TANDEM SEALS FOR NEAR ZERO HYDROCARBON EMISSIONS

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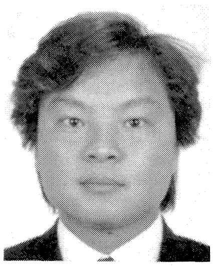

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ABSTRACT

More stringent regulatory limits are being placed on gaseous emissions from pumps handling hydrocarbon fluids. For example, new pumps in the Los Angeles Basin are required to have near zero emissions. The EPA is proposing a pump emission category of no detectable emissions (NDE), defined as being less than 50 ppm. Single seals have been developed that result in emission levels below 1,000 ppm, sufficient to meet regulations for existing equipment but not those for new pumps in the Los Angeles area. In response to these emission regulations, laboratory testing has been conducted on tandem mechanical seal arrangements in which the primary seal operated on propane and the secondary seal on various barrier fluids. Secondary seal emission rates of volatile organic compounds, on optimized seal designs, were determined to be well below 50 ppm. These results demonstrate that tandem mechanical seals provide an effective alternative to double seals and sealless pumps in the drive to achieve near zero emission rates for pumps handling light hydrocarbons.

INTRODUCTION

In the last decade, significant regulatory activity has occurred in the control of VOC (volatile organic compound) emissions for pumps and other fluid handling equipment. Typical fluids which generate VOC emissions are light hydrocarbons such as ethane, propane, and butane. Emissions from these fluids are in the form of vapors, which dissipate in the air and are a source of air pollution.

Prior to 1980, few regulatory controls were applied to these types of emissions. In 1983, in the Southern California area, Rule 466 [1] was enacted. The rule limited these emissions for pump seals to 10,000 ppm (parts per million) concentration in air. This emission limit became a national standard as Title 40, Part 60 of the Code of Federal Regulations [2].

The regulations have become much tougher in the Los Angeles Basin, consisting of Los Angeles, Orange, Riverside, and San Bernardino counties. The South Coast Air Quality Management District (SCAQMD) adopted the stringent Rule 1173 on July 7, 1989 [3]. Rule 1173, effective February 1991, reduces the allowable seal emission level to 1,000 ppm on existing equipment. New pumps will require best available control technology (BACT) which can be met by enclosing the shaft seal and venting to a vapor disposal or vapor recovery system. BACT also applies to existing pumps that have been subjected to five significant repair actions for major (> 10,000 ppm) gas leaks within a continuous twelve month period. The Environmental

Protection Agency (EPA) is proposing [4] a national emission standard of 1,000 ppm for pump seals. In addition, the EPA regulations describe a pump emission designation of no detectable emissions (NDE), as indicated by an instrument reading of less than 50 ppm above background. Pumps meeting this criteria are to be exempt from certain inspection requirements. Emissions are measured as defined by EPA Method 21 [2, Appendix A]. In this method, a portable organic vapor analyzer (also known as an OVA or sniffer) draws in a sample of air 1.0 cm (or less) from the interface between the rotating shaft and seal end plate. The sample is analyzed for the concentration of VOC and read as a ppm concentration of the VOC in the sample. Readings can be significantly affected by air flow in the measurement area.

TANDEM SEAL TECHNOLOGY

Optimized single mechanical seals for low specific gravity products have been successfully tested in a laboratory environment, resulting in emission levels well below 1000 ppm with low face wear. The optimized design variables include seal face deflection (pressure and thermal distortions), material selection for the mating faces, hydraulic balance ratio, and flush arrangement. The extensive testing [5] performed shows that a properly designed seal can even withstand short periods of "dry running," characterized by no liquid in the pump stuffingbox or seal chamber. In this mode of operation, no circulation flush is present to remove the heat generated at the seal faces, due to asperity contact.

Although single mechanical seals have been applied successfully in the field, with low emissions in light hydrocarbon applications, the new rules in Southern California will restrict their use. The South Coast Air Quality Management District (SCAQMD) Rule 1173 (effective 1991) states that after a pump has been repaired (to fix liquid leaks or major gas leaks) five times in a 12 month period, best available control technology (BACT) must be implemented. To meet BACT achieved in practice requirements, tandem or double seal systems must be used. Also, BACT is required on all new equipment. However, the technology applied to single seals for optimization in light hydrocarbon products is carried over to tandem seal arrangements, which provide additional benefits.

A tandem seal is typically two single seals in a series arrangement (Figure 1). In this paper, tandem seals will be defined as dual sealing arrangements with a nonpressurized barrier fluid. A double seal will be defined as having a pressurized barrier fluid which is at a higher pressure than the process fluid. (API 610-7th edition defines double seals as having two rotating faces per chamber, sealing in opposite directions, and tandem seals having two rotating faces per chamber, sealing in the same direction.) In the tandem arrangement, the primary (inboard) seal is in contact and pressurized by the process fluid. The secondary (outboard) seal will seal the barrier fluid, a nonhazardous fluid with good lubricating properties, at atmospheric pressure. Typical barrier fluids are ethylene glycol / water mixtures, kerosene, diesel fuel, 10 wt oil, and automatic transmission fluid (Table 1). A vent line to a vapor recovery system (VRS) from the seal reservoir must be provided.

Since the primary seal operates with a full pressure drop from OD to ID of the sealing interface, it will typically fail first, due to increased closing force at the seal faces and the nonlubricating properties of most light hydrocarbon products. The secondary seal is running with barrier fluid lubrication at atmospheric pressure and has greater life expectancy. If the primary seal fails, pressure and liquid level of the reservoir will increase, signalling an alarm and triggering the solenoid valves to close off the reservoir system (Figure I). The secondary seal will then be running as a single seal until time allows for an orderly shut down procedure. Therefore, in order to have a reliable minimum leakage tandem seal, its single seal components must have the same optimized design as the single mechanical seals for light hydrocarbons.

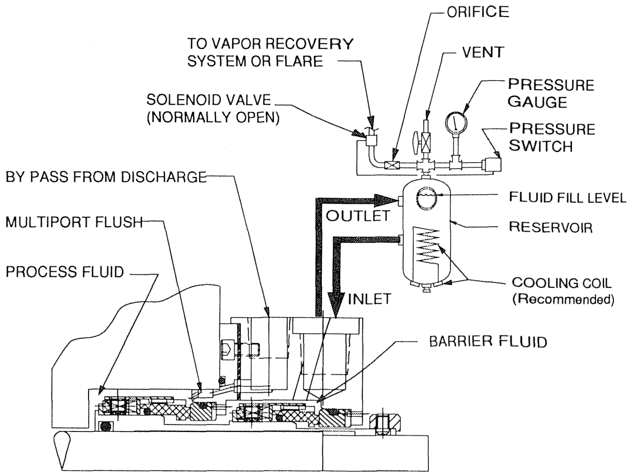


Figure 1. Tandem Seal and Auxiliary System Arrangement.

Table 1. Barrier Fluid Recommendations

Barrier Fluid	Low temp limit °F	High temp limit	Specific Gravity
Kerosene	0	275	0.80
Diesel #2	10	300	0.82
Hydraulic Oil 10 Wt	30	300	0.87
Mineral Oil	-13	400	0.87
ATF Dexron II	55	200	0.88
Silicone H.T. Fluid	-40	300	0.93
Water/Glycol 50/50	-50	220	1.06

In comparison to double seal systems, tandem arrangements are much simpler to install and maintain. Double seal systems were once the preferred method for meeting minimum leakage requirements due to the higher pressure of the barrier fluid compared to the process fluid (protects environment from the escape of any hazardous fluids). However, the simplicity and effectiveness of the modern tandem arrangement often outweighs the small leakage advantage of the double seal. Since the double seal system requires barrier fluid pressurization, its auxiliary support system can be quite involved, requiring separate pressure pumps with their own integral sealing system. Since barrier fluid pressure is higher than stuffingbox, a small amount of reverse leakage into the process fluid will occur. It also requires an operator to maintain the correct liquid level in the reservoir. Finally, the inherent nature of the double sealing method prevents its use as a backup system in case of a single seal failure. The secondary seal (outboard) will usually fail first due to the high pressure drop (process + approximately 50 psi to atmosphere). Most double seal systems need to be shut down immediately in case of barrier fluid loss.

Solubility of Hydrocarbon Gases in Barrier Fluids

A small amount of leakage occurs across the secondary seal faces to the environment. Emission rates from this leakage are higher when the barrier fluid is petroleum based due to evaporization of the leaking liquid and the greater concentration of dissolved light hydrocarbons in petroleum fluids as compared to water. The solubility of a gas in a liquid is approximately proportional to the partial pressure, P, of the gas, provided the total system pressure is on the order of 1.0 atmosphere or less (unpressurized barrier fluid). Mole fraction, X, of the dissolved gas can be calculated from Henry's law [6]:

$$X=P/H \qquad (1)$$

Henry's law constant, H, is a function of the gas and liquid. Generally H increases with temperature and, hence, solubility decreases with increasing temperature. The maximum quantity of hydrocarbon gas that can be dissolved in an unpressurized barrier fluid occurs when the gas partial pressure, P, is one atmosphere. (Space at top of reservoir is 100 percent hydrocarbon vapor).

Using information provided by Stephen and Stephen [7], calculated maximum concentration of several light hydrocarbons in a water barrier fluid are presented in Table 2. On a weight basis, maximum gas solubility is less than 0.01 percent in water. Hence, with a water barrier fluid, secondary seal emissions should be extremely low.

Table 2. Gas Solubility in Water (1 atm partial pressure)

Gas	Temp °C	Solubility Wt.%
Ethane	20 30 40	0.0064 0.0048 0.0039
Propane	20 30	0.0075 0.0055
Butane	20 30	0.0083 0.0059

Petroleum based barrier fluids can adsorb substantially greater amounts of hydrocarbon gases. A method to estimate solubility of gases in petroleum liquids is referred to by ASTM [8]. Solution concentration decreases with increasing liquid density and increasing temperature. Maximum dissolved ethylene concentration as a function of oil barrier fluid density and temperature is given in Table 3. (Concentration values for other light hydrocarbon gases are not given by ASTM [8]. Ethylene concentrations are less than 0.5 percent of the barrier fluid and, hence, the dissolved gas would have only a minor effect on secondary seal emission levels.

Solubility of propane and butane in methanol is given by Stephen and Stephen [7] and Table 4. Propane concentration at 0°C is almost three percent (wt/wt), while butane concentration is nearly six percent at 25°C. It would appear that solution gases in methanol could contribute a small, but detectable, level of emissions from the secondary seal. This effect might be significant at temperatures below freezing.

Table 3. Ethylene Solubility in Petroleum Liquid (1 atm partial pressure)

Liq.s.g.	Temp °C	Solubility wt.%
0.75	20 35 50	0.41 0.34 0.28
0.88	20 35 50	0.15 0.13 0.10

Table 4. Propane and Butane Solubility in Methanol (1 atm partial pressure)

Gas	Temp °C	Solubility wt.%
Propane	0 25 50	2.8 1.3 0.4
Butane	25 35 50	5.7 3.5 1.5

Barrier Fluid High Temperature Emissions

Fluid vapor pressure increases with temperature, and thus emissions due to barrier fluid vaporization might be of concern at high temperatures. A simple experiment was conducted to examine this effect. An open container of diesel fuel was heated slowly on a hot plate (Figure 2). Emissions were monitored with the sniffer tip located about one inch above the liquid surface.

Results of this experiment are shown in Figure 3. Vapor concentration level above the liquid surface was about 180 ppm at 100°F, 1400 ppm at 200°F, and 2700 ppm at 250°F. Actual emissions from a secondary seal would be substantially lower than these values, but the results demonstrate the desirability of using a cooled reservoir.

TEST SETUP

Low emissions testing of tandem mechanical seals required a seal tester specifically designed for operation in light hydrocar-

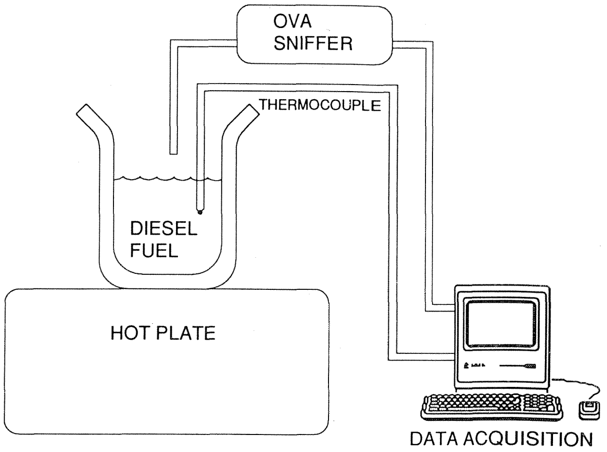


Figure 2. Test Setup for Barrier Fluid High Temperature Emissions Test.

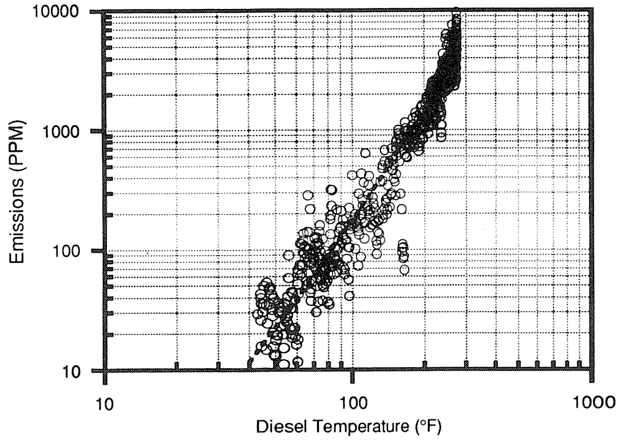


Figure 3. Diesel Fuel Emissions vs Fuel Temperature. (See Figure 2 for setup.)

bons (Figure 4). The seal tester is composed of a cylindrical test chamber penetrated by a rotating shaft. The shaft is belt driven by a 50 hp explosion proof motor. This testing arrangement is capable of simulating pump shaft speeds to 6000 rpm, stuffingbox pressures to 2200 psi, and shaft diameters to six inches.

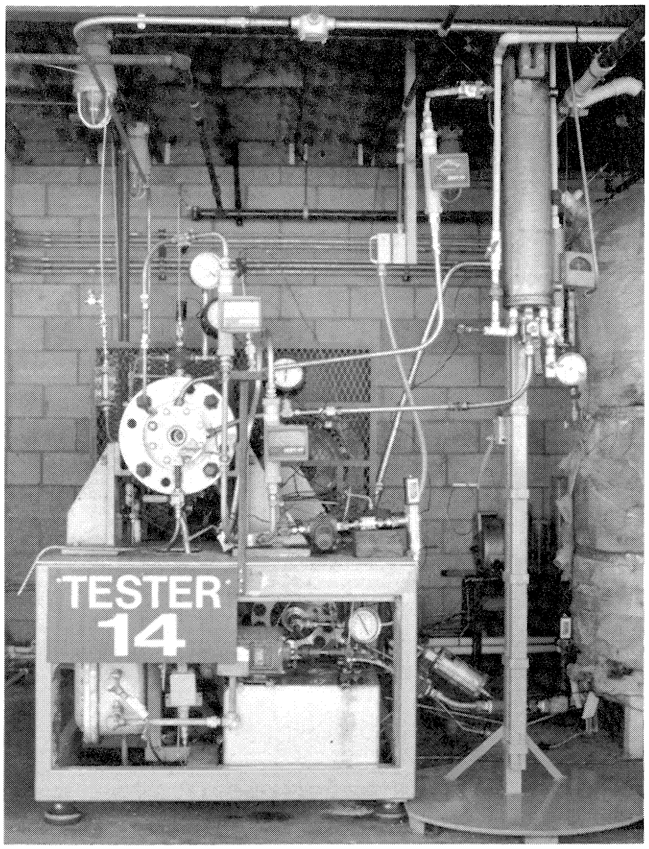


Figure 4. Test Rig for Evaluating Tandem Mechanical Seals for Low Emissions.

Three single mechanical seals could be evaluated simultaneously with the tester. A single mechanical seal was installed at the driver side (inboard) of the tester for the purpose of sealing that end. The outboard side of the tester was sealed by the tandem mechanical seal, which consisted of a primary and secondary seal. Test fluid flow for the driver side seal and primary seal on the tandem arrangement was provided by a circulation pump, while the secondary seal had circulation provided by the integral pumping action of the rotating assembly. Test chamber pressure was controlled by a pressure pump/accumulator system. Cooling to remove system heat was achieved through liquid/liquid heat exchangers allowing test temperature to be controlled. The temperature of the barrier fluid in the secondary seal cavity was controlled by adjusting the water flow rate through the cooling coil in the five gallon reservoir. The barrier fluid reservoir was vented to atmosphere, resulting in a full pressure drop from primary to secondary seal. In field application, the reservoir is vented to either a vapor recovery system or flare. Temperature and pressure regulation of the primary seal chamber were controlled manually due to safety issues regarding the volatile nature of the propane.

In order to monitor the test fluid and seal face conditions in a real time continuous manner, temperature and pressure sensors were used for transmission of signals to the host computer (every 10 seconds). Seal face temperature of the secondary seal was measured by a thermocouple inserted in the stationary face (Figure 5). It was then possible to have a graphical display of the difference between the seal face temperature and bulk fluid temperature, referred to as ΔT (Figure 6). During seal operation, the magnitude and stability of the ΔT gives a good indication of the seal's fluid film status, heat generation rate, and relative stability. *Note that* ΔT *was monitored for the secondary seal faces only in Figure 6, and the screen is showing startup of a tandem seal.*

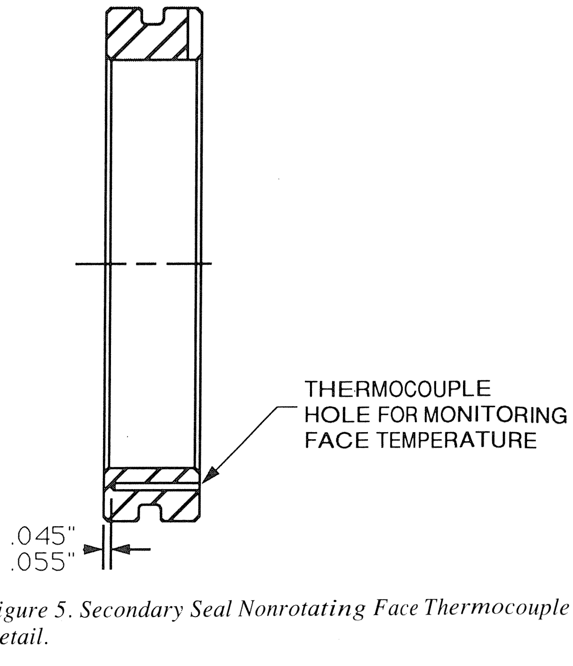
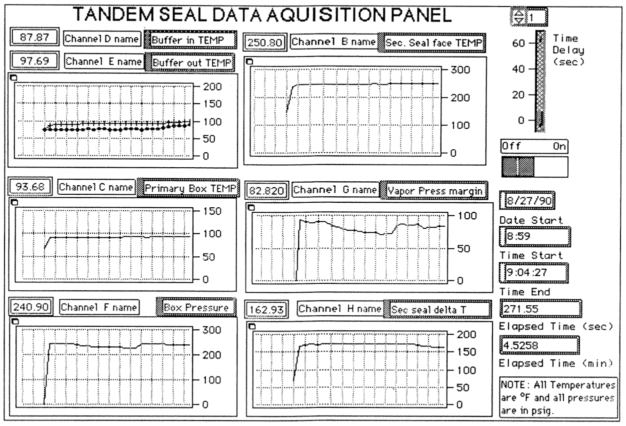


Figure 5. Secondary Seal Nonrotating Face Thermocouple Hole Detail.

Using fluid temperature and primary seal chamber pressure as input, continuous vapor pressure margin calculations were performed and displayed in real time via the data acquisition screen. Vapor pressure margin is defined as the difference between the seal chamber pressure and vaporization pressure of the product at chamber temperature. A vapor pressure margin of zero is the condition at which the fluid has made a transition into a gas.



TANDEM SEAL DATA AQUISITION PANEL

87.87 Channel D name: Buffer in TEMP

97.69 Channel E name: Buffer out TEMP

250.80 Channel B name: Sec. Seal face TEMP

93.68 Channel C name: Primary Box TEMP

82.820 Channel G name: Vapor Press margin

240.90 Channel F name: Box Pressure

162.93 Channel H name: Sec seal delta T

Time Delay (sec): 1

Date Start: 8/27/90

Time Start: 8:59

Time End: 9:04:27

Elapsed Time (sec): 271.55

Elapsed Time (min): 4.5258

NOTE: All Temperatures are °F and all pressures are in psig.

Figure 6. Computer Data Acquisition Screen for Tandem Seal Testing In Propane (Shortly after Startup).

TEST CONDITIONS

The primary mechanical seal was tested in propane at a seal chamber pressure of 240 psig. The temperature of the propane was approximately 112°F and the shaft speed of the motor was 3600 rpm. The secondary seal was evaluated using various barrier fluids and a five gallon reservoir. The 2-7/8 in primary seal and 2-5/8 in secondary seal were typically tested for a 20 to 30 hour duration. Both of these seals were spring-pusher types, employing a carbon rotating face and silicon carbide stationary face.

Emission readings were made using EPA Method 21 [2] with a portable organic vapor analyzer (sniffer). The instrument allows direct readout of total organic vapor concentration and features a flame ionization detector with three scale ranges of 0 to 10, 0 to 100, and 0 to 1000 ppm.

TEST RESULTS

Emissions

Secondary seal emissions were determined by first measuring the background level about three feet from the seal and then sampling within one centimeter of the shaft sleeve and seal flange. The sniffer tube was then moved circumferentially around the sleeve and the highest reading noted. Actual seal emission was recorded as the difference between the sleeve/flange value and background.

Emissions from the secondary seal were never higher than 50 ppm above the background reading and, for most barrier fluids, the average value was less than 10 ppm. Occasionally seal emissions were a few ppm less than background. Background emission levels generally ranged from 2 to 20 ppm and resulted from the difficulty in obtaining leak tight pipe fittings and flange connections.

There was one test, however, where apparent seal emissions were about 200 ppm. Using the sniffer, it was revealed that propane gas from the primary seal chamber was leaking through all eight flange/housing bolt holes. The test assembly was disassembled and a deteriorated flange gasket replaced. (This gasket had experienced several tests which also involved assembly and disassembly.) With the new gasket in place, very low secondary seal emissions were recorded.

Both the primary and secondary seals were the same type. When used as a SINGLE seal on light hydrocarbons this seal experiences less than 1,000 ppm emissions, with average emissions below 200 ppm. On nonvolatile liquids, it generally exhibits no visible leakage. Thus, in a tandem seal arrangement, expected emissions from the secondary seal should be near zero.

The possibility existed that the barrier fluid was not saturated with propane due to low primary seal emission rate. In an attempt to ensure saturation, near the end of testing on diesel oil, propane was bubbled through the barrier fluid reservoir at a rate of 10 scfh for a period of 10 minutes. Secondary seal emissions were below 10 ppm for the remainder of the test (1-3/4 hr).

Under the conditions of this test program, the choice of barrier fluid had no significant effect on secondary seal emissions. Our highest barrier fluid temperature averaged 150°F. At temperatures above 200°F (still below diesel's initial boiling point), diesel oil in an open beaker gives off emissions exceeding 1,000 ppm (Figure 3). Actual high temperature emissions from a seal would be substantially lower than that, but might exceed the EPA level of less than 50 ppm in no detectable emissions (NDE) applications. Hence, reservoir cooling is recommended to assure low emissions as well as to promote better lubrication for the secondary seal faces.

Face Temperature Rise

As noted earlier, secondary seal face and chamber temperatures were monitored with thermocouples. The difference, known as ΔT, is a very good indicator of seal performance. Delta T values are contained in Table 5 and a typical plot using barrier fluids 30 wt oil and water/glycol is shown in Figure 7. Lowest temperature rise of 40°F was obtained with the 50/50 mixture of water/glycol and a water cooled reservoir. When the reservoir was uncooled, ΔT was 55°F. All other barrier fluids resulted in significantly hotter running secondary seal faces.

Table 5. Tandem Seal Test Results

Fluid	Face design	Spring Load	Flow(gpm)	ΔT (°F)	Tbuff (°F)
Water/Gly	Standard	Standard	6.0^2	55	150^1
Water/Gly	Standard	Standard	6.0^2	40	110
30 wt oil	Standard	Standard	3.5^2	180	150
30 wt oil	Standard	0.5	3.5^2	100	140
30 wt oil	76% RW^3	Standard	3.5^2	145	150
30 wt oil	76% RW^3	0.66	3.5^2	119	139
30 wt oil	Hydropads	Standard	3.5^2	100	135
10 wt oil	Standard	Standard	2.7	127	137
10 wt oil	Hydropads	Standard	2.5	90	128
Silicone	Standard	0.66	3	116	122
Diesel#2	Standard	Standard	3.3	78	131
Diesel#2	Standard	0.66	3.3	74	125
Mineral oil	Standard	Standard	3.1	137	150
Mineral oil	Standard	0.66	2.8	118	142
ATF	Standard	0.5	2.6	138	140
Kerosene	Standard	0.66	3.0	70	115
Kerosene	Standard	Standard	3.0	78	115

1. Uncooled Reservoir
2. Axial screw pumping ring
3. Radial width

The barrier fluid reservoir is vented to atmosphere; thus, there is only about 4.0 ft of liquid head on the secondary seal, which is inadequate to provide a thick lubricating fluid film across the faces. Hence, secondary seals can be expected to run hotter than if on the same fluid, but at a moderately higher pressure.

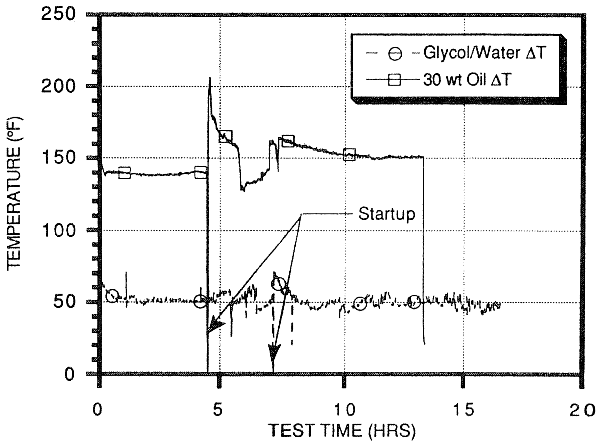


Figure 7. Secondary Seal Face ΔT vs Test Time.

It is anticipated that service life could be enhanced if seal face temperature could be reduced and, hence, the following methods to lower the ΔT were evaluated:

- Narrower than standard face width (76 percent of nominal)
- Hydropadded face (Figure 8)
- Reduced spring loading (1/2 or 2/3 of nominal)

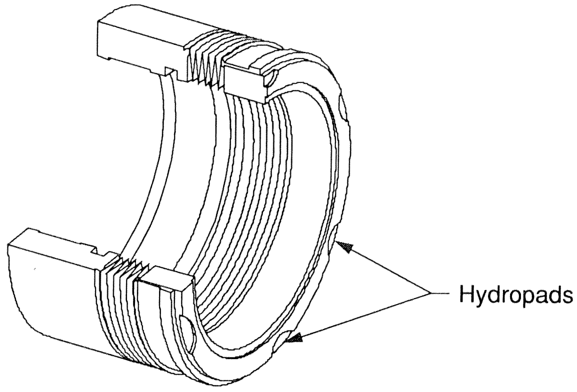


Figure 8. Hydropadded Face (for Bellows Seal).

All of these options did result in reduced face temperature (Table 5) and lower wear rates. Use of hydropads and reduced spring loading resulted in liquid "weepage" leakage on the secondary seal. Generally, this dried up in a few hours. Even with weepage, emission levels were well below 50 ppm. Since the same seal type is used for both primary and secondary seals, the use of nonstandard faces (narrow or hydropad) could result in misapplications in the field. If service life of the secondary seal is deemed to be insufficient, the recommendation is to use a standard seal with approximately 70 percent of nominal spring compression. Reduced spring loading in this test program was achieved by using either three or four instead of six springs. In the field this would be accomplished by designing in reduced compression rather than fewer springs to assure an even face load. Further evaluation of seal capability (with reduced spring loading) to hold pressure in the event of a primary seal failure is required.

Heat Generation and Need for Cooling

Heat generated by the secondary seal and heat soak from the primary seal is in the range of 20 to 80 Btu/min as determined from barrier fluid flow rate and temperature rise across the seal chamber. This heat is transferred from the seal to the barrier fluid. Rotating elements in the secondary seal were used to pump the barrier fluid through a five gallon reservoir containing cooling coils and back to the seal, thereby producing seal flush. Barrier fluid flow rates and temperature are contained in Table 5. Product (propane) temperature in the primary seal chamber was maintained at 112°F. Cooling water temperature ranged from 65 to 80°F at a flow rate (through cooling coils in five gallon reservoir) of about 2.5 gpm. The coolest barrier fluid, shown in Table 5, was the water/glycol mixture, at 110°F, with water cooling of the reservoir. With no reservoir cooling the water/glycol temperature was 150°F (Figure 9) and face ΔT increased by 15°F. Water cooling of the reservoir was used on all other barrier fluid tests. It can be seen from Table 5 that the use of reduced spring compression or hydropads resulted in lower temperature of the barrier fluid as well as lower face ΔT. Beneficial effects of lower barrier fluid temperature are longer seal life and reduced corrosion rates of system components.

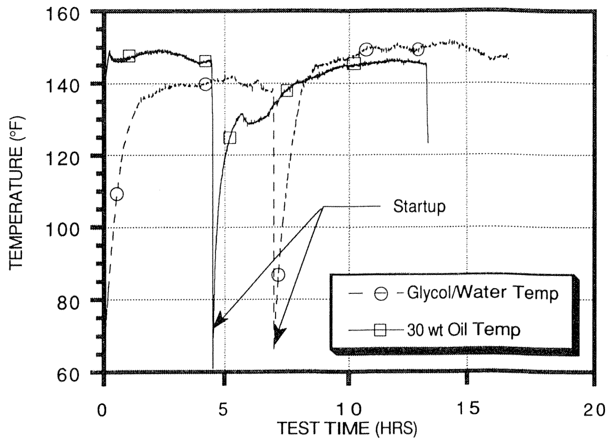


Figure 9. Barrier Fluid Temperature vs Test Time (Water/Glycol Had No Water Cooling for Reservoir.)

CHOICE OF BARRIER FLUID

Choice of a suitable barrier fluid depends on consideration of temperature limits, compatibility with the process fluid, cost, and ease of disposal. Secondary seal emissions were below 50 ppm with all of the fluids tested.

A 50/50 mixture of water/glycol results in very good heat transfer as evidenced by low face ΔT and the coolest barrier fluid temperature. One drawback is that in practice, water and glycol may be mixed in improper proportions, which changes the freezing and boiling points and could lead to poor seal performance. For this reason, it is recommended that water/glycol be premixed and stored in drums.

Diesel fuel and kerosene result in less face temperature rise compared to that obtained using other petroleum barrier fluids. The mineral oil heat transfer fluid might be considered due to its wider temperature range and good lubricating properties.

Pumping Ring Flow results for the standard rotating elements are shown in Table 5 and a typical plot shown in Figure 10. The axial screw impeller produced 6.0 gpm on water/glycol and 3.5 gpm on 30 wt oil. With just the standard rotating elements to produce flow, this was reduced to 2.5 gpm on 30 wt oil, about 1.0 gpm/in of seal balance diameter, more than adequate for seal cooling. Due to space constraints of most pump designs, axial screw arrangements to achieve flows above 2.5 gpm could not be fitted.

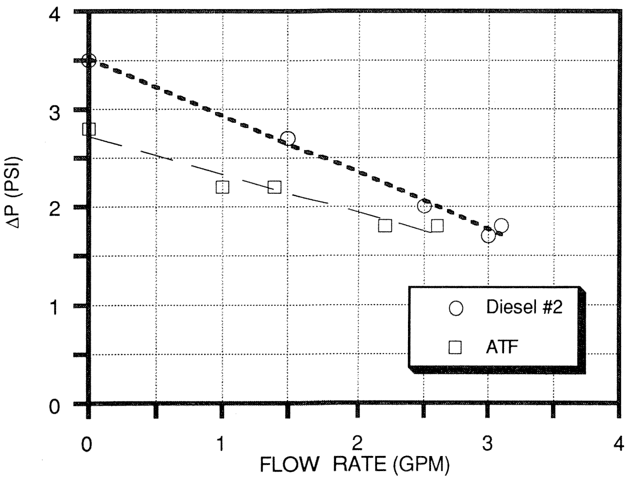


Figure 10. Seal Generated Head vs Flow Rate (No Pumping Ring).

The secondary seal flange had tangential inlet and outlet ports (Figure 11) to facilitate flow. Earlier tests with radial ports resulted in flows about 1/4 of that obtained with tangential ports.

Thermosyphon enhancement to flow was accomplished by having the flange outlet port on top and return port on the bottom. Also the reservoir inlet is higher than the outlet as shown in Figure 11. Thermosyphon flow results from the density difference between the "Hot" outlet line and "Cold" return line to the seal chamber. It can be estimated as follows:

$$\Delta P = (\rho_c - \rho_h)\ g\ h \tag{2}$$

Ignoring flow losses due to fluid friction, fittings, etc.,

$$1/2\ \rho u^2 = (\rho_c - \rho_h)\ g\ h \tag{3}$$

Solving for flow velocity,

$$u = (2\ (\rho_c - \rho_h)\ g\ h\ /\ \rho)^{1/2} \tag{4}$$

Volumetric flow rate is then,

$$Q = u\ A = u\ (\pi/4)\ D_i^2 \tag{5}$$

Equation (5) was used to estimate thermosyphon induced flow in the system. Tubing ID is 0.584 in (3/4 in tubing). For a light oil with a 3°F difference between the hot and cold lines, the upper limit on thermosyphon flow is 0.4 gpm. The actual value would be less due to flow losses. Nevertheless, it is apparent that plumbing for barrier fluid circulation flow should be designed to enhance thermosyphon flow rather than retard it.

FIELD RESULTS

Many tandem seals similar to the type tested were installed during 1989 and 1990 in Los Angeles area refineries. Pumped products are either propane, butane, or propane/butane mixture. Most systems use a five gallon reservoir with a diesel fuel barrier fluid. Six pumps with a total of seven tandem seals were monitored for emissions from one to three months after startup. Emissions from four of the secondary seals were less than 20 ppm above the background. (Background ranged from 4.0 to 10 ppm). Organic vapor levels near the remaining three secondary seals varied from 80 to 540 ppm. Two of the latter pumps had fitting or flange emissions in excess of 1,000 ppm, which almost certainly accounted for the high concentration of VOCs in the vicinity of the secondary seals. No such high emission sources could be located on the remaining pump. Follow up investigation will be done to determine why this pump had an emission level of about 300 ppm. These field results, with one unexplained exception, demonstrate that tandem mechanical seals can achieve near zero (< 50 ppm) emissions under refinery operating conditions.

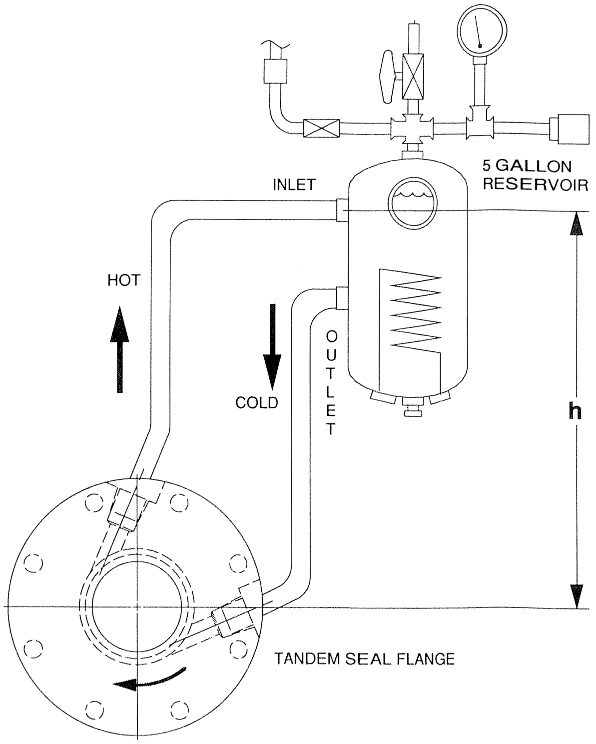


Figure 11. Thermosyphon Arrangement.

CONCLUSIONS

Laboratory testing and field evaluation have demonstrated that properly designed tandem mechanical seals can be used on light hydrocarbon pumps to maintain near zero emission levels (<50 ppm). Leak tight gaskets and pipe fittings are also required to achieve very low pump emission levels. Secondary seal life can be enhanced by reducing the spring loading on the seal and by using a cooled reservoir to maintain barrier fluid temperature at acceptable levels.

NOMENCLATURE

A	Tubing cross sectional area
D_i	Tubing ID
g	Gravitational constant
h	Vertical distance between pump centerline and reservoir inlet
H	Henry's constant
P	Pressure
Q	Barrier fluid circulation flow rate
T_{buff}	Temperature of buffer fluid
u	Flow velocity in tubing
ΔP	Pressure rise across secondary seal pumping ring
ΔT	Temperature difference between secondary seal face and barrier fluid

ρ	Fluid density
ρ_c	Fluid density in "Cold" line
ρ_h	Fluid density in "Hot" line

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