

MECHANICAL SEALS WITH WAVY SIC FACES FOR A SEVERE DUTY NGL/CRUDE PIPELINE APPLICATION

by

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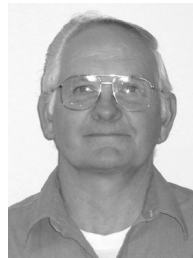
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ABSTRACT

Mechanical seals were developed for a pipeline with demanding operating conditions. Natural gas liquids (NGL), refined products, and crude oil are pumped in batch mode. Seal pressure varies over a wide range and the pumps are subjected to start/stop operation. The presence of abrasive pipe rouge (iron fines) is a further complication. Laboratory tests and field verification demonstrate that mechanical seals employing silicon carbide versus silicon carbide wavy faces can handle the severe service conditions. Successful seals have been running in the field since September 2000.

INTRODUCTION

Enbridge Pipelines Inc. operates a large system of cross-country liquid hydrocarbon pipelines. The pipelines generally originate in Western Canada, in Edmonton, Alberta. They deliver crude, synthetic crude, NGL, and refined products to customers in Canada and the United States. The pipelines are segregated primarily by the viscosity, density, and sulfur levels of the pumped liquids. The "clean" line, designated as Line 1, is dedicated to the transport of NGL, refined products, and synthetic crude in batch mode (Figure 1). Small amounts of mixing or "interface" are created between the batches. Most of the time, however, the fluids flowing through a pump have the full properties of one of these fluids.



Figure 1. Line 1 Route Map. (Not all Pump Stations Shown).

Pumping Stations

Flow and pressure in Line 1 are maintained by a series of regularly spaced pumping stations approximately 50 to 65 kilometers (30 to 40 miles) apart. Each pumping station consists of three to four double suction double volute horizontally split centrifugal pumps (API 610 BB1 designation). One pump is shown in Figure 2. The pumps are arranged in a series configuration such that the discharge of one pump becomes the suction of the next downstream pump (Figure 3). The pumps are also piped in parallel with a check valve such that nonoperating pumps are bypassed by the main flow. A pipeline operator located in a control center in Edmonton, Alberta, originates control—the starting and stopping of the pumps. Thus the suction pressure in any given pump is dependent on the position of the pump in the station (e.g., first, second, third, or fourth), the number of pumps running upstream, and the incoming station pressure.

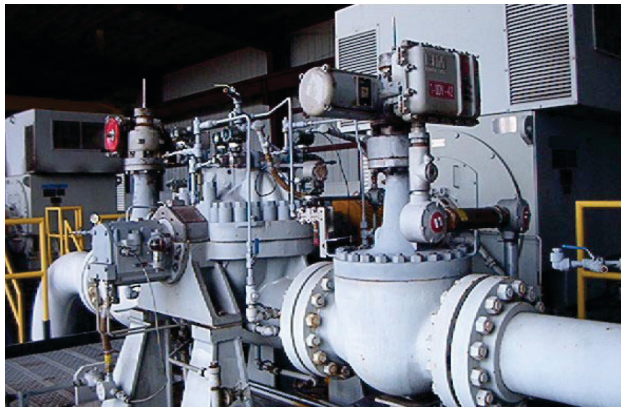


Figure 2. Pipeline Pump.

The pumps are driven by 3600 rpm electric motors. Some of the stations have variable frequency drives (VFD), which can vary the speed of any one pump from 1800 to 3960 rpm. A pressure control valve (PCV) is also available for backup pressure control in case the VFD is unavailable and for rapid pressure fluctuations (the VFD response speed is much slower than a PCV). Pipeline control is by a supervisory control and data acquisition (SCADA) system maintaining pressure set point at each of the pump stations. Control center operators input required pressures for each of the stations to achieve a desired flow rate. While starting and stopping of pumps is operator initiated, ramping of the VFD speed and stroking of the PCV are controlled automatically at the local station level.

Pumped Products

The pumped commodities vary considerably in fluid properties (Table 1). NGL is a gas at atmospheric conditions and must be kept under pressure to prevent vaporization and bubble formation in the

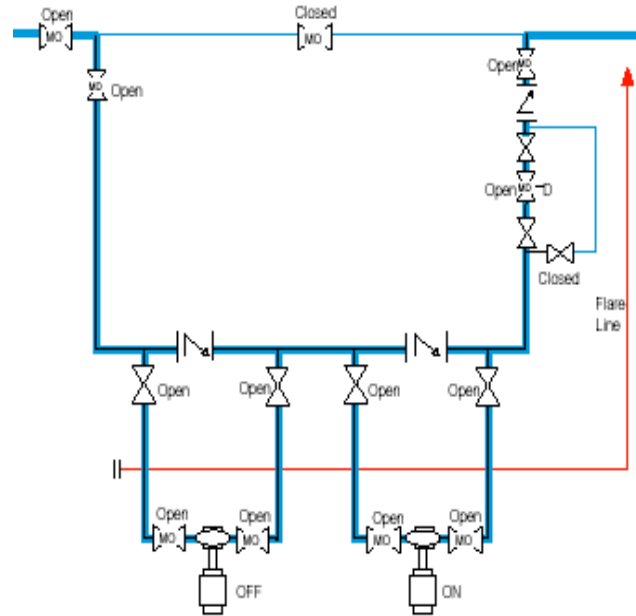


Figure 3. Series Station Schematic Configuration.

pipeline. Minimum station inlet pressures ensure adequate suction pressure is available to the first operating pump. NGL has a very low viscosity and, hence, low lubricity. Conventional mechanical seal selection suggests the use of carbon for one of the seal faces. The heaviest product is synthetic crude with a density 57 percent higher than NGL, and a viscosity, while still low, 20 times that of NGL. The density and viscosity changes when pumping alternately light and heavy material create pressure spikes and dips in the suction and discharge pressures seen in operating pumps (Figure 4). These pressure fluctuations create a difficult operating environment for the mechanical seals.

Table 1. Typical Fluid Properties.

Fluid Type	Density (kg/m ³) @ 15° C	Viscosity (cSt) @ 10° C	Reid Vapor Press. (kPa)
Synthetic Crude	865	7	35
Refined Prod.	700 - 850	1 - 10	Up to 107
NGL	550	0.35	1125

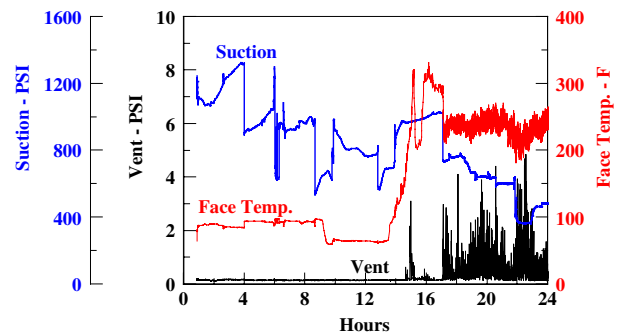


Figure 4. Field Performance—Original Seal. (Large pressure spikes are due to prior pumps being turned on or off.)

In addition to the typical electrical and vibration unit protection, the pumps are also monitored for seal leakage. Small quantities of leakage past the primary seal flow through a 3 mm (0.125 inch) orifice to a sump tank system. On excessive leakage, the pressure

in the cavity behind the primary seal will increase and the station programmable logic controller (PLC) will signal an alarm. On higher leak rates the PLC will shut the unit down and close all valves, completely isolating the pump. Separate pressure transmitters on the drive-end and nondrive-end seals are monitored continuously; excessive leakage of either seal will initiate isolation of the pump. A dry running, noncontacting secondary containment seal was chosen to provide greater sealing capability in the case of primary seal failure than a simple “disaster bushing” or throttle bushing. The dry running secondary seal provides two functions:

- Create a seal at the shaft so that the chamber pressure will increase to detect small leaks
- Run continuously under high pressure until the pump shuts down if the primary seal fails catastrophically

A dry running seal is also considerably less expensive than other dual seal systems since it requires no external buffer fluid system. A noncontacting design is used and will have virtually no wear.

Expansion Project

A major expansion project in 1997-1998 replaced all existing pumps on the pipeline with higher head and flow units. Several additional pumps were also installed, for a total of 74 new pumps in the existing stations. Two mechanical seals were installed in each between-bearings pump, for a total of 148 seals. The new design seals were successfully field tested for about six months in existing pumps prior to their acceptance for the major expansion project. Once installed in the new pumps, however, reports began surfacing of premature failure of the seals. As these reports increased, a systemic problem was suspected. Sporadic failures of some seals at some stations and not others made diagnosis difficult. Some seals failed within a few hundred hours of operation, while others ran for thousands of hours. Sometimes only the drive-end seal would fail regularly, and in other cases only the nondrive-end seal experienced multiple failures. It was evident that the expertise of the seal vendor would be required in determining the cause of the problem and solving the problem of short operating life.

ORIGINAL SEAL

The original dual seal arrangement consisting of a primary seal that handles full system pressure drop, and a dry running secondary containment seal, is shown in Figure 5. The flexible face of the 4.5 inch primary seal is nonrotating to facilitate tracking of the rotating mating face. Primary seal face materials are antimony impregnated carbon-graphite running against reaction bonded silicon carbide (SiC). Both primary seal faces (for the original seal) are flat, with no surface features. Face cooling is provided by an API Plan 11 flush through a multipoint distribution ring as shown in Figure 5. The seal is designed to handle NGL, refined products, and crude oil at pressures up to 1500 psig.

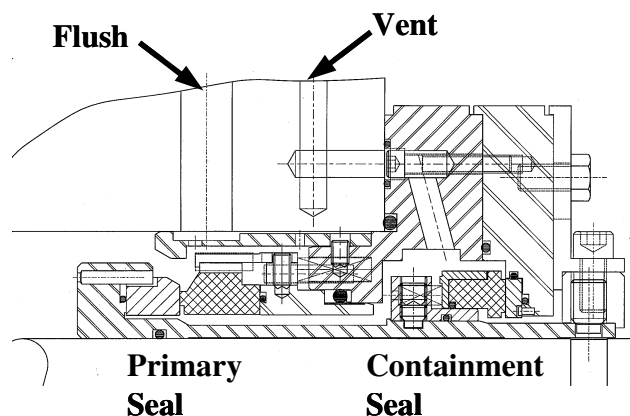


Figure 5. Dual Seal for Pipeline Service (Original Design).

Containment Seal

The containment seal employs a wavy face to provide noncontact operation in a gas or liquid environment at pressures from zero to greater than 600 psig (Young, et al., 1996). Face materials for the containment seal are resin impregnated carbon-graphite versus SiC. The wavy feature is bidirectional and is less likely to clog than designs that employ some form of grooves in the face. Also, because of the smooth wavy shape, the seal can withstand incidental contact without damage to the carbon face.

Leakage past the primary seal is routed through a 0.125 inch orifice to drain into a sump. Pressure on the containment seal (vent pressure on the upstream side of the orifice) is monitored to detect excessive primary seal leakage.

This tandem seal arrangement with a dry running secondary seal is easy to maintain as it does not require a liquid buffer fluid support system.

Seal Face Coning Deflection

Seal faces distort (cone) under mechanical and thermal loads (Lebeck, 1991). Net coning distortion must be less than a few helium light bands (HLBs) to assure long seal life and low leak rate. (One HLB is approximately 10 μin , or about 1/400th thickness of a sheet of paper.) It is imperative to not run with heavily loaded outer diameter (OD) contact that pinches-off the sealed fluid—particularly volatile liquids such as NGL. This mode of operation can result in intense face heating and “flashing” as the faces rapidly separate and contact, leading to face damage. On the other hand, the faces should not have a large OD gap as this results in a thick fluid film and high leakage.

Finite element analysis (FEA) is used to design seal faces to minimize coning deflections (Salant and Key, 1984).

Heat generation causes the faces to cone open to the OD fluid. This effect is due to the larger radial *thermal* growth of the hot sealing surface compared to the cooler back end of a face. Figure 6 shows FEA predicted thermal coning distortion (greatly magnified) of the primary faces when sealing propane at 1100 psig, 3600 rpm, and 80°F. Note the small OD gap.

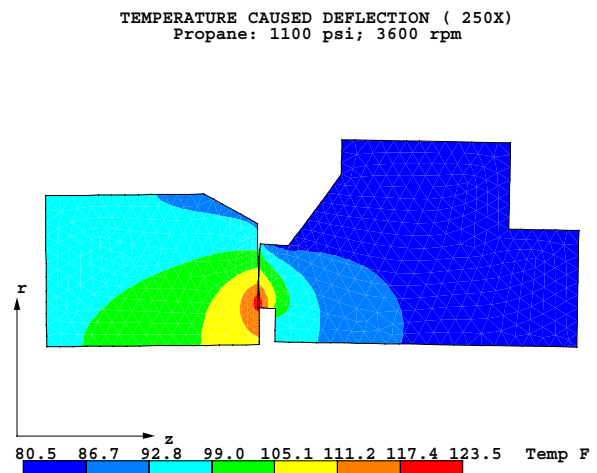


Figure 6. Thermal Deflection (Faces Cone Open at OD).

The faces were designed so that pressure acts to cone the faces toward OD contact in an attempt to cancel out thermal coning. Figure 7 shows predicted *pressure* distortion results in a slightly divergent taper of the fluid film in the direction of leakage from OD to inner diameter (ID).

Net pressure and thermal coning deflection are computed to be 20 μin (open at OD) for this seal on propane at 1100 psi, 3600 rpm. Computed film thickness is 10 μin at face ID and 30 μin at OD. The nearly flat running faces from the FEA model are shown in Figure 8. The faces are predicted to run with lightly loaded asperity

PRESSURE AND CONCENTRATED LOAD CAUSED DEFLECTION (250X)
Propane: 1100 psi; 3600 rpm

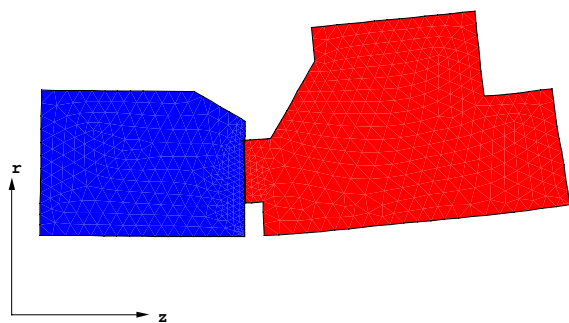


Figure 7. Pressure Deflection (Faces Cone Toward OD Contact).

contact. Average face temperature is calculated to be about 30°F hotter than the seal chamber. Lab testing on propane (refer to the section, *Lab Testing on Propane*) verified flat running faces with low wear rates and low temperature rise.

TOTAL DEFLECTION WITH FILM THICKNESS (250X)
Propane: 1100 psi; 3600 rpm

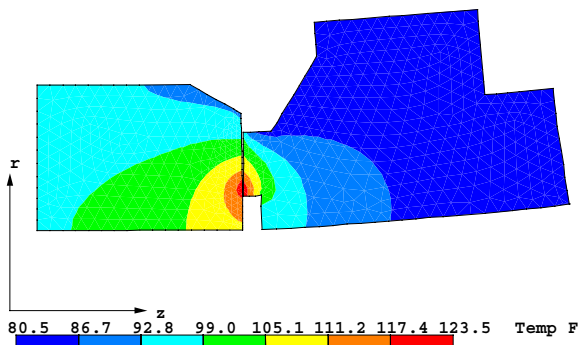


Figure 8. Net Deflection on Propane.

Viscous shear heating is significantly higher for an oil film compared to running on NGL. Figure 9 contains model results for operation on a 10 cSt oil. The model predicts that temperature rise of the sealing interface is 110°F. The large thermal induced radial taper results in a moderately thick fluid film: 28 μin at face ID and 153 μin at OD.

TOTAL DEFLECTION WITH FILM THICKNESS (250X)
10 cSt Oil: 1100 psi; 3600 rpm

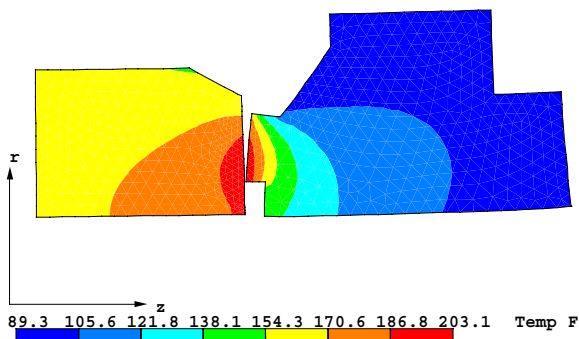


Figure 9. Net Deflection on Oil.

Face—Seat Interaction

Interaction between a face and its seat contributes to coning deflection (Metcalf, 1980). Axial and radial force components on the back of the face produce tipping moments (Figure 10). A traction force arises from radial slip due to relative contraction (or growth) of the face and seat under pressure and thermal loads. Note that carbon stiffness is about 1/8th that for stainless steel. OD pressurization causes a carbon face to be compressed more than the seat—resulting in an outward force on the backside of the face (Figure 10). Traction force magnitude is proportional to the axial contact force and coefficient of friction between the face and seat.

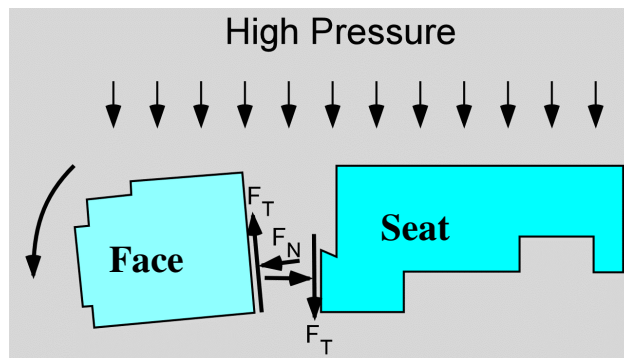


Figure 10. Tipping Moment Due to Seat Interaction.

To minimize tipping moments the pipeline seal is designed to function with low contact force between each face and its seat. Employing a “long” face further reduces the amount of face coning due to seat forces. Face coning angle, for a given tipping moment, is proportional to the inverse of length cubed. A small increase in axial length results in a significant stiffening of a seal face to tipping moments.

Lab Testing of Original Seal

Lab tests of the original seal were conducted on propane and on a mineral oil chosen to nearly match the crude oil viscosity. The seal was tested as a tandem seal on the outboard end of the test machine. A single seal, similar to the primary, served as a closure seal at the aft end of the tester. Face temperature was monitored with a thermocouple epoxied to the bore of the stationary face. Propane and oil tests were conducted on separate test machines. Each tester was equipped to run automatically 24 hours per day. Pressures and temperatures were monitored with a computerized data acquisition system. Closure and containment seal emission readings were periodically measured with a portable volatile organic compound (VOC) analyzer.

Lab Testing on Propane

Propane testing (Figure 11) of the *primary* seal was conducted with a nearly constant pressure of 1100 psi, 3600 rpm, and 80°F chamber temperature. Face surface profiles after a 173 hour propane test are contained in Figures 12 and 13. The traces go from OD to ID, and then continue from ID to OD halfway around the face. The up-tick at the start of each trace is an artifact of the measurement instrument. Face wear is less than 10 μin, indicating that the seal operated with lightly loaded and nearly flat running contacting faces. Emissions from the single *closure* seal (on inboard end of tester) were less than 1000 ppm. Face temperature measured at the thermocouple location on the stator bore was about 10°F hotter than the bulk fluid, in good agreement with that position on Figure 8.

Propane emissions from the dry running *containment* seal were less than 100 ppm with chamber pressure less than 1 psig. Face wear was about 10 μin for the containment seal faces.

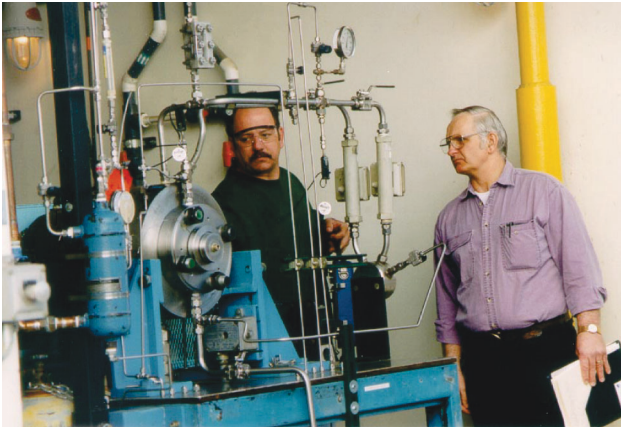


Figure 11. Propane Lab Test.

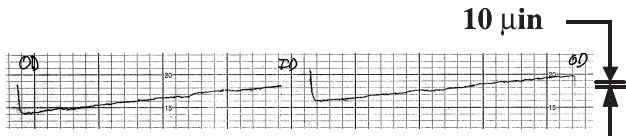


Figure 12. SiC Surface Trace after 173 Hour Propane Test.

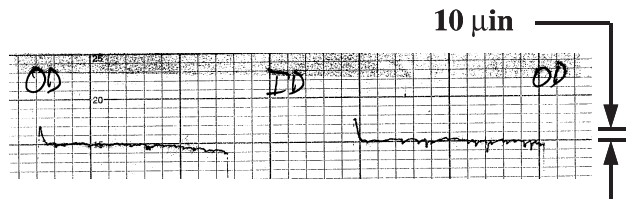


Figure 13. Carbon Surface Trace after 173 Hour Propane Test.

The containment seal was designed to operate as a backup seal, capable of handling pressure to 600 psig and with minimal leakage in the event of primary seal failure (Young et al., 1996). Further testing was conducted with propane pressure up to 1100 psi (at 3600 rpm) on the pipeline secondary seal. There was no visible leakage from the containment seal on 1100 psi propane, but emissions exceeded 10,000 ppm.

Lab Testing on Oil

A different test machine was used to evaluate primary seal performance on a mineral oil. A 10 cSt (at 104°F) oil was selected to approximately match the pipeline crude oil viscosity. Testing was conducted at 1100 psi, 3600 rpm, and 80°F. Face surface contours after a total run time of 206 hours are contained in Figures 14 and 15. The carbon face experienced about 30 μin (average of each side) wear near its ID. Examination of the faces after the initial 42 hours revealed nearly identical surface wear. The faces were reinstalled, without lapping, and run for an additional 164 hours. Most of the carbon wear occurred during the first few hours of run-in. The SiC face had no measurable wear. Both faces were in excellent condition, with no signs of blistering, pullouts, or grooves.

Primary seal face temperature (measured on stator bore) was about 50°F hotter than the bulk oil, in good agreement with Figure 9.

FIELD PROBLEMS WITH ORIGINAL SEALS

A number of the original seals failed with short run times, often less than 200 hours. Other original seals ran well, accumulating over 10,000 hours operation time. The original primary seals employed flat faces (no hydrodynamic features) of carbon versus SiC. The high suction pressure pumps (units in the second, third,

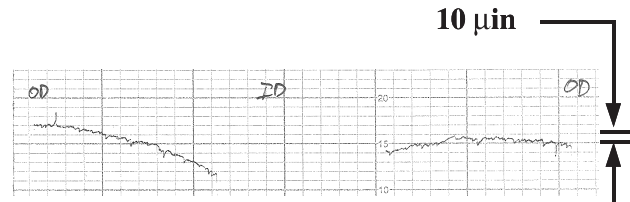


Figure 14. Carbon Surface Trace after Oil Test.

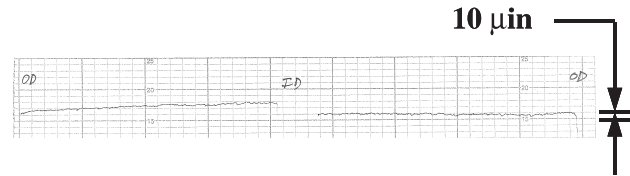


Figure 15. SiC Surface Trace after Oil Test.

and fourth position) at each pumping station experienced the most problems. Failures occurred primarily when running on NGL. The pumps operated near their best efficiency point (BEP), experienced insignificant vibration levels, and had acceptable run out. Short seal life was thus attributed to inability of the seals to handle actual operating conditions. A few failures were due to upset conditions such as NGL suction pressure close to vapor point. Some seals were packed with crude oil sediments behind the rotating face. Lai, et al. (2003), also note difficult service conditions for mechanical seals in pipeline applications.

Seal failure did not appear to be a gradual wearing-in process, but rather an abrupt phenomenon. Several seals were pulled for inspection and were found to be in excellent condition. The replacement seal might then fail in a fraction of the prior seal run time.

Pipe Rouge

Pipe rouge—magnetic iron fines—was found in all of the disassembled seals. The iron fines likely contributed to abrasive face wear. Debris was found in the spring pockets and between the rotating face and its holder (shaft sleeve).

Carbon Stationary Face Condition

In most failures the carbon “nose” was severely worn; in some cases as much as its full length of 0.062 inches. Other seals, with less loss of carbon, experienced heavy ID chipping. OD chipping was minor or nonexistent in most cases. Generally there was significantly more loss of material at the carbon ID. An example of radial taper wear is shown in Figure 16. The amount of worn-in convergent taper is about 0.0008 inches. The large magnitude of ID wear is a strong indication of excessive thermal taper caused by high friction, mechanical loading, or poor cooling.

The backside of some carbons was worn in the area supported by the stainless steel holder. Figure 17 contains a surface trace of the back of a carbon for a seal that failed after 3017 hours run time. The trace shows two radial segments, 180 degrees apart, joined in the middle. Note that the face support area has a wear depth up to 40 μin (this carbon face had one of the largest wear grooves observed in the seat area). None of the SiC rotating faces had a backside wear pattern. This groove could cause resistance to radial motion at the face/seat interface and thus induce substantial face coning distortion (refer to Figure 10). FEA calculation shows that carbon face *radial* movement relative to the stainless support is about 0.003 inches at 1300 psi if the two surfaces are allowed to slip (low contact force and low coefficient of friction). Traction induced face coning is about 1 HLB, provided the face/seat friction coefficient is less than about 0.2. However, the seat area wear groove might dramatically increase traction friction between the carbon face and its stainless steel seat. Then a large change in pressure would cause substantial coning deflection of the face.

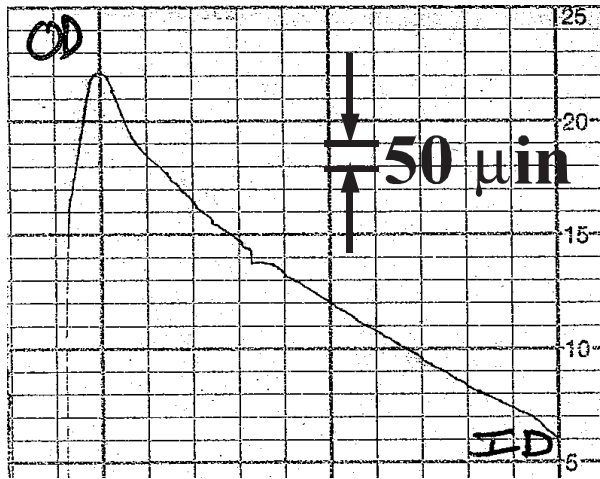


Figure 16. Field Result—Radial Taper Wear of Carbon Face.

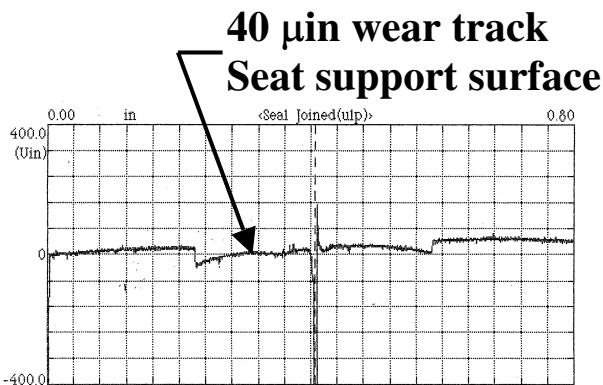


Figure 17. Back of Carbon Face—Surface Trace.

Silicon Carbide Rotating Face Condition

The SiC rotating face typically exhibited a wear track with a small groove near the wear track ID and a much deeper groove near the wear track OD. Random pullouts were sometimes observed. An example of a failed seal SiC wear pattern is shown in Figure 18. Face ID groove is about 100 μin deep and the OD groove about 650 μin .

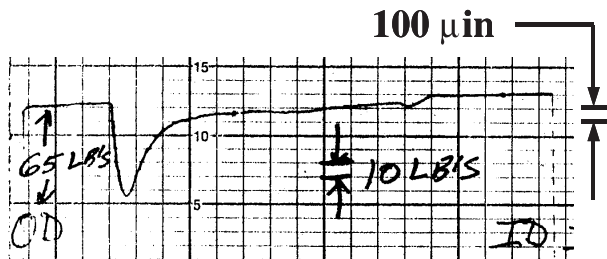


Figure 18. Surface Trace of Failed SiC Face.

Containment Seal

The containment seals were covered with black sticky oil consisting of primary seal carbon wear debris and pipe rouge. Nevertheless, these seals functioned well—preventing visible release of NGL.

FIELD OPERATING CONDITIONS

The primary seals were designed for operation on NGL at 1100 psi. Field observation, however, revealed that pump suction pressure varied over a wide range. To determine actual seal operating

conditions, the pump that suffered the most seal failures was monitored using instrumentation hard wired to a computer in the station control room (Figure 19). This pump is the fourth at the station; it experiences the widest pressure range and is exposed to pressure spikes caused by on/off operation of the prior three pumps. A modem was used to access performance in real time and also download data to the seal vendor's site. Monitored parameters include:

- Pump suction and discharge pressures
- Seal chamber pressure
- Seal face temperature
- Containment seal (vent) pressure
- Motor load



Figure 19. Pump Station Seal Performance Computer.

Face temperature was measured with a thermocouple bonded to the bore of the stationary face. The data reveal that:

- Chamber pressure is nearly identical to suction pressure (pumps have no throat bushing).
- Suction pressure range for this pump is from about 150 to over 1300 psi.
- Suction pressure jumps as much as 350 psi (up or down) when a prior pump (1, 2, or 3) is turned on or off when pumping NGL, and spikes as much as 500 psi on crude oil.
- Seal face temperature is about 25°F hotter on crude compared to NGL.
- Seal failure is noted by increasing (and oscillating) face temperature and vent pressure spiking.
- NGL and crude oil are pumped in batch mode.
- Interface between NGL and crude lasts for several minutes.
- Pump is subject to start/stop operation
- Continuous run time (between standby) can vary from less than one hour to more than 70 hours.
- Standby time can be less than one hour to several days.

An example of seal monitored performance culminating in failure is shown in Figure 4. The pump handles crude the first nine hours, then NGL the remainder of time shown. The change from crude to NGL operation is determined by a large decrease in face temperature and drop in motor load (not shown). Incipient seal failure is noted at about 13.5 hours by rapidly increasing face temperature. Containment (vent) seal pressure spiking due to increased primary seal leakage is noted at about 14.8 hours. The primary seal was subjected to 13 pressure spikes before the start of detectable seal failure.

Several more seals failed in a similar manner—after a number of substantial pressure changes. All failures were on NGL. The

containment seal functioned well as a safety backup. Most of the failures for the original seal in this “worst case” pump occurred in less than 200 hours run time. Some original seals had run times exceeding 10,000 hours in other pumps.

Lab Tests to Simulate Field Conditions

Since field seal failures occurred after a number of pressure spikes, it was decided to conduct propane lab tests with similar rapid pressure changes. Results for one test are shown in Figure 20. An original seal with slightly lower balance ratio was tested. Flat faces of carbon-graphite versus SiC were employed. Periodically, propane pressure was abruptly changed about 400 psi up or down. Maximum up pressure jump was 900 psi, while maximum down spike was almost 600 psi. Seal chamber pressure ranged from 200 to 1300 psi during this test. Seal face temperature was about 85°F to 92°F after the first hour, with propane temperature at 80°F. The start of detectable failure occurred at four hours as the seal face got progressively hotter subsequent to the 11th pressure spike. The test was terminated after about 6.5 hours when face temperature reached 140°F.

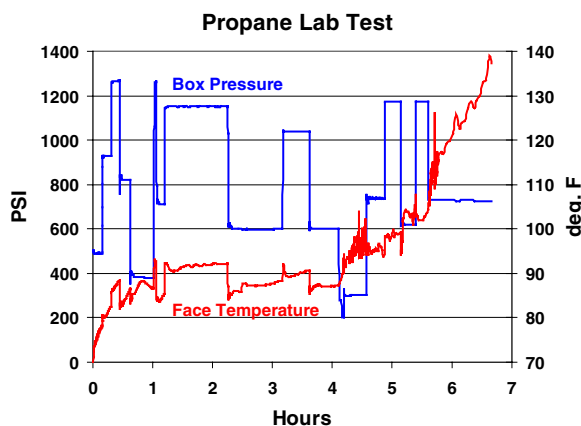


Figure 20. Original Seal—Propane Lab Test with Severe Pressure Spikes.

Post-test inspection revealed higher wear near the ID of both faces. The hot faces thermally coned into ID contact where wear rate was highest. Surface traces showed 90 μm concave wear of the carbon-graphite face. The SiC face had an ID groove about 10 μm deep.

Tests of other faces showed that incipient failure could occur at either high or low pressure (200 to 1300 psi). Some failures occurred after a down pressure spike, others on an up spike. A number of failures occurred several hours after a pressure step change.

This type of laboratory test, with severe pressure spikes while sealing propane, was utilized in all further development of the pipeline seal. Redesigned seals that performed well in this test were then evaluated in the field.

INITIAL UPGRADED SEALS

A number of seal modifications were investigated in order to obtain satisfactory performance.

Liquid/Liquid Tandem Seal

A tandem seal with liquid buffer fluid was considered as an option to improve seal life. Such an arrangement works well in a Middle East NGL (40 percent ethane) pipeline application as reported by Lavelle and Woods (1995). Liquid buffer fluid enhances face cooling and thus acts to reduce thermal coning of the primary seal. Also, some liquid “wicks” between the primary faces providing wet lubrication at face ID in spite of NGL vaporization. However, a liquid tandem seal was ruled out for the current application due to maintenance required to support a buffer fluid system.

Traditional Technologies

Traditional solutions for enhanced seal life were evaluated on the propane test stand. Seals that performed well in the laboratory were installed in field pumps. Technologies investigated included:

- Increased flush flow rate
- Lower balance ratio
- Throat bushing
- Longer carbon face for increased stiffness
- SiC/SiC hydropad faces

Enhanced Flush

Initial focus was to increase the flush flow rate to enhance face cooling. This was achieved by enlarging the Plan 11 orifice hole. Increased flush had no significant effect.

The stainless steel ring containing the multipoint flush ports was redesigned to locate the ports closer to the seal interface. There was no significant improvement to seal life. This change, for seals that did run for an extended time, resulted in jetting erosion of the carbon OD areas in line with the eight ports.

Part of the flush flow was directed to the back end of the stationary assembly to flush out the spring pocket area. Turbulent flow in this area minimizes potential for debris accumulation.

Another area of concern was debris packing of the space between the rotating face and its holder (shaft sleeve). Debris buildup could dislodge or tilt the rotor. A number of radial holes were incorporated into the sleeve extension over the rotating face OD to allow through-flow, thus nearly eliminating debris packing in that area.

Balance Ratio

The balance ratio was reduced (as much as 7 percent from original design) to lighten face contact loading. This change initially had a minor effect on seal life. Later, in combination with other design changes, lower balanced faces were utilized—resulting in some improvement. Faces still showed signs of high heat generation.

Throat Bushing

The seal chamber is wide open to the impeller (Figure 5). Thus seal chamber pressure is equal to suction pressure. During low pressure operation on NGL there might not be sufficient vapor margin for the seals. A bronze throat bushing was designed and installed between the seal chamber and suction. On NGL operation this bushing resulted in about a 60 psi increase for the seal chamber. There was no improvement to seal life.

Longer Stationary Face and Modified Seat Support

Longer stationary carbon faces were employed to provide increased stiffness to bending moments. There was no significant improvement in seal life until seat support geometry was modified to reduce backside of face axial and traction loads by 80 percent. While most installations improved, some short runs to failure still occurred.

Silicon Carbide Versus Silicon Carbide Hydropad Faces

Many of the failed seals had a worn area on the backside of the carbon stationary face (Figure 17). None of the SiC rotating faces had this backside wear groove. To eliminate the possibility of this groove in the carbon binding with its seat, several seals were assembled with a SiC stationary face in place of carbon. Hydropads on the rotating face (Figure 21) facilitated lubrication with hard/hard faces on NGL. Two hydropad designs were evaluated, conventional “deep” hydropads (0.062 inch axial depth) and microhydropads (120 μm depth).



Figure 21. Hydropad Face.

Silicon carbide is about 15 times stiffer than carbon and, thus, much less sensitive to pressure induced distortions. SiC faces tend to run cooler as they conduct heat about 12 times better than carbon. SiC is also highly resistant to abrasion by pipe rouge.

Both types of SiC/SiC hydropad faces performed well on lab propane tests. In field evaluations, however, four of 10 micropad faces failed with run times from 40 to 4430 hours. Nine of 10 seals with deep hydropads failed with run times from 125 to 6152 hours. The deep pad faces tended to have sharp edges (in spite of filing), which chipped, thus inducing hard debris between the faces, leading to damaging face wear in some cases. The micropad faces sometimes ran well in several of the most difficult to seal pumps. Nevertheless, further performance enhancements were required to improve seal reliability.

WAVY SILICON CARBIDE FACES

Field performance of the seals with SiC versus SiC hydropad faces was encouraging, but this technology still did not provide satisfactory seal life. Hydropads primarily promote *hydrodynamic* lift when one face rotates (Key, et al., 1989). They do not significantly enhance lubrication under *static* conditions, such as at startup or during slow roll operation. Furthermore, hydropads may allow ingress of solid particles between the faces.

Seals with a wavy face have advantages over those with a hydropad face. Figure 22 shows a wave pattern (greatly magnified). The proprietary shape consists of circumferential waviness with radial tilt and seal dam (Lebeck and Young, 1989a and 1989b). Depending on required operating conditions, waviness amplitude (h) can range from about 2.5 to 10 microns (100 to 400 μin). The radial taper aspect provides *hydrostatic* load support under all operating conditions. Circumferential waviness provides *hydrodynamic* load support during shaft rotation. The seal dam restricts leakage.

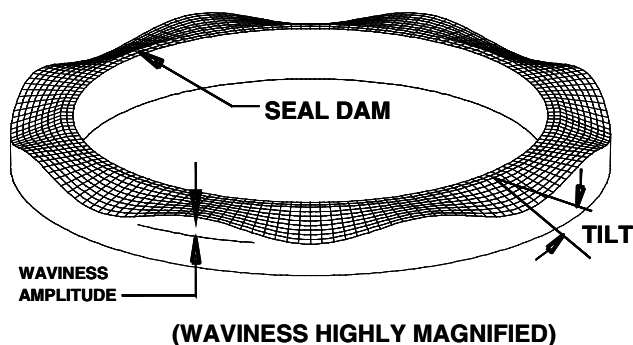


Figure 22. Wavy Face Geometry.

Contamination Resistance

Another positive feature of wavy faces is contamination resistance. As Figure 23 illustrates, fluid entering the valley portion of the wave has three paths it can take during dynamic operation. As the fluid travels toward the wave peak it is compressed and localized pressure is created. A small portion of the fluid can cross

the seal dam as leakage. Another part of the fluid continues over the wave peak as hydrodynamic load support, and the third path the fluid can take is back into the seal chamber generating a recirculation effect. This action removes contaminants from the interface. Other advantages to this wavy shape include bidirectionality, high fluid film stiffness, long life, and high reliability.

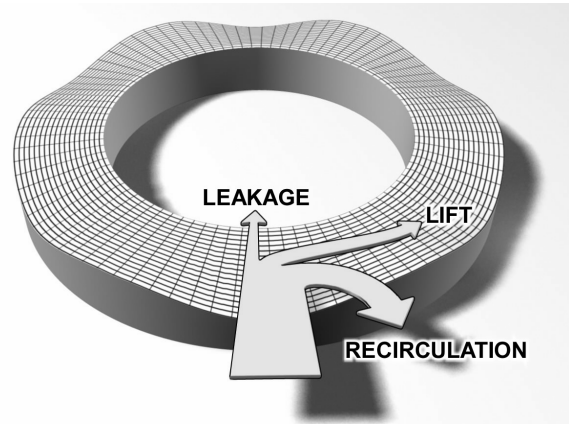


Figure 23. Recirculation Effect of Wavy Technology.

Circumferential waviness by itself was first explored over 30 years ago by Iny (1971a, 1971b), Stangham-Batch and Iny (1973), and Snapp and Sasdelli (1973). These researchers found waviness to be a source of hydrodynamic lubrication. The addition of radial tilt and a seal dam is established in a wide range of applications (Young and Lebeck, 1986, 1989; Young, et al., 1996, 1999, 2003; Young and Huebner, 1998).

The recent advent of laser micromachining (Young, et al., 2003) allows the manufacturing of a wide range of microfeatures in a silicon carbide seal face. Relatively smooth topographical features, those required for wavy faces, are now achievable. Complete control of waviness amplitude, seal dam location, and number of waves are just a few of the feature elements that are possible with the laser manufacturing process. Figure 24 shows a laser machined wave pattern in one of the pipeline seal faces that ran 4276 hours (Figure 25 shows the mating face).



Figure 24. SiC Wavy after 4276 Field Hours.

Wavy Face Computer Model

Various wave patterns were analyzed to assure both noncontact operation and low leak rate. Special software models the fluid mechanics of the fluid film between the faces. The model computes load support for seals having surface features such as waves. An iterative scheme is used that incorporates FEA and the fluid film model. Program inputs include seal geometry, wave shape, and operating conditions. Outputs include face deflection, film thickness, and leak rate.

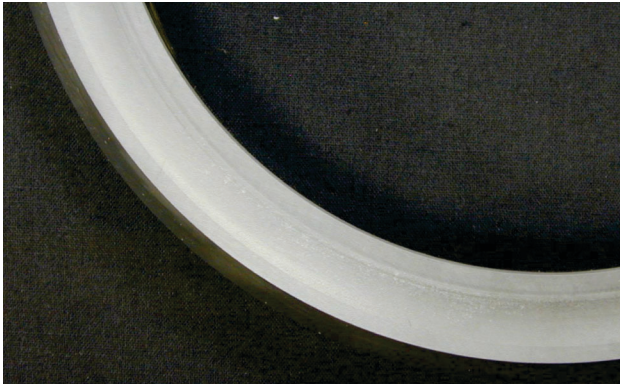


Figure 25. SiC Mating Face after 4276 Field Hours.

Figure 26 shows exaggerated face distortion and temperature distribution for propane at 1200 psig, 70°F, and 3600 rpm. Thermal and pressure induced radial taper is computed to be less than 3 μin; i.e., the faces run “flat.” Predicted film thickness is 15 μin at 1200 psi. This face separation gap is sufficient to assure low wear rate in liquid sealing applications. Film thickness, for this seal, increases with decreasing pressure due to enhanced load support provided by the phase change from liquid propane to vapor. At 150 psig, computed film gap is 37 μin. Sealing interface temperature is predicted to be about 4°F hotter than flush temperature. Temperature rise is about 2°F at the thermocouple location on the stationary face bore (right face in Figure 26). Computed leak rate is 128 grams/hour propane at 1200 psi.

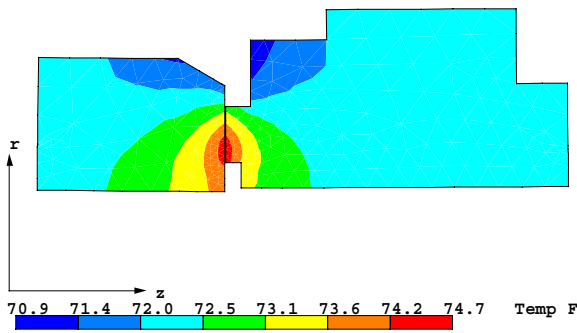


Figure 26. Deflection and Temperature Distribution. Wavy Seal on 1200 PSI Propane.

Two dimensional pressure distribution for one wave is shown in Figure 27. Peak film pressure, 2180 psi, is almost 1000 psi greater than the sealed pressure.

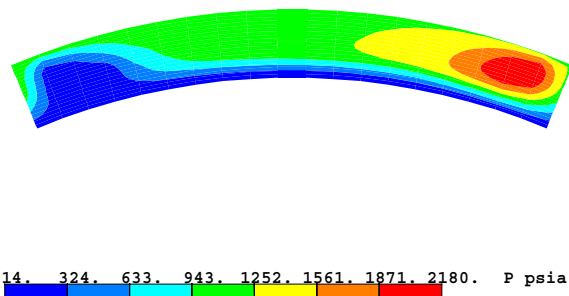


Figure 27. Pressure Distribution over One Wave. Seal Pressure is 1200 PSI Propane.

Wavy Seal Lab Tests

The wavy silicon carbide seal was tested on propane in the laboratory. Shaft speed was 3600 rpm and flush fluid temperature

was about 70°F. Periodically, chamber pressure was rapidly stepped up or down to simulate field operation. Pressure and seal face temperature during a 123 hour test are shown in Figure 28. The variation in face temperature is primarily due to daily ambient temperature cycle. Figure 29 shows a six hour segment of the 123 hour test. Face temperature is barely affected by the pressure spikes. Note that the increased frequency of data collection for a short time subsequent to each step change in pressure. There was no measurable face wear.

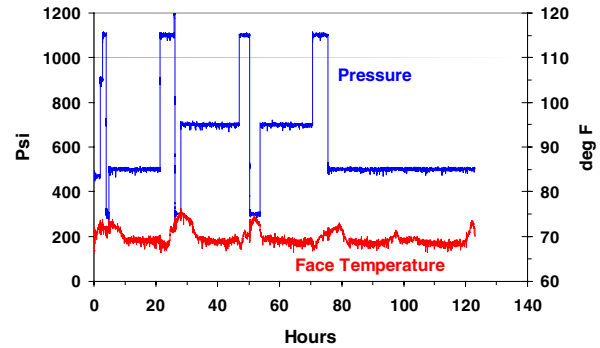


Figure 28. Propane Lab Test of Wavy SiC Seal.

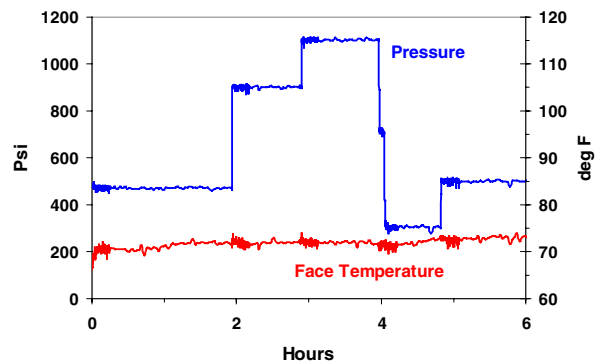


Figure 29. First Six Hours of Wavy SiC Lab Test.

Wavy Face Field Results

The first two wavy SiC/SiC seals were installed in the worst actor pump in September 2000. The seals remain in service as of October 2003. Figure 30 shows a 20 hour time span of field performance for the seal installed in the monitored pump. Pumped fluid appears to be a mixture of crude and NGL during the first two hours, then crude. Fluid is NGL from eight to 16 hours. Pump is on pressurized standby the final four hours. Suction pressure varied between 505 to 1257 psi over the 20 hour time span. Seal face temperature is about 20°F hotter on crude compared to NGL. The seal had accumulated over 4000 total run hours at the time shown and had been subjected to over 200 start/stops.

As of October 2003, wavy face SiC seals have been installed in 67 pumps (134 seals). Four of these seals have failed with high leakage on NGL. Three of those cases had faces in excellent condition (1037, 2149, and 4276 run hours). The third failure had heavily damaged SiC faces (1697 hours). Figures 24 and 25 show the stationary and rotating faces, respectively, for the seal that ran 4276 hours. Both faces experienced negligible wear. Cause of these failures is undetermined.

Final Seal Configuration

The final tandem seal configuration is shown in Figure 31. The primary seal employs silicon carbide versus silicon carbide faces. A lab and field proven wave pattern is laser machined in the stationary face. The containment seal is unchanged from the original design. Other enhancements to the primary seal include:

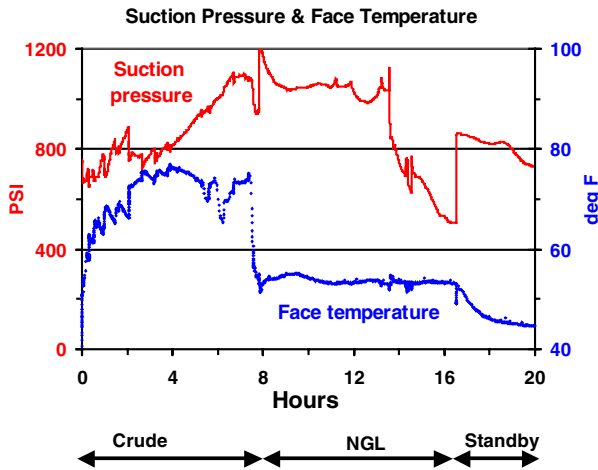


Figure 30. Field Performance for Wavy SiC Seal.

- Multipoint flush with baffle to direct flush to seal interface region for improved cooling
- Portion of flush directed to spring pockets to prevent settling of debris in that area
- Debris expeller holes in sleeve extension over rotating face OD
- Seat/face support loads are low in order to minimize face tipping moments

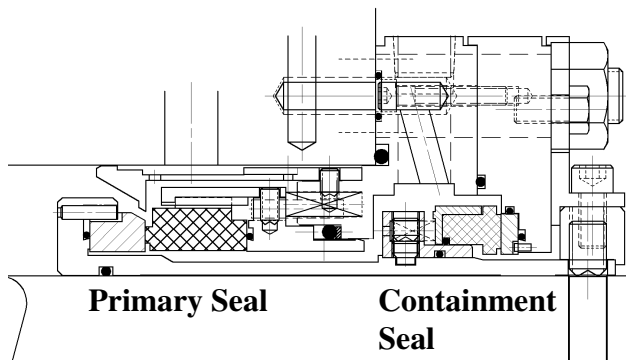


Figure 31. Final Seal Configuration.

CONCLUSION

Extensive analysis, laboratory testing, and field experience has shown that a mechanical seal incorporating laser machined wavy silicon carbide faces can successfully perform in a demanding NGL/crude oil pipeline application. The wavy SiC seals are achieving significantly longer life compared to conventional seals employing nonfeatured faces of carbon versus silicon carbide.

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