

TESTING AND FIELD PERFORMANCE OF A HIGH DUTY NGL PIPELINE SEAL

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design has completed over one year service life with very low leakage rates. Field data over the first year of operation is included.

INTRODUCTION

The pipeline pumps were originally commissioned in 1982. A tandem mechanical seal design optimized on propane testing with development tools available in the late 1970s and early 1980s, was being used in the pumps with limited success. This design was characterized by occasional short term failures and a six month life cycle. Lack of stability and accelerated failure rate at shaft speeds over 3600 rpm also presented problems to the user.

Since the original design was validated on propane testing and shown to have good performance, a development program was initiated based on testing in an NGL mix approximating the actual product mix in the pump. This program started with a baseline test of the original seal under simulation of field conditions in the test lab. Weak points in the design were then identified, analyzed, and a new design produced for final validation testing.

Goals for the development program included the elimination of short term failures, attainment of an average seal life of two years, allowance for stable operation of the pump over the full range of shaft speeds, and maintenance of low leakage rates of the pumped product.

Field Conditions

Pumping conditions for the service are as shown in Table 1. Typical properties of the pumped fluid are shown in Table 2, with actual stream measurements from February 1992. Weight percent breakdowns of product were converted from barrel per day and mole percent data. A cross sectional view of the pump design is shown in Figure 1.

Test Program Design

A laboratory test set was developed to simulate the field operating conditions as closely as possible. The test machine

Table 1. Pumping Conditions.

Suction Pressure	775 PSIG
Speed	2000 - 5100 RPM
Discharge Pressure (2000 RPM)	1350 PSIG
Discharge Pressure (5100 RPM)	2175 PSIG
Temperature (Max)	115° F

ABSTRACT

An improved multiple seal design was developed for a critical high performance pipeline pumping application with a difficult range of conditions. The pumps are four stage construction, with a 7000 hp electric motor and variable speed gear box driven with a range from 2000 to 5000 rpm.

The mechanical seal has a nominal balance diameter of 6.5 in, and is a dual seal design with a buffer fluid. The sealed product is a mixture of ethane, propane, butane, and some heavier liquids, and typically ranges from 28 to 37 percent ethane concentration with a mixture of other hydrocarbons and a temperature up to 115°F. The combination of conditions makes this a demanding sealing application.

Information is presented on the seal design, development, and test data on a 40 percent ethane NGL mix. The improved seal

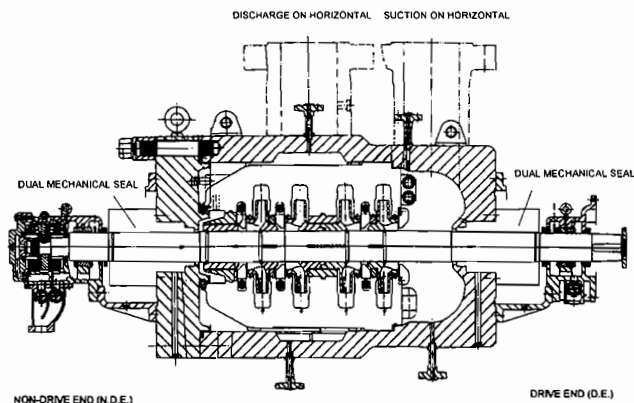


Figure 1. Pump Cross Section.

used is shown in Figure 2. It incorporates a 150 hp motor designed to provide high start up torque, a belt drive system capable of speeds up to 6000 rpm, and a seal housing overhung on a process pump type bearing bracket. Maximum pressure capability is 2000 psig with seal sizes of up to 8.0 in balance diameter. A pressurization system, as shown in Figure 3, provides for the circulation of light hydrocarbon products.

Test conditions were established per Table 3, using a test fluid to simulate the product as shown in Table 2. The buffer fluid was diesel fuel.

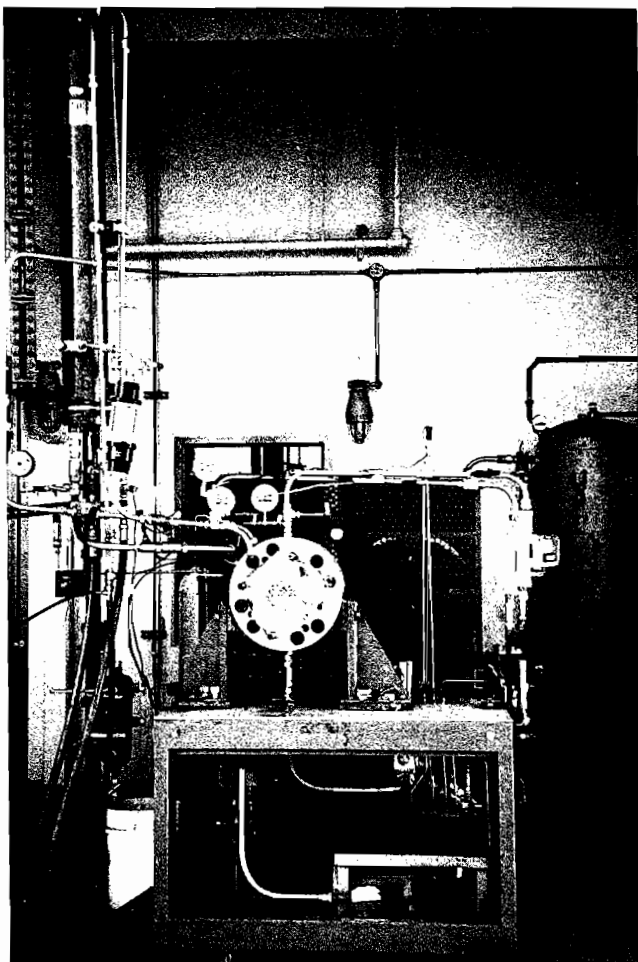


Figure 2. Test Unit.

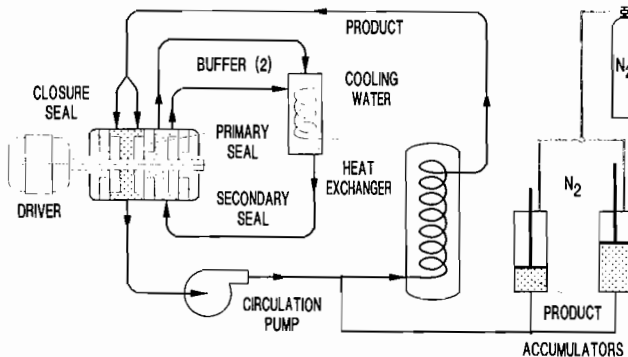


Figure 3. NGL Test Loop Circulation Plan.

Table 2. NGL Product Analysis—Typical Weight Percent of Components.

Component Pumpage / Test Mix	Test Mix	February 92 Product Analysis	Summer Typical	Winter Typical
Methane CH ₄	0.0%	.4%	Trace	Trace
Ethane C ₂ H ₆	40.0%	40.3%	28%	37%
Propane C ₃ H ₈	50.0%	42.1%	35%	38%
Butane C ₄ H ₁₀	10.0%	17.2%	19%	16%
Heavier C ₅ +	---	---	18%	9%
Calculated Vapor Pressure at 115° F, PSIA	500	510	-----	-----
Calculated SG at 60° F	.45	.45	-----	-----

Table 3. Seal Test Conditions.

Test Speed	2200, 3600, and 4800 RPM
Product Pressure	850 PSIG
Product Temperature	90 - 115° F

Measurement points in the system were as follows:

- Product Inlet and Outlet Temperatures
- Seal Chamber Pressure
- Buffer Fluid Inlet and Outlet Temperatures
- Buffer Fluid Pressure
- Primary and Secondary Seal Face Temperatures
- Primary Seal Flush Flowrate
- Buffer Fluid Circulation Flowrate
- Cooling Water through Reservoir Flowrate, Inlet, and Outlet Temperatures

Seal face temperatures were measured using a thermocouple inserted into a drilled hole in the carbon—graphite stationary face located within 0.040 in of the sealing surface.

Baseline Testing—Original Seal Design

The original seal design (Figure 4) was baseline tested on the NGL mix per the Table 3 conditions.

The seal design was a tandem arrangement, stationary spring design seal with a nonpressurized buffer fluid. A single port

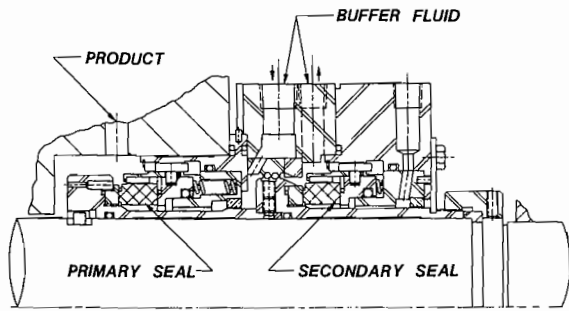


Figure 4. Original Seal Design.

product flush was utilized over the primary seal, and an axial screw type pumping ring with grooved rotor and stator fed from a plenum cavity was used to circulate the buffer fluid through a reservoir. The buffer fluid reservoir contains cooling coils that are used for cooling fluid obtained from product circulation in the field installation. This cooling fluid was simulated by cooling water circulation on test. Seal face materials were carbon-graphite-resin vs reaction bonded silicon carbide.

On initial testing, the NGL mixture was noted to be difficult to handle as it was prone to developing gas bubbles during system venting. Four tests were conducted with repetitive results. Conclusions were as follows:

- Initial primary seal face performance was erratic, and the seal face temperatures and leakage showed wide variations and evidence of face flashing.
- After the first 24 hr of running, the seal faces showed more stable performance.
- Seal leakage and even minor flashing could induce loss of buffer fluid circulation due to gassing of the secondary seal cavity. Loss of buffer fluid circulation resulted in high primary and secondary seal face operating temperatures. Once lost, buffer circulation was difficult to reestablish.

The results of this testing were comparable to actual operating experience on the seal.

ANALYSIS OF ORIGINAL SEAL DESIGN PERFORMANCE

Primary Seal Performance

Primary seal face performance was seen to be erratic during testing. Two reasons were established for this performance.

First, the flush flow of the NGL mixture tends to easily form gas bubbles. This tends to occur in areas where heat is added, such as seal faces. The single port injection at the faces is inadequate to sweep away bubbles and prevent hot spots. The material combination of carbon-graphite-resin vs reaction bonded silicon carbide showed wear and blistering.

Second, the faces were seen to go into outer diameter contact due to pressure coning effects, sealing off the fluid film. As face temperature increases, thermal rotation brings the faces into contact at the inner diameter.

The net result of these effects was unstable face performance until an initial wear in pattern was established.

The NGL mix appeared to be able to provide adequate seal face load support. Previous theories had been proposed that the ethane component of the mix would act partially as a supercritical vapor (packed gas) and would not adequately lubricate the seal faces. Test results combined with analysis of returned parts from the field, however, indicate that the NGL mix could provide good seal face life.

Secondary Seal Performance

The secondary seal performance is dependent on the maintenance of buffer fluid circulation. On baseline testing, the buffer fluid circulation was subject to stalling, especially during periods of high leakage through the primary seal. The relative expansion rate from liquid to gas transition of the product is shown in Figure 5. A small amount of liquid leakage expands greatly in volume in the low pressure buffer fluid cavity, causing stalling of the axial flow pumping ring. Since only the head in the reservoir was available to reestablish suction pressure to the pumping ring, flow was extremely difficult to reestablish once lost.

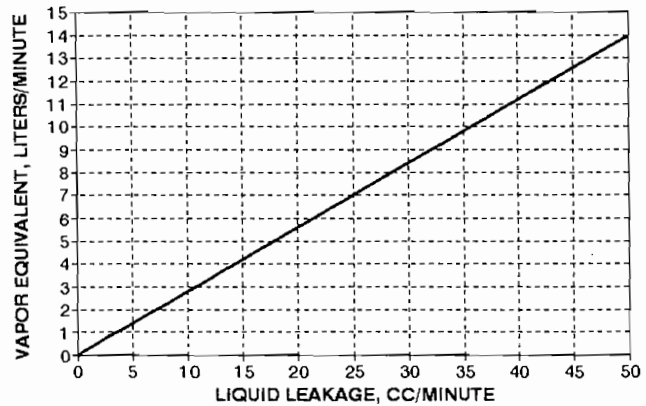


Figure 5. Equivalent Vapor Volume from Liquid Leakage.

NEW SEAL DESIGN

Given the limitations of the existing design, a completely new design was developed which would eliminate the problems. This design is shown in Figure 6.

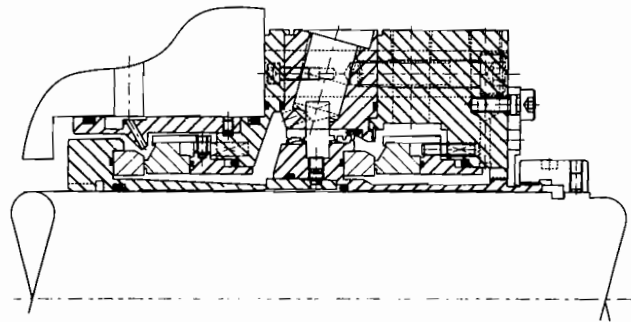


Figure 6. New Seal Design.

Primary Seal Faces

Existing primary seal faces were analyzed on a computer aided seal analysis program incorporating film theory and finite element analysis [5]. The faces were found to have negative coning (tending towards outer diameter contact) in the analysis matching well with the test results.

New faces were designed and optimized using the same computer aided seal analysis model. The performance of a mechanical seal is principally related to the load supporting film between the faces. The film must be sufficient to provide load support, but not excessive leakage. The film is affected by

surface distortions, pressure and temperature gradients, surface roughness, and dynamic influences, such as vibrations.

The computer aided seal analysis model is a group of programs, centered about finite element analysis, which calculates pressure and temperature distortion for a stationary and rotating seal face combination, including the operating conditions. The system is a tool that allows faces to be designed so that their coning deflections at operating conditions are nearly optimized for best performance.

In most mechanical seals, the stationary and rotating faces are designed to form a convex shape when operating dynamically. In this case, the fluid film is in a wedge shaped annulus with the widest distance across the outer diameter. This assumes the highest pressure is on the seal's outer diameter. However, it is also desirable for the faces to distort to a concave condition (heavier contact at the faces outer diameter) when not rotating, to prevent static leakage. These two conditions are achieved by controlling the seal face's pressure and thermal distortions.

The original seal face design was found to have a pressure deflection of $-140 \mu\text{in}$ at the operating pressure. The new seal was designed to have a pressure deflection of $-37 \mu\text{in}$. Pressure sensitivity of the carbon graphite stationary face was reduced by changing the carbon shape to balance out the cross section and provide optimum rotation. The thermal deflection for both the original and new face designs remained virtually identical at $70 \mu\text{in}/\text{hp}$ input. The pressure deflection results over the range of pressures are shown in Figure 7.

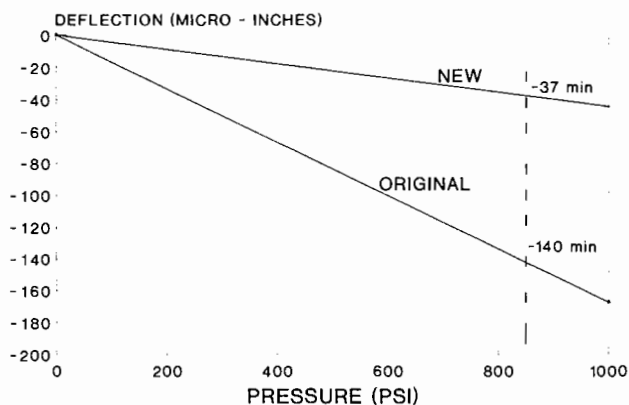


Figure 7. Seal Face Deflection Comparison.

Primary Seal Flush

The primary seal flush was changed to a multiport injection type flush incorporating eight evenly spaced holes opening into a plenum sized to ensure even distribution of the flush, and provide adequate velocity to penetrate the sealing interface and wash away gas bubbles that may form. Success of this flushing method and its effect on stabilizing face temperatures has previously been documented [2, 3].

Secondary Seal Faces

A seal face combination identical to the primary seal was chosen for the secondary seal faces. This allows for safety in the event of a primary seal failure, and ease of installation as the faces are interchangeable.

Secondary Seal Buffer Flow System

The secondary seal buffer system was redesigned to prevent stalling due to vapor expansion of leakage from the pumped product. A 50/50 mix of ethylene glycol and water was chosen

as the buffer fluid due to its favorable combination of properties for the service [4].

In the new buffer flow system, a novel design incorporating two axial flow pumping rings pumping in opposite directions was used (Figure 8). One ring provides cooling and lubricating flow to the secondary seal while the other ring directs gaseous primary seal leakage away from the secondary seal chamber to a port located in the vapor space of the buffer fluid reservoir.

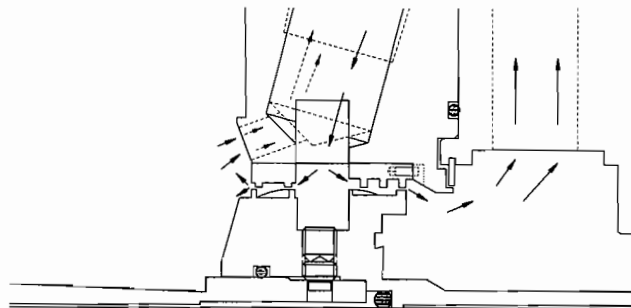


Figure 8. Dual Axial Flow Pumping Ring Construction.

Effects of Buffer Fluid Pressurization

As a further safeguard against stalling of the barrier fluid flow with variations in primary seal leakage, to ensure adequate suction head to the pumping rings, and to improve face performance of the secondary seal, a backpressure regulator was used in the vent line to the vapor recovery system. This device was designed to control reservoir pressure to 75 psig, as pressurized by accumulation of the primary seal leakage.

The volume of a gas is proportional to absolute pressure. By pressurizing the secondary system to 75 psig, the gaseous volume of the primary seal leakage is reduced to one-sixth the volume at atmospheric pressure conditions.

The potential problems in buffer fluid pressure regulation are few. The primary concerns are the effects on secondary seal performance, possibility of reverse pressurization of the primary seal, and safety concerns for failure detection.

Secondary seal performance is normally enhanced by pressurization of the buffer fluid. When the buffer fluid is maintained at atmospheric pressure, the centrifugal force caused by rotation of the faces and the axial closing force of the seal spring load work against development of a lubricating film at the face. Increasing the buffer pressure assists in developing a lubricating film at the faces and the faces tend to run cooler than they would at atmospheric pressure. Hence, secondary seal performance will be more reliable with pressurization control.

If the buffer fluid is pressurized, at shutdown and depressurization of the pump the primary seal can be subject to reverse pressure. The primary seal design, therefore, must have reverse pressure capability to withstand this condition. The problem can also be avoided by venting the barrier fluid system before the pump case is vented as a standard procedure, but in this case the primary seal was designed for reverse pressure to provide an extra safety margin.

Safety concerns in monitoring the reservoir for failure detection of the primary seal can be addressed by monitoring the pressure increase caused by vapor passing through the buffer fluid pressure regulator. Use of a pressure gage, pressure switch, and orifice provides the same level of detection as in a nonpressurized system.

If the buffer fluid pressure cannot be maintained due to low leakage of the primary seals or a gas leak in the system, the

pressurized buffer fluid system will operate in the same mode as a nonpressurized system. Extended running times in the nonpressurized mode, however, can result in cavitation of the pumping rings and shortened seal life. If buffer level is lost, a level switch will signal the loss. The primary seal will safely operate as a single seal until shutdown.

Other Seal Design Features

The seal drive and spring arrangement was redesigned to a more compact package to provide additional space for the seal improvements while maintaining the same overall cartridge dimensions. An axial pin arrangement was used to drive the stationary face assemblies with hardened pins to prevent galling and hangup. A tungsten carbide type overlay was used on the dynamic gasket sliding surfaces shoulders to prevent fretting type wear.

Face materials chosen were high grade reaction bonded silicon carbide vs antimony impregnated carbon graphite. These materials provide a low coefficient of friction when in direct contact and superior blister resistance in flashing hydrocarbon service [1, 2].

NEW DESIGN VALIDATION TESTING

A number of short term tests were run with teardowns and inspections to validate the seal face design and optimize the flow split and capacity of the pumping ring. Head flow curves for the gas side pumping ring and secondary seal circulation pumping ring at tested speeds are shown in Figures 9 and 10. When these tests were completed and an optimized design was obtained, a 200 hour endurance test of the seals was performed.

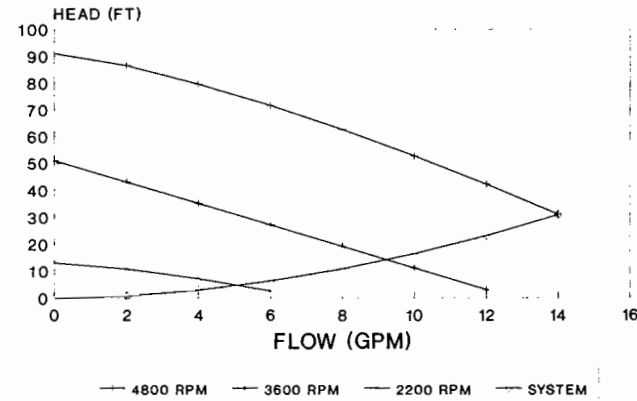


Figure 9. Main Pumping Ring Performance Curve.

The test cycle for the 200 hour test was as shown in Table 4. Other operating parameters were as previously shown in the baseline testing, Table 3.

Primary seal flush flows were varied with speed changes per calculations of actual API Plan 11 flow that will vary with changes in discharge pressure. The test cycle was based on a normal mixture of running speeds anticipated in field service. Start stop cycles at full pressure were conducted at each speed change.

The performance throughout the test was very satisfactory. The leakage performance of the primary seal is shown in Figure 11. No visual signs of distress could be observed in the face profiles. Surface analysis profiles indicated the primary sealing face wore 6.5 lightbands concave, proving stable stationary face pressure and thermal coning.

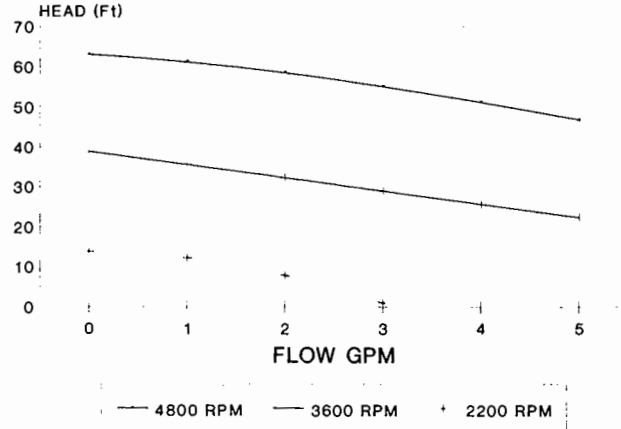


Figure 10. Gas Side Pumping Ring Curve.

Table 4. 200 Hour Test Cycle.

HOUR (TOTAL)	SPEED (RPM)	PRIMARY IN	BUFFER IN	PRIMARY PRESSURE	PRIMARY INJ FLOW
0-15 (15)	4800	100° F	115° F	850 PSIG	15 GPM
15-25 (10)	4800	110° F	125° F	850 PSIG	15 GPM
25-95 (70)	3600	100° F	118° F	850 PSIG	8.5 GPM
95-110 (15)	3600	110° F	128° F	850 PSIG	8.5 GPM
110-180 (70)	3600	100° F	120° F	850 PSIG	8.5 GPM
180-195 (15)	3600	110° F	128° F	850 PSIG	8.5 GPM
195-200 (5)	2000	100° F	120° F	850 PSIG	3.0 GPM

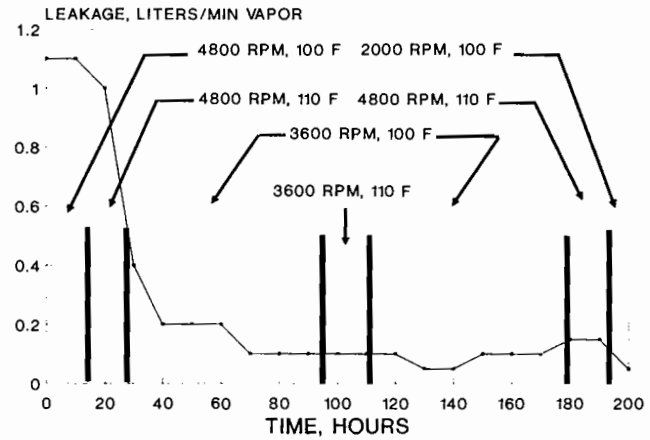


Figure 11. 200 Hour Test Cycle Vapor Leakage.

Several additional tests were run to ensure adequate operating limits for the seal. At a primary seal pressure of 1200 psi and product temperatures of 140°F, the design continued to run well. A failure simulation sequence for complete primary seal failure and full pressurization of the secondary seal cavity by NGL was run with good results.

FIELD PERFORMANCE

The reservoir/heat-exchanger and seal installation took place between June 12, 1993 and June 16, 1993. The installation and subsequent static pressurization were uneventful.

The installation, startup, and initial running period for the seals was similar to test loop performance. During the initial

running period, the pump flow was through the recirculation loop, rather than being put directly onstream. The initial operating parameters are shown in Table 5.

Table 5. Initial Field Operating Parameters.

Speed	4200 RPM
Suction Pressure	700 PSIG
Discharge Pressure	1820 PSIG
S'box Pressure (NDE)	770 PSIG
S'box Pressure (DE)	760 PSIG
Buffer Fluid Temperature (NDE)	130 F
Buffer Fluid Temperature (DE)	135 F
Reservoir Pressure (Both Ends)	75 PSIG

Ten days after the initial start, and after five stoppages for minor nonseal related repairs, the machine was restarted on June 28, 1993. Twenty-four hours later the data as shown in Table 6 were taken. The machine was on line and shipping product.

Table 6. Operating Conditions After Shutdown.

Speed	4100
Suction Pressure	700 PSIG
Discharge Pressure	1800 PSIG
Seal Chamber Pressure (NDE)	790 PSIG
Seal Chamber Pressure (DE)	770 PSIG
Buffer Fluid Temperature (NDE)	128 F
Buffer Fluid Temperature (DE)	135 F
Reservoir Pressure (NDE)	75 PSIG
Reservoir Pressure (DE)	50 PSIG

During the term of more than 8500 total running hours, 225 data sets were taken, a minimum of 24 hr between sets. The data included the barrier fluid temperature for both seals and the pump's shaft speed. The recorded fluid temperature was at the seal flange discharge port, the point of highest temperature in the barrier fluid flow. The statistical range of the data is presented in Table 7. The pumps had several startup and shutdown cycles for normal maintenance with the longest outage being 700 hr. The barrier fluid temperature and the shaft speed data for the hours of operation are shown in Figure 12. Over the 8500 running hours, there were approximately 10 gallons of barrier fluid used.

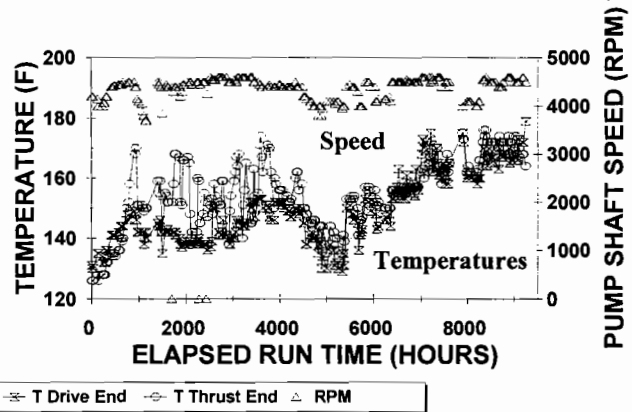


Figure 12. Field Performance Data, Fluid Temperature-Shaft Speed.

Table 7. Range of Buffer Fluid Operating Conditions.

Variable	Drive End Temperature (°F)	Non-Drive End Temperature (°F)	Speed (RPM)
Maximum	175	176	4600
Minimum	129	126	3000
Average	148	154	4332
Standard Deviation	10.8	11.2	305

CONCLUSION

A mechanical seal operating in a flashing fluid such as propane provides a challenging set of conditions. NGL mixtures, particularly with very light hydrocarbon components such as ethane, are even more challenging. In this case, by developing a laboratory test that closely simulated field conditions, areas of a seal design prone to operating problems on NGL mixtures were identified and corrected. The corrections were validated in laboratory testing and have shown to offer significant improvements over the original seal design.

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