DEVELOPMENT AND FIELD TEST RESULTS OF HIGH PERFORMANCE SEALS FOR AN NGL INJECTION PUMP

by

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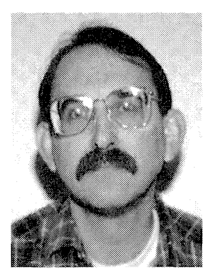

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ABSTRACT

Improvement in the efficiency of enhanced oil recovery techniques at the Kuparak, Alaska, oil field required replacement of a reciprocating pump with a larger capacity centrifugal pump. During the specification of a high speed two-stage centrifugal pump, the mechanical seals and seal support system were identified as key factors for a successful installation. While not all encompassing, the mechanical seal pressure velocity (PV) parameter, does allow the user to gauge the duty level of a mechanical seal. Since low leakage mechanical seals typically operate with a very small fluid film between the stationary and rotating faces, significant friction and heat may be generated when the pressures and rubbing velocities become high. Thus, the higher the PV parameter, the more crucial the physical seal design and material selection become to minimize heat and maintain an acceptable life.

Although the pump and seal manufacturer had successful seal experience up to PV = 2.1 million (psi–ft/min), the chosen pump configuration resulted in a PV = 2.7 million, or 30 percent above the previous experience. Venturing beyond the current state-of-the-art seal, PV parameter necessitated an extensive qualification

program to ensure that pump reliability and process safety management practices were met. To accomplish these objectives, a stringent set of seal qualification requirements were prepared by the contractor and end user to satisfactorily demonstrate: No catastrophic failures during normal and transient operation, A predicted seal life in excess of one year, and a maximum leakage rate of 150 ml/hr.

A total of 600 operating hr were accumulated, including 75 start/stops to prove the robustness of the mechanical seals. Analytical verification, test lab results, and field operating experience are presented. In addition, a method for grading the success of the test program results against the desired design criteria is included.

INTRODUCTION

Ever increasing sealing demands continually challenge users, seal manufacturers and pump manufacturers to extend the mechanical seal's application envelope. Such was the case during the replacement of a reciprocating pump used for injecting NGL into a North Slope enhanced oil recovery field. Kuparak, Alaska, is a maturing, solution-drive oil field producing approximately 290,000 BPD. For enhanced oil recovery, a method has been developed that uses dry gas separated from the production stream and enriches it with the heavier ends produced on and offsite as a miscible injectant into the reservoir. The required gas-to-liquid blending requires an injection pressure of approximately 4000 psi or nearly 14,000 ft of head at a flowrate of 400 gpm, which exceeded the capacity of the existing reciprocating pump. Conversion of the reciprocating pump would have been impractical. Delivery by truck and onsite module size constraints tended to favor smaller, high energy equipment. The low flow high pressure process requirements, physical size criteria, and high cost of an alternate diaphragm pump, led to a high speed two-stage centrifugal pump being selected.

While the overall pump design conditions were within previous operating experience, the mechanical seal performance requirements would present the most demanding seal operating parameters yet attempted. Due to the combined pump and seal configuration resulting in a seal PV of 2.7 million (psi–ft/min), reliability and safety concerns mandated that the seal and seal system undergo a rigorous qualification test program prior to field installation. Previous seal experience by the pump manufacturer and seal manufacturer had been limited to a PV level of approximately 2.1 million (psi–ft/min).

It must be cautioned that the PV parameter is not the only criteria to measure the duty level of a mechanical seal. Leakage, life, and stability under adverse operating conditions must also be taken into account to establish a successful application. The PV value, however, does allow one to draw useful comparisons from prior experience for a particular seal design and set of materials. Recognizing these limitations, the seal PV parameter is a valuable tool. Lebeck [1] provides further detail and typical PV limits for a variety of examples.

The seal test program developed that enabled the PV parameter to be extended, was conducted on propane and diesel due to their similarities to the actual process fluids, and validated:

- No catastrophic failures during normal and transient operation.
- A predicted seal life in excess of one year.
- A leakage rate of less than 150 ml/hr.

The owner had recognized that this application was outside the API 682 scope, and considered the above constraints to be acceptable. In addition to reviewing the seal design and development test program, a unique and specific seal grading system methodology is proposed for assisting other users considering areas of unproven experience.

EVOLUTION OF DESIGN

The chosen centrifugal pump, shown in Figures 1 and 2, utilizes a single high speed pinion shaft with impellers on each end. The two stages are piped in series to develop the necessary pump head rise. Since the first stage impeller discharge feeds the second stage impeller, a very high second stage seal chamber pressure is created. The pump operating conditions, shown in Table 1, result in a Stage 2 seal chamber pressure of 1500 psi, which has been reduced by the use of seal rotor, depicted in Figure 2.

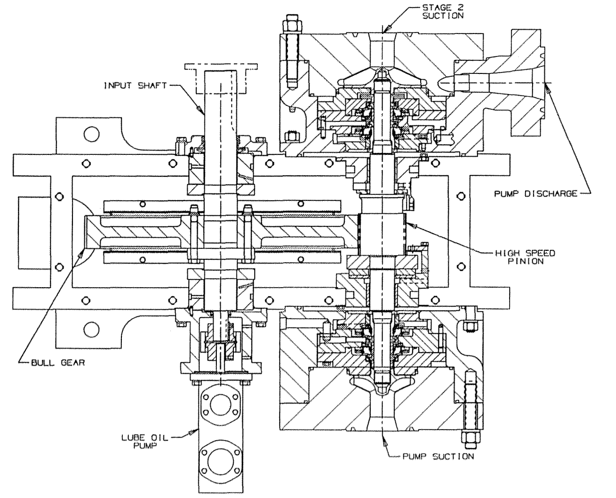

Figure 1. High Speed Two-Stage NGL Injection Pump—Cross Section.

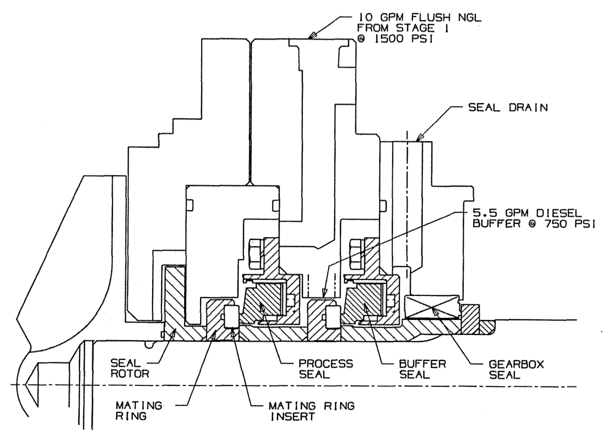

Figure 2. Dual Seal Orientation

Table 1. NGL Injection Pump Operating Conditions.

	STAGE 1	STAGE 2
Fluid	NGL	NGL
Specific Gravity	.586	.586
Flowrate, gpm	220	220
Total Pump Head Rise, ft	--	13,961
Temperature, F	47	47
Suction Pressure, psig	497	497
Discharge Pressure, psig	1900	4039
Speed, rpm	21,230	21,230
Shaft Diameter at Seal, in	2.00	2.00
Total Pump Horsepower, BHP	--	1095

Due to the explosive nature of the NGL process, it was imperative not to have leakage to the atmosphere. Therefore, a tandem oriented dual seal arrangement with a buffer fluid between the primary and secondary seals was mandated. Furthermore, a commonly used seal system technique that "splits" the total seal pressure differential between the seals with a high pressure buffer fluid supplied by a separate buffer pump was employed. By providing the buffer fluid at a predetermined pressure, the individual seal differentials can be better controlled to manageable level along with buffering the process fluid (NGL) from the atmosphere. An Arctic diesel fuel was chosen as the buffer fluid due to its availability and ease of handling at the local site. The seals and seal system with the external buffer pump are shown in Figures 3 and 4.

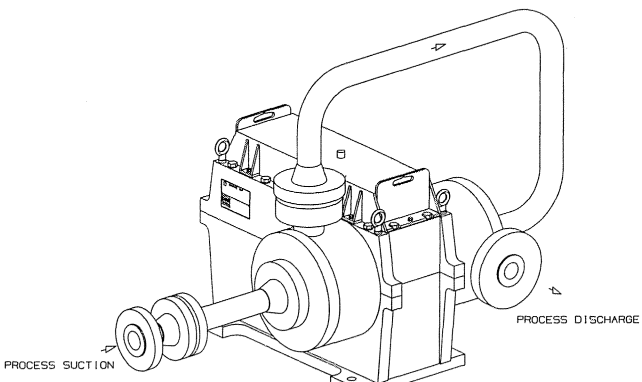

Figure 3. High Speed Two-Stage NGL Injection Pump–Orthographic.

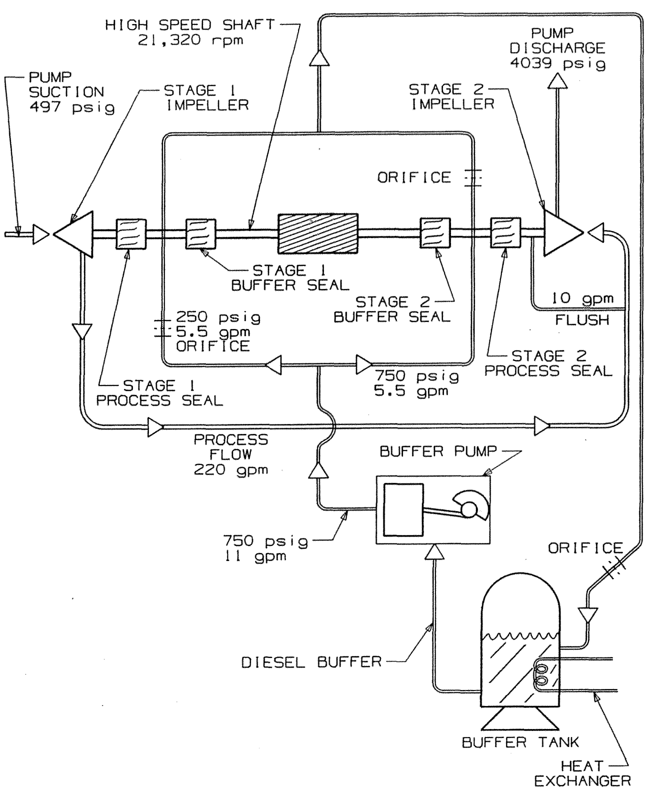

Figure 4. Pump and Seal System Schematic.

Although controlling the individual seal pressure differential helped stabilize the sealing operating environment, the high surface speed and sealing pressure caused several design and material iterations during the test program. The PV levels cited in Table 2 are based upon the seal face contact pressure rather than the seal differential pressure [1], and calculated from Equation (1).

Table 2. Seal Operating Conditions.

	STAGE 1	STAGE 2
Seal Chamber Pressure, psi	500	1500
Buffer Pressure, psig	250	750
Shaft Speed, rpm	21,230	21,230
PROCESS SEAL		
Sealed Fluid	NGL	NGL
Pressure Differential, psi	250	750
Seal Balance Ratio	.76	.76
Seal Mean Diameter, in	2.24	2.24
Surface Velocity, ft/sec	12,450	12,450
Process Seal PV, psi-ft/min	1.12 x 10E6	2.74 x 10E6
BUFFER SEAL		
Sealed Fluid	Diesel	Diesel
Pressure Differential, psi	250	750
Seal Balance Ratio	.76	.76
Seal Mean Diameter, in	2.24	2.24
Surface Velocity, ft/sec	12,450	12,450
Buffer Seal PV, psi-ft/min	1.12 x 10E6	2.74 x 10E6

$$PV = P_c \times V_m \qquad (1)$$

where:

P_c = Contact pressure = p (b−k) + p_s
P_c = 750 psi (0.76 − 0.5) + 25 psi = 220 psi
V_m = Mean seal face velocity = Pi × D_m × N / 12
V_m = Pi × 2.24 in × 21,230 rpm / 12 = 12,450 fpm
PV = 220 × 12,450 = 2.74 × 10E6 psi−ft/min

Previous seal experience by the pump manufacturer and seal manufacturer had been limited to a PV level of approximately 2.1 million (psi−ft/min). Therefore, the 30 percent increase or extrapolation in the PV parameter would require unique solutions to achieve the necessary performance. Since the pressures were most extreme on Stage 2 with respect to Stage 1, virtually all testing and analytical work focused on these seals. Since previous successful operating experience at slightly lower PV values with a carbon stationary and tungsten carbide rotating face had been demonstrated, these configurations and materials provided the starting point. As one can imagine with a project of this magnitude, several difficulties were identified during the test program and had to be overcome. Only the major variables changed during the test program are described.

Stationary Face

• *Pressure Balance*–Seal pressure balance or balance ratio is a commonly used term that reflects the ratio of the net hydraulically loaded area to the seal face area. Generally, the higher the pressure balance ratio, the higher the contact forces, the lower the leakage, and unfortunately, the shorter the seal life. The starting point in this test program was, of course, based upon prior experience. This led to a pressure balance of approximately 76 percent and a face geometry utilizing hydropads being selected (Figure 5). Hydropads act as small scoops to assist in drawing in fluid for face lubrication. In an attempt to minimize seal wear and increase life, the pressure balance was decreased to 73 percent, but seal face separation occurred at low pressures. This design was abandoned, and the pressure balance was left at 76 percent for both the process (NGL) and buffer (diesel) seals.

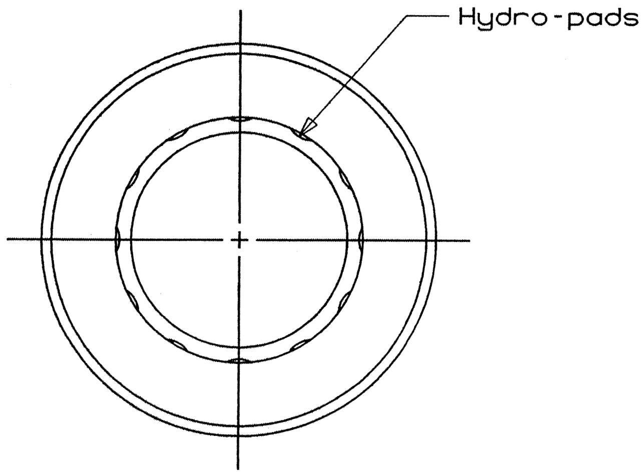

Figure 5. Stationary Face With Hydropads.

• *Material*–The diesel buffer fluid generates more heat than the process NGL fluid due to its higher viscosity, and necessitated a change in material to accommodate the higher temperatures recorded. Temperatures were reduced and seal life improved by upgrading the diesel buffer seal face material from carbon to silicon carbide with 15 percent graphite to assist in removing the additional heat.

• *Cooling Holes*–It was also apparent that excessive heat was causing premature wear on the carbon process (NGL) seal, and the ability to provide a cool flush to the faces was hindered by the high rotational speed. With an OD pressurized seal, the sealed fluid must overcome the rotational velocity to provide circulation at the seal face. To improve seal face cooling, a unique (patent pending) set of axial cooling holes were drilled into the stationary face to take advantage of the mating ring pumping action. This enhancement, shown in Figure 6, significantly reduced the seal face heating as evidenced by the reduced buffer fluid temperature rise and increased life to meet the design objectives.

Rotating Face or Mating Ring

Although past experience utilized tungsten carbide as the rotating face, the material was changed to silicon carbide to minimize the heat generation at the seal faces. To account for the brittle nature of silicon carbide, several iterations on the mating ring carrier were necessary to avoid breakage. The final configuration utilized a metal band shrunk directly on the silicon carbide to reduce the rotational induced stresses. Three flats (Figure 7) on the band outside diameter were used between the mating ring insert and carrier for antirotation throughout the final qualification tests. For strength, the carrier material was 17–4PH stainless steel.

Buffer Pressure Split

Several early tests evaluated the buffer pressure split to determine if the process seal pressure differential could be reduced.

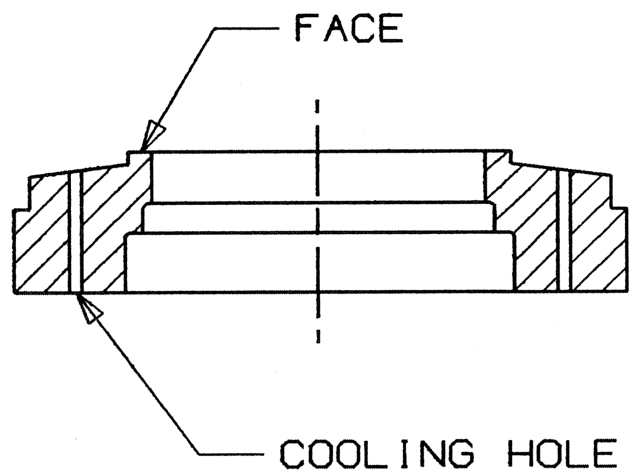

Figure 6. Stationary Face With Cooling Holes.

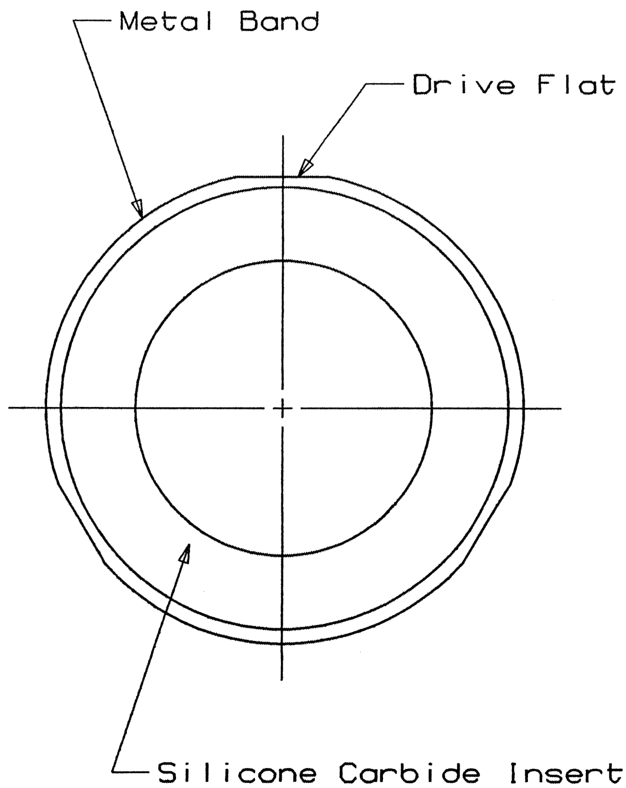

Figure 7. Rotating Face Drive Flat Mechanism.

Successful steady state operation up to 850 psi was achieved, but transient testing revealed problems such as overheating and pullout of the carbon. Ultimately, the buffer pressure split was limited to 50/50, i.e., both the diesel buffer and NGL process seal pressure differential was 750 psi.

ANALYTICAL VERIFICATION

Stationary Face

A full 3D finite element stress analysis (FEA) model with pressure at the OD of the seal, which places the stationary face into compression, was developed. Temperature gradients were obtained from the seal analysis program and verified on test. As seen in Table 3, the maximum stresses for both seals result in a minimum safety factor of 3.4. A typical stress distribution in the stationary face from the FEA is shown in Figure 8.

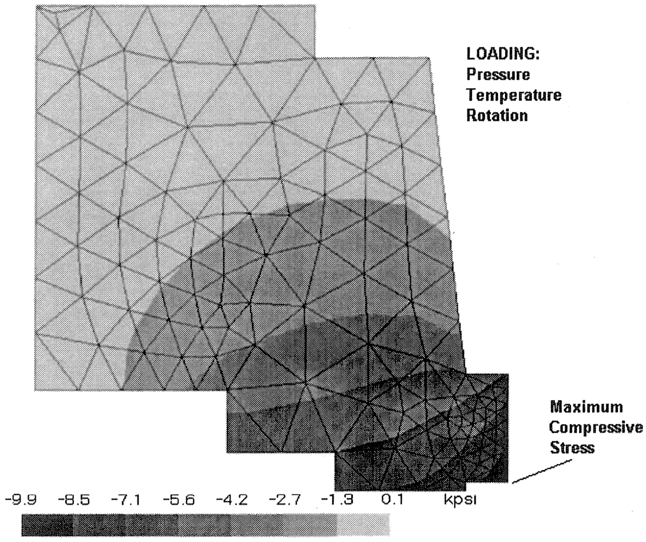

Figure 8. Stationary Face Stress Profile.

Table 3. Stationary Face Stress Analysis.

	STAGE 1		STAGE 2	
	Buffer Seal	Process Seal	Buffer Seal	Process Seal
Pressure, psi	250	250	750	750
Material	Silicon w/ Graphite	Carbon	Silicon w/ Graphite	Carbon
Maximum Compressive Stress, psi	3000	3000	10000	10000
Compressive Strength, psi	94000	34000	94000	34000
Safety Factor	31.3	11.3	9.4	3.4

Rotating Face or Mating Ring

By shrinking a metal band to the silicon carbide rotating face, stresses induced by rotation were significantly reduced. A material was chosen with a low thermal expansion coefficient nearly identical to the silicon carbide. This matching of the thermal expansion coefficients simplified and provided a high degree of integrity of the shrink fit process. Shown in Figure 9 is the mating ring stress profile from the FEA after the installation of the metal band which reduced the mating ring hoop stress approximately 25 percent. Reduction of the hoop stress was an important improvement due to the brittle nature of silicon carbide in tension. All cracking of the silicon carbide mating ring experienced during the initial test phase was eliminated with the metal band.

Process Seal Cooling Holes

The addition of the unique cooling holes in the process seal provided the added robustness needed to reduce wear and improve life. Although complicated to model, a simplified computational fluid dynamics (CFD) flow model was generated to estimate the cooling flowrate. Buffer fluid temperature measurements of the seal face heat input confirmed the CFD model and estimated flowrate through the cooling holes reasonably well.

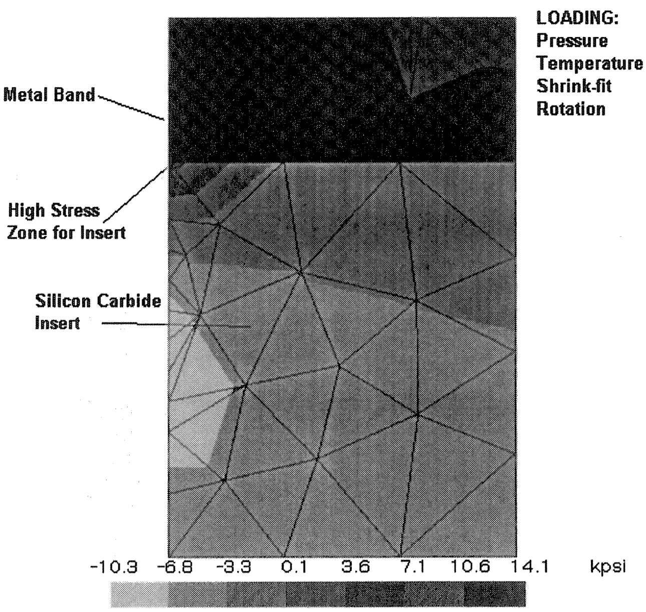

Figure 9. Banded Mating Ring Stress Profile.

DEVELOPMENT TESTING

Twenty-eight tests were conducted between the seal and pump manufacturer's facilities to establish: the avoidance of catastrophic failure, seal life, and seal leakage. Both companies used dedicated seal test rigs that were coupled to a speed increasing gearbox to obtain the necessary operating speed of 21,230 rpm. The test plan was modelled after the API 682 qualification procedure and involved a series of steady state dynamic and upset dynamic conditions. Typical pressure and temperature duty cycles are shown in Figures 10 and 11.

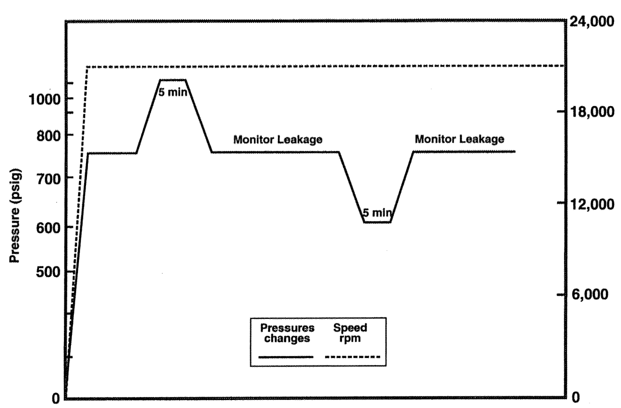

Figure 10. Typical Testing Duty Cycle—Pressure.

Avoidance of Catastrophic Failure

As detailed in the EVOLUTION and ANALYTICAL VERIFICATION sections, several iterations on the seal face materials and mating ring insert containment methods were necessary to get past the infant mortality stage. After many successful start/stop transients, testing preceded to quantify seal life and seal leakage.

Seal Life

Undoubtedly, determining the long term seal life proved to be the most challenging. Initially, mechanical seals have a high rate of

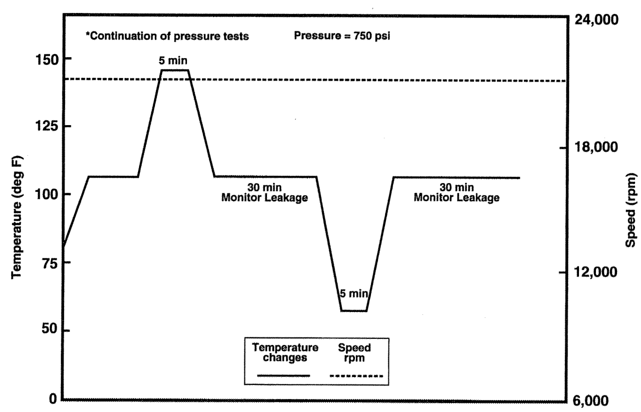

Figure 11. Typical Testing Duty Cycle—Temperature.

wear as the faces align themselves due to pressure and temperature. To reliably establish the long term wear rate, the test program had to ensure that the seal had passed through the high wear rate break-in period and reached a steady state condition. Accurately determining the amount of material removed also required development of consistent measurement techniques, since, typically, the amount of material removed from the seal face had to be resolved to 50 millionths of an inch. By plotting the seal nose height wear at intervals of 0, 25, 50, and 100 hr, it was discovered that at least 50 hr of operation were necessary to reach the steady state material removal rate. Long term life was estimated by substituting the amount of material removed (MR) between 50 and 100 hr and the available face material for wear into Equation (2).

$$\text{Projected Life} = [1/(\text{MR})] \times \text{Test Time} \times \text{Face Height} \quad (2)$$

Most of the difficulties encountered during the life testing were related to the measurement accuracy of the amount of material removed from the seal face. Small variations in the seal height change could dramatically affect the long term seal life calculation. In fact, several tests indicated that the seal face actually grew or swelled under propane operation. As shown in Table 4, the expected seal life for the buffer and process seals far exceeded the minimum one year life requirement. This was due in part to the generous leakage allowance contributing to a large film between the seal faces, and not allowing the faces to rub causing excessive wear.

Table 4. Projected Seal Life.

	STAGE 1			STAGE 2
	Buffer Seal	Process Seal	Buffer Seal	Process Seal
Pressure, psi	250	250	750	750
Material	Silicon w/ Graphite *vs* Silicon Carbide	Carbon *vs* Silicon Carbide	Silicon w/ Graphite *vs* Silicon Carbide	Carbon *vs* Silicon Carbide
Life, years	Infinite - No Wear	25.5	Infinite - No Wear	12.5

Seal Leakage

Seal leakage measurements proved to be more straightforward than the life estimate measurements. The diesel fuel was simply collected over the duration of the test. Propane leakage was collected and measured by a total hydrocarbon analyzer and converted to mass flow by using formulas provided in the literature [2], or bubbling the propane gas into a holding container and converting the displaced volume into a mass flow. Either method proved viable once the techniques were refined.

Recognizing that adequate life was more important than leakage, and a self contained buffer system would be employed, the 150 ml/hr leakage rate was easily achieved for both the process and buffer seals.

SEAL GRADING METHODOLOGY

Due to the highly loaded nature of the Stage 2 process and buffer seals, and limited history on this design, the user required a method be developed to prove that long term seal performance would be successful. This pump was also in an OSHA 1910 Process Safety Management affected area, which stipulated that the mechanical integrity be fully established, and formed the methodology of determining the acceptance of the pump and seal system. Recommended seal maintenance intervals were also to be determined.

Reiterating, the pump and seal system had to demonstrate: 1) Avoidance of catastrophic failure, 2) Sufficient life, and 3) Low leakage. The first item was clearly a go/no go decision: The seal and seal system had to be designed to tolerate all of the fluids, temperatures, velocities, and loads imposed upon it by the pump and process. Any remaining risk of catastrophic failure was addressed through normal process hazard analysis techniques to determine the supporting pump skid design features. The second and third items, however, had to be empirically based, which led to a grading system that quantified the acceptability of the seal.

By "splitting" the process and buffer seal with a buffer fluid, each seal can be independently evaluated, i.e., the process seal had to perform reliably on the NGL process fluid, and the buffer seal had to fulfill its duty on the diesel fluid.

A matrix was developed and agreed upon up front by all parties. As shown in Table 5, a grading scheme based on "A" through "F" would ultimately determine the success of the new seals. The grading scheme helped to place some objectiveness to the nature of the test program being undertaken. For example, a projected seal life less than one year was rated an "F." Leakage rates were similarly graded (Table 6).

Table 5. Stage 2 Process Seal Grading.

Seal Design: Narrow Face Carbon w/Holes -76 Percent Balance

By: S. Murphy Date: 1/26/95

Item	Grade	Based on C-Stedy, Test, Judgment	Comments
1. Has seal experienced a major failure? Yes = F. No = A.	A	Test	
2. Does analysis/test indicate "Peak" limits exceeded? Yes = F. No = A.	A	Test	
3. Does analysis/test indicate "Maximum Average" limits exceeded? Yes = F. No = A.	A	Test	
4. Grade the seal for run-in leakage.	A	Test	
5. Grade the seal for pressure transient leakage.	A	Test	
6. Grade the seal for temperature transient leakage.	A	Test	
7. Grade the seal for any other relevant upset leakage.	A	Test	
8. Grade the seal for normal leakage/seal life.	B	Test	
9. If an outer seal, grade the seal for an inner seal failed condition.	NA	NA	
10. If an inner seal, grade the seal for an outer seal failed condition.	A		
11. Final overall seal utility grade equals the poorest grade of items 1 through 10 above.	B		

Final grading results for the buffer seals on diesel and process seals on propane are presented in Table 7. The buffer seal results were very positive. However, the process seal outcome was less

Table 6. Stage 2 Buffer Seal Grading.

Seal Design: D96 - No Holes

By: S. Murphy Date: 1/26/96

Item	Grade	Based on C-Stedy, Test, Judgment	Comments
1. Has seal experienced a major failure? Yes = F. No = A.	A	Test	
2. Does analysis/test indicate "Peak" limits exceeded? Yes = F. No = A.	A	Test	
3. Does analysis/test indicate "Maximum Average" limits exceeded? Yes = F. No = A.	A	Test	
4. Grade the seal for run-in leakage.	A	Test	
5. Grade the seal for pressure transient leakage.	A	Test	
6. Grade the seal for temperature transient leakage.	A	Test	
7. Grade the seal for any other relevant upset leakage.	A	Test	
8. Grade the seal for normal leakage/seal life.	A	Test	
9. If an outer seal, grade the seal for an inner seal failed condition.	A	Test	
10. If an inner seal, grade the seal for an outer seal failed condition.	NA	NA	
11. Final overall seal utility grade equals the poorest grade of items 1 through 10 above.	A		

conclusive using the established grading criteria. Due to the timing of events, it was agreed upon to install the seals in the pump and start up the oil recovery production stream on a conditional and closely monitored basis. Final conclusions would be evaluated in the field.

Table 7. Final Seal Grading.

	Seal Overall Utility Grade			
	First Stage (250 psid)		Second Stage (750 psid)	
Seal	Inner	Outer	Inner	Outer
Process	A		B	
Buffer		A		A

FIELD RESULTS

Like many projects that push boundaries, both satisfactory and unsatisfactory results were seen under actual field operating conditions. At the first pump/seal disassembly, both the process and buffer seal faces were generally acceptable as there were no signs of overheating. However, the problems encountered were centered on:

• *Stage 2 Process Stationary Face–Cracked carbon.* The failure mode suggested a pressure reversal as the carbon was broken in a shear or transverse condition, indicative of a brittle material failure in tension. Also, the crack boundaries had not worn, suggesting the failure occurred just prior to pump shutdown. This failure mode shape was confirmed by conducting a reverse pressure test on a carbon to fracture. The pressure at which the failure occurred also agreed with the finite element analysis predictions. Design changes included better control of the buffer system pressure during pump shutdown.

• *Stage 1 and 2 Buffer Seals–Excessive drive lug wear.* To accommodate the higher heat generated in the diesel buffer seals, a silicon carbide carbon graphite material had been utilized that has a much higher hardness than the 316 stainless steel seal retainer. The high hardness differential appeared to have caused the wear on the antirotation drive lugs. Redesign entailed eliminating the drive lugs and using a single hardened pin as the drive mechanism. The before and after drive methods are shown in Figure 12.

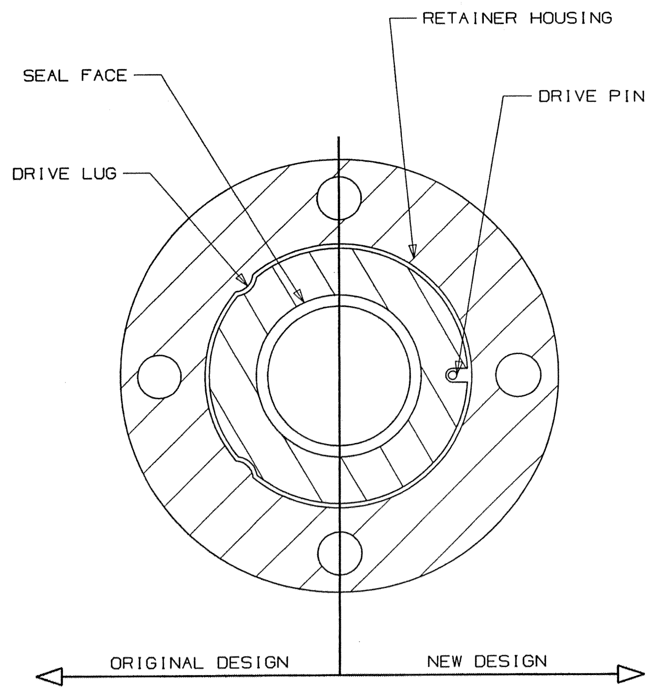

Figure 12. Stationary Face Drive Method.

• *Stage 1 Buffer Seal–Cracked mating ring.* The pump and seal accumulated approximately 1200 steady state hr with multiple starts/stops before this particular disassembly and, surprisingly, one of the least loaded seals had a cracked silicon carbide hard face. Review of the nonpressure side of the mating ring suggested some motion was present. Further analysis determined that the O-ring on the back side of the mating ring was not completely compressing under load. This allowed relative motion on the mating ring and its carrier.

Corrections of these anomalies led to another successful run of over 3600 hr without any failures.

CONCLUSIONS

With the strategic use of analysis, testing, and an objective rating method, a reliable seal has been developed that achieved an operating PV level of 2.7 million (psi–ft/min) that was 30 percent above previously known levels. A unique seal face cooling hole geometry was discovered to provide the added robustness necessary for ultimate success, and has proven to enhance seal performance on other less rigorous applications. Although development testing demonstrated that the design input criteria: Resistance to catastrophic failure, life in excess of one year, and leakage less than 150 ml/hr were achieved, it should not be forgotten that there always remain further unknowns in the field. However, it is believed that the rigorous qualification test program lead to a more reliable and robust design than past practices.

For any company to succeed with new developments, some risk is involved to take advantage of new products. Apart from the difficult technical problems presented by this application, an organized approach had to be maintained by all parties to keep focused on the overall objective of providing a safe reliable seal and seal system. With considerable cooperation and constant vigilance, these objectives have been met.

NOMENCLATURE

b	Balance ratio (in decimal)
D_m	Mean seal face diameter
k	Pressure gradient factor
MR	Material removed during time interval
N	Rotating speed, rpm
p	Seal pressure differential
Pi	3.14159
p_s	Spring pressure on seal face

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2. *Handbook of Environmental Data on Organic Chemicals*, Van Nostrand (1980).

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