



*The Development, Testing, and Successful Application
of Arrangement 2 Seals for CO₂ Pipelines*

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ABSTRACT

The use of CO₂ for Enhanced Oil Recovery (EOR) has been around since the early 70's and was developed to enable oil extraction from declining fields. The process involves injecting CO₂ from either natural or man-made sources into the ground to free up trapped crude oil that would otherwise not be possible to obtain. This process significantly improves the lifetime production of oil from mature fields. Moving the CO₂ from one field to another is accomplished through a series of pipelines and pumps using centrifugal technology. In the early days of this process, pumps were sealed using double pressurized mechanical seals. This particular sealing arrangement was selected because of the very poor lubricating qualities of CO₂, the desire for high reliability, and the lack of capability to seal this fluid in the liquid state using a single seal. The downside to this arrangement was the cost and complexity.

This paper will focus on new technology that allows for contacting seal faces with minimal seal face temperature rise, minimal wear, and leakage almost an order of magnitude less than non-contacting gas seal technology. Details of the design, testing and successful field application of a contacting type liquid seal using engineered seal face micro features in a CO₂ pipeline application are presented.

INTRODUCTION

Moving CO₂ through pipelines can be accomplished either as a low pressure gas or as a supercritical fluid. It is most cost effective and reliable to move CO₂ as a supercritical fluid in the range of 1073 – 3046 psig (7.4 – 21 MPa). In this condition the fluid possesses the properties of both a liquid and a gas where the compressibility of the gas is retained and the density of a liquid is also present. The advantage from a cost standpoint is that a greater mass flow of CO₂ can be achieved through a given pipe size when compared to the lower pressure gas phase. This also helps the pipelines from surging and reduces the chance of hydraulic shock (Battelle 2005).

CO₂ applications require some special considerations when designing mechanical seals for this service. These include:

- Water and hydrocarbons if present in carbon dioxide can combine to form hydrates that can contaminate seal faces that incorporate micro-features for hydrodynamic lift, plug multiport injection holes, or facilitate hang-up of dynamic O-rings.
- Supercritical CO₂ dissolves into and damages elastomer sealing materials through blistering and cracking.
- Many lubricants, both synthetic and petroleum-based, harden in contact with CO₂ and become ineffective.
- CO₂ cools dramatically during decompression so pressure and temperature must be carefully controlled during depressurizing. This is a key concern when considering leakage across the seal face to atmospheric conditions. This cooling affect can alter face distortion in an unfavorable way and potentially form solid phase dry ice.

Over the years several approaches have been taken to seal pumps in liquid CO₂. Early on this was accomplished using double or even triple pressurized liquid mechanical face seals

for pressures above 3000 psi (20.5 MPa). However, the required support systems are complex and many times are as large as the pump skids. In the 80's the use of dry gas seal technology commonly found in compressor applications was introduced. This design option requires the liquid CO₂ to be conditioned with gas panels so as to provide the seals with gaseous phase CO₂. While this technique has proved to be quite successful, it too has the drawback of additional cost and complexity of a support system.

One of the more recent developments has been a move to seal CO₂ in its liquid phase using a dual unpressurized seal with a containment seal with engineered *macro* features on the seal face. Hydropads, as shown in Figure 1,



Figure 1. Hydropadded Face

is an example. This approach, although simple in concept and application, has been met with limited success. If sufficient cooling of the faces is not maintained, the liquid tends to flash to vapor early as it moves across the interface. Since this

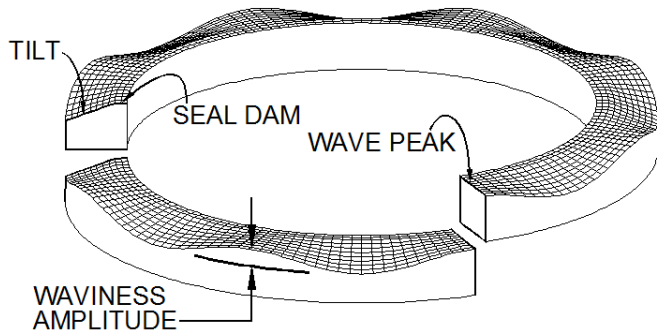


Figure 2. Wavy Face

sealing option is not designed to run as gas seal, face temperatures can increase leading to fluid flashing which in turn results in dry running, excessive wear, and seal failure. The other disadvantage to the macro feature design is the requirement for significant seal flush for cooling to be provided, sometimes in the range of 20 gal/min (76 l/min) or more. With the current regulatory direction to reduce CO₂ emissions from all sources, the direction from industry to simplify support systems and at the same time improve reliability, a new approach to existing technology needed to be made. The following is one approach.

It has been previously shown (Young, et al. 1996, 1997) that the use of hydrodynamic producing *micro* features, such as waves (Figure 2), can be incorporated using laser machining methods. Seal designs that operate in all liquid, all vapor, or combination (mixed phase) is then possible. These micro features can be on the order of 60 to 300 μin (1.5 – 7.6 μm) in depth. In those previous applications the micro feature design was developed to assure complete lift-off of the faces under dynamic operation so that no face contact would occur. Only under static conditions would face contact occur thereby providing very low to unmeasurable leakage. Since these designs were full lift-off face configurations, low leakage was not expected.

While full lift-off technology has gained a significant installed base in the field, many end users were concerned about the high normal leakage rates. In addition, some end users experienced poor reliability due to contamination in the process stream and attempted to meet these challenges with more effective process filtering for the seal injection. While pipelines were operating with acceptable reliability, there was room for improvement. This however required a change in sealing strategy from operating the seal in a non-contacting mode to operating in a contacting mode. The film thickness would need to be reduced so there was asperity contact between the faces. This required a hydrodynamic feature which would allow operation with lightly contacting seal faces (required to reduce the leakage) while minimizing the effective face contact pressures (required due to the poor lubricating properties of CO₂). This strategy required a significant change in seal design parameters that would need to be optimized for these specific application conditions. It would also require an extensive laboratory test and field trial to prove the concept. This approach was needed to answer the following questions:

- Can the seal faces be designed to minimize or eliminate distortions over the wide range of application conditions while using both hard and soft face materials? This was essential to both minimizing leakage and face wear in operation.
- How would the poor lubricating properties of supercritical CO₂ affect the tribological interaction of the contacting faces under the application conditions?
- How would contaminants or heavier hydrocarbon components in many process streams impact the performance of the seal or degrade the conditions of the micro hydrodynamic features?

The challenge for this project and the problems that needed to be addressed would be to create a sealing solution which provided the simplicity of a dual unpressurized seal with containment seal, the low leakage rate of a contacting wet seal, and the low heat generation and power consumption of a gas seal. The following describes the development effort incorporating *contacting* wave technology in CO₂, test results, and field experience.

INITIAL / REVISED DESIGN AND LABORATORY RESULTS

Analysis - As with other hydrodynamic face technology

designs, extensive computer modeling was performed during the design phase of this project. The key parameters that were used included:

- Balance
- Face Width
- Number of Waves
- Waviness Amplitude (peak to valley)
- Width of seal dam
- Spring Load
- Effects of pressure and temperature distortion on seal faces

The starting design used for this application (shown in Figure 3) is a 4.5 inch (114.3 mm) balance diameter primary

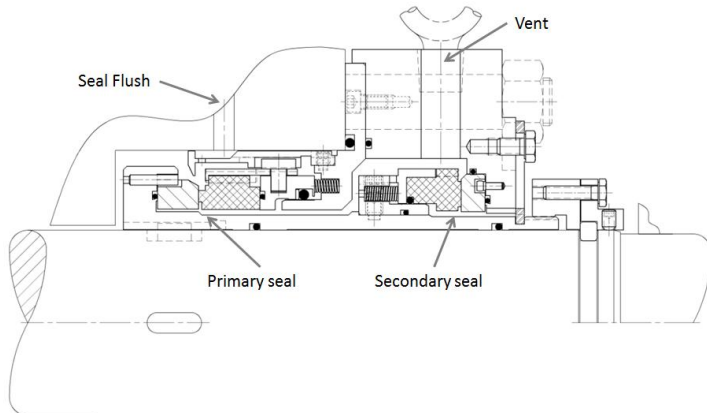


Figure 3. Initial Seal Design for CO₂ Applications

mechanical seal with a 4.437 inch (112.7 mm) balance diameter containment seal as a backup. Stationary face material for the primary seal is an antimony impregnated carbon and the rotating face is self-sintered silicon carbide. The secondary (containment) seal has a resin bound carbon rotor and a self-sintered silicon carbide stator. Extensive coupled finite element analysis (FEA) with interface fluid mechanics was performed on the primary seal to minimize face distortion and leakage by taking into account the factors listed above. Figure 4 shows

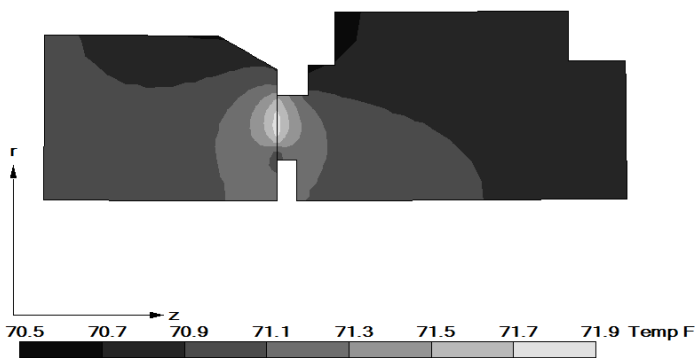


Figure 4. FEA Deflection and Temperature Results

FEA results for total deflection profile (100X magnification) of the faces at 2000 psig (138 barg) seal chamber pressure, 3600 rpm shaft speed and 70°F (21°C) fluid temperature. This figure also shows slight inside diameter seal face cooling due to CO₂ leakage as it expands across the face (Joule Thomson effect). Running fluid film thickness predicted between sealing surfaces at normal operating condition was 14 μin (~0.36 μm) which would indicate very slight asperity contact based on the

combined roughness of the two faces. Figure 5 shows a two dimensional plot of fluid pressure distribution on the face. This plot indicates that the waves generate a very strong hydrodynamic effect with pressures at the wave peaks in the range of 3300 psig (227 barg) thus providing increased face stability due to significant film stiffness and damping. These high pressure zones are also beneficial from the standpoint that under near vapor conditions

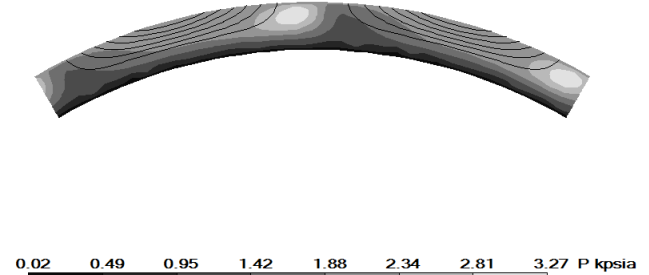


Figure 5. Pressure Distribution in Seal Interface

the fluid can be forced back into the liquid state at the regions of the face (wave peaks) where face contact would occur. Leakage was predicted to be 0.75 scfm (21 slm).

Testing - Operating conditions were set based on typical CO₂ application conditions. Pressure ranged from 1300 to 2000 psig (89.6 – 137.8 barg). Test temperatures were 40 – 60°F (4–16°C) based on ambient conditions during testing. The flush flow was set at 0 gpm (worse case cooling condition) and shaft speeds ranged from 3200 – 3600 rpm. These conditions were changed throughout the initial 240 hour test to observe what affect they might have on seal leakage. Any leakage past the primary seal is contained in the secondary seal chamber and vented through a 10 psig (0.7 barg) check valve in a vent line and then through a mass flow meter. The installed seal in the tester is shown in Figure 6.

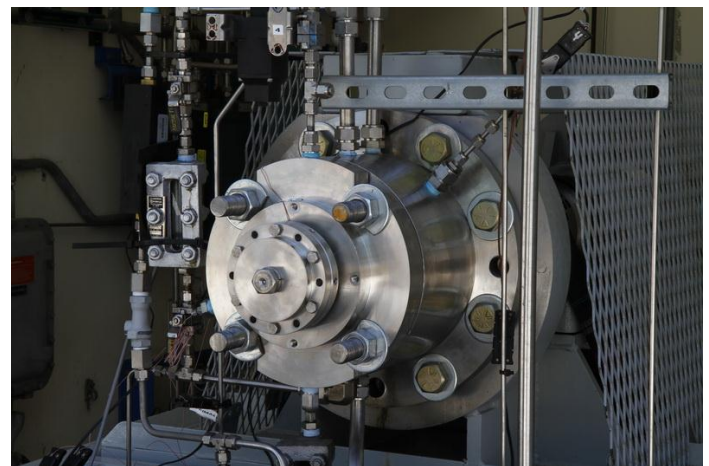


Figure 6. Seal Installed in Test Rig

Initial Design Test Results – Figure 7 shows a plot of shaft speed, pressure and primary seal leakage as a function of time. This first test conducted on the seal design ran well with 1.2 scfm (34 slm) average leakage, which was slightly higher than expected according to analysis. Predicted seal leakage by FEA at maximum pressure and shaft speed was 0.75 scfm (21 slm).

At the conclusion of testing, the seal assembly was taken apart for post-test inspection. During disassembly the faces were seen to be covered with oil and debris from the test fluid (see Figure 8 and 9). Although the oil and contamination are not desirable from a testing standpoint, it was decided it represents a more realistic field environment. These photographs also show that the wave shape is contamination resistant. No debris was found in the micro feature portion of the waves.

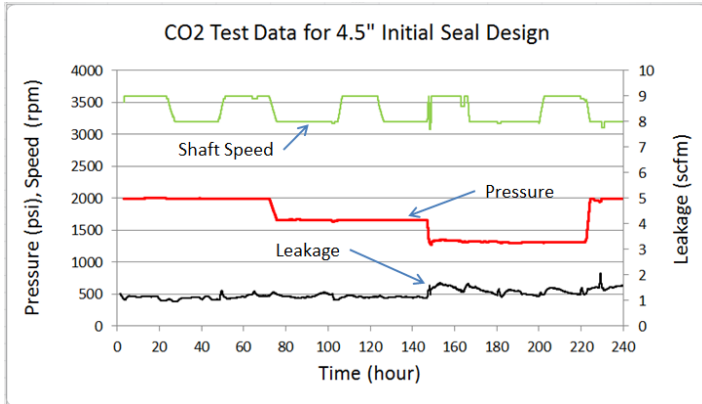


Figure 7. Test Data Results for Initial Design

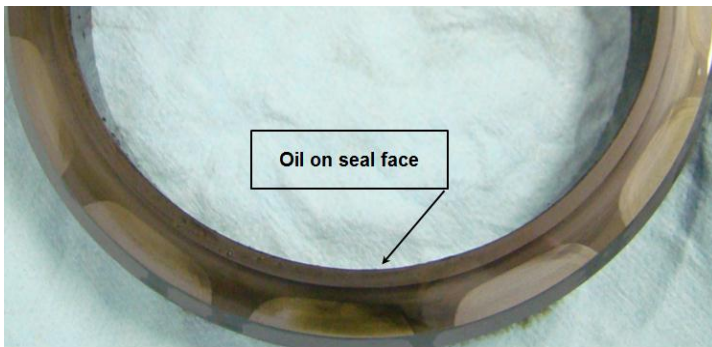


Figure 8. Post-Test Photograph of Wavy SiC Face

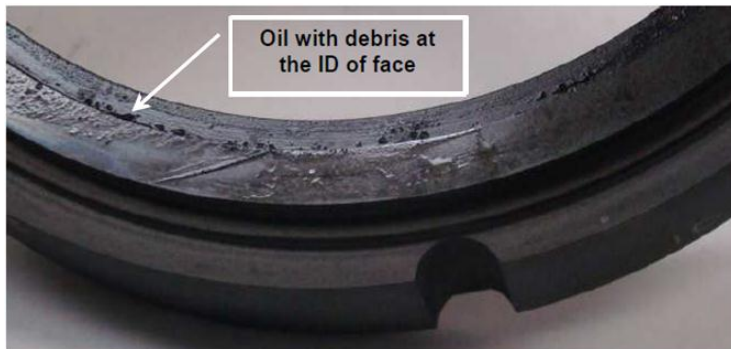


Figure 9. Post-Test Photograph of Carbon Face

Post-test seal face traces showed localized face distortion regions at the drive notch locations of the carbon ring. The distortions were believed to be the result of the high pressure loading on the seal ring. Using carbon seal faces at pressure above 1500 psig (104 barg) can be challenging due to the low modulus of the material. It is common to transition the face material combination under these conditions to SiC vs. SiC to minimize potentially undesirable face deflections. The choice to use carbon vs. SiC in this application was driven by the fact

that, to achieve low leakage, contact of the faces would be required. Running SiC vs SiC would pose a risk due to the very poor lubricating properties of CO₂.

Figure 10 shows the circumferential and radial profile measurements performed on the stationary carbon face. The circumferential trace taken on the running face toward the OD shows three distortion regions corresponding to the three drive pin slots made at the OD. A radial profile measurement (also shown) was taken across the diameter of the carbon face from

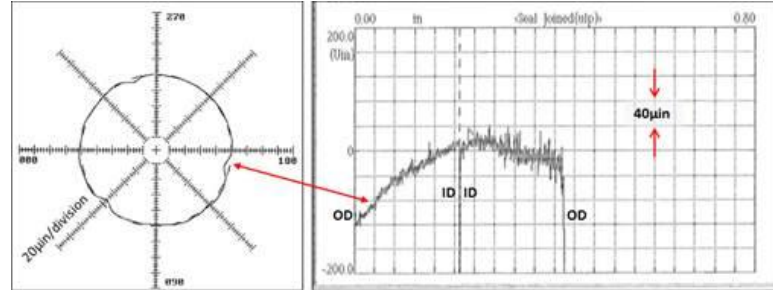


Figure 10. Post-Test Circumferential and Radial Trace Results

OD to ID and then ID to OD. The space between the two ID locations of the face has been removed to consolidate the entire trace. The left side of this trace was aligned with the OD notch region and shows the worn-in distortion. As a result, these distortions on the face were believed to be the reason for higher than expected seal leakage.

Based on the test results further analysis was needed to understand the extent of the problem and to help develop a solution. As part of this analysis a full 3D FEA was performed utilizing the same pressure boundary conditions as seen in test. Figure 11 shows the original carbon seal ring and Figure 12

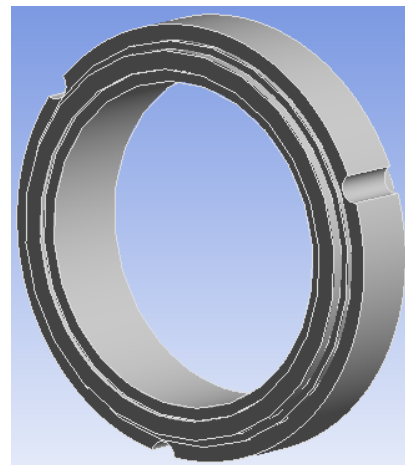


Figure 11. Original Carbon Seal Ring Design

shows the deflection results for just the seal face region, the remaining portion of the ring is shown as a ghosted image. This analysis shows that localized distortion due to the slots can be as high as 121 µm (3 µm) at the higher pressures. This is virtually identical with measured of Figure 10, also at approximately 120 µm (total drop from ID to OD).

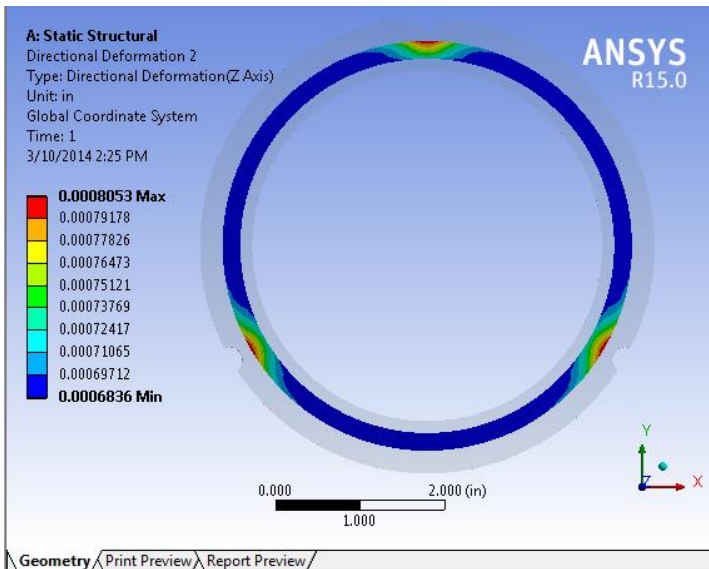


Figure 12. 3D FEA Deflection Results

Revised Design - Undesirable distortion and non-uniform wear of the carbon face required a design change. Several options were explored using a 3D FEA tool. One solution was to increase the number of drive slots from three to six and then



Figure 13. Modified Carbon Seal Ring Design

twelve, the idea being that trying to force the ring into higher modes of distortion becomes increasing more difficult. This option did reduce the localized distortion but at the maximum number of twelve slots it was evident that the majority of the OD had been removed which then led to the option shown in Figure 13. 3D FEA for this configuration is shown in Figure 14 and shows that the localized distortion has been significantly reduced and the remaining distortion is mostly uniform radial taper, a distortion that can be readily addressed through a simpler geometric change of the ring.

Revised Design Test Results – The revised seal design was installed back in the tester and run through various pressures from 1300 to 2100 psig (89.6 – 144.8 barg) while speed was held constant at 3600 rpm. Fluid temperature ranged between 58 and 93°F (14.4 – 33.9°C). Figure 15 shows the results over a

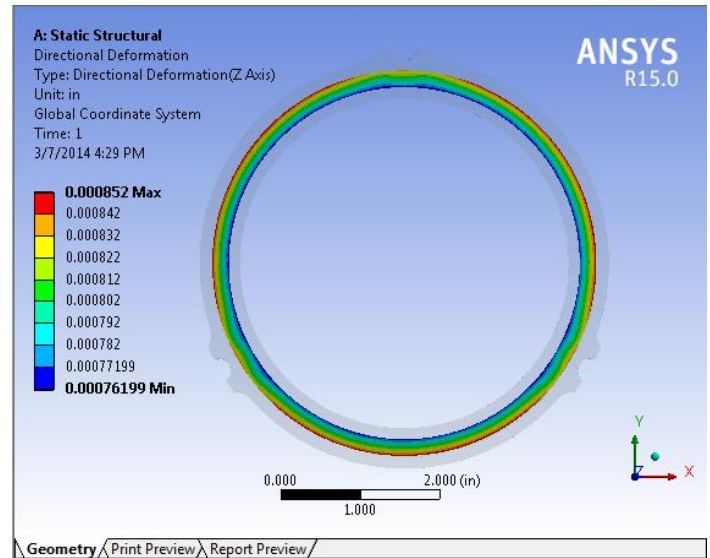


Figure 14. 3D FEA Deflection Results

100 hour test. Leakage has been reduced into the range predicted by analysis and is 40% of the original measured. This chart also shows that when approaching the fluid critical point the leakage tends to increase. Post-test face traces are given in Figure 16. The localized distortion has been eliminated and only slight uniform OD contact and wear is present.

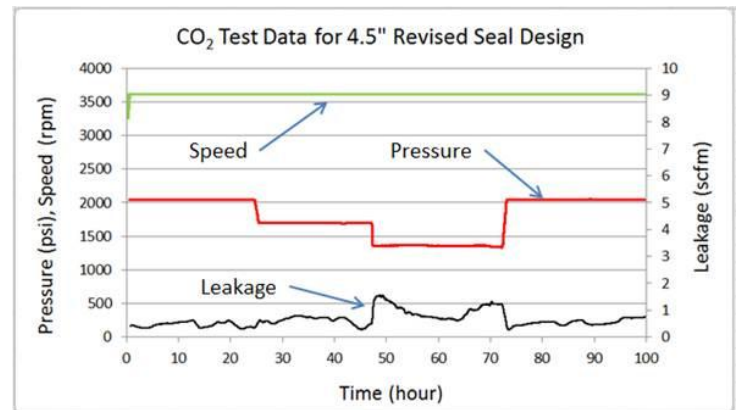


Figure 15. Test Data for Revised Design

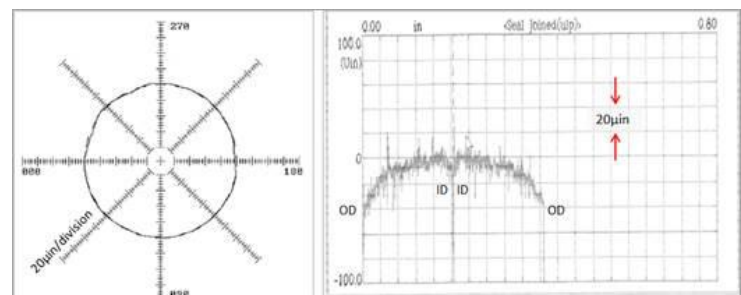


Figure 16. Post-Test Circumferential and Radial Trace Results – Revised Design

This design concept was extended to a 6.000 inch (152.4 mm) balance diameter seal, a 33% increase in seal size. Figure 17

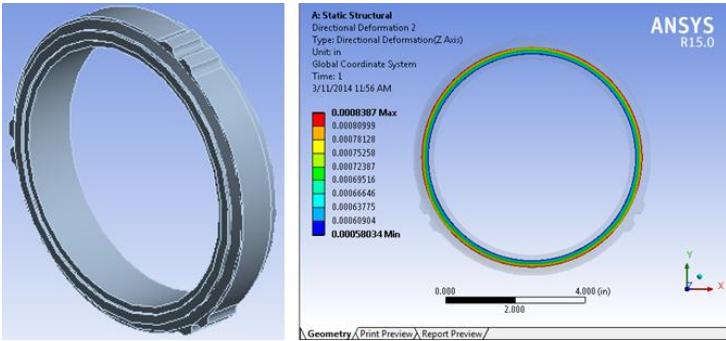


Figure 17. 6" Seal Design and 3D FEA Results

shows this design and the resultant 3D FEA. Again, the localized distortion has been removed and only uniform taper is remaining. This seal was installed in the test rig and run through a specific series of operating conditions as specified by a customer prior to installation in their pump. Conditions ranged from static, 1250 to 2150 rpm, pressures from 1600 to 2200 psig (110.3 – 151.7 barg), and fluid temperature between 40 - 60°F (4.4 – 15.6°C). Figure 18 shows a plot of pressure,

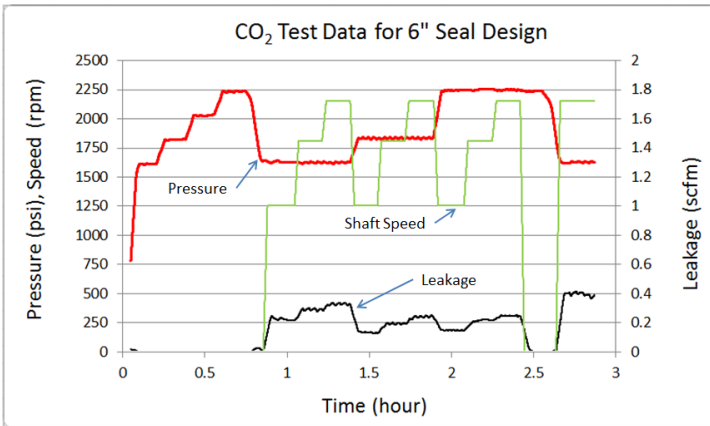


Figure 18. Test Data Results for 6" Seal Design

speed, and leakage as a function of time. Leakage is low and very stable over the various operating conditions. Under static conditions there is no measureable leakage. Figures 19 and 20 are post-test photographs of the SiC and carbon seal faces. Figure 21 shows the post-test traces of the carbon face. No measureable localized distortion can be seen.



Figure 19. Post-Test 6" Seal Design Wavy SiC Face



Figure 20. Post-Test Carbon Face

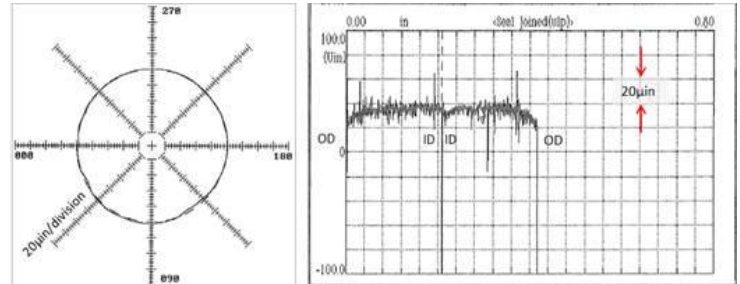


Figure 21. Post-Test Circumferential and Radial Traces of 6" Seal Design

FIELD RESULTS

The six inch balance diameter seal design was installed in pumps in three different CO₂ pump stations: one in May 2012, one in October 2012 and one in February 2013. All three field applications are part of the same 30 inch (762mm) diameter pipeline that stretches 500 miles (805 km) long from southwestern Colorado to western Texas. Additional pumping capacity put in service in 2008 increased the output to 1.3 BCFD (368 BLD). Figure 22 and 23 shows one of the pump



Figure 22. Pump Station (Courtesy Kinder Morgan)

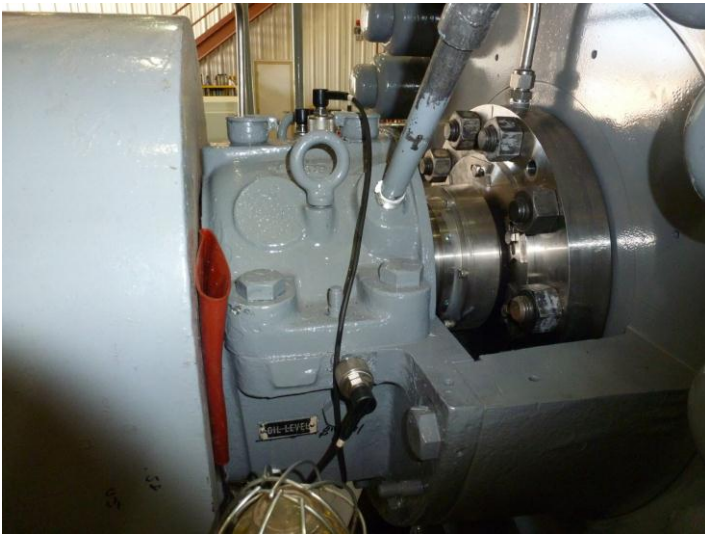


Figure 23. Installed seal on Pump Drive end

stations and installed seals during the initial installation in May 2012. The pump is an API BB2 16X25 radial split case, single stage, between bearings unit running at 1600–2200 psig (110–152 barg) chamber pressure, 65 –115 F (18 - 46 C) fluid temperature, and 1250–2150 rpm. Seal flush was set at 3 gal/min (11.4 L/min). Primary seal leakage is captured by the secondary containment seal (as shown in Figure 3) and directed out of the secondary seal containment region through a 10 psi back pressure check valve in the vent line and then through a support panel flow meter. There are also pressure transducers in the vent line prior to the check valve to monitor secondary seal chamber pressure due to primary seal leakage. Initial startup showed no icing or sweating on the vent tubing which would have indicated excessive leakage. Leakage was measured between 0 and 1 scfm (0–28.3 slm). The adjacent pump had non-contacting (compressor style) gas seals installed and were reading 4–6 scfm (113–170 slm) during operation. At the end of February 2014 the longest running seals had logged more than 16,000 hours of successful operation. While all installed seals are reported as running well, seal performance was reviewed at the May 2012 install site and found that the pressure in the vent line was measuring .17 psig (inboard seal) and .78 psig (outboard seal) psig (1.1KPa and 5.4KPa). With a 10 psig check valve in the vent line this shows that the leakage is well below 1 scfm which is in the same range observed at start up.

CONCLUSIONS

It has been demonstrated through successful laboratory testing and field installations at multiple locations that it is possible to design a dual unpressurized seal with containment seal to run in supercritical CO₂ with low leakage and low wear. Using carbon as a seal face material for pressures required for CO₂ applications can be successful but careful attention to ring design to include 3D FEA is recommended. The other important design consideration is the application of specific micro features to help create load support in the range from all liquid to all gas conditions and maintain minimal face contact load throughout thereby keeping wear to a minimum. These features should also be contamination resistant based on the

level of contamination seen in field applications. Wave technology has been shown to be one design option.

NOMENCLATURE

BCFD = Billion Cubic Feet per Day
 BLD = Billion Liters per Day
 ID = Inside Diameter of seal face
 OD = Outside Diameter of seal face

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