MAGNETIC FLUID SEAL SYSTEM FOR CONTROL OF PUMP FUGITIVE EMISSIONS

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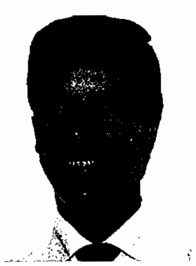

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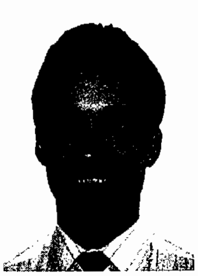

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ABSTRACT

Pump fugitive emissions can now be controlled to near zero levels by the use of a simplified dual seal system. A magnetic fluid secondary seal traps any vapors that leak past the primary mechanical seal. This arrangement requires no external support equipment other than venting the vapors to a suitable disposal or recovery system.

Magnetic fluid seals have provided hermetic sealing for various gas applications over the last 20 years. One common use is as exclusion seals for magnetic disk drives. Adaptation of this technology to centrifugal pump applications required an extensive development effort. Laboratory tests show maximum emission rates of less than 3.0 ppm of organic vapors. Field evaluations on process pumps confirm the feasibility of using this technology to achieve strict emission control.

Magnetic fluid seals are nonwearing devices. Improved technology allows renewal of the magnetic fluid while the pump is in operation. The magnetic fluid barrier provides a hermetic seal even if the mechanical seal leak rate increases due to wear.

INTRODUCTION

Government agencies in the United States are imposing stricter limits on emissions of volatile organic compounds (VOCs) from centrifugal pumps. Many of these compounds change from the liquid to the vapor phase upon leaking to atmospheric pressure and thus become air borne pollutants. Leakage of VOCs from valves, fittings, and pump seals is referred to as "fugitive" emissions.

EPA Method 21 [1, Appendix A] is the standard field measurement technique for determining emission levels. It employs a portable hydrocarbon detector, commonly referred to as a "sniffer." A probe draws in a continuous sample of gas from a region within 1.0 cm of the seal end plate/shaft interface. VOC concentration is measured in parts per million (ppm). Emission levels are somewhat dependent on variables such as wind speed, shaft rotation rate and proximity of the probe to the emitting source. Schaich [2] has a correlation relating ppm emissions to mass leakage rates.

Emission regulations for process pumps in VOC services were first enacted in the early 1980s in Southern California [3]. Thereafter, a similar regulation was developed as a national standard [1]. Gaseous emissions from pump seals in these services were limited to 10,000 ppm.

In 1991, a more stringent regulation, Rule 1173 [4], went into effect in Southern California, lowering allowable emissions to 1,000 ppm for existing pumps. Best available control technology (BACT) is required for new equipment. BACT can be met by either of two equally acceptable technologies [5]:

- Sealless pumps if available for service, or
- Dual mechanical seals

The Environmental Protection Agency proposes [6] an emissions limit of 1,000 ppm for most pumps in volatile hazardous air pollutant (VHAP) service. The regulations are to be applied primarily to chemical manufacturing facilities. Refineries and other industries that emit VOCs will be covered by similar rules within a few years. Almost 200 chemicals and over 400 chemical processes are covered by the proposed rules.

Under the proposed EPA regulation, pumps with single mechanical seals must be monitored monthly. However, the EPA offers an incentive to achieve near zero emissions. An exemption from monthly monitoring is proposed for pumps equipped with dual mechanical seals that include a barrier fluid system [6]. Such systems must be constructed to ensure essentially zero emissions to the environment, namely one of the following conditions must be met:

- Barrier fluid pressure is always higher than the pump stuffing box pressure.
- Barrier fluid reservoir is vented to an approved vapor disposal means (e.g., flare).
- Barrier fluid is purged into a process stream.

In addition, the barrier fluid system must be equipped with a sensor that will detect failure of the seal system, the barrier fluid system, or both.

Both the Southern California BACT regulation and the EPA dual seal monitoring exemption offer strong incentives for seal technology to achieve near zero emissions from pumps. Low emission single seals are capable of sealing VOC services to the 500 ppm level [7, 8]. Dual seal arrangements (with a barrier fluid) achieve emission levels less than 50 ppm [9].

Initially, an overview is presented of the various seal technologies proposed to meet the emission regulations. Next, a detailed account is given of a particular dual seal arrangement that uses a magnetic fluid secondary seal to contain emissions. Finally, lab tests and field evaluations confirm that the magnetic fluid seal is a viable technology for control of pump VOC emissions.

SEAL TECHNOLOGY TO MEET THE NEW EMISSION STANDARDS

In considering the following seal technologies, bear in mind that not only should the seal system meet emission limits, but it should also be reliable for at least three years. This is specified by the Mission Statement of API Standard 682 [10], "...This Standard will result in sealing systems which will have a high probability of operating for three years of uninterrupted service while complying with environmental emission regulations." Some of the following technologies may not meet the three year service criteria.

Low Emission Single Seals

Improved single mechanical seals can meet the 1,000 ppm regulations in most applications [8]. These seals employ a face combination of antimony filled carbon graphite running against high grade silicon carbide. Multiport flush injection to the interface area assures uniform cooling of the faces. It also acts to sweep away any vapor bubbles that might form an insulating blanket around the carbon nose.

The extremely thin fluid film between the faces, on the order of 0.4 μm [8], ensures low leakage. It also promotes longer seal life by keeping larger abrasive particles, which may be contained in the flush, from getting between the faces. After two years service in Southern California refineries, these low emission single seals continue to meet the 1,000 ppm limit on propane and ethane pumps.

Single mechanical seals provide reliable sealing for most applications provided the following conditions are met:

- Fluid specific gravity > 0.45;
- Vapor pressure margin in the seal chamber > 25 psi;
- Bulk fluid temperature < critical temperature;
- Pumpage provides adequate lubrication of the seal faces.

A dual seal with a barrier fluid is recommended if any of these conditions are not met.

Dual Seal Technology

Dual seals consist of two single mechanical seals per seal chamber. A barrier fluid is usually located in the volume between the primary and secondary mechanical seals. The barrier fluid may be either pressurized or nonpressurized. It is typically circulated through a reservoir which provides cooling and makeup of lost fluid. Some secondary seals have been designed to run on gas, and thus do not require a barrier fluid.

Dual Seals with Nonpressurized Barrier Fluid

In this arrangement the inner (primary) seal is lubricated by the pumpage and the outer (secondary) seal is lubricated by the barrier fluid. Barrier fluids should have good lubricating properties, be compatible with the pumpage and not be on the EPA VHAP [6] list. Typical barrier fluids are ethylene glycol/water mixtures, kerosene, diesel fuel, or light hydraulic oils of either hydrocarbon or synthetic bases. Circulation of the barrier fluid through an external reservoir is usually accomplished by pumping action of the rotating seal element.

Process fluid migrates into the barrier fluid in a nonpressurized barrier fluid system. In volatile liquid service this leakage is typically a vapor, since the barrier fluid is at low pressure. Barrier fluid circulation transports the vapor into the external reservoir, where it is vented to either a vapor recovery system or flare. A typical nonpressurized barrier fluid has a volatile gas solubility less than 0.5 percent by weight [9]. Thus, any barrier fluid leakage contains a negligible amount of process fluid VOC. Secondary seal leakage to the environment is low, due to the small pressure difference across this seal. The secondary seal, however, should be designed to accommodate full system pressure for safety.

Laboratory tests and field surveys in refineries show that secondary seal emissions are less than 10 ppm for typical barrier fluids [9].

Advantages of this type of sealing system include very low emissions and a safety backup seal. Also, instrumentation to monitor the seal system is relatively simple and reliable. Disadvantages include initial cost, daily inspection of the barrier fluid system, water cooling of the reservoir, and the need to vent to a vapor disposal system.

Dual Seals with Pressurized Barrier Fluids

In this arrangement, both the inner and outer seals are lubricated by the barrier fluid, which is at a higher pressure than the pumpage. Thus, some barrier fluid migrates into the process fluid. Face wobble and other effects, however, can cause reverse migration of the product into the barrier fluid, causing barrier fluid contamination. Also, a pressurized barrier fluid will trap more of the volatile gases in solution, since solubility is approximately proportional to absolute pressure.

Pressurized systems usually experience a larger rate of barrier fluid loss to the environment compared to nonpressurized systems. Nonetheless, emissions of VOCs are near zero ppm from properly maintained pressurized dual seals.

Advantages of a dual seal arrangement with a pressurized barrier fluid include near zero emissions and no requirement for a vapor disposal system. Disadvantages include high initial cost and maintenance of the pressurization system. Somewhat more complex instrumentation (compared to nonpressurized systems) is required to detect primary seal failure.

Dual Seal with a Dry Running Secondary Seal

A dual seal arrangement with a dry running secondary seal is a simpler system compared to liquid/liquid dual seals. Eliminated is the support system: barrier fluid, reservoir, and provision for cooling water. In some designs the secondary gas seal can support full system pressure (often for only a limited time) if the primary seal fails.

Near zero emissions are attainable if the secondary seal faces are contacting *and* the space between the two seals is vented to a vapor disposal system. If no vapor handling system is provided, then emissions are solely determined by the effectiveness of the primary seal.

There are two basic types of dry running secondary seals:

- Seals with flat faces, and
- Seal faces with lift augmentation.

Flat face seals usually run in a contacting mode, while seals with lift augmentation will typically have a gas film between the faces. Flat face gas seals thus provide lower leakage, but also experience more rapid wear than seals with lift enhancement.

An extensive lab test program was conducted [11] to evaluate dry running secondary seals. Over 40 tests were conducted using 17 different face material combinations and several different face designs, including some with lift augmentation. Test gases were air, nitrogen and propane. Pressure ranged from 0.0 to 3.0 psig. Very low spring loads were involved.

Test results showed that flat-faced dry running seals consistently displayed unacceptably high wear rates. A typical post test surface trace of a carbon face is shown in Figure 1. The flat face carbon was run against a silicon carbide face for 24 hours. Face spring loading was 3.0 psi and gas pressure 0.0 psi. Note that roughness average, R_a, is 24 μ-in. A lapped carbon has a surface roughness of about 2.0 μ-in. Experimental work [12, 13] shows that a well performing seal on liquids should have a post-running surface roughness of 2.0 to 3.0 μ-inches. Dry running *contacting* faces, however, experience aggressive wear as illustrated in Figure 1.

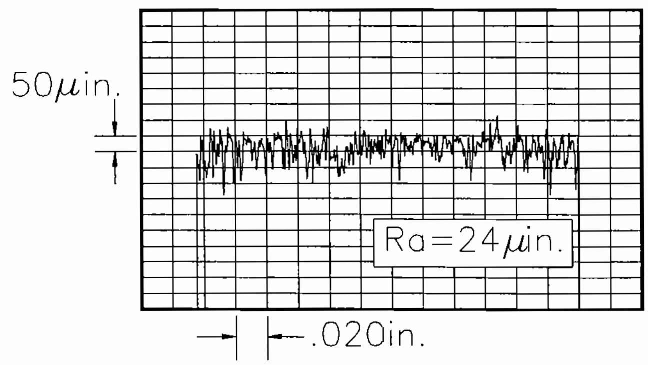

Figure 1. Post Test Surface Trace of a Flat Face Dry Running Carbon. Seal was run for 24 hours on air at 0.0 psi. Face spring pressure was 3.0 psi and shaft speed 3,600 rpm. R_a for a lapped carbon is only 2.0 μ-in.

Life expectancy of about one year might be achieved with flat face dry gas seals, depending on amount of sacrificial face material. The developed face roughness (Figure 1) provides significant leak pathways and thus cannot be expected to maintain near zero emission levels. Also, a severely worn secondary seal should not be relied upon to safely backup the primary seal. Furthermore, API users are insisting on seal life of at least three years [10]. For these reasons, it is doubtful that flat-face gas secondary seals will find broad acceptance in the market place.

Some of the tested designs with lift augmentation (such as gas pads or hydropads) displayed very low wear rates [11], indicative of more than three years life. Such seals may meet 1,000 ppm emission regulations, but are unlikely to achieve near zero emissions (less than 50 ppm) status. A gas seal with lift augmentation can be a viable secondary containment seal capable of limiting (for 24+ hours) leakage if the primary seal fails. Such seals, however, do not qualify for either the EPA dual seal monitoring exemption or BACT technology, because they do not have a barrier fluid.

Dual Seal with Spiral Groove Primary and Secondary Seal Faces.

The primary seal has spiral grooves on the outer portion of one of the faces and a sealing dam on the inner portion of the face [14]. The secondary seal has a similar, but generally deeper, groove pattern. In volatile liquid service, vapor is created through phase change due to shear heating of the fluid as it migrates across the primary seal. This vapor is claimed to be contained by the spiral groove secondary seal. The space between the two seals is vented to a vapor disposal system.

Lift augmentation produced by the spiral grooves causes the floating face to "lift-off." Gap thicknesses are typically on the order of groove depth [15]. The preferred form of the primary seal has groove depth of about 50 μ-in [14]. Resulting film thickness is also on the order of 50 μ-in, essentially triple the gap for low emission single seals [8]. Leakage is approximately proportional to film gap cubed. Thus the spiral groove primary seal should experience on the order of 27 times the emission rate of a flat face primary seal.

The downstream vapor seal has a preferred groove depth of about 100 μ-in [14] which should result in a gas film gap of comparable magnitude. A seal with such a large gap is unlikely to provide much of a barrier to environmental emissions.

Contamination in the process fluid may collect in the grooves and thus degrade lift augmentation. In some applications thermal distortion of the faces leads to reduced performance [16].

Dual Seal with Gas Pad Secondary Seal.

A gas seal which incorporates "gas pads" (radial slots) over the outer portion of the face and a sealing dam on the inner portion is shown in Figure 2. Analytical investigation [17] of slotted (hydropad) faces shows that pressure and thermal distortions result in a circumferential wavy pattern on the seal face. Lift augmentation is developed in the converging sections of the wavy fluid film.

The seal in Figure 2 is a secondary containment seal designed to handle gas pressures from 0.0 to 40 psi. It will restrict liquid leakage to the environment upon failure of the primary mechanical seal, thus allowing an orderly shutdown of the pump.

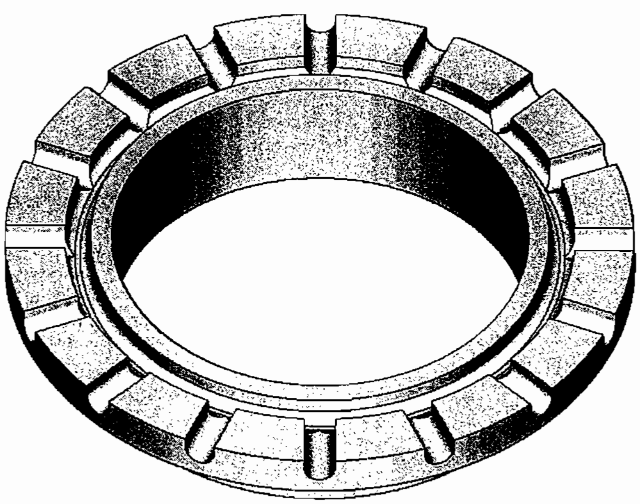

Figure 2. Gas Pad Dry Running Secondary Seal.

Dual Seal with Hydrostatic Radial Taper Secondary Seal

This option uses a conventional mechanical primary seal. The secondary seal incorporates a radial taper of one of the faces which is convergent from OD to ID. Experimental work shows that load support can be generated using this method [18]. Limitations to this concept are that radial taper must be large to ensure film stiffness stability. Also, minimum film thickness must be small to minimize leakage. This combination may be difficult to achieve in practice.

Dual Seal with an Upstream Pumping Inner Seal

This dual seal arrangement incorporates a primary mechanical seal with shallow spiral grooves (similar to those described earlier) over the inner surface of one of the faces. The secondary seal is a conventional mechanical seal. Rotation of the primary seal results in low pressure barrier fluid being pumped, against a pressure gradient, into the process stream. Pumping rates are reported to range from 0.1 to 16 ml/min [19]. Dilution of the product is usually not a concern, provided the barrier fluid is compatible with the pumpage.

Theoretically, the upstream pumping concept should result in no leakage of the process stream into the barrier fluid or environment. However, the pressurized product may find leakage pathways against the pumping action.

At a typical pumping rate of 2.6 ml/min [19], barrier fluid makeup can be on the order of 1 gal/day. Contamination of the shallow grooves, whose depth ranges from 2.0 to 6.0 μm [20], results in deteriorated performance. In addition, cavitation effects may occur if the barrier fluid is not pressurized [21]. Pressurization of the barrier fluid imposes additional system complexity.

Comments on Current Technology to Control Emissions from Pump Seals

This survey of emission control technology shows that dual seals are required to achieve near zero emissions along pump shafts. No one sealing method can be considered "best" for all applications. Some of the above seal systems are unlikely to find wide acceptance in the marketplace.

In the next section, an alternative technology that achieves hermetic sealing, is easy to maintain, and is cost competitive with other dual seal systems is presented.

MAGNETIC FLUID SEALS—GENERAL BACKGROUND

Basic Operation

Ferrofluids are magnetically permeable liquids which are responsive to magnetic fields [22]. The fluids consist of three primary components; the carrier (typically a low vapor pressure oil), the surfactant (a chemical bonding agent similar to a soap), and magnetic particles (extremely small spheres of iron oxide). The surfactant bonds the magnetic particles to the carrier oil in a colloidal suspension giving the resultant fluid magnetic properties. All three of the primary components are engineered to create a ferrofluid with the desired properties for a specific application [23].

Magnetic fluid seals are created by trapping the ferrofluid in a high flux density magnetic field between the rotor and stator creating a liquid O-ring. The magnetic field is engineered using permanent magnets and pole pieces to focus the magnetic flux at a specific air gap between the rotor and stator (Figure 3). The shaft or sleeve (typically the rotor) is magnetically permeable to provide a return path for the flux to complete the magnetic circuit. The housing which contains the magnets and pole pieces (typically the stator) is nonmagnetic to prevent the circuit from shorting out where it is in contact with the north and south pole pieces.

A seal is created by inserting ferrofluid into the air gap where it is retained by the extremely strong magnetic field. Pressure capacity is determined by the resistance of the ferrofluid to leave the magnetic field of the air gap. The strength of the magnetic field in the air gap is engineered through the type and size of the magnets, dimensions of the air gap and the size, shape and material of the pole pieces and rotor.

Magnetic Fluid Seal Applications

Magnetic fluid seals have been successfully applied to a wide variety of diverse applications over the past two decades. One of the largest scale commercial applications of this technology has been exclusion seals for computer disk drives. Low pressure magnetic fluid seals are utilized to separate the grease lubricated ball bearings of the drive motor spindle from the disk head. This prevents contaminants from reaching the disk area and causing loss of data. Over 90 percent of the hard disk drives manufactured today utilize this exclusion seal technology.

Magnetic fluid vacuum rotary feedthroughs are used in semiconductor and vacuum processing to transmit rotary motion into

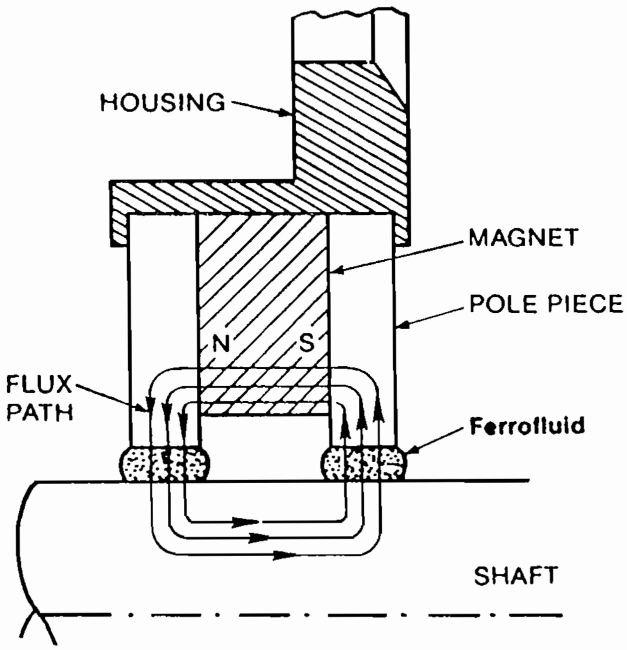

Figure 3. Magnetic Fluid Exclusion Seal.

environmentally controlled process chambers. In these applications, the seal is supplied as an integral part of a precision spindle that is used for applications ranging from large diameter fans and mixers to intricate robotic actuators. Magnetic fluid seals are involved in the processing of most silicon based integrated electrical circuits.

Other applications include vacuum and pressure seals for the manufacture of high efficiency electric light bulbs, the sealing of sensitive optics for military aircraft, and seals for lasers and X-ray generators.

MAGNETIC FLUID SEAL SYSTEMS TO MEET PUMP EMISSION REGULATIONS

General Description

A development effort was begun to adapt this hermetic seal technology to control fugitive emissions from pumps and other rotating machinery. A dual seal system was designed utilizing a conventional mechanical primary seal and a magnetic fluid secondary seal. Vapors leaking past the mechanical seal are contained by the magnetic liquid in the secondary seal. The vapors are vented out the quench tap (found on API 610 seal glands) to a suitable disposal system. A schematic representation of the dual seal arrangement is shown in Figure 4.

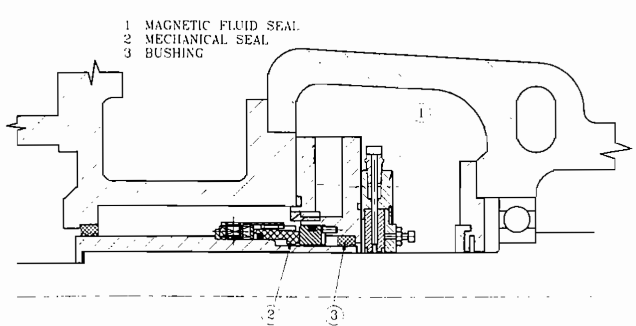

Figure 4. Cross Sectional View of a Centrifugal Pump, Showing the Pimary Mechanical Seal and the Secondary Magnetic Fluid Seal.

Even though the quantity of magnetic fluid is typically less than 1.0 cc, and it is not circulated through a reservoir, the fluid blocks escaping vapors, and thus can be considered a barrier fluid.

Any liquid leakage past the mechanical seal must be prevented from accumulating and diluting the magnetic fluid. This is accomplished by using the drain tap, required on API 610 glands, for liquid leakage removal.

An antispark bushing is located between the primary and secondary seals. The carbon labyrinth style bushing has a smaller radial clearance (over the sleeve) than the pole pieces of the magnetic fluid seal.

The shaft or sleeve under the secondary seal must be magnetically permeable. Typically, it is manufactured from a 400 series stainless steel.

To confirm the feasibility of using this technology, a test was run to verify that a commercially available magnetic fluid seal could effectively contain hydrocarbon vapors. Next, two prototype magnetic fluid seals were developed to fit on API pumps.

Development of Secondary Seal for Control of Pump VOC Emissions

Off Shelf Standard Magnetic Fluid Seal

A commercially available magnetic fluid seal was evaluated in a lab test to determine its VOC containment capability. The test configuration is shown in Figure 5. The mechanical seal had a 2-3/8 in balance diameter and the magnetic fluid seal fit on the 2.0 in shaft. The vapor space between the mechanical and magnetic fluid seals was vented to atmosphere through an adjustable check valve.

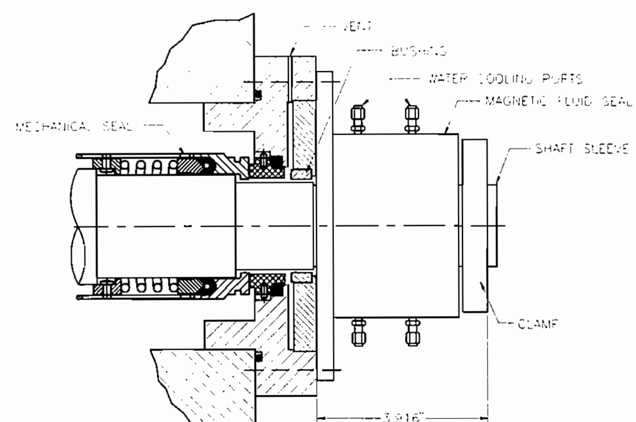

Figure 5. Test Arrangement to Evaluate Emission Containment Capability of an "Off-the-Shelf" Magnetic Fluid Seal.

Each end of the 4.0 in long magnetic fluid seal had radial bearings to minimize runout on the tight clearances between pole pieces and shaft. Taps on the seal housing were provided for water cooling (recommended for 3,600 rpm shaft speed applications).

The magnetic fluid seal was rated at 40 psi capacity resulting from its multistage construction composed of 28 separate liquid O-rings interspaced with air. Each stage can sustain a differential pressure of about 1.5 psi, giving a net pressure rating of 40 psi.

The mechanical seal was run on propane at 240 psi, 110°F, and 3600 rpm. Emissions from the mechanical seal fed into the cavity between the two seals. Pressure level in the cavity was established by adjustment of the check valve to atmosphere. Tested cavity pressures ranged to 26 psi. Sniffer readings were taken every 30 minutes at the shaft/flange junction. There were no detectable emissions over the 99 hour test.

This test proved the feasibility of using magnetic fluid seals on light hydrocarbon vapors, even though the actual seal was too large to fit on typical pumps. Other limitations included the need to water cool the assembly, due to built-in bearings and the unit could not be field recharged with magnetic fluid. Nevertheless, these early results encouraged an effort to develop a practical seal.

First Prototype

The first prototype magnetic fluid seal was designed to fit on a specific refinery pump. The dual seal system combined a primary mechanical seal with a specially configured magnetic fluid secondary seal. Design parameters for the field application are contained in Table 1.

Development proceeded using only existing technology on the first prototype. This resulted in secondary seal pressure capacity shown in Table 2.

Table 1. Design Parameters for First Prototype Seal.

Shaft Size	1.875″
Shaft Speed	3570 rpm
Temperature	100 - 200°F
Cooling Water	Available
Pressure Differential	0.5 - 10 psig
Target Service Life	2 - 3 years
Sealed Medium	Butane
Vapor Disposal Means	Flare

Table 2. First Prototype Pressure Capacity.

Radial Gap (in)	Pressure (psi)
0.0035	20
0.0060	10

Runout is of some concern with seal gaps that are on the order of 0.006 in. An early decision involved the mounting configuration for the magnetic fluid seal: either attachment to the bearing housing, or attachment to the primary seal gland.

Mean shaft deflection was estimated to be 0.000 to 0.003 in next to the bearing housing and somewhat larger inside the seal gland. Hence, the more conservative bearing box mounting was chosen. A multistage magnetic fluid seal was designed to handle 0.003 in runout and to meet the Table 1 conditions. All design parameters were met and verified in a laboratory test program. Review of the design by refinery personnel, however, revealed that the first prototype was impractical for field conditions. The following problem areas were identified:

- Mounting on the bearing housing limits application of the seal to pumps that have a concentric mounting provision.
- If ferrofluid is displaced due to an upset, recharging the seal requires pump disassembly.
- Complex assembly procedures require extensive training of installation personnel.
- Water cooling is not available in all locations, and not desirable in any location.

For these reasons, the first prototype was not submitted for field testing.

Second Prototype (Current Design)

To overcome the objections encountered by the first prototype, the second prototype had to be simple in form and implementation. A second prototype seal was commissioned with the following adjustments to the design parameters:

- Cartridge secondary seal that fastens to the primary seal gland
- Increased tolerance to shaft deflection and runout
- Provision for centering the secondary seal to the shaft
- Increased radial clearance of the pole pieces
- Lower maximum design pressure to 4 psi due to larger radial gap
- Develop means to replenish ferrofluid on a running application
- Eliminate need for cooling

A study was performed to determine the pressure requirement of the seal by monitoring flare system background at the refinery. It was determined that 4.0 psi capability was safely above the maximum expected pressure, except during plant upset conditions. Occasional over pressurization due to upsets could be accommodated by the fluid replenishment feature.

In consideration of the reduced pressure requirement, a multistage seal was no longer necessary. This meant a two pole, field rechargeable seal was feasible. A self refilling design was evaluated *vs* a manual refill design. Due to wide distribution of fluid, the self refilling design resulted in lower sealing pressure capacity. Hence, a manual refilling configuration was adopted.

The seal was designed to tolerate up to 0.010 in TIR of runout. Experience gained in design and testing of the first prototype provided an empirical baseline for the second. Note that the first design represented a significant extrapolation from established design rules used to create magnetic fluid seals for conventional applications. The second prototype represented a similar extrapolation from the first. In both cases the extrapolation involved approximately doubling the operating clearance between the shaft and the seal.

Computer modelling and lab testing were used in designing the second prototype. The result was only slightly less pressure capacity than the first prototype, and with increased tolerance to eccentricity. Innovative design features were incorporated to facilitate concentric mounting and a fill port was added.

Lab Testing of Field Prototype Seal

Lab Test of Magnetic Fluid Seal Only

An intensive lab test program was conducted to characterize the seal's performance under simulated field conditions. The 1.875 in shaft diameter seal was evaluated using butane vapor. Testing included:

- Pressure capacity *vs* ferrofluid volume.
- Pressure capacity *vs* eccentricity.
- Pressure capacity *vs* shaft speed.
- Seal temperature *vs* ambient temperature.
- Start/stop testing.
- Seal refilling.
- Repeated over pressurization.
- Endurance.
- Hydrocarbon containment.

Key findings of the lab test program are:

Pressure capacity: The seal reliably holds 7.0 psi.

Emissions: VOC emissions (butane) measured from 0.0 to 3.0 ppm, essentially equal to background.

Tolerance to runout: Runouts to 0.010 in have little effect on pressure capacity.

Shaft speed capability: There is no loss in sealing capacity at test speeds to 55 ft/sec. Maximum speed limit is estimated to be more than 100 ft/sec.

Recharging of fluid: It is easy to refill the seal with magnetic fluid while the shaft is turning. Sealing capacity can thus be easily renewed if over pressurization occurs.

Pressure recovery: The seal sustains about 1.0 psi sealing pressure capacity after five over pressure events.

Ease of installation: The seal can be installed with normal pump shop tooling and mechanical skills.

Process fluid compatibility: The seal is fully compatible with petroleum products in the vapor phase. Compatibility of the ferrofluid with chemical vapors is under investigation. Sour or caustic vapors may require more frequent ferrofluid replacement. The magnetic fluid must be protected from more than incidental contact with liquids.

Temperature limit: The seal is capable of operating at temperatures to 200°F continuously. Higher temperatures are possible with water cooling or with a special ferrofluid.

Size: The secondary seal is about 0.75 in long (axially). Its diameter is similar to that of a standard mechanical seal gland plate.

After the secondary seal design was verified, lab testing was done on the complete dual seal system.

Lab Test of Field Prototype Dual Seal System

This phase of lab testing was geared to address the following questions:

- Will the mechanical seal temperature be raised by operation of the magnetic fluid seal?
- What is the pressure capacity of the magnetic fluid seal under the following conditions?
    - Clean and dry with 0.002 TIR concentricity
    - Rain or water spray from maintenance clean-up
    - Contact of liquid leakage with the magnetic fluid
    - Runout greater than 0.002 TIR
- Will liquid leakage migrate past the outboard flange bushing?

To evaluate these questions, a test setup was devised that approximated a seal installation in a petroleum refinery. This arrangement is shown in Figure 6. In the test setup, the shaft sleeve extends beyond the seal to simulate the pump shaft. Due to the unsupported shaft sleeve, a bearing bracket was added to eliminate

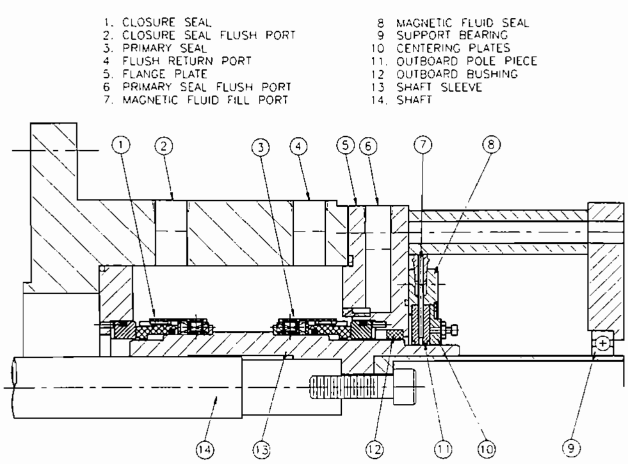

Figure 6. Lab Test Arrangement Simulating Initial Field Installation of the Magnetic Fluid Seal.

any deflection of the sleeve resulting from magnetic attraction to the pole pieces. In the refinery application the seal is located over the pump shaft while the sleeve extends no further than the outboard bushing.

A piping schematic drawing for measuring primary seal leakage and cavity pressure between the primary and the magnetic fluid seals is shown in Figure 7. Pressure was measured using a high accuracy test gage (0.1 psi resolution) and also with a pressure transducer connected to a computerized data acquisition system. Primary seal leakage was measured with a 0 to 15 ml/min gas rotameter. Leakage past the secondary seal was monitored with an organic vapor analyzer.

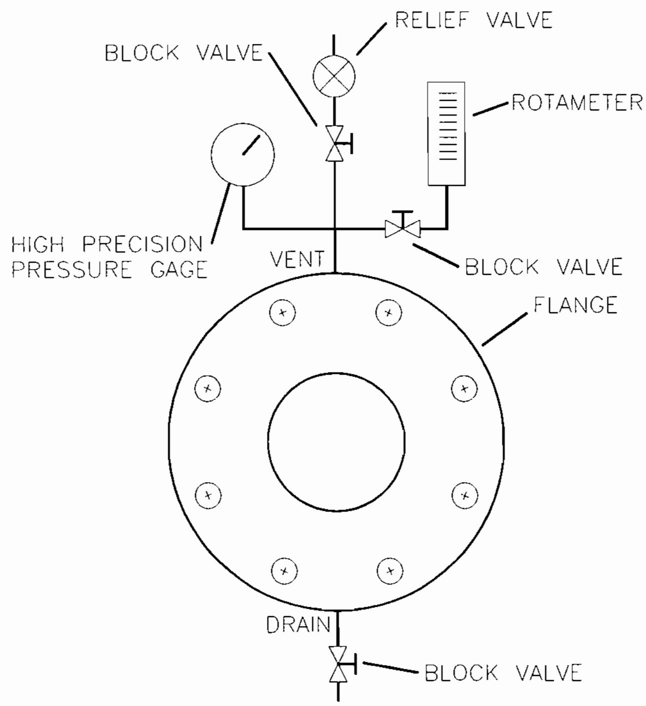

Figure 7. Piping Schematic Representation of Lab Test Arrangement.

Primary seal stationary face temperature was measured with a thermocouple located in a hole drilled in the ring to within 0.050 in of the running surface. Cavity temperature, outboard pole piece temperature (near its ID), and the inboard process fluid temperature were also monitored by computerized data acquisition. Test conditions are contained in Table 3.

The first test was run to determine if heat generated by the magnetic fluid seal would increase the temperature of the primary mechanical seal. The test was started without charging the secondary seal with magnetic fluid. Injection occurred after the mechanical seal face temperature reached equilibrium. Initial data sampling

Table 3. Lab Test Conditions for Field Prototype Seal.

Primary Seal	Spring-pusher; carbon vs SiC; 2.375″ balance diameter
Magnetic Fluid Seal	1.875″ shaft diameter
Shaft/Pole Piece Radial Gap	0.015″
Sealed Fluid	Propane
Sealed Pressure	240 psig
Fluid Temperature	110°F
Shaft Speed	3600 rpm

rate was set at every three minutes and then reduced to 20 seconds when the seal was injected with magnetic fluid.

Plots showing the results for two cases are given in Figures 8 and 9. In Figure 8, the spike in the box temperature at approximately 1.8 hours was done by opening the thermocouple connector to indicate injection of magnetic fluid. On the second test, magnetic fluid was injected at 5.4 hours as shown by the spike on Figure 9. The delay in recorded face temperature measurement of Figure 9 was the result of adjustments to the thermocouple connector at the start of testing.

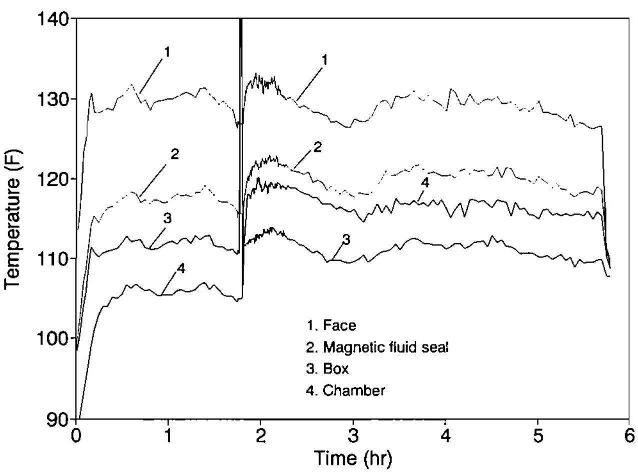

Figure 8. Temperatures During Lab Test Run, Case 1. Temperature spike is due to opening thermocouple connector to indicate fluid injection at 1.8 hrs.

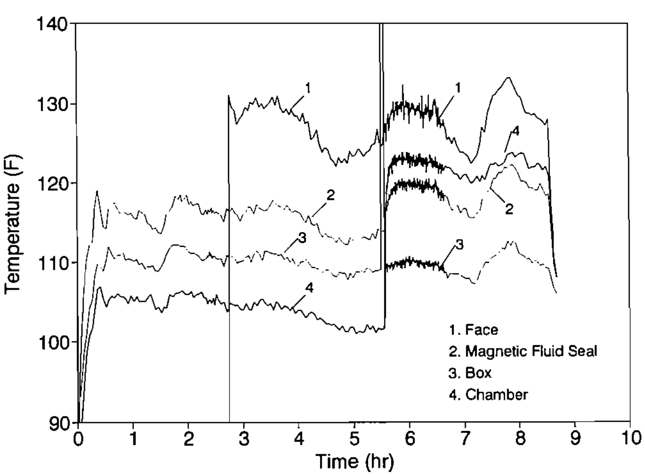

Figure 9. Temperatures During Lab Test Run, Case 2. Magnetic fluid injected at 5.4 hrs.

These results show that the magnetic fluid seal has a negligible effect on the mechanical seal face temperature. Addition of the magnetic fluid did result in an increase of about 5°F to the outboard pole piece and 10°F to 15°F to the vapor cavity space between the mechanical and magnetic fluid seals. These temperature increases are due to heat generated from viscous shear of the magnetic fluid. Heat transfer between the pole pieces and shaft sleeve is facilitated by the magnetic fluid which acts as a thermal conductor.

The next test was conducted to determine the pressure capacity of the magnetic fluid seal under dynamic and dry operating conditions. This was accomplished by closing all block valves shown on Figure 7 and using propane vapor leakage (from the primary seal) to pressurize the secondary cavity. Pressure increase of the cavity was approximately 2.0 psi/minute. (In a prior test, leak rate of the primary seal was about 3.0 cc/min propane vapor). Cavity pressure continued to increase until the magnetic liquid O-rings burst. Some of this fluid was ejected from the seal. Subsequent bursts took the form of a continuous popping of the fluid until a minimum sustained pressure was attained. The time between first leak and minimum holding pressure was less than five minutes. Consistent results for three tests are shown in Table 4

Table 4. Pressure Threshold (dynamic/dry - 0.002in TIR).

	Test #1	Test #2	Test #3
Maximum Pressure at first burst (psig)	7.3	7.5	7.5
Minimum Pressure after many bursts (psig)	1.0	1.0	1.0

A fourth test showed a maximum holding pressure of only 4.0 psig. The filler cap had not been tightly screwed down, which prevented pressure build up between the two pole pieces.

To simulate the effect of rain or water spray due to maintenance cleanup, a water spray was directed at the seal from 10 ft away and at an angle of 45 degrees. This was done for 15 to 20 seconds while the test apparatus was in operation. Cavity pressure was maintained at 2.0 psig (propane vapor). After this process was completed, it was noticed that some small amount of the magnetic fluid had been dislodged from the outboard pole piece. Pressure threshold was evaluated using the procedure outlined before. The results (Table 5) show about 1.0 psi loss in pressure capacity. This most likely resulted from the small amount of magnetic fluid that was washed away during the spray cleaning operation.

Table 5. Pressure Threshold (Simulated Clean-up, 0.002in TIR).

	Test #5	Test #6
Max Pressure (psig)	6.5	6.8
Min Pressure (psig)	1.0	1.0

The next experiment was done to determine if liquid leakage (from the primary seal) would migrate past the outboard bushing. A 5.0 liter container was installed above the flange to deliver auxiliary fluid to the flange vent. Two fluids, diesel fuel and automatic transmission fluid (ATF), were dripped onto the shaft sleeve in the space between the mechanical seal and bushing. The secondary seal was removed for this test. Test fluid was dripped onto the rotating (3,600 rpm) sleeve at a rate of 2.5 ml/min for two hours. Liquid could exit through the drain port on the bottom of the flange. In both cases a small amount of test fluid migrated along the sleeve past the bushing. The next test was run to determine the effects of liquid leakage on magnetic fluid seal performance.

After thoroughly cleaning the flange and shaft sleeve of any fluid residue, the magnetic fluid seal was installed. The previous two experiments using diesel and ATF were repeated. After two hours of fluid exposure all block valves were closed and the pressure threshold determined. Results for the two fluids tested are shown in Table 6. There is a small reduction in maximum pressure capability and a slight change in minimum pressure capacity.

Table 6. Pressure Threshold (dynamic/wet, 0.002in TIR).

	Test #7 (#2 Diesel)	Test #8 (ATF)
Max Pressure (psig)	6.9	7.1
Min Pressure (psig)	0.3	0.9

To address liquid migration past the bushing, a simple device was designed to restrict fluid movement along the shaft. The device, called a slinger, is located just behind the bushing and is both sealed along the shaft sleeve and driven by an O-ring. This configuration is illustrated in Figure 10. Liquid leakage along the shaft is propelled to the flange bore where it can exit the drain port. The amount of force exerted by the slinger against the bushing is that due to O-ring windup during installation on the shaft sleeve. There was no measurable increase in bushing temperature resulting from slinger contact loading.

The last three tests performed were done with the magnetic fluid seal off-center by 0.004 in TIR. Pressure threshold under dry and dynamic operation is shown in Table 7. These results show only a slight drop in pressure capacity due to this range of runout.

Table 7. Pressure Threshold (dynamic/dry - 0.004in TIR).

	Test #9	Test #10	Test #11
Max Pressure (psig)	6.3	5.8	5.9
Min Pressure (psig)	1.1	1.1	1.5

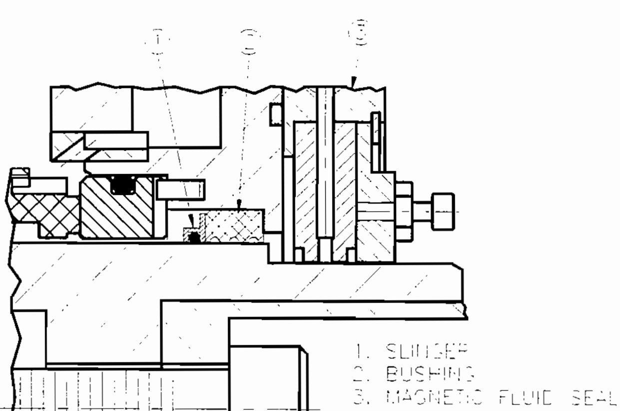

Figure 10. Slinger to Deflect Liquid Leakage Out the Drain Port (not shown) at Bottom of Flange.

Field Tests

The second prototype seal was installed in a Southern California refinery in November, 1991. Operating conditions are given in Table 8.

Connection to the refinery flare line took place about two weeks after seal installation. During this time, the pump was on pressurized standby. Since the vent port was blocked in, leaking vapors across the primary mechanical seal built up sufficient pressure to burst the secondary magnetic fluid seal. After attachment to the flare line, the secondary seal was recharged with ferrofluid and the pump started.

Table 8. Operating Conditions for First Field Test.

Pumpage	Light naphtha
Flush	Butane (plan 32)
Suction pressure	118 psig
Shaft speed	3570 rpm
Temperature	110°F
Mechanical seal size	2.375" balance diam.
Magnetic fluid seal	1.875" shaft diam.

Shortly afterwards the vent line pressure gauge indicated 2.0 psi, then this pressure dropped to about 1.0 to 1.5 psi. About 2.0 psi was needed to open the check valve in the vent line.

Emission readings near the secondary seal registered in the area of 20 to 50 ppm. An inspection of the pump installation was then performed with the sniffer. This examination revealed that the fitting to the seal flange flush line was leaking more than 1000 ppm. The fitting was made leak tight in January 1992. Subsequent secondary seal emission readings have ranged from a low of 0.0 to a high of 20 ppm. It is likely that emission measurements above 1.0 ppm are due to variations in background levels. Previous work [8, 9] has shown that leakage at fittings or through a damaged flange gasket can result in high ppm readings near a mechanical seal. It should be standard practice to survey such potential leak sources if high emission levels are detected near a seal.

In March 1992, the magnetic fluid seal ruptured due to over pressurization. A recording pressure gauge showed that pressure between the primary and secondary seals had reached 7.0 psi. It is suspected that flare line over pressure held the check valve closed long enough for primary seal leakage to produce enough pressure (7.0 psi) to displace the magnetic fluid. The seal was recharged with magnetic fluid and readings for emissions continue to register near 0.0 ppm. This seal remains in service, and has accumulated nine months of continuous operation to date (September, 1992).

A second installation became operational in a petrochemical facility in the U.S. Gulf Coast area in September 1992. No measurable emissions have been detected from the cyclohexane pump. Any liquid leakage is routed to a chemical collection system.

Seven additional units have been engineered for field installation. Data should soon be available from these sites.

Seal System Monitoring

The magnetic fluid seal contains emissions even if primary mechanical seal performance deteriorates due to wear. Primary seal face wear and secondary seal performance can be ascertained using simple instrumentation. An indicator of face wear is increased leakage, which can be detected with a flow switch in the vent line. A cavity pressure reading of 1.5 to 6.0 psi and a maximum recording less than 7.0 psi confirms the secondary seal is leak tight. For cavity pressures less than 1.5 psi, a sniffer is needed to verify emissions containment. Finally, if peak recorded pressure is greater than 7.0 psi, an over pressure event has likely occurred and the seal needs to be refilled with magnetic fluid.

CONCLUSIONS

The dual seal arrangement of a primary mechanical seal and a secondary magnetic fluid seal is shown to provide near hermetic sealing (<3.0 ppm) along a pump shaft. This system has been adapted to fit on most process pumps. It meets the most stringent proposed government regulations on VOC and VHAP emissions. Pumps equipped with such seal systems can qualify for reduced (or are exempted from) emissions monitoring.

Advantages of Mechanical/Magnetic Fluid Seal System

Besides near zero measurable leakage, the seal system described in this paper has several advantages over other dual seal arrangements, including:

- Magnetic fluid secondary seal has no rubbing or wearing parts.
- Secondary seal can be recharged with magnetic fluid on a running pump.
- Does not require a liquid reservoir.
- Does not require purge gases.
- No need for cooling.
- Simple instrumentation (pressure gauge and flow switch) to detect system failure.
- Initial and operating costs are less than for most dual seal systems.

The only support equipment required is a vapor disposal means such as a flare, carbon filter, or refrigeration system.

Limitations of Seal System

The seal system is not a panacea for all pump emission problems. Pressure to the vapor disposal system must not exceed 7.0 psi. An occasional pressure surge above 7.0 psi will result in displacement of the magnetic fluid. Such over pressure events can be detected with a high pressure recording gauge. Renewal of the seal is easily done by refilling it with magnetic fluid, even on a running pump. A floating bushing is recommended to minimize emissions to atmosphere if the primary seal fails catastrophically.

Exposure of the magnetic fluid to liquids results in a loss of pressure capacity. Hence liquid leakage from the primary seal must be disposed of by drain-off or other means. Compatibility of magnetic fluids with various VHAP chemicals is undergoing evaluation. Alternate compatible fluids may result in reduced pressure capability. Temperature limit of the seal is between 200°F to 250°F, depending on the choice of magnetic fluid. Higher temperature can be accommodated by water cooling or frequent refilling with ferrofluid.

Final Remarks

The leakproof nature of the magnetic fluid secondary seal makes it practical to monitor the condition of both the primary and secondary seals by measuring vapor flow to the recovery line. Such monitoring assures emissions compliance, allowing the system to be exempt from inspection (according to the proposed EPA Clean Air Act Amendments). Furthermore, it provides a means of identifying a degraded primary seal with sufficient advance notice to schedule a shutdown for repair.

These results show that a dual seal arrangement incorporating a primary mechanical seal and a secondary magnetic fluid seal is a practical alternative to traditional dual seals and sealless pumps for strict emission control.

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