditions. The liquid propane consistently flashed across the inboard seal faces during stable, full fluid film operation. The outboard seal faces, likewise, were noncontacting under a combination of nitrogen and propane vapor. The primary and secondary purge flows of 10 scfh (4.7 slpm) and

2 scfh (0.94 slpm), respectively, provided emissions levels at the seal's collar of less than 500 ppm. These low emissions were achieved in a seal assembly without an interstage labyrinth between the primary purge inlet and outlet. Although the methane service is at a higher pressure than the propane testing, low methane emissions were expected because of the following:

- The methane vapor pressure is so marginal in the actual pump seal chamber that the presence of vapor can be assured. Under the specific operating conditions, established gas seal analytical methods predict inboard seal methane leakage at less than 15 scfh (7.1 slpm).
- The implementation of an interstage labyrinth would provide an additional hydrocarbon leakage restriction with the same 10 scfh (4.7 slpm) purge.
- The outboard interference fit labyrinth would have a greater flow restriction due to the colder operating temperatures. The thermal expansion of the teeth relative to the wear ring provides greater overlap and a more torturous leakage path.

FIELD INSTALLATION

The installation of the nonpressurized dual gas seal was somewhat hindered by equipment problems. The purpose in presenting these experiences is to allow the user to learn from actual installation events and to highlight the performance of the nonpressurized dual gas seal. First, the initial gas seal could not be installed in the target spare pump. The discharge valve for the spare pump had failed and could not be replaced without shutting down the entire plant. This presented a difficult situation for the conventional seal operating in the primary pump, whereas a seal failure would force a shutdown of the plant.

Circumventing the primary and spare pumps, a third pump was installed to fulfill the need for a spare pump. The gas seal was installed into this third pump with 10 scfh (4.7 slpm) and 10 psig (69 kPa) primary nitrogen purge conditions. The seal chamber was an unmodified design with a close clearance throat and no flush that easily causes vapor conditions in the seal chamber. Unfortunately, the piping had so many restrictions that the suction head was too low and the vapor-locked pump could not start. Enough time elapsed trying to get the third pump working that a routine plant shutdown allowed an opportunity to replace the faulty discharge valve. The gas seal in the third pump had experienced multiple short duration stop/starts and operated in all vapor conditions. The seal was disassembled, inspected, and the seal faces were as new, confirming full fluid film operation.

The gas seal from the third pump was installed in the original spare pump, while the conventional seal remained in the primary pump. When the process was switched to the spare pump for startup, the motor could not turn the shaft. The seal sleeve had shifted out toward the bearing frame under the hydraulic thrust loading. This movement caused both the inboard and outboard seal faces to close together heavily and create excessive startup torque. The seal was removed, inspected, and found to have repairable seal face wear.

A spare gas seal that did not have an interstage labyrinth was then installed in the spare pump. The pump started up without incident on January 1, 1997. The primary nitrogen purge was set at 10 scfh (4.7 slpm) with 10 psig (69 kPa), and the secondary purge was not turned on. Even without the secondary purge, the methane emissions are regularly measured at less than 250 ppm around the seal. As of the submission of this paper in November 1998, the pump and seal have been running continuously without failure.

Upon visiting the plant, one would notice an inconspicuous mechanical seal gland bolted to a large cylinder of ice, as shown in Figure 8. (The control panel is shown in Figure 9.) The pump icing does not affect the seal performance, because it does not enter the seal area due to the purging and relative warming of nitrogen. There was at least one monitored occasion when the primary nitrogen purge was temporarily lost, but it did not cause any apparent outboard seal problems. The available methane vapors at the outboard seal faces were sufficient to provide seal face liftoff, and icing did not affect the secondary seals. In the primary vent, there are no measurements taken to capture the actual methane leakage through the primary seal faces. Simply based on the emissions monitored outside the seal and the normal operation of the flare vent system, the methane leakage to flare is low and acceptable.

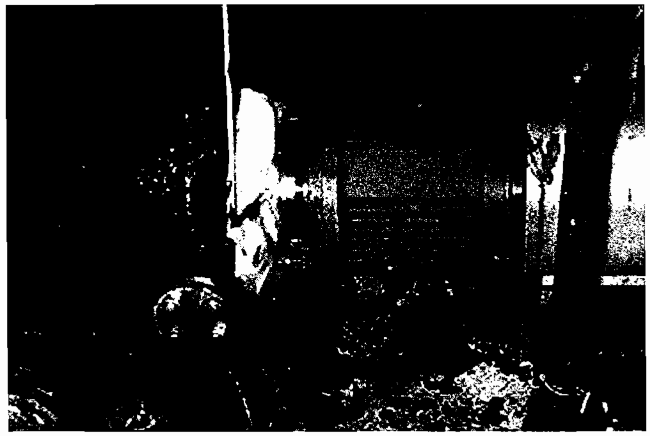

Figure 8. Dual Nonpressurized Seal Installation at the Plant.

Figure 9. Dual Nonpressurized Seal Control Panel.

VALUE ASSESSMENT

Analysis of the relative cost advantage of dual nonpressurized gas seal technology over dual nonpressurized liquid lubricated seal technology is shown in Table 1. The cost differential between a four month MTBR with liquid seals versus 1.7 years with a gas seal includes the costs associated with pump removal, seal repair, reinstallation, and system downtime. The improvement of MTBR alone easily accounts for the largest share in cost savings.

Table 1. Hardware and Operating Cost of Liquid Lubricated Versus Gas Lubricated Dual Nonpressurized Seal Technology.

Component	Liquid Lubricated Seal	Gas Lubricated Seal
Initial seal costs	1 unit	2 units
Initial support system cost	1 unit	1 unit
Support system operation	3 gallons n-propanol per day	10 SCFH (4.7 SLPM) N_2
Maintenance requirements	High	Low
Methane losses	High	Low
Power consumption	10 units	0.5 units
MTBR	4 months, high	1.7 years, low

FUTURE DEVELOPMENTS

The assumptions required to accurately predict gas seal performance in flashing hydrocarbons will continue to require laboratory test confirmation until the analytical technology matures. Fortunately, gas seal technology is robust such that a single face pattern can work satisfactorily in both hydrocarbon vapor and liquid. To ease the technical efforts required to design a custom seal face for each application, it may be more effective to follow the approach outlined in this paper and design for primarily vapor in the seal chamber. Even with a normally generous vapor margin in the seal chamber, the introduction of a close clearance throat bushing, labyrinth, or external heat can produce vapor at the inboard seal faces. In the meantime, gas seal technology will continue to develop and advance the use of nonpressurized dual gas seals into higher vapor margin liquid hydrocarbon applications.

Although the performance of the nonpressurized dual gas seal has satisfied expectations, a direct measurement of the methane vapor leakage and correlation with analytical methods has not been performed yet. Historically, comparisons between predicted and actual gas leakage for similar seal face technology have been very favorable, and the same is expected for this application. Future installations may include vent flow sampling devices for confirmation purposes.

This technology can be readily adapted to other light hydrocarbon processes with designs developed specifically for each application. At the same plant, nonpressurized dual gas seals are being supplied for a 340 psig (2346 kPa) ethylene splitter reflux pump and a 230 psig (1587 kPa) propylene pump. Both of these pumps are double suction, single stage pumps with 3.250 inch (82.6 mm) shafts. The design analysis efforts include matching the shallow groove technology to the specific vapor characteristics. These applications are destined to have vapor in the seal chamber, a proponent of seal failure for the existing liquid lubricated seals.

CONCLUSIONS

The purpose for presenting this information is to inform pump users about the reliability improvements attained through applying nonpressurized dual gas seal technology to light hydrocarbon services. In hydrocarbon pumps where the vapor pressure in the seal chamber is marginal, the performance of liquid lubricated seals is dramatically limited by poor lubrication of the seal faces. A dual nonpressurized gas seal with noncontacting seal faces provides stable, reliable operation in both hydrocarbon vapor and liquid conditions. The impact of various operating characteristics on the application of liquid and gas lubricated seal technology in light hydrocarbon services is shown in Table 2.

Table 2. Comparison of Dual Nonpressurized Seal Technology in Light Hydrocarbon Service.

Operating Characteristic	Liquid Lubricated Seal	Gas Lubricated Seal
Vapor in seal chamber	Unreliable operation	Reliable operation, stable film
Liquid in seal chamber	Satisfactory operation	Reliable operation, stable film
Barrier fluid	Required	Low nitrogen purge flow
Seal chamber flush	Required	None
Seal chamber cooling	Required	None
System maintenance	Frequent supply tank filling	Constant nitrogen purge
Hydrocarbon losses with vapor in seal chamber	10 units	2 units
Hydrocarbon losses with liquid in seal chamber	1 unit	4 units
Power requirements	10 units	0.5 units
Atmospheric emissions	Less than 500 ppm	Less than 500 ppm

Performance improvements achieved with dual nonpressurized gas seal technology were the result of recent laboratory testing and field experience in propane and methane services. The charter application at a major petrochemical plant improved the MTBR from four months with conventional liquid lubricated seals, to over 1.8 years and counting with a gas seal. The conversion to gas seal technology eliminated the daily maintenance duties of monitoring seal performance and adding barrier fluid. The elimination of n-propanol as the barrier fluid, while maintaining low methane emissions, fulfilled the plant's compliance with environmental regulations. The success and reliability of this dual nonpressurized gas seal have satisfied the operator's confidence, and the pump is no longer the poorest performer.

Laboratory testing also confirmed that a gas seal face designed for hydrocarbon vapor conditions operates equally well with a hydrocarbon liquid. With a gas seal face designed for hydrocarbon liquid conditions, the hydrocarbon leakage can be further reduced. Shallow spiral groove technology can be optimized for satisfactory performance under any specific design condition and is flexible enough to tolerate off-design conditions. The inclusion of labyrinths and nitrogen purges guarantees low environmental emissions. In fact, the inaugural nonpressurized dual gas seal operating at the plant allows hydrocarbon emissions lower than plant limits, even without using a secondary nitrogen purge.

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