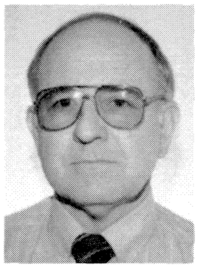
DESIGN AND TESTING OF DUAL SEAL SERIES ARRANGEMENTS FOR PRESSURIZED BARRIER FLUID

by
Kenneth E. Lavelle
Manager of Engineering
William E. Key
Manager of Research
and
Al C. Holman
Senior Design Engineer
BW/IP International, Inc.
Temecula, California

Kenneth E. (Ken) Lavelle, Manager of Engineering at BW/IP International, Inc., received his B.S. degree (Mechanical Engineering) from the University of Illinois. He is a member of the STLE Seal Technical Committee, a member of the STLE Annual Programming Committee, and an Associate Editor for Lubrication Engineering Magazine. In addition, Mr. Lavelle is a past Chairman of the STLE Seal Education Course and member of SAE.

William E. (Bill) Key, Manager of Research at BW/IP International, Inc., received his B.S. and M.S. degrees (Mechanical Engineering) from the University of California at Berkeley and did graduate work in Biophysics at the University of Western Ontario. Prior to joining BW/IP (1974), he held positions in North American Aviation and TRW Systems Group. Mr. Key is member of STLE Seals Technical Committee and the Advanced Projects Subcommittee. At BW/IP, he is responsible for development of new sealing technology and the generation of mathematical models of seal performance.



Al Holman is a Senior Design Engineer in the Product Development Group at BW/IP International, Inc. He received his B.S. degree (Mechanical Engineering) from Brigham Young University. He has been with BW/IP since 1978, involved in mechanical seal testing and development. Prior to joining BW/IP, he held positions at Aurora Pump Co., and Fluor Corp.

ABSTRACT

Dual mechanical seals are operated on pressurized barrier fluid systems when the pumped product properties do not support reliable single seal operation. This provides added protection against environmental release of the pumped product. Traditionally, many dual seal arrangements for pressurized barrier fluids consisted of two standard rotating flexible unit single seals mounted in a back to back orientation. Reliability concerns with the traditional designs include component unseating due to pressure reversals, inadequate provisions for heat removal, and collected debris causing damage to the inboard seal faces.

The draft version of API 682 specifies standard pressurized barrier fluid dual seal designs (API 682 Arrangement 3) with two rotating elements and two mating rings in series (tandem arrangement). In addition, it includes design provisions for flush arrangements and component design to prevent unseating due to pressure reversals.

The successful design and test validation of two dual seal series designs is described. One design is of the multiple spring type rotating flexible element (API 682 Type A), and one design uses a rotating welded metal bellows (API 682 Type B) with 0.008 in thick heavy plate construction. Both designs were developed for use in HF acid services. Design features include multiple port injection at the primary seal, and provisions for continued seal operation under a range of pressure reversals and flashing conditions of the pumped product.

The testing described includes operation on propane, isobutane, and unleaded gasoline (to simulate motor alkylate) fluids in a laboratory light hydrocarbon test loop. Test validations for specific customer requirements were run to validate the ability of the designs to perform safely under various conditions, including loss of primary seal pressure and loss of barrier fluid pressure. The effects on seal face temperature and post test face condition are shown for the variations in operating conditions. Designs meet strict emission standards. Both seal types are performing in refinery services since 1991.

INTRODUCTION

Two dual seal designs have been developed and tested for specific customer applications in pumps handling HF acid. One design uses a rotating spring seal, the other a welded metal bellows seal. Both designs were constructed as a retrofit to existing pumps, with space limitations, and safety of the installations was considered critical. Test programs were designed and performed under laboratory conditions to simulate normal service and anticipated possible fluctuations in operating conditions for the designs. Emissions testing was included as part of the test programs.

SPRING SEALS IN SERIES ARRANGEMENT

This dual seal is in agreement with API Standard 682 [1] for the standard pressurized dual (arrangement 3) mechanical seal. It is an inside balanced type cartridge mounted mechanical seal (with two rotating flexible elements and two mating rings in series). Each seal has multiple springs. The design is made to retrofit existing horizontal process type pumps with limited seal chamber space. Wetted machined metal components are Monel, with alloy C-276 springs. Components not directly wetted, such as setting plates, are type 316 stainless steel. O-ring type gaskets are perfluoroelastomer.

Primary Seal Design Features

The primary seal is a rotating flexible unit multiple spring type seal pictured in Figure 1. The primary seal has a 2.250 in balance diameter for outer diameter pressure. The seal is designed for outer (positive) or inner (reverse) pressure conditions. The following design features allow the seal to function under dual pressure conditions:

- The O-ring type stationary face is positively retained against unseating by the outer and inner flanges, as shown in Figure 2.
- The balance diameter O-ring is captured in both directions by the sleeve design, as shown in Figure 2.
- The rotating face is balanced for proper hydraulic load under positive or reverse pressure. (The face remains closed under either OD or ID high pressure).
- Both the stationary and rotating faces are constructed of sintered silicon carbide. While this material was chosen primarily for its excellent resistance to chemical attack, it also provides high stiffness for resistance to pressure coning which could cause contact at the inner diameter and face damage under reverse pressure conditions. The faces were designed to seal light hydrocarbon products under positive or reverse pressure. A microstructure specification assured high quality silicon carbide with minimal anomalies [2]. A special hydropad design was used to ensure lubrication under a variety of pressure and fluid conditions as shown in Figure 3. The primary seal also features a multiple port [3, 4] injection type flush connection (Figure 2) directed at the outer diameter of the sealing faces for use with the API Plan 32 [5] flush.

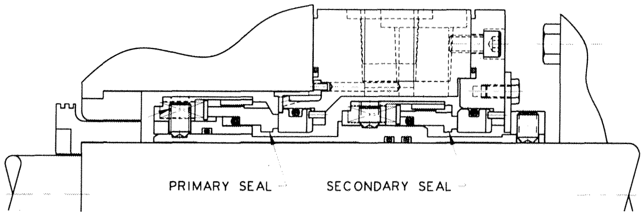


Figure 1. Dual Seal, with Multiple Springs, in Series Arrangement.

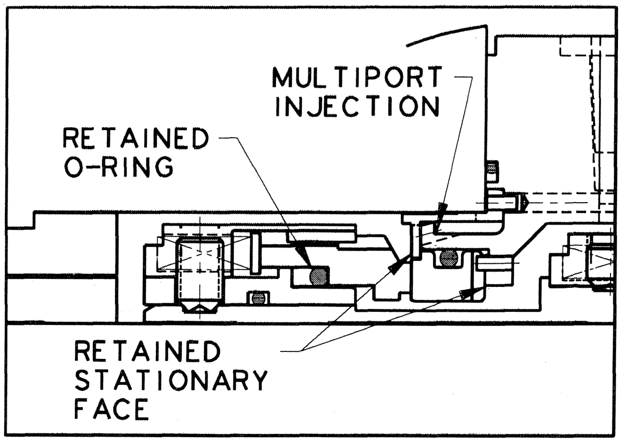


Figure 2. View of Primary Seal Showing: Positive Retention of Stationary Face, Retained O-ring, and Multiport Flush Injection.

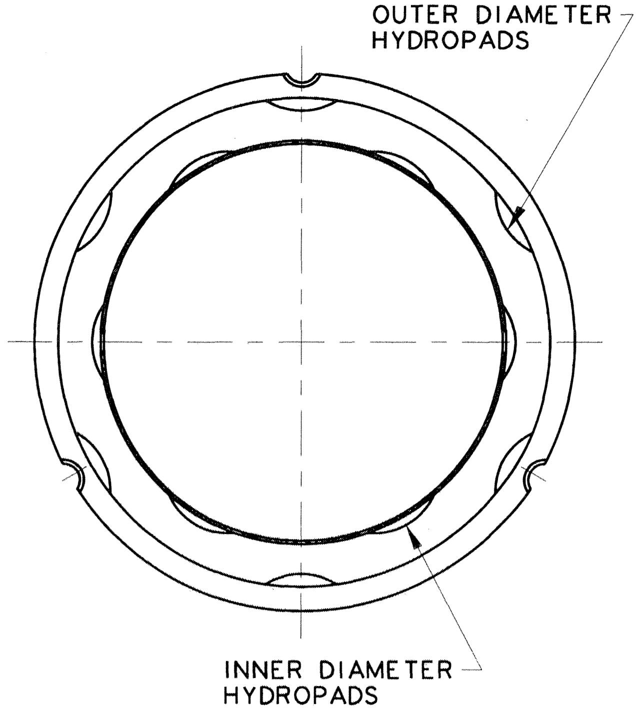


Figure 3. Hydropad Face Design.

Secondary Seal Design Features

The secondary seal is also a rotating flexible element multiple spring type design. This design does not include reverse pressure capability, since the inner diameter is exposed only to atmospheric pressure. Face materials consist of an acid resistant type carbon graphite (low ash) rotating face, and a sintered silicon carbide stationary face. This seal is designed for forced flow (API Plan 54) or circulating flow (API Plan 52 or 53). The spring cage itself acts as a pumping ring, combined with tangential pipe taps in the seal flange. The outlet tap breaks through the cavity at the top center for venting. Previous work [6] has shown this feature to be required to obtain adequate flow and prevent vapor build up in the cavity.

Test Set Up and Conditions

Testing was conducted in an outdoor light hydrocarbon type test rig using a process pump type bearing bracket with a test housing (Figure 4). The shaft was belt driven, and all testing was conducted at nominal speed of 3600 rpm. All fluid temperatures were approximately 100°F for all tests. A piping arrangement designed and built to facilitate testing with a wide range of conditions is shown in Figure 5. The intent of the testing was to simulate site conditions as accurately as possible in a laboratory environment. A witnessed series of eight tests was arranged to validate the seal performance under a variety of conditions. Test conditions are as shown in Table 1. Seals were disassembled and inspected after each test. Reconditioned or new faces were used to start each new test. The following test parameters were recorded:

- Primary seal chamber inlet and outlet temperatures
- Primary seal chamber pressure
- Primary seal flowrate
- Secondary seal chamber inlet and outlet temperatures
- Secondary seal pressure
- Secondary seal flowrate
- Secondary seal emissions (EPA Method 21)
- Secondary seal face temperature (measured using a thermocouple inserted in a drilled hole in the stationary face 0.040 in from the running surface)

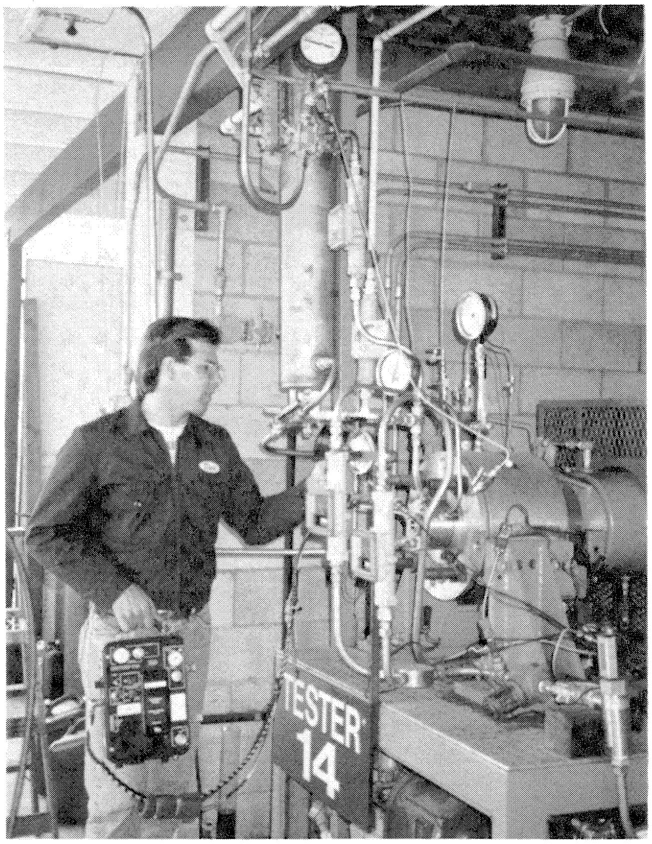


Figure 4. Light Hydrocarbon Test Unit.

Table 1. Primary and Secondary Seal Chamber Conditions — Dual Seal with Spring Loaded Rotating Faces.

		Primary Seal			Secondary Seal		
Test	Time (Hours)	Fluid	Press (psig)	Flow (gpm)	Fluid	Press (psig)	Flow (gpm)
1	18.5	Propane	225	4	95% iC4 / 5% C3	250	3
2	15.0	Propane	300	4	95% iC4 / 5% C3	325	3
3	17.5	Propane	300	4	95% iC4 / 5% C3	325	3
4	0.25	None	0	-	95% iC4 / 5% C3	325	3
5	3.0	None	0	-	Unleaded Gasoline	325	3
6.1	3.0	Propane	300	4	Unleaded Gasoline	325	3
6.2	0.5					115	
6.3	1.0					325	
6.4	0.25					10	
6.5	1.0					325	
7	3.0	Propane	275	4	95% iC4 / 5% C3	325	3
8.1	0.25	Propane	275	4	95% iC4 / 5% C3	115	3
8.,2	0.25					125	0
8.3	0.25					125	3

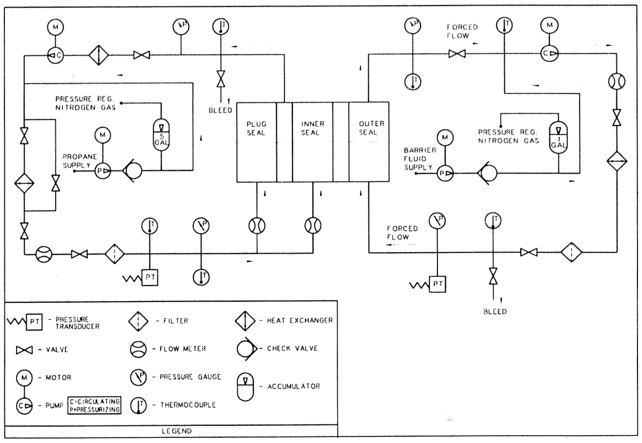


Figure 5. Test Unit Hydraulic Schematic.

Test Results

Test 1 was a run in test to review performance of the seals under anticipated normal operating conditions. Barrier fluid (95 percent iC4, five percent C3) pressure was 25 psi higher than product (propane) pressure. Faces were in good condition after the test, with minimal indications of wear and no distress. Secondary seal emissions averaged less than 100 ppm with a maximum of 400 ppm.

Test 2 evaluated high pressure excursions of the seal chamber conditions. Primary seal ID pressure was 325 psi and OD pressure 300 psi. Again, faces were in very good condition, and average emissions through the outboard seal were less than 200 ppm. Secondary seal face temperature was 137°F to 177°F. Leakage was noted through the primary seal, and traced back to set screw clearance in the sleeve holes resulting in excessive axial motion of the seal components. The parts were redesigned to a tighter tolerance.

Test 3 was a repeat of test 2. Faces were again in very good conditions, with emission readings averaging less than 100 PPM.

Test 4 was a short term test designed to simulate a complete loss of suction pressure in the pump. The iC4/C3 barrier fluid pressure was 325 psi and the primary seal OD was exposed to air at 0 psi. Emissions of the primary seal exceeded 10,000 ppm.

The primary seal faces were intact and operating, but scored. Secondary seal faces were in good condition.

Test 5 was a repeat of test 4, except using unleaded gasoline as the barrier fluid. Primary seal emissions were reduced to 60 to 120 ppm. Seal faces were in good condition.

Test 6 was a cycle test designed to cycle the seal through various pressure and flow conditions, including a loss of barrier pressure and flow. This is a difficult test for the seal because the primary seal is alternately pressurized from positive to reverse pressure. Primary seal propane pressure remained constant at 325 psi, while the unleaded gasoline barrier pressure ranged from 325 to 10 psig. Faces were in good condition after the test, and emissions were typically less than 100 ppm. A data plot of secondary seal chamber pressure, flush inlet temperature, and secondary seal face temperature is shown in Figure 6.

Test 7 was designed to check the effect of a barrier fluid pressure differential of 50 psi above primary seal pressure. Again the faces were in good conditions, with emissions averaging under 100 ppm.

Test 8 was a cycle test designed to test the effects on performance using a light hydrocarbon barrier fluid, reduced down to vapor pressure. Propane product pressure on the OD of the primary seal was held at 275 psig. A data plot of secondary seal chamber pressure, flush inlet temperature, and secondary seal face temperature is shown in Figure 7. Barrier fluid (95 percent iC$_4$ / 5 percent C$_3$) was dropped from 325 psi to near vapor conditions of 115 psig for approximately 15 minutes. Squealing of the seal faces was noted under the vapor conditions, but the faces recovered immediately when pressure was increased to 125 psig. The circulation line block valves were then closed to achieve zero flowrate of the barrier fluid. This condition was run for about 15 minutes and resulted in fluid vaporization due to temperature increase. Again, face squealing was noted. The block valves were then opened and, with commencement of flow circulation, the squealing immediately stopped. Post test face condition was good and no damage was found. Post test surface traces of the primary seal faces are shown in Figure 8.

Conclusions

The seal performed well under all variations of operating conditions. Face damage and high emissions only occurred after extended running time simulating a light hydrocarbon barrier fluid with a complete loss of pump suction. In this case, the damage would not have caused a catastrophic failure but would likely have caused excess leakage over time that may eventually required a seal change out.

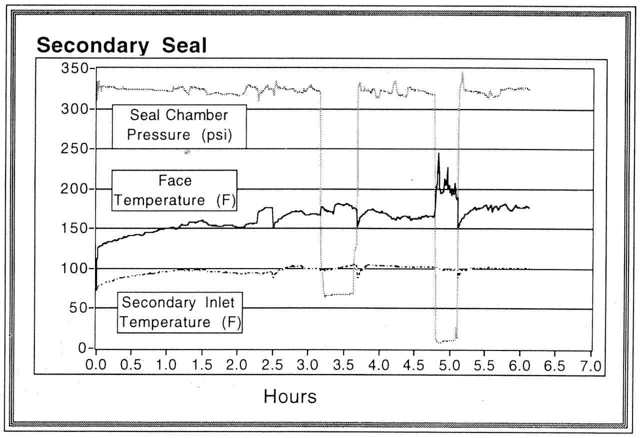


Figure 6. Test Data for Secondary Spring Seal under Cyclic Conditions, Including Loss of Barrier Pressure and Flow (Test #6).

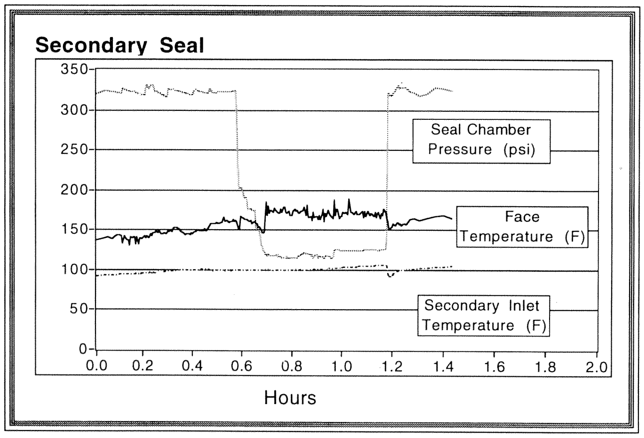


Figure 7. Secondary Spring Seal on Test #8. Pressure of iC4/C3 barrier fluid reduced to vapor point of 115 psi. Pressure then increased to 125 psi and barrier flow stopped with valve closure.

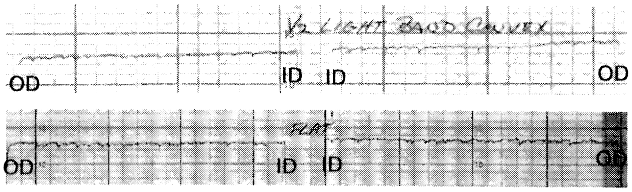


Figure 8. Post Test Surface Profiles of Primary Faces (Spring Seal) After Test #8. Trace for stationary SiC face shown on top and rotating SiC face on bottom. Traces taken from OD to ID, then continued 180 degrees around the ring from ID to OD. Faces flat to one light band.

BELLOWS SEALS IN SERIES ARRANGEMENT

The dual bellows seal is shown in Figure 9. The design consists of two welded metal bellows rotating units in a series (tandem) arrangement. The seal is a preset cartridge design for a between bearings horizontal type pump. This design also complies with API Standard 682 for a pressurized dual (arrangement 3) seal. The bellows was constructed of 0.008 in thick Monel plates. All other metal components, with the exception of hardware in non-wetted areas, was Monel. O-ring gaskets were perfluoroelastomer for basic seal and sleeve parts and Teflon for flange gaskets. The secondary seal was designed to work only with a Plan 54 flush and provisions for pumping were not included.

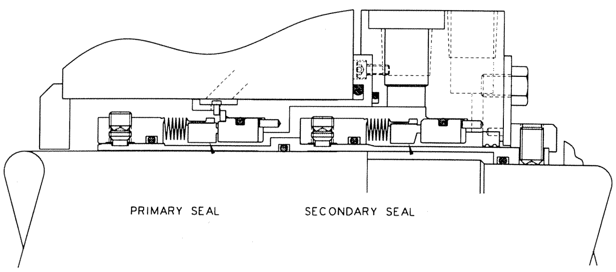


Figure 9. Dual Bellows Seal in Series Arrangement.

Primary Seal Design Features

The primary seal stationary face is retained for positive or reverse pressure in the flange design. The mechanical seal bellows construction and face design allow the seal to operate successfully with either inner or outer diameter high pressure. Face materials were sintered silicon carbide vs sintered silicon carbide. The inner diameter of the bellows adapter was 3.5 in. Hydropads are used at the inner and outer diameter of the balanced rotating face similar to Figure 3. Multiport injection is used on the primary seal.

Secondary Seal Design Features

The secondary seal is a metal bellows seal similar to the primary seal, without the provisions for pressure reversals. No provision is made for self circulation and the cavity is designed for forced barrier flow only, but uses a top outlet for the barrier fluid to avoid entrapment of vapors. The secondary seal faces were acid resistant carbon-graphite *vs* sintered silicon carbide.

Test Set Up and Conditions

Test conditions are shown in Table 2. Shaft speed was 1750 rpm. All fluid temperatures were in the range of 100°F. The test mix of isobutane and propane was customer supplied to simulate actual service fluid. Unleaded gasoline was chosen for the barrier fluid to simulate the use of alkylate supplied as a Plan 54 bypass stream flush in refinery service. The test equipment used was similar to that described in the spring seal testing earlier in this report.

Test Results

Test 1 simulated the performance of the seal under anticipated operating conditions. Barrier pressure was 25 psi above primary seal pressure. Condition of the seal faces was good following the test, with a light contact pattern and no signs of distress. Surface profiles of the primary seal faces are shown in Figure 10. Emissions from the secondary seal averaged less than 50 ppm. Ten start stop cycles were performed during the testing with no damage to the seals or severe temperature fluctuations.

Test 2 simulated a loss of partial loss of pump suction pressure to place a reverse pressure differential of 125 psi on the primary seal. The seal faces were again in good condition following the test, with no signs of damage or distress. Emissions from the primary seal averaged 100 ppm. No indications of high leakage from barrier to product side were seen.

Test 3 simulated a severe failure with a complete loss of pump suction pressure, placing a reverse pressure differential of 200 psi on the primary seal. The ID of the primary seal was exposed to unleaded gasoline at 200 psi and the OD to ambient air at 0 psi. Visual inspection of the faces indicated a polished contact pattern on the stationary face about 20 percent of the face width. No signs of damage were seen on the faces. Emissions average 126 ppm, with no visible leakage from either the primary or secondary seals. After shutdown, the barrier fluid was increased to 300 psig and the reservoir blocked in. Over one hour, barrier pressure dropped from 300 psig to 160 psig with no visible leakage observed.

Table 2. Primary and Secondary Seal Chamber Conditions — Dual Bellows Seal.

		Primary Seal			Secondary Seal		
Test	Time (Hours)	Fluid	Press (psig)	Flow (gpm)	Fluid	Press (psig)	Flow (gpm)
1	62	95% iC4 / 5% C3	275	6	Unleaded Gasoline	300	3
2	6	95% iC4 / 5% C3	175	6	Unleaded Gasoline	300	3
3	2	None	0	-	Unleaded Gasoline	200	3

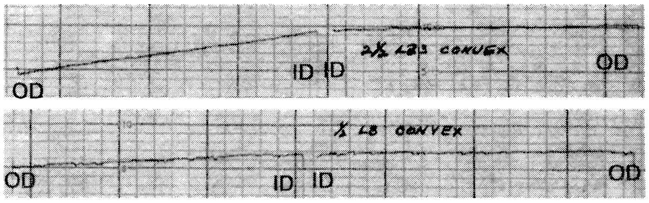


Figure 10. Post Test Surface Profiles of Bellows Seal Primary Faces After Test #1. Stationary face on top and rotating face on bottom.

Conclusions

The seal performed well over the range of operating conditions. No indications of damage or heavy leakage were observed during the testing. Test 3 results show a polished contact pattern at the inner diameter of the primary faces, which could become worse in time and possibly result in face damage. Since this operating condition was a shutdown condition and not expected for extended running times in service, this was not seen to be a problem.

DISCUSSION

Both types of designs studied have been in successful operation in refining environments. Based on the testing, for the range of conditions evaluated, the following significant conclusions are drawn:

- Sintered silicon carbide *vs* sintered silicon carbide can be run successfully as a mating pair combination, and operate in a variety of running conditions. Careful attention must be placed on the lubricating film between the faces.
- For pumped products under flashing conditions, the sintered silicon carbide mating face combination should be run with a barrier or buffer fluid to ensure liquid conditions on one side of the face. This helps create a lubricating film, and acts to remove seal generated heat that could result in severe face damage.
- The series arrangement seal can be designed to give a high degree of safety for containment of pumped products and low emissions. Series design seals can operate under a variety of conditions with good performance.

Seal designs of both the spring and bellows types have been installed in refinery service with good results since 1991, with both pressurized barrier fluids and nonpressurized buffer fluids. Through this time, no failures relating to leakage of pumped product to the environment have been recorded.

REFERENCES

1. API Standard 682, Draft of First Edition, "Shaft Sealing Systems for Centrifugal and Rotary Pumps," American Petroleum Institute, Washington, D.C. (1992).
2. Klimek, E. J., "Selection, Properties and Quality Assurance of Face Materials for Rotating Mechanical Seals," Lubrication Engineering, 44 (9), pp. 719-726, (September 1988).

3. Lavelle, K., and Wang, G., " Emissions Control of Volatile Hydrocarbons with Mechanical Seals," First International Congress on Fluid Handling Systems, GrugaHalle Essen, West Germany, (1990).
4. Key, W.E., Lavelle, K.E., and Wang, G., "Mechanical Seal Performance for Low Emissions of Volatile Organic Compounds," In Proceedings of the 13th International Conference on Fluid Sealing, BHRG, Cranfield, U.K., (1992).
5. API Standard 610, Seventh Edition, "Centrifugal Pumps for General Refinery Service," American Petroleum Institute, Washington, D.C. (1989).
6. Key, W.E., Wang, G., and Lavelle, K.E., "Tandem Seals for Near Zero Hydrocarbon Emissions," *Proceedings of the Eighth International Pump Users Symposium*, The Turbomachinery Laboratory, Texas A&M University, College Station, Texas (March 1991).