

# DUAL GAS SEALING TECHNOLOGY FOR PUMPS

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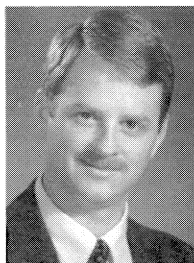
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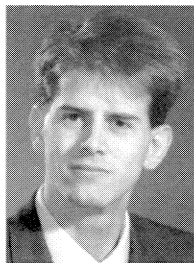
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## ABSTRACT

Dual gas seals are gaining acceptance by industrial users as a product that offers advantages over conventional liquid lubricated seals. These advantages are measured in terms of controlling emissions, minimizing product contamination, and reducing system installation and operating costs while being capable of handling both normal and off-design pump operations. Test programs were conducted to document the performance characteristics of various dual gas seal arrangements and technologies while handling thermal sensitive fluids, fluids containing debris, and off-design pump operating conditions. Variances in leakage, projected seal life, and sensitivity to secondary seal hangup were evaluated as a function of the seal design. Results from these test programs, user case histories, and application guidelines provide users with a means for selecting dual gas seal arrangements and technologies to improve process operations.

## INTRODUCTION

The sealing of process pumps has traditionally been accomplished by using compression packing or mechanical seals, both of which require liquid lubrication for their survival. Previous investigative work was conducted to evaluate factors affecting the life of liquid lubricated mechanical seals. These seals were evaluated under normal operating conditions [1, 2] and off-design operating conditions [3] to provide application guidelines for extending operating life.

Seal users in industries worldwide are now being restricted by government legislation limiting the level of fugitive emissions leakage allowed from process equipment. Single liquid lubricated mechanical seals can be very effective in controlling fugitive emissions. Some services, however, require the use of dual mechanical seals, either in a pressurized (double) or a non-pressurized (tandem) seal arrangement, to ensure containment of fugitive emissions. Liquid lubricated double mechanical seals require the use of a pressurized barrier liquid that is circulated within the cavity between the inner and outer seals. Such liquid barrier systems require investment in additional equipment and monitoring time for proper maintenance. Moreover, the barrier liquid must be selected with care, since it must be compatible with the process fluid and must not contaminate the process fluid under normal operating conditions. The barrier liquid must also be safe for release to the atmosphere if it leaks from the barrier system.

Gas sealing technology is now available for application to pumps. This technology offers a choice of designs. One basic design category incorporates low-friction, contacting seal faces and provides key performance benefits for a specific range of

applications frequently found in the process industries. The second basic design category incorporates zero-friction noncontacting seal faces, based on seal technology used to date in the compressor industry. In general, gas sealing technology offers distinct advantages over liquid lubricated sealing technology in terms of system operation and configuration. The application and resulting benefits of gas lubricated dual seals vs liquid lubricated dual seals are discussed. Source data are based on product test evaluations and user case histories.

### TECHNOLOGY BENEFITS

Typical liquid barrier and gas barrier double mechanical seals used in industry today are shown in Figure 1 and Figure 2, respectively. At first glance, one might feel that these two systems are identical because they perform essentially the same purpose. Both use a compatible fluid, either a liquid or a gas, under pressure to provide a barrier against process liquid leaking into the cavity between the mechanical seals. This barrier also provides a clean, cool fluid around the mechanical seals and prevents the escape of process fluid to the atmosphere. This meets one of the most important requirements of double mechanical seals today, zero emissions.

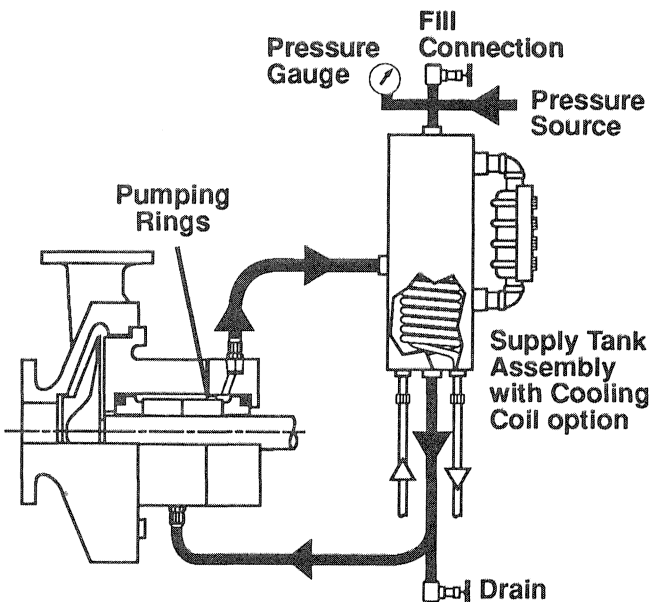


Figure 1. Double Liquid Barrier Seal.

From here, the differences between the two systems are substantial. Comparisons of the system characteristics for double liquid and double gas barrier seals are shown in Table 1.

**Supply Tank**—Liquid lubricated seals require a supply tank or reservoir to provide the barrier fluid between the double mechanical seals. Gas lubricated seals require a constant source of supply for the gas barrier between the double mechanical seals.

**Pumping Device/Cooling Coils**—For API 682 specifications, all liquid lubricated double mechanical sealing systems are required to incorporate a pumping device to circulate the liquid through the cavity between the double mechanical seals, out through the supply tank, and back into the cavity between the double mechanical seals. Depending on the operating conditions, cooling coils may need to be added to the barrier fluid supply tank. Generally, pumps operating either at speeds greater than 2900 rpm, with shaft diameters over 2.000 in (51 mm) or on

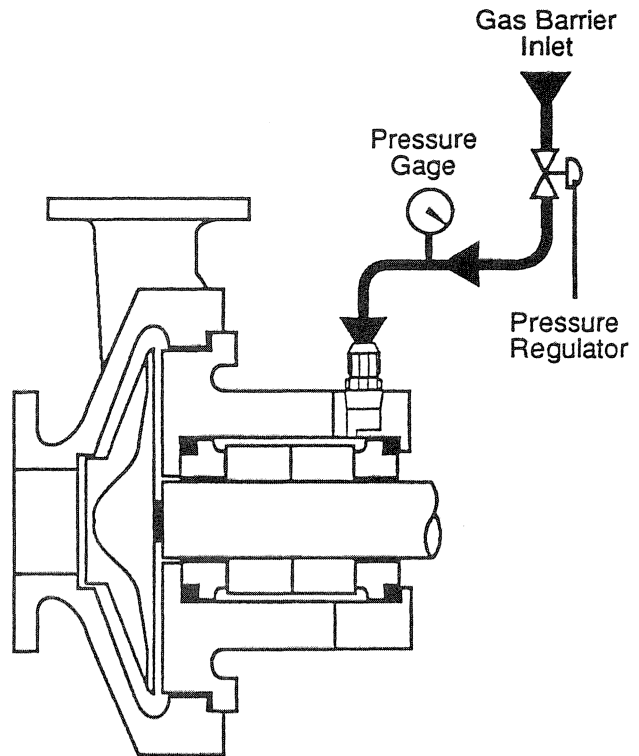


Figure 2. Double Gas Barrier Seal.

higher temperature service applications, will require cooling coils. Gas lubricated double mechanical seals usually operate with the gas barrier dead ended into the cavity between the double mechanical seals, thus requiring no circulating devices or cooling mechanisms.

**Additional Maintenance**—Liquid lubricated sealing systems with pumping devices and supply tank reservoirs have measured leakage rates of 2.0 to 5.0 ml/day, with this leakage going into both the process stream and the atmosphere. These minute amounts of leakage require periodic maintenance of the supply tank reservoir. Normally, these maintenance checks are made every week in the process plant with periodic refills. Gas lubricated sealing systems use a continuous source of supply for the dead ended gas barrier. As leakage of the gas occurs, this loss is automatically replenished from the constant supply source, thus requiring no maintenance checks or adjustments.

**Elimination of Process Liquid Contamination**—Dual gas barrier seals will typically be pressurized with nitrogen, an inert gas, which is not considered as a contaminate to most process systems. This is viewed by many users as having substantial advantages over liquid lubricated systems. In a failure mode, liquid lubricated systems can leak large amounts of the barrier fluid into the process system causing contamination of the process purity or leak barrier fluid into the ground or atmosphere.

**Horsepower Consumption**—Operation of typical liquid lubricated seals requires from 10 to 20 times the horsepower of standard dual gas barrier seals. This difference in horsepower consumption can oftentimes offset the cost of retrofitting equipment from liquid lubricated to gas barrier seals. The effects of horsepower consumption are shown in Figure 3 and Figure 4.

### CONTACTING AND NONCONTACTING SEAL FACE DUAL GAS SEALS

Before discussing the types of double gas barrier seals for pumps, it is important to understand why liquid lubricated seals

Table 1. Technology Comparison.

SYSTEM CHARACTERISTICS	DOUBLE LIQUID BARRIER SEAL	DOUBLE GAS BARRIER SEAL
Meets EPA Clean Air Act Requirements	Yes	Yes
Requires Supply Tank	Yes	Yes
Requires Pumping Device	Yes	No
Requires Cooling Water	Yes	No
Requires Periodic System Maintenance	Yes	No
Eliminates Process Fluid Contamination	No	Yes
Reduces Horsepower	No	Yes
Reduces Energy Consumption	No	Yes
Simplified Piping System	No	Yes
Reverse Pressure Capabilities	Yes	Yes

present problems when conditions cause them to run dry. Liquid lubricated mechanical seals require high seal face closing loads to overcome O-ring secondary seal drag and to prevent leakage of the process fluid while accommodating for seal face runout and fluctuating seal face loads. The seal face closing loads are supported by both the process liquid that is being pumped and by seal face contact. In many cases, the process liquid being pumped is also effective in carrying away the seal face heat generated.

When the process liquid is removed from the seal faces due to pump operations that result in dry running, cavitation, etc., the high seal face closing loads are carried by seal face contact alone. If the process liquid is totally removed from the seal chamber, the heat dissipation capabilities provided by the pro-

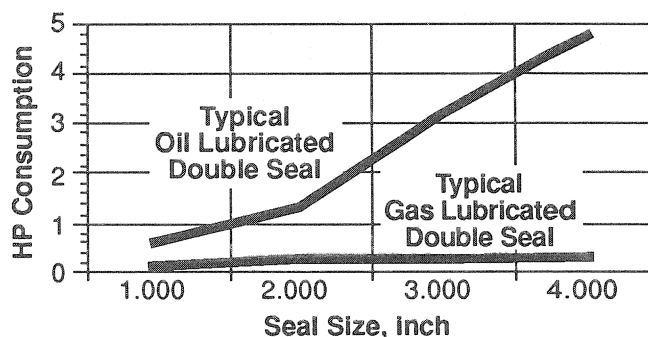


Figure 3. Horsepower Requirements.

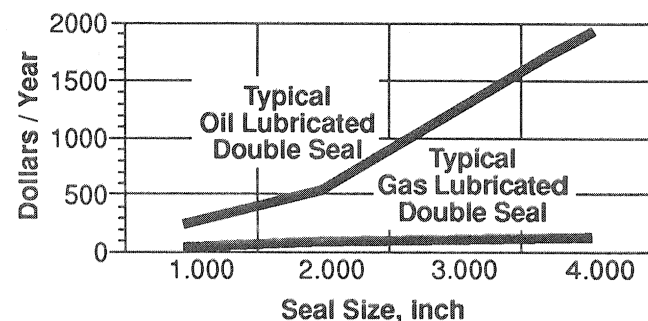


Figure 4. Annual Energy Cost.

cess liquid are also removed. The lack of adequate heat dissipation and the increase in rubbing contact at the seal faces will cause a rise in the seal face temperatures. These temperatures may exceed the ratings of both the seal face materials and the elastomer O-rings. Excessive temperatures can cause high seal face wear and thermal damage to O-ring secondary seals.

There are two basic types of double gas barrier seal designs for centrifugal pumps. One uses contacting seal faces and the second uses noncontacting seal faces.

Contacting seal face designs have been in operation for many years on equipment such as mixers, blowers, and fans. Seal face materials can operate in the contacting dry mode as long as the seal face loads are reduced and the heat generated by the seal faces is adequately dissipated. Contacting seal face gas seals use hydrostatic forces to reduce the seal face closing loads, but these hydrostatic forces do not create seal face separation. Sometimes, a grooved seal face pattern is used to assist in providing these hydrostatic forces and in cooling the seal faces. Contacting seal face designs also use wide seal faces to assist in reducing the overall unit loading on the seal faces.

Noncontacting seal face designs have been in operation on compressors since the mid 1970s. Noncontacting seal face gas seals use various configurations of shallow grooves to provide seal face separation. These shallow grooves provide both hydrostatic and hydrodynamic lift to create seal face separation. When pressure is applied to the seal, the forces exerted on the seal faces are hydrostatic and are present when the seal is either stationary or rotating. Hydrodynamic forces are generated by viscous shear of the gas film by the nongrooved counterface during rotation. Wide seal faces are desired for noncontacting seal face designs to provide maximum load support at slow peripheral speeds and low pressures.

Both designs require that deformation of the seal faces be minimized so that both seal faces can be kept as parallel as possible over the total range of operation. Both designs provide good performance in a gas medium where there is a gas at both

the inside diameter and outside diameter of the seal faces. For centrifugal pump applications, the inner seal has the process liquid at one side of the seal faces. In many of these applications, the fluid at this sealing interface is constantly changing due to dry running, cavitation, and off-design pump operation.

**SEAL HEAT GENERATION**

Noncontacting seal face designs have less seal generated heat compared to contacting seal face designs. Contacting seal face designs are subject to gas film viscous shear heat generation as well as heat generated by the rubbing contact between the seal faces. Noncontacting seal face designs are subject to only heat generated from viscous shearing of the fluid between the seal faces. The spring force applied in both noncontacting and contacting seal face dual gas seal designs is significantly smaller than in liquid lubricated designs. This, along with precise hydrostatic balancing, allows the contacting seal face designs to run adequately with light seal face contact and also allows the noncontacting seal face designs to survive seal face contact that might occur during pump startup or upset conditions. This reduction in heat generation can be translated directly to horsepower savings.

**DUAL GAS SEAL LEAKAGE CONSIDERATIONS**

Leakage is approximately proportional to the cube of the gap between the sealing faces. Noncontacting seal face designs have sealing gaps ranging from 0.00005 in to 0.00015 in (0.0013 mm to 0.0040 mm), while contacting seal face designs have sealing gaps [4] approximately three times the combined roughness of the as-worn seal faces. This combined roughness has been measured to be 0.000005 in to 0.000010 in (0.00013 mm to 0.00025 mm), which would result in a sealing gap of 0.000015 in to 0.000030 in (0.00038 mm to 0.00076 mm).

The comparison of leakage for contacting vs noncontacting seal face designs with equal seal face widths for a 2.000 in (50.8 mm) diameter shaft operating at 50 psig (345 kPa) and from 0 to 3600 rpm is shown in Figure 5. Notice the increase in leakage for the noncontacting seal face design vs the shaft speed. This is a result of the hydrodynamic lift created by the shallow seal face groove geometry which, in turn, increases the leakage.

The effect of entrained gas on the performance of centrifugal pumps was reported [3] and is illustrated in Figure 6 [5]. The maximum amount of inert gas entrainment is recommended to be less than three percent by volume. Even the highest nitrogen leakage rate shown in Figure 5 for the noncontacting seal face design would produce only 0.075 percent entrained gas in a pump that is circulating 50 gpm of water. This entrained gas

must, however, be removed from the pumping system at some time. Therefore, removal of the nitrogen that could collect in the system piping should be considered when specifying a double gas barrier seal.

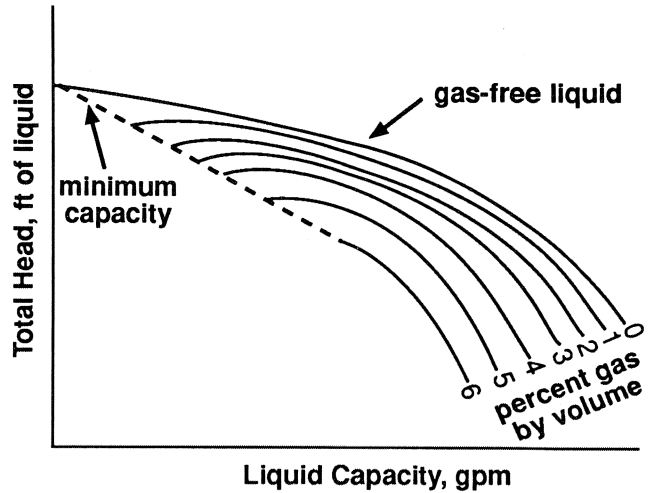


Figure 6. Effects of Entrained Gas.

**SEAL FACE GROOVE PATTERNS**

Many types of seal face groove patterns are used for contacting and noncontacting seal face dual gas seal designs. The two most common groove patterns are used for noncontacting seal face designs. They are the pumping groove design and the pocket groove design as shown in Figure 7 [6].

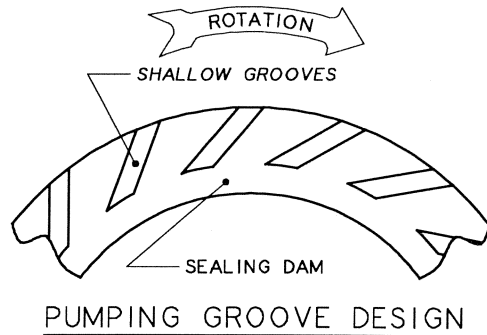
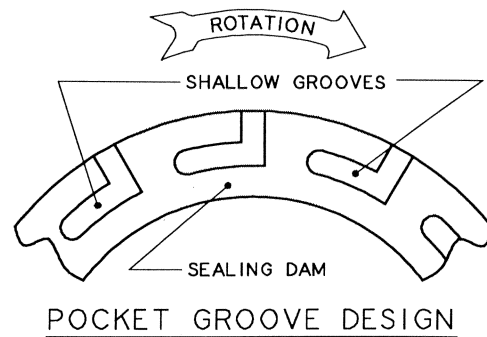


Figure 7. Noncontacting Seal Face Groove Patterns.

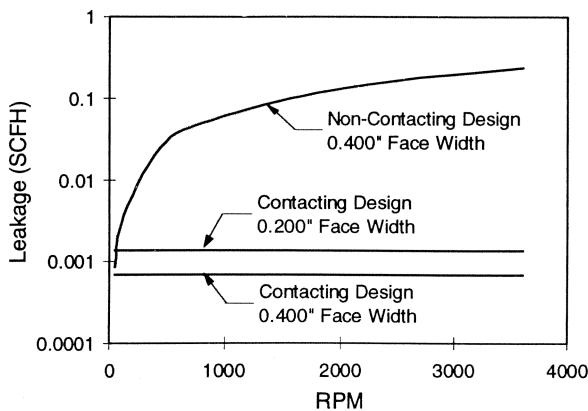


Figure 5. Gas Seal Leakage.

With the pumping groove pattern, grooves are slanted in one direction around the seal face. As the pump rotates, the grooves pump gas inward while the dam at the groove's inner dimension restricts its flow. This generates a pressure at the dam area that provides a hydrodynamic lift. With the pocket groove pattern, pockets are grooved into the seal face. As the pump rotates, these pockets scoop up the gas while the lands around the pockets restrict escape of the gas increasing its pressure and providing hydrodynamic lift. Both of these designs along with most other noncontacting seal face groove designs require shallow groove depths of less than 0.0003 in (0.0076 mm) to provide proper face separation [7].

Another important parameter that should be considered when comparing the performance of noncontacting seal face gas seal designs is seal face film stiffness. If the sealing gap between the seal faces were to suddenly decrease, it is important for optimum performance that the seal face opening force increases sharply to prevent contact of the seal faces and to allow the seal faces to open back up to the original operating gap. The rate at which axial opening and closing forces change with seal face gap is measured in lbs/in and is called stiffness. In any noncontacting seal face design, the stiffness should be as large as possible to prevent seal face contact. When going from unidirectional seal face patterns, as shown in Figure 7, to bidirectional seal face patterns, the stiffness is reduced by as much as 50 percent in most designs. This is due to the fact that, with a bi-directional groove pattern, fewer grooves can be allocated to accommodate seal rotation in either direction [7].

In comparison, the seal face grooves used for contacting seal face designs do not need to be as shallow because the grooves do not provide a significant dynamic lift effect. Contacting seal face groove depths can be as deep as 0.062 in (1.57 mm). The contacting seal face grooves are mainly used for directing the pressure down to the sealing dam to produce a hydrostatic effect. Some designs incorporate straight radial feed grooves while other designs incorporate angled feed grooves to force the barrier gas through the seal face for a cleansing and cooling effect. The pads between the grooves provide a bearing support for the seal to reduce the overall unit load. These bearing pads typically have a waviness that generates a slight hydrodynamic lift. However, the majority of the contact load reduction is due to hydrostatic forces. Typical contacting seal face angled and straight radial feed groove designs are shown in Figure 8.

**DUAL GAS SEAL BARRIER SYSTEMS**

Nitrogen and instrument air are the most common barrier gases used for double gas seals. If compressed air (plant air) is used, it is important to make sure that oil and any particulates are removed from the feeder lines. The simplified barrier system for a double gas seal is shown in Figure 9. The minimum components required for this system are a regulator and a pressure gage to ensure that the barrier gas pressure remains from 20 psig to 50 psig (138 to 345 kPa) greater than the process fluid being pumped. A flowmeter can be used as an indicator of the seal's performance. With a quick glance at the flowmeter, the operator can determine how well the seal is performing by noting any changes in gas flow. A needle valve or an orifice may be used to limit the maximum amount of nitrogen leakage in the event of a seal failure. A pressure switch and flow switch can be used as an alarm or shutdown mechanism for loss of the barrier pressure or excessive nitrogen flow, respectively. A 15 micron inline filter is recommended for gas barrier systems that may have foreign particulates present. A three way valve allows the pump and seal to continue operation while maintenance is being conducted on the barrier gas system by connecting an external gas supply to the three way valve to keep the barrier cavity pressurized.

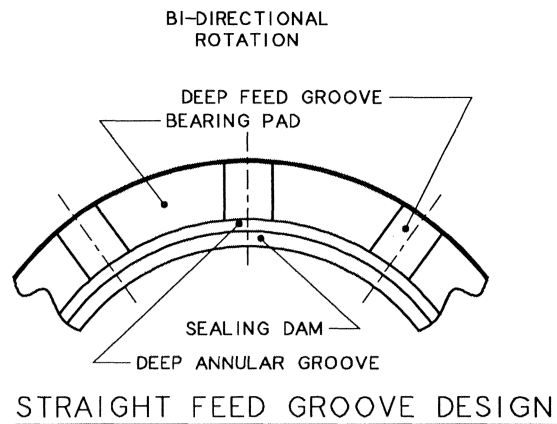
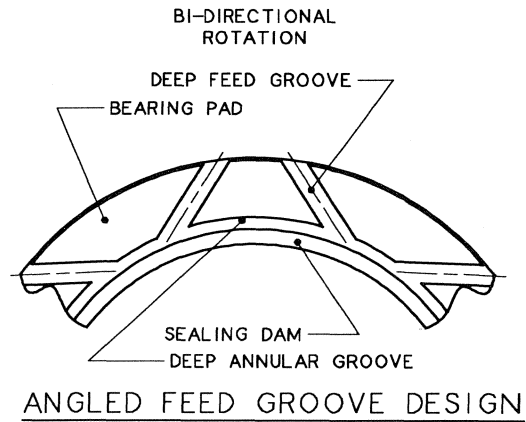
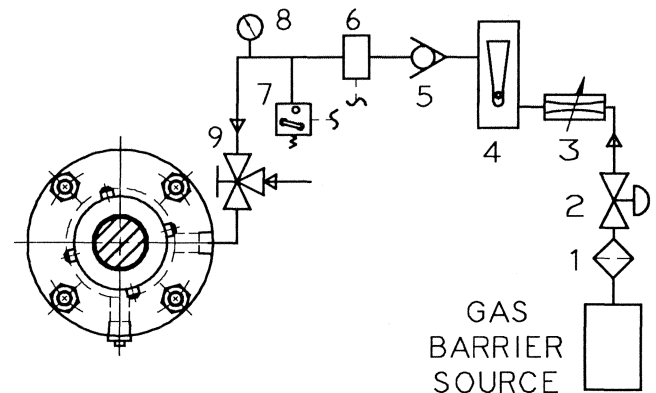


Figure 8. Contacting Seal Face Groove Patterns.

For some process fluids that require their temperatures to be maintained above 200°F (93°C) to prevent solidification, a steam barrier system may be beneficial. A typical steam barrier



- 1-FILTER
- 2-REGULATOR
- 3-NEEDLE VALVE
- 4-FLOW METER
- 5-CHECK VALVE
- 6-FLOW SWITCH
- 7-PRESSURE SWITCH
- 8-PRESSURE GAGE
- 9-THREE WAY VALVE

Figure 9. Double Gas Seal Barrier System.

system schematic is shown in Figure 10. A pressure reducing valve is required to maintain an adequate barrier cavity differential pressure above the pump seal chamber pressure and steam traps are required to channel the condensate from the piping and the seal to a drain or a condensate return. If maintenance is required on the pressure reducing valve, a bypass line can be installed.

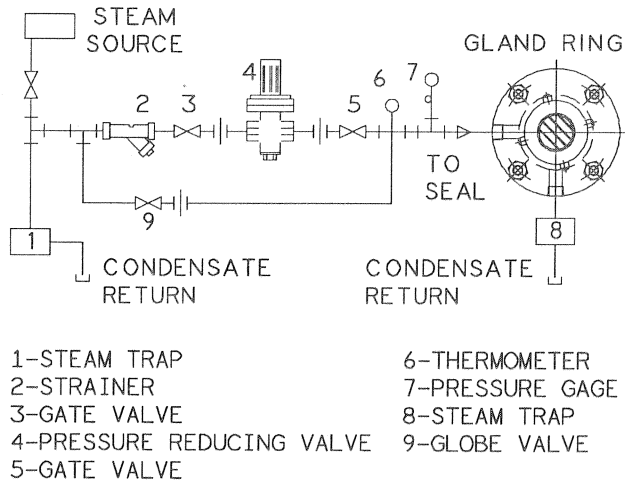


Figure 10. Double Steam Seal Barrier System.

## SEAL ARRANGEMENTS

There are several basic seal arrangements for double gas seals as illustrated in Figure 11. The most common design is the canister arrangement which utilizes back-to-back inner and outer seal designs. Either contacting or noncontacting seal faces can be used in this arrangement. When using this arrangement, it is imperative that the fluid being pumped be evaluated for the size and type of solids present. Computational fluid dynamics modelling, confirmed through actual testing, indicates that solids from the process fluid tend to get trapped and accumulate under the inner seal and can prevent the inner seal from tracking properly.

Some canister arrangements utilize a conventional contacting seal face without seal face grooves for the inner seal. This design relies on the cooling of the process fluid being pumped to obtain good performance and is not usually recommended for continuous dry running operation. The major advantages of this design are its capability to operate as a single seal during pressure reversals and a low level of leakage into the process fluid.

Use of an inline seal arrangement in conjunction with seal chamber flow modifications is one way of reducing the potential for seal hangup due to polymerization of the process fluid or the presence of solids. Use of conventional contacting seal faces without seal face grooves on the inner seal allows this design to operate with good single seal performance during pressure reversal conditions.

For some process fluids, there may be an advantage in using a repeller type pump as shown in Figure 12. The repeller evacuates the process fluid from the seal chamber during dynamic operation. This may be beneficial for canister seal arrangements used with process fluids that tend to polymerize or process fluids that contain solids. When using a repeller type pump, it is important that a gas seal design, that does not rely on the process fluid for cooling be applied since the removal of seal generated heat is low. This can be accomplished by using either

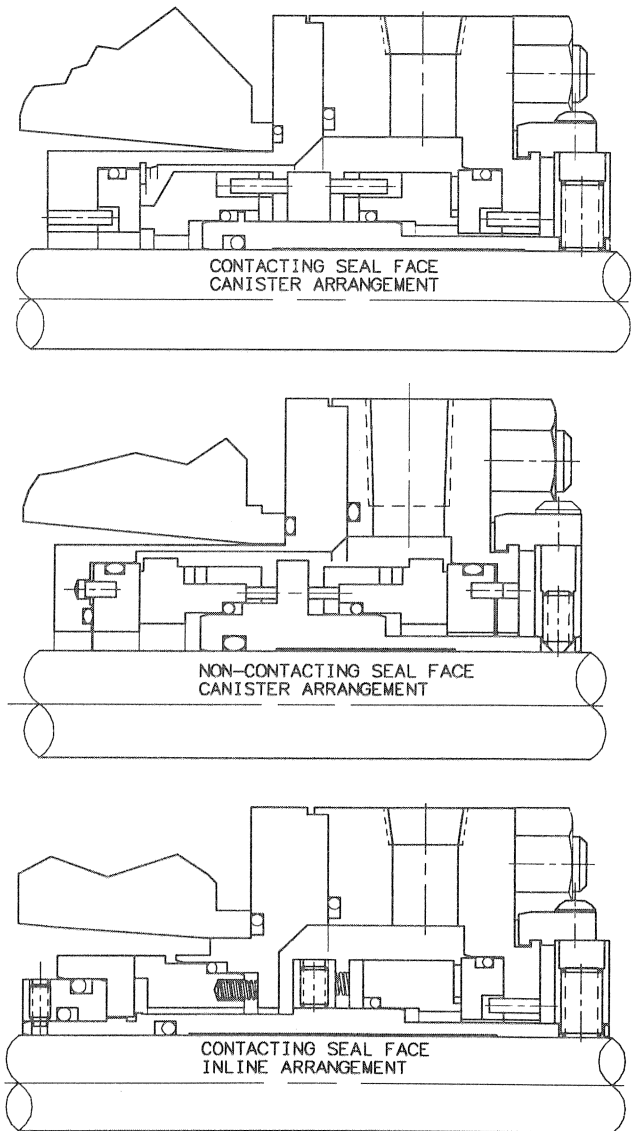


Figure 11. Double Gas Seal Arrangements.

a noncontacting seal face canister arrangement or a contacting seal face canister arrangement that utilizes a grooved inner seal face pattern.

## PROCESS FLUID CONSIDERATIONS

Double gas seals applied to pumps are different from those applied to compressors, mixers, fans, and blowers in that the inner seal is immersed in the process fluid being pumped either at the inside or outside diameter depending on the seal arrangement chosen. Evaluation of seals operating at customer sites and through research testing indicates that, even though the barrier system gas pressure is greater than the process fluid pressure, there is still a portion of the gap between the seal faces that contains process fluid due to either capillary action or the wiping effect from shaft or sleeve runout, as illustrated in Figure 13.

When dealing with process fluids that contain solids or that tend to polymerize, coke, or salt out, it is important to remove the entrained solids by either applying a flush over the seal faces, providing a seal chamber that flushes the solids away, or using

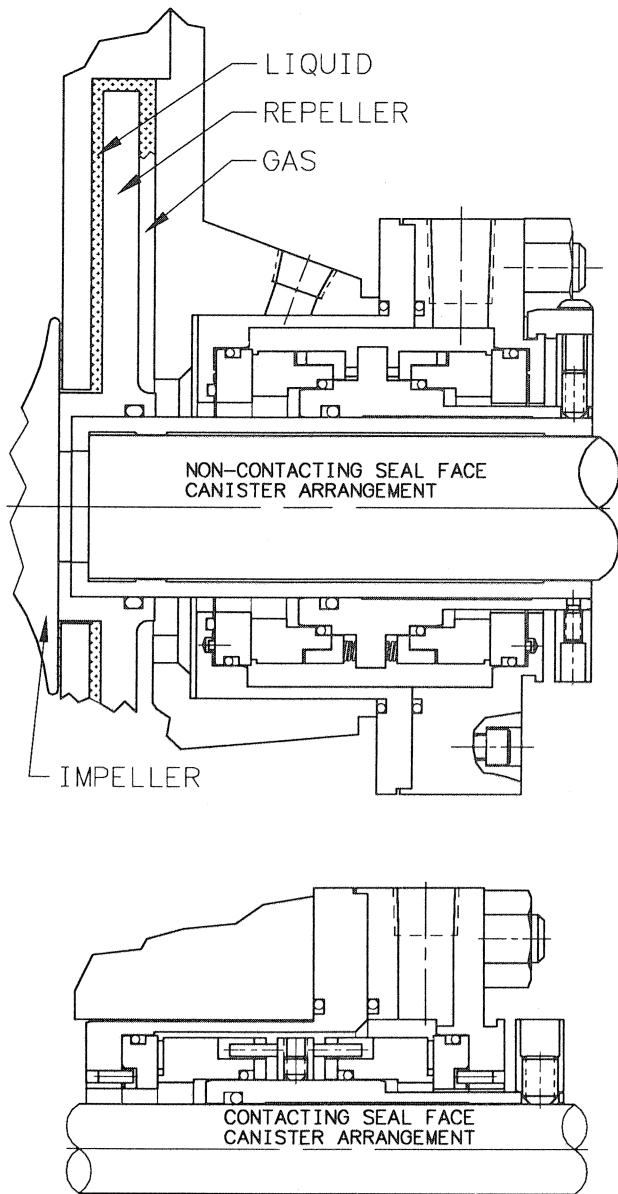


Figure 12. Double Gas Seals for Repeller Pumps.

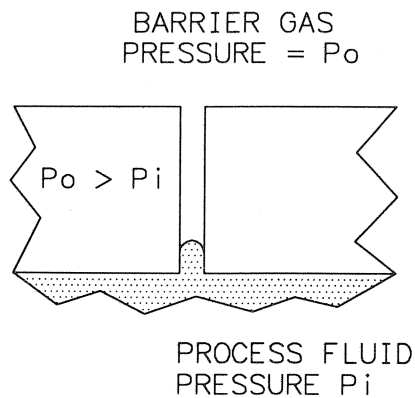


Figure 13. Fluid Phase Between Gas Seal Faces.

a repeller pump that keeps the process fluid away from the seal during dynamic operation. If a carbon material is used for one of the seal faces, such solids could slowly erode this seal face until excessive gas leakage occurs. In some applications, use of hard-on-hard seal faces such as silicon carbide vs silicon carbide may be beneficial. Hard-on-hard seal faces may also be beneficial when highly corrosive (strong oxidizing) process fluids are being pumped.

When applying double gas seal designs for light fluid applications that have low boiling point margins, one should choose a design that provides the lowest seal generated heat to prevent vaporization of the process fluid. Although vaporization of the process fluid will not usually affect the performance of the gas seal, it could lead to pump vapor lock.

OFF-DESIGN PUMP OPERATION

Contacting and noncontacting seal face double gas seals were operated on a standard 2X1-10 ANSI B73.1 pump equipped with an enlarged tapered bore seal chamber during off-design pump operation. The contacting seal face design was a canister arrangement with an inner seal that did not contain seal face grooves. The noncontacting seal face design was also a canister arrangement but did contain shallow seal face grooving in both the inner and outer seals.

The test circuit used during all testing is shown in Figure 14. All tests were conducted with softened municipal water as the process fluid. The suction reservoir had a 400 gallon capacity with a water level approximately 4.0 ft (1.2 m) above the pump center line. The pump was equipped with a fixed speed, 40 hp motor operating at 3600 rpm. All seal designs were sized to fit the 1.875 in (47.62 mm) shaft diameter of the pump. Both seal arrangements contained seal faces of carbon graphite vs silicon carbide and fluorocarbon compound O-rings. The process fluid temperature was maintained at 150°F (66°C) by use of a cooling coil placed in the suction reservoir. Nitrogen was used as the barrier gas and was maintained at 50 psig (345 kPa) for all tests. Vibration was monitored by mounting accelerometers in the vertical, horizontal, and axial locations on the bearing housing near the gland ring and on the pump casing. These monitoring locations were the same as those used during previous evaluations of mechanical seals under off-design operating conditions [3].

Volumetric flow through the pump was recorded manually using a differential-pressure flowmeter. The nitrogen barrier

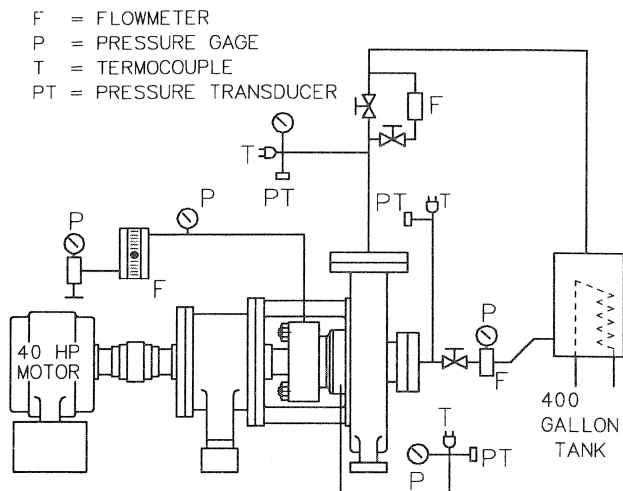


Figure 14. Off-Design Pump Operation Test Circuit.

flowrate was measured using a mass flowmeter. Seal face temperatures were monitored with thermocouples mounted 0.06 in (1.52 mm) back from the stationary seal face. The pump was equipped with appropriate transducers to record pressures, temperatures, and vibration levels, using a computer based data acquisition system.

In order to document the performance of the double gas seal designs vs pump operating conditions, the pump was operated at various points on and off the established pump performance curve. Pump operation was conducted for 12 hr at the best efficiency point (bep), 12 hr at a low flow high head condition (10 percent of bep flow), 12 hr at a high flow, low head, low NPSH condition (127 percent of bep flow), 12 hr at bep flow with a low NPSH cavitation condition, and 12 hr at a dry running condition. Average seal face temperatures recorded during this off-design testing are shown in Figure 15 and compared to similar testing performed with liquid lubricated seals [3]. Testing was also performed to simulate loss of the gas barrier.

#### Pump BEP Operation

Testing at the pump bep condition was achieved by fully opening the suction valve and throttling the discharge valve to obtain a flow of 150 gpm. This stage was important as it established baseline operating data to which succeeding off-design conditions could be compared. This testing produced an average seal chamber pressure of 13 psig (90 kPa). The pump ran with average vibration levels of 0.18 in/sec (4.6 mm/sec) for both seal arrangements.

The contacting seal face design ran with average inner seal face temperatures of 154°F (68°C) and outer seal face temperatures of 158°F (70°C). The nitrogen consumption rate was 0.2 standard cubic feet per hour (scfh). The noncontacting seal face design ran with average inner seal face temperatures of 151°F (66°C) and outer seal face temperatures of 149°F (65°C). The nitrogen consumption rate was 0.73 scfh.

#### 10 Percent of BEP Flow Operation

Testing at the 10 percent bep flow condition was achieved by throttling the discharge valve to obtain 15 gpm flow. This produced an average pressure of 18 psig (124 kPa) in the seal chamber. The pump ran with average vibration levels of 0.22 in/sec (5.6 mm/sec) for both seal arrangements, which was 22 percent higher than bep operation.

The contacting seal face design ran with average inner seal face temperatures of 159°F (71°C) and average outer seal face temperatures of 220°F (105°C). The noncontacting seal face design ran with average inner seal face temperatures of 147°F (64°C) and average outer seal face temperatures of 150°F (66°C). The nitrogen consumption rate was 0.15 scfh for the contacting seal face design and 0.75 scfh for the noncontacting seal face design.

#### 127 Percent of BEP Flow and Low NPSH Operation

Testing at 127 percent of bep flow and a low NPSH was achieved by fully opening the suction valve and throttling the discharge valve to obtain a 190 gpm flow. This produced a pressure of 10 psig (69 kPa) in the seal chamber and created a low NPSH condition. The pump ran with average vibration levels of 0.21 in/sec (5.3 mm/sec) for both seal arrangements which was 17 percent higher than bep operation.

The contacting seal face design ran with average inner seal face temperatures of 152°F (67°C) and average outer seal face temperatures of 185°F (85°C). The noncontacting seal face design ran with average inner seal face temperatures of 151°F (66°C) and average outer seal face temperatures of 150°F (66°C).

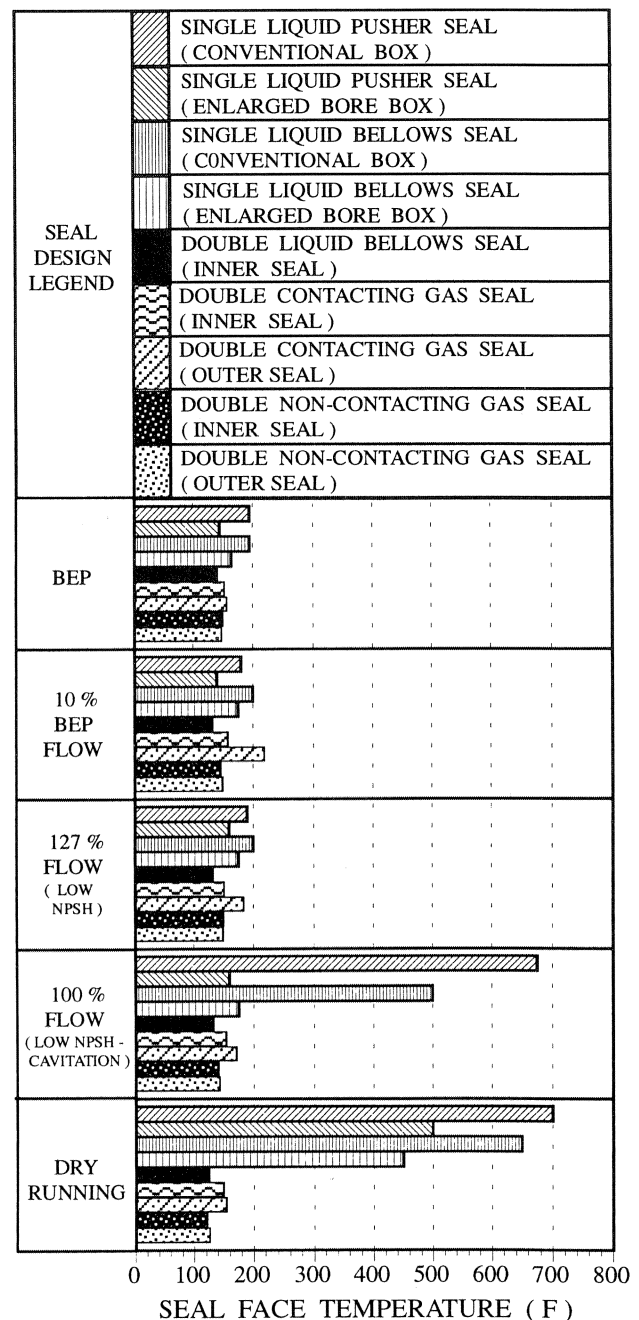


Figure 15. Off-Design Operation Seal Face Temperatures.

The nitrogen consumption rate was 0.15 scfh for the contacting seal face design and 0.73 scfh for the noncontacting seal face design.

#### Low NPSH at 100 Percent of Bep Flow Operation

Testing at the low NPSH at 100 percent of bep flow condition was achieved by fully opening the discharge valve and throttling the pump suction valve to obtain a 150 gpm flow. This produced cavitation at the pump suction valve and a vacuum of 22 in (0.56 m) of Hg in the seal chamber. The pump ran with average vibration levels of 0.30 in/sec (7.6 mm/sec) for both seal arrangements, which was 67 percent higher than bep operation. Radial and axial vibrations as high as 0.37 in/sec (9.4 mm/sec) occurred during this operation.



The contacting seal face design ran with average inner seal face temperatures of 155°F (68°C) and average outer seal face temperatures of 172°F (78°C). The noncontacting seal face design ran with average inner seal face temperatures of 142°F (61°C) and average outer seal face temperatures of 144°F (62°C). The nitrogen consumption rate was 0.10 scfh for the contacting seal face design and 0.94 scfh for the noncontacting seal face design.

#### *Dry Running Operation*

Dry running operation was achieved by fully closing the suction valve in the suction line to stop water flow from the suction reservoir and opening the suction line to atmosphere through a small bleed line. The seal chamber pressure dropped to zero psig and the average vibration readings dropped to 0.08 in/sec (2.0 mm/sec) for both seal arrangements, which was 56 percent lower than bep operation.

The contacting seal face design ran with average inner seal face temperatures of 150°F (66°C) and average outer seal face temperatures of 155°F (68°C). The noncontacting seal face design ran with average inner seal face temperatures of 123°F (51°C) and average outer seal face temperatures of 127°F (53°C). The nitrogen consumption rate was 0.03 scfh for the contacting seal face design and 0.71 scfh for the noncontacting seal face design.

#### *Vibration Measurements*

The vibration velocities encountered during these various cycles of pump operation were the same as those recorded during previous evaluations of mechanical seals under off-design operating conditions [3]. The lowest magnitudes of vibration were measured during the bep and dry running cycles with values of 0.23 in/sec (5.8mm/sec) or less. The greatest magnitudes of vibration were measured during the low NPSH cavitation cycle in the axial direction on the pump casing with values reaching 0.37 in/sec (9.4mm/sec).

#### *Loss of Gas Barrier*

Contacting and noncontacting seal face gas barrier seals were tested to simulate the loss of barrier fluid. The pump was run at bep and then the gas barrier supply was removed to see how well the inner seal was able to withstand pressure reversals. Typical gas seal designs rely on an O-ring shift during pressure reversals to provide an adequate seal balance for keeping the seal faces closed. The contacting seal face arrangement with conventional nongrooved inner seal face technology provided a process fluid leakage rate of 0.000002 gpm. The noncontacting seal face canister arrangement was designed with adequate pressure reversal balancing across the dam portion of the inner seal face as shown in Figure 7. However, when a pressure reversal was applied to this seal arrangement, the inner seal leaked at a rate of 0.007 gpm. This indicates that for noncontacting seal face arrangements with shallow seal face grooves, pressure reversal balancing must be applied across the entire sealing face and not just the dam area.

## USER CASE HISTORIES

Double gas seals are growing in popularity in the chemical industry as an economical means of reducing emissions and increasing equipment reliability. A few case histories are presented to indicate the success of this technology.

A chemical manufacturer in Michigan was having difficulty sealing a rail car unloading pump that was unloading fluids such as chloroform, Freon 114, and carbon tetrachloride. The pump was an ANSI Group II with a 1.875 in (47.62 mm) diameter shaft

rotating at 1750 rpm. The seal chamber was pressurized to 20 psig (138 kPa). A single bellows seal design was being used. Typically, these seals were able to operate for less than one month due to the long periods of dry running experienced after the rail car was emptied. The customer did not want to install a double liquid barrier seal due to the possible contamination of the product by the barrier liquid and the need for selecting different barrier fluids for the chemicals being pumped. A double gas barrier seal was installed on this pump and performed well for nine months until the pump was taken out of service.

A chemical manufacturer in Tennessee had a critical chemical process pump that cavitated continuously. This manufacturer decided to evaluate a double gas barrier seal for this application. To test out this concept, they decided to first operate it on a condensate pump that had similar cavitation problems. The condensate pump was an ANSI Group II with a 1.750 in (44.45 mm) diameter shaft rotating at 1750 rpm. The condensate was at a temperature of 250°F (121°C). A single pusher seal had been providing less than one month's life. The customer went back to using packing which lasted for three months. They then installed the double gas barrier seal with a 40 psig (276 kPa) nitrogen barrier pressure. This seal has been running well for over 22 months.

A chemical manufacturer in Texas had an application that was pumping 93 percent sulfuric acid at 140°F (60°C). The pump was an ANSI Group II with a 1.875 in (47.62 mm) diameter shaft rotating at 1800 rpm. The seal chamber pressure was 9.0 psig (62 kPa). The pump was operating at minimum flow conditions which caused single pusher seals to last only one week and single bellows seals to last from three to five weeks. A double gas barrier seal was installed on this pump with a 40 psig (276 kPa) nitrogen barrier that operated nine months.

A second chemical plant in Texas had an application that was pumping a mixture of 90 percent acrylic acid and 10 percent acetic acid at 200°F (93°C). The pump was an ANSI Group I with a 1.375 in (34.92 mm) diameter shaft rotating at 3600 rpm. The pump was running off-design, which caused it to cavitate and run occasionally with negative seal chamber pressures. Single bellows and single pusher seals were lasting from one week to one month. A double gas barrier seal was installed on this pump. This seal has been performing well for six months.

Another chemical plant in Texas had an application that was pumping 95 percent sulfuric acid at 100°F (38°C). The pump was an ANSI Group I with a 1.375 in (34.92 mm) diameter shaft rotating at 1750 rpm. The user wanted to reduce the emissions and reduce the possibility of personal injury in this application. However, they did not have a liquid barrier fluid that was compatible with this process fluid to apply double liquid lubricated seals. The existing single pusher seal was providing approximately one year's life. They installed a gas barrier seal on this pump and performance has been good for six months.

A chemical manufacturer in Indiana had a solvents recirculation pump that was pumping primarily methylene chloride but the tank was also used for toluene, methyl ethyl ketone, and other solvents. The typical product temperature was 100°F (38°C). The pump was an ANSI Group II with a 1.750 in (44.45 mm) diameter shaft rotating at 1750 rpm. The typical seal chamber pressure was 20 psig (138 kPa). The existing double pusher seals with a liquid barrier system were failing every three months. A double gas barrier seal was installed on this pump. This seal has been operating well for 16 months.

## CONCLUSIONS

The purpose for presenting this information is to better inform pump users about the availability and operating characteristics of dual gas seal technology. Recent testing and field experience

have indicated that double gas barrier seals provide some advantages over double liquid barrier seals and single liquid mechanical seals. Contacting and noncontacting seal face designs as well as alternate design arrangements provide different advantages and disadvantages. The impact of various operating characteristics on the performance of four double seal arrangements is shown in Table 2 supporting the following conclusions.

- Double gas barrier seals can eliminate product contamination, reduce power consumption, and reduce auxiliary system and maintenance costs when compared to double liquid barrier seal designs.

- Noncontacting seal face gas barrier seals have lower horsepower consumption than contacting seal face gas barrier seal designs.

- Noncontacting seal face gas barrier seals have higher consumption rates of the barrier gas than contacting seal face gas barrier seals. Typical leakage rates of noncontacting seal face designs do not affect pump performance.

- With process fluids containing solids, canister seal arrangements tend to trap the solids around the inner seal design which can lead to seal hangup. Inline seal arrangements are less susceptible to this effect. Use of repeller pumps are one way to keep solids away from the seal during dynamic operation.

- Double gas barrier seals work reasonably well in thermal sensitive fluids. Noncontacting seal face designs work better than contacting seal face designs due to their lower seal generated heat. Inline arrangements work better than canister arrangements due to the greater access of the inner seal faces to the process fluid to aid in reducing seal generated heat. Double liquid barrier seals can also work well in these fluids when adequate cooling is applied to remove the seal generated heat through the circulating barrier liquid.

- Double gas barrier seals use lower spring loading than double liquid barrier seals which increases the potential for seal hangup when pumping process fluids containing solids or thermal sensitive process fluids.

- Contacting seal face double gas barrier seals can withstand pressure reversals better than noncontacting seal face gas barrier seals due to their narrower seal face widths and deeper seal face grooves. Pressure reversal balances must be calculated across the entire seal face on noncontacting seal face designs. Double liquid barrier seals can withstand a pressure reversal and return to normal operation. A small amount of process fluid mixing into a barrier liquid is not as destructive as a small amount of process fluid mixing into a barrier gas. In many instances, a double gas barrier seal will not be able to return to normal operation after a pressure reversal.

- Double gas barrier seals perform very well during off-design pump operations. Noncontacting seal face designs generate the lowest seal face temperatures and are the best suited for total dry running operation.

- Double liquid barrier seals can also perform well under these conditions as long as adequate barrier liquid cooling is applied and as long as the barrier liquid level is properly maintained, since higher vibration levels associated with off-design operations tend to increase the barrier liquid leakage rate.

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Table 2. Application Guide.

TYPICAL APPLICATION OPERATING CHARACTERISTICS	IMPACT ON THE PERFORMANCE OF VARIOUS SEAL ARRANGEMENTS			
	A	B	C	D
Limiting of Process Fluid Contamination	5	1	1	1
Horsepower Consumption	5	2	2	1
Energy Consumption	5	2	2	1
Dirt, Debris, and Process Fluid Solids	3	4	3	4
Thermal Sensitive Process Fluids	3	4	3	2
Resistance To Seal Hang Up	2	3	3	3
Pressure Reversals	2	3	3	4
Off - Design Pump Operations	4	3	3	2
Pressure Requirements	2	3	3	1
Temperature Requirements	3	4	4	3
Speed Requirements	1	1	1	1
Seal Size Requirements	2	2	2	2
A = Double Liquid Barrier Seal : Contacting Seal Face Design B = Double Gas Barrier Canister Seal : Contacting Seal Face Design C = Double Gas Barrier Inline Seal : Contacting Seal Face Design D = Double Gas Barrier Canister Seal : Non-Contacting Seal Face Design				
IMPACT RATING SCALE 1 ----- 2 ----- 3 ----- 4 ----- 5 ( No Impact ←-----→ Great Impact )				

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