DESIGN AND TESTING OF SEALS TO MEET API 682 REQUIREMENTS

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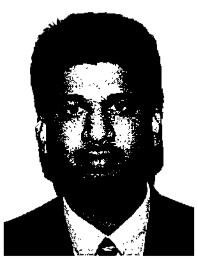

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ABSTRACT

In recent years, manufacturers of mechanical seals have been faced with meeting more stringent leakage requirements along with reliability and extended MTBPM. Regulations by the Environmental Protection Agency and standards by the American Petroleum Institute, have quantified these sealing requirements. Some of the test results are described along with analysis performed to verify sealing performance on pusher and nonpusher seals in flashing and nonflashing hydrocarbons conducted per API Standard 682," Shaft Sealing Systems for Centrifugal and Rotary Pumps," testing requirements.

API Standard 682's mission was to create a specification for seals that would have a good probability of meeting emission regulations and have a life of at least three years. The testing requirements are set up to simulate refinery pump operation that entails continuous duty, pump shutoffs, fluid vaporization, or low pressure operation and to run the seals without a flush. While there are no pass/fail criteria established, the seals are expected to perform within regulated emission limits and demonstrate a minimum three year life.

Testing seals to meet this standard required some face material evaluation and some optimization of the seal design parameters to improve performance. The testing was verified with finite element analysis to establish proper guidelines, in order to maintain design integrity with intermediate seal sizes.

Two classes of mechanical seals were studied:

- Process or liquid lubricated seals
- Gas barrier lubricated seals

Both single and dual liquid lubricated, or contacting face seals, were tested to the standard's requirements. In addition, tests were conducted to verify seal performance limits and to demonstrate stable operation, while being exposed to varying steady state pressures.

API 682 does not currently cover the application of noncontacting gas barrier, seal designs. Testing was conducted on these seals to demonstrate their performance and reliability in real world conditions. Again testing was conducted that went beyond the standard's requirements.

In addition, field operation on some of the seals tested are demonstrated and discussed.

INTRODUCTION

API 682 [1] attempts to cover 90 to 95 percent of all refinery services with existing technology. The goal of testing seals per the standard's requirements are to evaluate and improve seal performance characteristics, so that the final product meets the intent of the standard and its mission statement. Single mechanical seals must be capable of sealing products ranging from light hydrocarbons to heavy ends without customizing each application for the particular service. In addition flashing hydrocarbons with either high or low vapor pressures are to be sealed from pressures ranging from zero to 515 psia, according to the standard.

To meet these requirements, certain principles must be followed. The ability of any mechanical seal to provide satisfactory performance depends upon the integrity of the materials and a design that limits the distortion of the sealing faces. Two primary distortions that must be accounted for are mechanical distortions resulting from hydraulic pressure and thermal distortions resulting from seal frictional heat. The goal of the seal manufacturer is to design a sealing system that results in a slightly positive or converging seal gap that allows the fluid to enter the sealing faces to control the generation of heat without leaking product. At steady state operation, mechanical seals will eventually wear into a parallel face condition. Changes in pressure, speed, and to a smaller degree temperature will modify this parallel face condition and can cause instability in the operation of the seal. Testing must show that the seal will still operate satisfactorily.

The design of the mating ring is equally important where both the shape of the ring to control mechanical distortions and the flush arrangement to control thermal distortions must be controlled. In the past, some mating ring designs were almost entirely enclosed within the seal gland. This limits the exposure of the mating ring to the sealed fluid, hence limiting the transfer of heat to the surrounding fluid. New designs were required to improve the operating performance of the mating ring.

API 682 defines three basic sealing arrangements. Arrangement 1 is a single cartridge seal. Arrangement 2 is a dual unpressurized seal, which in the past has been termed a tandem seal. Arrangement 3 is a dual pressurized seal with the seal faces in a tandem arrangement, but with the outer seal chamber at a higher pressure than the product. Arrangements 1 and 2 are common designs. The new Arrangement 3 design places the product on the outer diameter of the inboard seal so it is not stagnant and centrifugal effects assist the barrier fluid in lubricating the inboard seal faces. The standard also defines basic seal types. The Type A seal is a pusher seal incorporating O-ring secondary seals. Type B seals are low temperature metal bellows seals and Type C seals are high temperature metal bellows seals incorporating Inconel bellows as a secondary seal.

The scope of API 682 is currently limited to mechanical seals that are lubricated by the product pumped or a buffer/barrier liquid. It was the feeling of the users who created the standard that sufficient plant experience did not exist for incorporation of other seal types. For this reason, the standard did not specifically address noncontacting or dry running sealing technology. Testing was done in accordance to the standard to illustrate the performance ability of this type of seal for future consideration.

SEAL DESIGN EVALUATION

Prior to running qualification tests per the standard's requirements, certain seal design parameters were evaluated. The results of these evaluations were then incorporated into the seals that underwent qualification testing.

Flush Arrangement Testing

A series of tests was run to evaluate flush arrangements that would meet the 682 Standard's distributed flush requirement for rotating single seals. The tests consisted of running a 4.0 in (balance diameter) Type A seal in water at 100 psi and 3600 rpm within a large bore chamber. The chamber bore was 6.0 in, resulting in a radial clearance over the seal OD of 0.562 in. Mating ring temperatures were taken with thermocouples located at the primary ring mean face diameter approximately 0.050 in away from the sealing interface.

A closed loop circulating system was used in the test. Flow was provided by a positive displacement pump and controlled by a variable orifice valve. Flow measurements were monitored using a calibrated rotometer. A heater was placed in the flush line to control the inlet temperature and a cooler in the return line to remove heat generated in the seal chamber. As the tests were conducted over a period of weeks, the cooling water temperature varied resulting in stabilized inlet temperatures between 128°F and 140°F. In each test, new primary and refurbished mating ring faces were used. Each test was run for a period of approximately seven hours, until an equilibrium temperature was reached. This condition was achieved when the mating ring and inlet temperature remained constant for a minimum period of 1.0 hr.

Various flush and mating ring styles (Figure 1) were evaluated in the tests.

The circumferential flush uses a rectangular mating ring with the O-ring near the back of the ring, to increase the amount of wetted area available for heat transfer. The flush fluid is circulated around the circumference of the mating ring through a groove and exits through a large slot 180 degrees from the inlet.

The distributed flush uses essentially the same mating ring and gland design as the circumferential flush, but the mating ring is modified by adding a large chamfer near the sealing faces. This allows the flush to exit past the "plate" in addition to the exit slot.

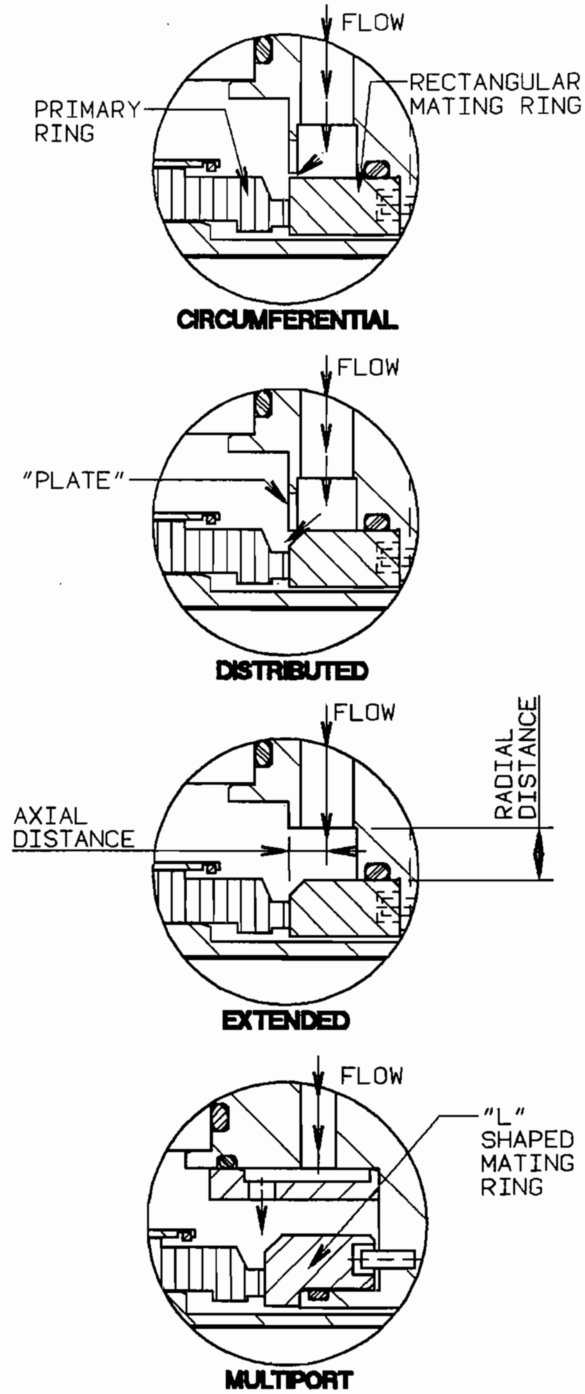

Figure 1. Mating Ring and Flush Arrangements Evaluated.

The extended flush uses a rectangular section mating ring with the O-ring near the back end of the ring. The gland is designed to expose approximately 65 percent of the mating ring's total axial length to the fluid in the seal chamber.

The multiport flush arrangement tested uses a sleeve with a circumferential groove on the outer surface and multiple holes around the periphery, to introduce the flush at multiple locations around the mating ring. This flush arrangement was used with both the extended style mating ring and with an "L" shaped mating ring design.

The initial test objective was to set up and evaluate each of the four flush arrangements, using a rectangular section mating ring. The concept of using an "L" shaped mating ring for dual seal arrangements modified this test sequence. The outcome of the tests was expected to confirm that the multiport flush would prove to be the best arrangement. This was the preliminary flush arrangement chosen for use with the "L" shaped mating ring. The initial test results did not confirm that the multiport flush arrangement provided the best cooling for the mating ring. Further testing of the multiport arrangement was continued using the "L" shaped mating ring. During this sequence of tests, the multiport sleeve was modified to reduce the radial clearance from 0.390 in, to 0.250 in, to a final clearance of 0.156 in. Smaller radial clearances were not attempted as API 682 requires a minimum radial clearance between the rotating seal assembly and the gland or seal chamber of 0.125 in. Also modified during this test sequence was the number and axial location of the inlet hole centerline with respect to the sealing interface. The final test, using the "L" shaped mating ring with a single point injection, resulted in lower face temperatures than the multiport, but higher temperatures than the extended flush.

The test parameters as well as a summary of the test results are shown in Table 1. The conclusion from this testing is that the distributed and extended flush arrangements performed significantly better than any of the other designs. Variations in performance between the other designs were less significant. It is assumed that the differences in performance are primarily due to turbulence and/or flow patterns that develop as a result of the injection method, angle of injection, flowrates, and resultant injection velocities. The inlet velocity of the multiport arrangement, especially with a larger number of inlet holes and/or larger hole diameters, is significantly reduced. The velocity of the fluid in the seal chamber at 3600 rpm is 4595 fpm. The radial velocity at the inlet of the multiport arrangement ranges from 72 to 870 fpm, assuming an even distribution of the flush fluid through the multiple holes.

Table 1. Test Results from Flush Arrangements Evaluation.

Type Flush	Flow Rate (GPM)	Mating Ring Temp (F)	Inlet Temp (F)	M.R. (-) Inlet Temp (F)	Avg. Flush Velocity (fps)	Radial Clear (in)	No.-Size Inlet Holes (in)	Axial Distance to Face (in)	Mating Ring Style
Distributed	1.5	156	128	28	9.8	.468	1-1/4	.265	Rect.
Extended	1.5	170	140	30	9.8	.468	1-1/4	.265	Rect.
Circumferential	1.5	179	138	41	9.8	.468	1-1/4	.265	Rect.
Multi-Port	1.5	178	139	39	1.2	.390	8-1/4	.260	Rect.
Multi-Port	2.2	175	131	44	7.2	.390	8-1/8	.260	Rect.
Multi-Port	2.2	175	134	41	7.2	.250	8-1/8	.437	"L"
Multi-Port	2.2	187	141	46	14.5	.156	4-1/8	.062	"L"
Multi-Port	2.2	186	140	46	14.5	.156	4-1/8	0	"L"
Multi-Port	2.2	190	144	46	14.5	.156	4-1/8.	.421	"L"
Single Point	1.0	180	140	40	6.5	.468	1-1/4	.390	"L"

The circumferential flush protects the flush fluid from the higher temperature seal chamber temperature by the front "plate." However, the plate also restricts the flow of flush fluid from part of the mating ring OD as well as the vertical plane directly above the seal interface. The distributed flush arrangement works like the circumferential flush, but has a chamfered surface to allow more flow close to the sealing interface. Based upon the results, it was concluded that the distributed flush arrangement would be used for all single seal qualification tests.

Due to the changes made during the test program and the fact that various mating ring arrangements were used to evaluate the multiport design it should be noted that this was not a definitive optimization of this flush arrangement. Compared to a single point injection with limited exposure of the mating ring surface, with optimized design parameters, the multiport design should show improved performance. Seal chamber studies by Adams [2], showed that just increasing the radial clearance over the seal faces decreased the seal face temperatures and improved the ability of seals to perform during off duty pump operation. The extended design that opens the radial clearance directly over the sealing faces models this setup and did perform well during this testing. More development work is needed in this area to optimize flush arrangements in both small and large bore chambers in both single and two phase fluids.

Material Evaluation

API 682 requires the use of a "premium grade, blister resistant carbon graphite with suitable binders and impregnants to reduce wear and provide chemical resistance" [1]. Carbon with antimony binders has proven, from both lab tests and field installations, to be a superior material for both water and light hydrocarbon service. It exhibits relatively good, time limited, dry running characteristics and is a higher strength material than carbon grades with resin binder. The disadvantage of this material is its poor chemical compatibility with both bases and acids, so its use is restricted. Carbon with resin binders is the material of choice for all other services, unless a service arises that requires a high degree of corrosion resistance, where a more chemically resistant carbon will need to be selected. Both the antimony and resin binder carbon materials used in the evaluation tests had been previously qualified under a company standard hot water PV test program.

Of the two general types of silicon carbide, alpha sintered and reaction bonded, reaction bonded silicon carbide was chosen as the default mating ring material by the standard. A series of evaluation tests was conducted to sort out the differences between some of the reaction bonded silicon carbide grades available. The test consisted of running a 4.0 in seal with one select grade of antimony carbon against the silicon carbide mating ring in propane for 100 hr at 250 psi, 90°F and 3600 rpm. These are the basepoint conditions specified per API 682 for propane qualification test. The flowrate for all the tests was 5.0 gpm. Each test consisted of running two seals, one in each chamber of the test rig. The test results are summarized in Table 2. The indexes are the changes in slope, waviness, and wear compared to the base grade, A. In all cases, post test evaluation of the primary rings showed a concave wear pattern.

Based upon its superior performance over the other grades tested, Grade A was chosen as the sole source for API 682. The difference between Grades A and B occurs because of process differences between the two manufacturers, as the grain structure is similar. A comparison of physical properties showed minor differences with Grade A having a slightly lower reported thermal conductivity. Examination of the microstructure did show Grade B to have some larger islands of residual silicon. The Grade C material is a bimodal material with two distinct grain sizes. The larger grain material is on the order of 40 μm, while the smaller grain material is on the order of 5.0 to 10 μm. As noted by Wallis [3], the fine grain structure tends not to be bonded to one another, but are isolated and prone to pullouts. In the conducted tests, some minor pull outs did occur. The carbon wear that occurred appeared to have been adhesive wear due to the presence of larger pools of silicon. The grade D material is classified as a fine grain material, but has a slightly larger structure than the other fine grain materials at 15 μm. This material had islands of unreacted graphite and also some surface porosity that was not noted by the unaided eye. These two conditions resulted in high carbon wear with subsequent high mating ring wear.

Table 2. Summary of Reaction Bonded Silicon Carbide Material Evaluation.

Silicon Carbide Grade	A	B	C	D
SiC Structure	Fine Grain 9-12 μm	Fine Grain 10 μm	Coarse Grain Bi-modal	Fine Grain 15 μm
Carbon Slope Index	1.0	1.0	1.4	0.86
Carbon Waviness Index	1.0	0.91	0.73	1.73
Carbon Wear Index	1.0	1.95	2.46	>10
SiC Waviness Index	1.0	1.8	2.0	2.5
SiC Wear Index	1.0	1.95	2.0	2.5

Face Width and Balance

There are multiple schools of thought on seal face width and balance for seals handling light hydrocarbons. One theory is that narrow seal faces with an 80 to 85 percent balance generate less heat. An opposite theory is that wide seal faces with 75 to 85 percent balance provide for some hydrodynamic lift, even when a two phase condition occurs at the seal faces, and thus generates less heat. In between these conditions there are endless possibilities of seal face widths and seal balances that can be considered. Individual design analysis is appropriate for high pressure applications or where the sealed fluid is very near its vapor point. For the seals undergoing API 682 qualification testing, there is an established window of pressures and temperatures along with specific guidelines for how close the fluid should be away from its vapor point, to achieve good performance in flashing hydrocarbon service. The intent of API 682 is to standardize on seal designs. Thus, a single design compromise is needed.

Three combinations of seal face widths and balances were tested. The test parameters and results are shown in Table 3. The qualification test parameters for propane were used in this evaluation. Only the narrow face design caused any measurable mating ring wear. The mating rings on the two other designs did show some localized grooving, corresponding to the primary ring face OD and ID, on the order of 20 μin or less. Due to operational difficulties, the wide face seal design was run only in the steady state dynamic mode. Post test inspection of this primary ring showed a 350 μin concave condition. This wear was considered excessive for a steady state condition.

The results from these tests show that narrower face designs, while exhibiting low leakage, generate more heat. The high heat loads cause the seal to have a greater percentage of the face in a vapor state than seals with wider faces. As a result these designs have higher wear rates. Also, very wide faces coupled with higher balances do not generate sufficient hydrodynamic lift to offset the additional heat generated by the increased face area. Intermediate face widths with lower balances provide better performance and exhibit less wear. Lower balance seals were not tested as past experience has shown that unless the primary ring is designed for a specific range of conditions, the seal can leak excessively.

Table 3. Test Parameters and Results of Face Width and Balance Evaluation. Operating Conditions: 250 psi, 90°F, 3600 rpm.

Face Width (in)	Seal Balance (%)	Average Dynamic Leakage (PPM)	Average Cyclic Leakage (PPM)	Primary Ring Wear (μ in)	Mating Ring Wear (μ in)	Calculated Coefficient of Friction (μ)
0.125	85	<20	<50	0.003	20	0.18
0.250	85	125	N/A	0.001	nil	0.11
0.200	80	<50	<40	<0.0005	nil	0.08

In Table 3, the calculated coefficient of friction was arrived at by taking the horsepower from the test and subtracting the churning losses due to fluid turbulence, to arrive at a net horsepower figure. With these known values, the coefficients were derived from the basic equation for seal generated heat,

$$hp = (PV)(\mu)(A_o)/33{,}000 \qquad (1)$$

Finite Element Analysis

Finite element analysis (FEA) was performed for the various combinations of seal balance and face widths. A proprietary FEA software was used for the analysis. The solution involves solving the axisymmetric, fluid flow, heat transfer and structural equations, using continuously updated fluid properties, which are temperature and pressure dependent. The FEA results were compared with the actual results to obtain a better understanding of the input variables. Various inputs were modified to evaluate the effects upon the net distortions. Using this the face widths, seal balances and primary ring geometries were optimized. The comparison of the analytical results to the actual test data is shown in Table 4. The initial inputs for the narrower face design overstated the hydrodynamic effects, so the calculated net distortion was lower than the actual net distortion of the primary and mating ring pair.

TYPE A PUSHER SEAL DESIGNS

The philosophy behind the development of the pusher seal was to design a range of seals for low, medium, and high pressures that could be used for fluids with both low and high vapor pressures (Figure 2). The geometry, face width and balance of the designs are different in order to control mechanical and thermal distortions. In light hydrocarbon service, the low pressure design (LPD) is suitable for pressures up to 650 psi. The medium pressure design (MPD) has the addition of a hammerhead on the front end OD to modify the centroid of the primary ring and provide additional stiffness. This design is also suitable for pressures up to 650 psi, but with approximately 30 percent higher ratings in larger seal sizes. The high pressure design (HPD) utilizes the geometry of the MPD, with the addition of hydropads on the sealing face to enhance face lubrication. There is also a modification to the seal's area balance to control distortion. This design is suitable for pressures up to 1000 psi depending upon size, shaft speed, and the fluid sealed.

Single Pusher Seal

The single Type A Arrangement 1 seal (Figure 3) is a cartridge design that can use any one of the three designs noted previously, depending upon the fluid to be sealed and the condition of the fluid in the seal chamber. The mating ring is a rectangular cross section with a chamfer on the seal face to allow the flush fluid to cascade towards the seal faces and remove any vapor pockets that may exist from seal generated heat. The flush arrangement selected as described above is the distributed flush.

Table 4. Seal Balance vs Face Width Optimization: Comparisons of Calculated and Experimentally Measured Values. Operating Conditions. 250 psi, 90°F, 3600 rpm.

Seal Config.	Performance Parameters		Finite Element Analysis Results	Experimental Results
Narrow Face Design	Primary Ring Face Distortion (minutes)	Due to Press.	-1.98	N/A
		Due to Temp.	1.36	
	Mating Ring Face Distortion (minutes)	Due to Press.	-0.07	
		Due to Temp.	1.70	
	Net Face Angle (minutes)		1.01	2.43
	Face Temperature (^{o}F)		179	193.5
Optimized Face Design	Primary Ring Face Distortion (minutes)	Due to Press.	-1.89	N/A
		Due to Temp.	1.07	
	Mating Ring Face Distortion (minutes)	Due to Press.	-0.07	
		Due to Temp.	1.30	
	Net Face Angle (minutes)		0.41	0.57
	Face Temperature (^{o}F)		153.5	157
Negative value indicates divergent faces whereas positive value indicates convergent faces.				

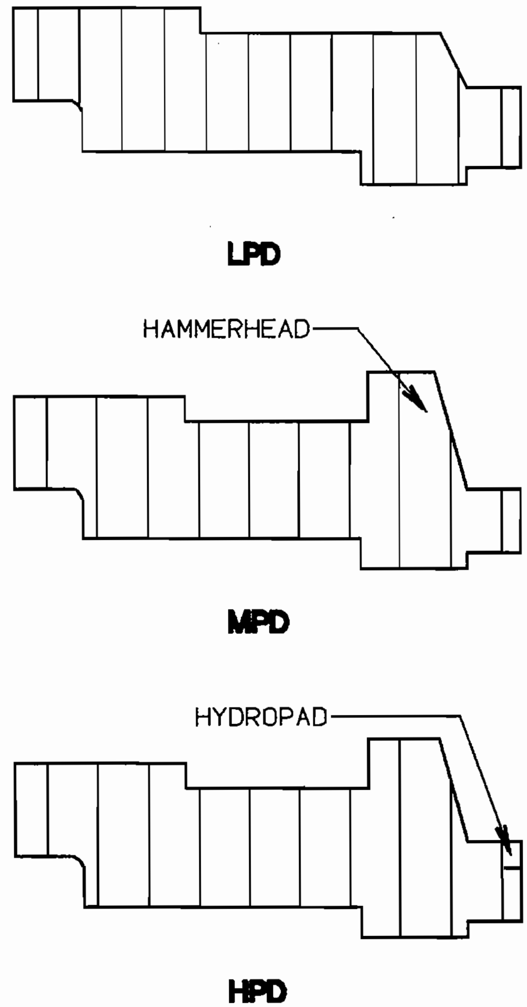

Figure 2. Type A Primary Ring Geometries.

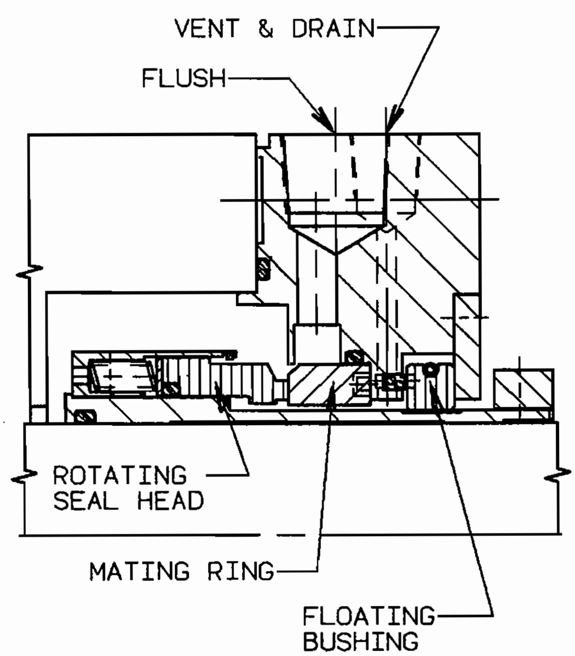

Figure 3. Type A Single Cartridge Seal Arrangement.

Dual Pusher Seals

The dual pressurized Type A Arrangement 3 seal (Figure 4) has an additional step in the sleeve for the inboard seal that allows it to operate with ID pressure without unloading the springs. The seal is double balanced, due to the shift in the balance diameter, so it can also operate with OD pressure. Depending upon the size and service conditions, the seal can be utilized as either an Arrangement 2 dual unpressurized or Arrangement 3 dual pressurized seal. When used in an Arrangement 2, it has lower OD pressure limitations than a standard Arrangement 2 seal, as the deeper primary ring O-ring counterbore results in a higher divergent face condition, from pressure distortion. The mating ring used is an "L" shaped design. It is hydraulically retained against the gland support surface with either ID or OD pressure. Due to the balance shift on the inboard seal, there is always a net positive thrust load on the ring, due to slight differences between the mating ring and primary ring O-ring sealing diameters.

QUALIFICATION TESTING

Overview

Qualification testing was conducted in two separate locations. Propane testing was conducted in the United States and hot oil testing was conducted in France. Qualification testing on propane was conducted on a rig that has two test chambers with a central chamber between them (Figure 5). The seal chamber dimensions on the rig conform to API 682. Cooling flow can enter each chamber individually and then is mixed in the center chamber to return to the reservoir. While flow can be introduced into the center chamber, this feature was not utilized during any of the Type A seals tested in propane.

Product temperatures, pressures, flows, along with other seal performance parameters are monitored once every second by programmable logic controllers (PLC). These data are then utilized to provide near real time performance trends while the seals are undergoing tests. Safety limits are also set for critical parameters that automatically shut the test down should preset limits be exceeded. During a qualification test period, over 80,000 data points are collected for each parameter monitored. This data is then clipped and converted into a spreadsheet format. Dynamic and static results are converted to 10 min intervals and cyclic data to five second intervals.

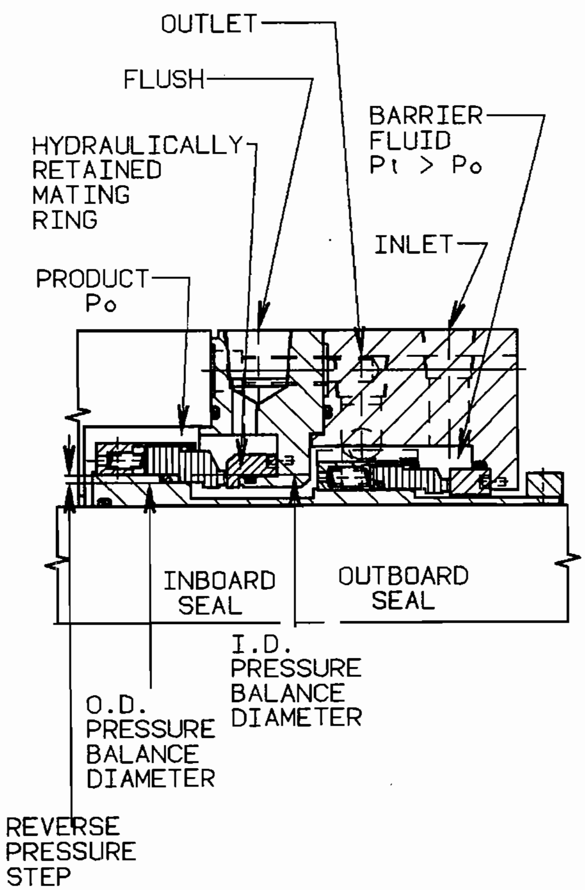

Figure 4. Type A Dual Pressurized Seal Arrangement.

It is a requirement that seals undergoing qualification testing be run, as a minimum, at the maximum allowable angular and radial misalignment. Per the standard, the maximum allowable angular pump misalignment is 0.0005 in/in of seal bore and the maximum radial misalignment is 0.005 in. The seals tested met or exceeded the standard's requirements.

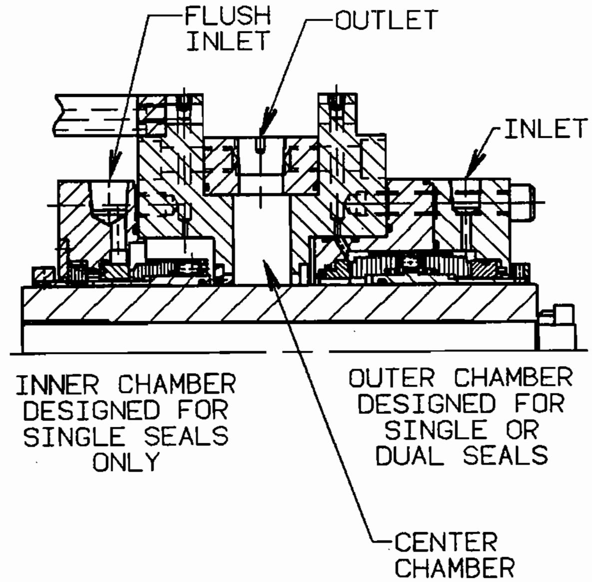

Figure 5. Test Arrangement for Single and Dual Seals Undergoing Propane Test.

Seal emissions in light hydrocarbon service is monitored using EPA Method 21 with a base mounted flame ionization analyzer. These values, while sometimes termed leakage, are actually screening values, indicating a hydrocarbon concentration. The values reported herein are net values. Background readings have generally been in the vicinity of 40 to 60 ppm, which is not unusual in an enclosed cell. There are over 40 potential leak sources from various packed valves, flanged connections, and pressurizing or circulating pumps in each test cell. The instrumentation is NIST traceable and is calibrated on a quarterly basis to maintain accuracy. Calibration is done with certified gases at zero and 1000 ppm to verify span and with 500 ppm gas to verify linearity.

Recording of the test results must be done at a minimum of eight points. Data are recorded at the beginning and end of the dynamic and static test portion. The fifth recording point is at the base point condition near the start of the first cycle. The last set of recording points are at the end of the fifth "no flush" cycle, at base point conditions just prior to stopping the dynamic portion of the cycle test and finally at the end of the 10 min static portion to end the test. These data can be used as comparisons of various designs. These show the user if the seal is stable during the steady state operation and if the seal recovers after undergoing upset conditions that can occur in refinery operation.

Single Seal Test Results

Qualification testing was conducted on Type A designs described. The dynamic, static, and cyclic test results of the 4.0 in LPD seals are shown in Figure 6. The results of the MPD seals exhibited similar results. This is due to the fact that at a test pressure of 250 psi, the net distortions for both designs are fairly similar. Leakage of the HPD seals during steady state operation was similar to the other designs, but was higher throughout the cycle test phase. The results of the 2.0 in HPD seal are shown in Figure 7. In the static mode following the dynamic test, seal leakage rose, as the thermal distortion of the faces was no longer present. With only mechanical distortion present on the primary ring, a divergent face condition exists, allowing some leakage past the hydropads.

The test results of the various designs are shown in Table 5. The results shown in Figures 6 and 7 are indicative of the other seal sizes tested. These show that the plain face seals, LPD and MPD, were more tolerant of upset conditions during the test, having

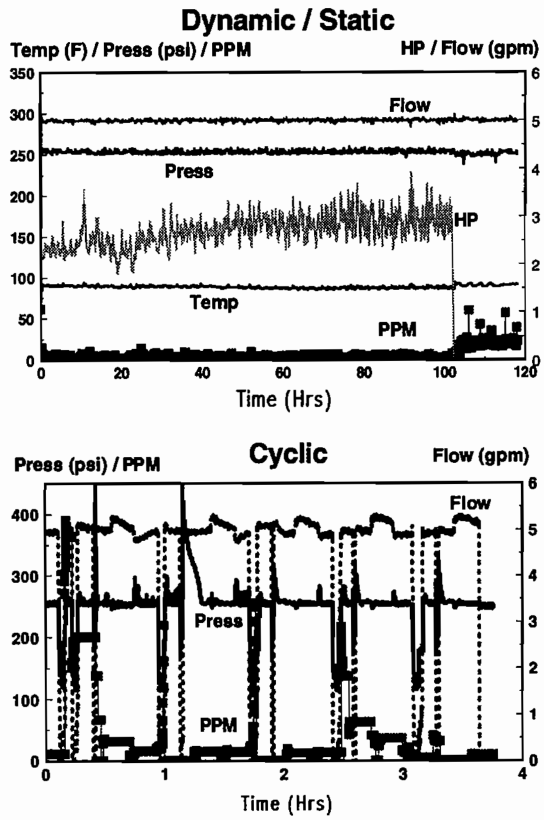

Figure 6. Dynamic and Cyclic Test Results for 4.0 In Type A LPD Design.

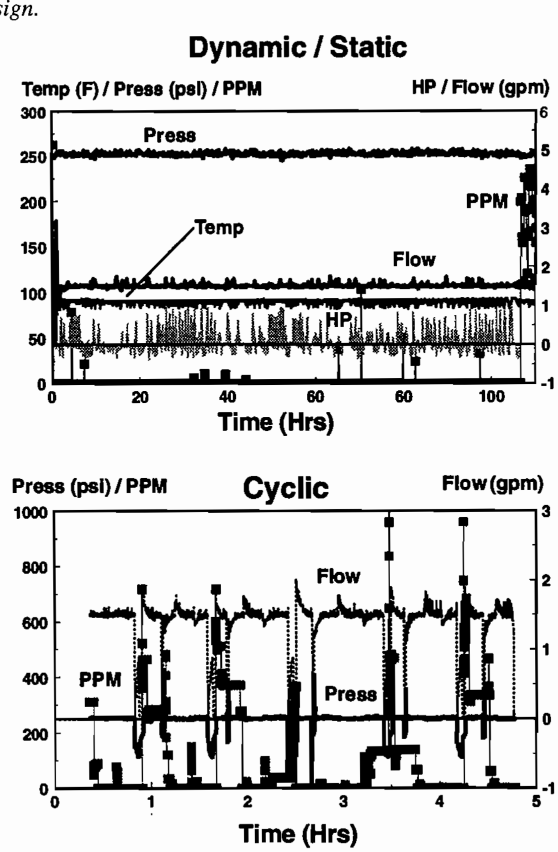

Figure 7. Dynamic and Cyclic Test Results for 2.0 In Type A HPD Design.

lower peak leakages than the HPD seal. In all cases, the seals recovered between upset conditions and had static leakage rates less than 50 ppm at the end of the test. Examination of seal face temperatures, during vaporization and no flow conditions in the cyclic phase, showed full recovery to preupset temperatures in 30 sec to 1.0 min. The main differences between the plain face and hydropaded face designs are power generated and wear. While a minimum operating life of three years can be projected for the plain face designs, the wear on the HPD primary rings was considerably less, as shown in Table 5. In all cases, the mating rings had less than 10 μin deep wear tracks. On the plain face designs, there was some local grooving at the primary ring face extremes (≈20 μin). The post test talysurf traces of the 2.0 in HPD primary and mating ring faces are shown in Figure 8.

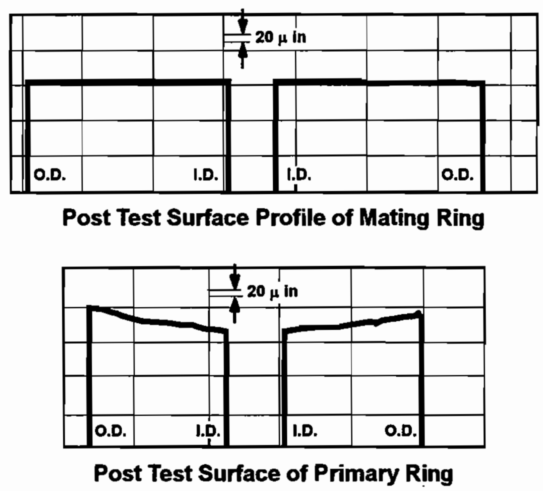

Figure 8. Surface Trace of Carbon Primary Ring and Silicon Carbide Mating Ring of Type A HPD Design after Complete API 682 Qualification Test.

Table 5. Single Type A Seal Propane Qualification Test Results.

Seal Type	2" LPD	4" LPD	2" MPD	4" MPD	2" HPD	4" HPD
Flow Rate (GPM)	2.0	5.0	3.5	5.0	1.5	5.0
Flush ΔT (oF)	7.3	8.9	7.8	8.9	1.7	5.2
Avg. Face Temperature (oF)	156	162	170	167	101	105
Avg. Dynamic Leakage (PPM)	25	<10	20	25	<10	15
Average Power (Hp)	0.8	2.3	1.2	1.7	0.22	0.8
Primary Ring Wear (μin)	<0.0005	<0.0005	<0.0005	<0.0005	<0.0001	<0.0003
Mating Ring Wear (μin)	<10	<10	<10	<10	nil	nil

The data shown in Figures 6 and 7 as noted earlier are snapshots in time. The horsepower values shown are calculated based upon temperature rise and flow of propane through the seal chamber. Variations of regulated flow, thermocouple accuracy and to a large extent system cooling operation (on/off) influence the calculated value. Net torque was not used to define horsepower, as this would be an average of the two seals installed on the test rig.

Stability Tests

Equipment problems and loss of electrical power voided some qualification tests, but provided additional data and an opportunity to run additional tests. Buck [4] reported that most seal failures were not caused by wearout and suggested that instability leading to increased leakage was a major cause of failure. Using seals from the voided tests, operating conditions were modified to 125 psi and 45°F to evaluate seal performance after a wear pattern had been established. An FEA analysis at the reduced conditions showed the combined pressure distortion reduced by 47 percent while the thermal distortion reduced by only 25 percent. The new net distortion was 0.73 min, compared to 0.41 min shown in Table 4 for the optimized design. Thus, the seal would operate at a larger converging sealing gap.

On one 4.0 in LPD test, power was lost after 98+ hr of dynamic operation. Upon disassembly the wear was less than 0.0004 in on both primary rings. After the standard cycle test was run the test rig was shutoff for a period of 2.0 hr. The stability test was then run at the reduced operating conditions. The results of this test are shown in Figure 9. The seal performed well with very limited leakage. The cycle test, along with the lower pressure stability tests, resulted in approximately 0.0001 in. of primary ring wear.

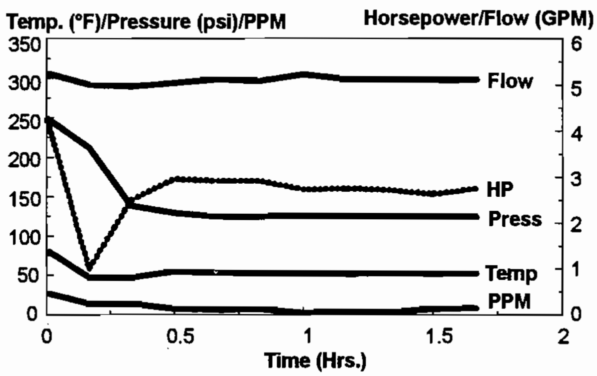

Figure 9. 4.0 In Type A LPD Design Stbility Test Results in Propane.

In a similar situation, a 2.0 in LPD design seal was stopped just short of a 100 hr dynamic running. Upon disassembly and inspection, the wear was measured at less than 0.0005 in. The seal was reinstalled and run through the cycle test. Starting from an ambient condition, the stability test was run for approximately 3.0 hr. The results of this test were similar to the 4.0 in. seal. Upon restarting the test, leakage rose slightly above 100 ppm, but stabilized to approximately 20 ppm within 10 min. Primary ring wear for the cycle test portion and the stabilization test was again on the order of 0.0001 in.

These two tests show that even beyond being able to withstand a number of upsets within a short period of 3.0 hr during cycle testing, that the seal can operate successfully when run at varying pressure conditions that seals can be exposed to in refinery operation. In both cases, the propane temperature was lowered to maintain roughly the same vapor point margin between the two operating pressures, 250 and 125 psig. Attempting to run at the low pressure at a higher temperature would have resulted in a vapor condition throughout the entire circulation loop.

Dual Type A Seal Qualification Testing

The dual pressurized seals tested in propane were 2.5/2.0 in and 4.5/4.0 in inboard/outboard. The reservoir was an API 682 unit that was positioned according to the guidelines in Standard 682, except that the axial location was approximately eight feet away from the test chamber to facilitate maintenance in the test cell. The standard recommends against using nitrogen to pressurize barrier fluids above 150 psi as gas entrainment could result. The tests were run in this condition as a worst case scenario. Upon draining the system after the tests, gas entrainment was noticed in the barrier fluid. However, this condition did not affect performance or wear of the seals. In the tests, the inboard seal of the cartridge was tested as a single seal on the inner test chamber while the complete dual cartridge assembly was tested on the outer end, Figure 5. The barrier fluid was kerosene with a viscosity of 35 SSU, similar to a #2D diesel. The barrier fluid pressure was set at 275 psig, 25 psi above the propane pressure. Flow of the barrier fluid was measured by an NIST traceable ultrasonic flow meter. This type of measuring device was selected as it provides a minimal pressure drop in the system.

A summary of the Type A seal test results is shown in Table 6. The seal face temperature of the inboard seal run as a single seal, were similar to the single LPD designs. Single seal leakages were both under 20 ppm during steady state dynamic testing and did not exceed 125/300 ppm, respectively, during the four hour static test. Face wear and profiles were comparable to the single LPD designs. In the cycle test phase, peak leakage values did not exceed 250 ppm throughout the tests with average leakage rates between upsets staying below 50 ppm.

Table 6. Dual Pressurized Type A Propane Qualification Test Results.

Seal Type	2" Type A Dual Press.	2" Type A Dual Press.	4" Type A Dual Press.	4" Type A Dual Press.
Arrangement	1	3	1	3
Test Fluid	Propane	Propane / Kerosene	Propane	Propane / Kerosene
Flow Rate Inbd / Outbd (GPM)	2.0	2.0 / 1.56	5.0	4.9 / 0.78
Flush ΔT (°F)	7.5	7.5	9.6	9.6
Avg. Face Temperature (°F)	160	122	166	142
Avg. Barrier Temp / ΔT (°F)	N/A	100 / 7	N/A	140 / 40
Avg. Dynamic Leakage	< 10 PPM	< 1 PPM	10 PPM	< 1 PPM
Avg. Seal HP	1.2	1.4	2.3	2.3
Comments	Inbd. Sgl. Seal	Dual Cartridge Inbd. Face Temp.	Inbd. Sgl. Seal	Dual Cartridge Inbd. Face Temp.

The dual cartridges exhibited near zero emissions during the dynamic tests. The inboard seals had no measurable wear (within instrument accuracy) on either the primary or mating ring face. The outboard seals, running in the barrier fluid, had less than 0.00015 in of wear on the carbon face and no wear or grooving on the mating ring.

The complete dual cartridge seals never exceeded 100 ppm during the cycle test. The leakage of the outboard seal is a combination of propane leakage from the inboard seal and in part emissions from the barrier fluid. The barrier fluid used was not a deodorized kerosene. The kerosene used contains some light ends that will be detected by the vapor analyzer. Using a nonhydrocarbon barrier fluid would reduce fugitive emissions to still lower levels. No visible outboard seal leakage was noted on either dual seal during the test.

The dual outboard seal arrangement was tested extensively without the aid of a separate pumping ring. Testing in water and kerosene on both 2.0 in and 4.0 in seals at 3600 rpm and the specified 0.125 in radial clearance produced sufficient flowrates to cool either a dual unpressurized or dual pressurized seal within the guidelines of the qualification testing. The 2.0 in seal produced an average of 1.56 gpm, while the 4.0 in seal produced an average of 0.78 gpm in kerosene. In a separate test in water, the seal produced 1.1 gpm in a flow circuit having reduced system resistance. The pumping of the barrier/buffer fluid is enhanced by the indentations in the seal retainer along with a large diameter tangential outlet in the gland. It is theorized that the higher flowrate from the smaller seal assembly is due to the reduced turbulence in the seal chamber and the land/width ratio of the indentations. For higher temperature and/or pressure services and applications involving a higher viscosity fluid a pumping ring is recommended. In these cases, the pumping ring will be needed to overcome the additional system resistance from the higher viscosity fluids.

TYPE C DUAL METAL BELLOWS SEAL DESIGN

The dual unpressurized and dual pressurized Type C metal bellows seal (Figure 10) is a rotating seal that was designed to be utilized in either pressure condition. The pressure area shift of the metal bellows capsule allows the seal to operate in this mode as the balance of the seal shifts with changes in pressure from either direction. The inboard and outboard seals are designed with different primary rings for the types of conditions they are exposed to. The inner seal has a small retention lip on the front adaptor that retains the ring from coming out under extreme ID pressure conditions. The outboard seal is a shortened version of existing metal bellows seal technology. It uses an axial flow pumping ring to provide flow for the buffer/barrier fluid in the chamber.

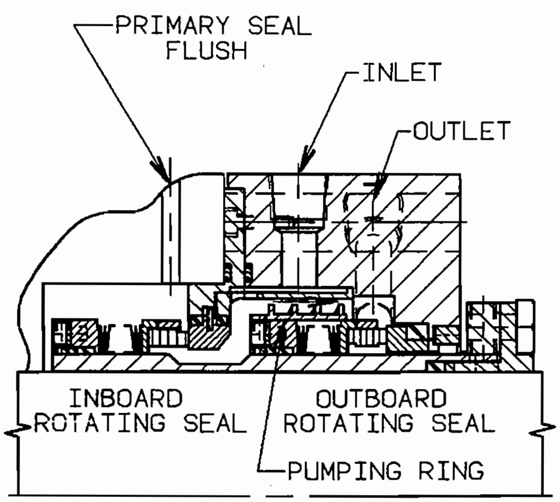

Figure 10. Type C Dual Pressurized/Unpessurized Cartridge Arrangement.

The retention of the inboard mating ring is critical in the new design. Unlike an O-ring used in pusher seal designs, the flexible graphite secondary seals used in the Type C seal, limits the type of mating ring mounting that can be utilized. API 682 discourages the use of clamped mating rings unless specifically approved by the user. In the past, typical clamped mating rings used thin, 0.062 in. to 0.125 in. gaskets to seal the upper section of the mating ring. The associated gland plates were designed to fit over the register fit of the seal chamber. Due to the slight amount of allowable compression for the thin gaskets along with the tolerance stackup between the gaskets, mating ring, and the length of the seal chamber lip, the gland was designed not to have metal-to-metal contact with the pump case. This resulted in distortion of the gland plate, especially in the location of the bolting. This distortion was transmitted to the mating ring, resulting in waviness and a change in slope around the circumference of the face. The axially retained mating ring used in the dual Type C design uses longer flexible graphite secondary seals with controlled compression loading of the gaskets that greatly reduces the distortion of the earlier designs.

To verify the integrity of the mating ring design, a fixture (Figure 11) was made to simulate the mating ring installation. The mating ring face was measured in its free state and again in its final assembled position. In addition, the fixture allowed for measurements of the ring under ID pressure conditions to verify that pressure distortion was minimal. In a free state, the mating ring was almost perfectly flat with a waviness of 4.0 µin. In an assembled condition the slope changed to one helium lightband (HLB) concave with an average waviness of 14 µin. Next, the mating ring was pressurized to 300 psig. In this condition, the flatness of the ring was less than 0.5 HLB concave.

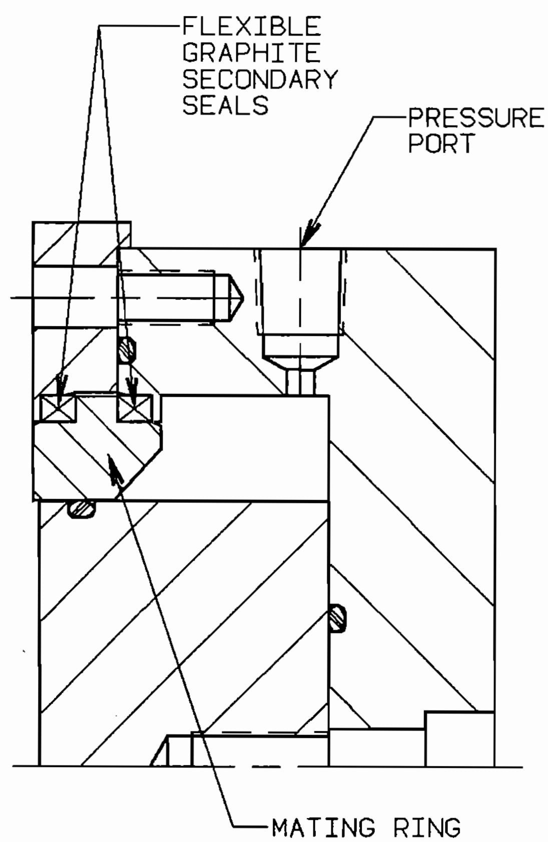

Figure 11. Type C Mating Ring Test Fixture.

Dual Type C Qualification Testing

Dual metal bellows seals were tested in a high temperature hydrogenated terphenyl. The conditions for this test are a base point temperature of 500°F and pressure of 100 psig, with temperature cycling down to 300°F and pressure cycling up to 250 psig. The original test rig was designed with a foot mounted seal chamber separate from the bearing frame. The 2.0 in seal tested with this arrangement had a concentricity and sleeve runout of 0.004 in. The squareness of the seal chamber face to the shaft was 0.0067 in, which exceeded API 682's specification of 0.002 in by a factor of over three. To supplement the cooling capacity of the five liter reservoir used, a heat exchanger was placed in the barrier fluid circuit. With the additional resistance from the heat exchanger, the barrier fluid flowrate was 0.5 gpm.

The face material combination was tungsten carbide *vs* reaction bonded silicon carbide for both the inner and outer seal assemblies. The leakage for the inboard seal is calculated by subtracting the outboard seal leakage from the reservoir volume change over a particular test sequence. The average leakage for the inboard seal was 3.5 cc/hr. The outer seal started with a leakage rate of 2.5 cc/hr, but was rapidly reducing in the first quarter of the dynamic test to a final value of 0.5 cc/hr. Due to the limited visible level in the reservoir, further inboard leakage measurements were not possible during static and cyclic test phases. With the heat exchanger in the barrier fluid circuit, the average temperature was low, but the flow was restricted, which resulted in a high temperature differential. Post test inspection showed no wear on the mating ring, but the inner surface of the inboard mating ring visually showed a more distinct wear pattern.

The second test involved running the inboard seal as a single seal. The results of this test with a higher pressure differential improved significantly. Dynamic leakage averaged only 0.1 cc/hr, with average static and cyclic test phase leakages of 0.008 and 0.12 cc/hr, respectively. The improved seal leakage is attributed to the higher closing forces at 100 psi. Post test examination of the seal faces again showed no wear and only a small amount of coked oil was noted on the mating ring ID.

The seal chamber squareness was considered to be a problem, especially on the inboard seal of the complete cartridge, that runs with a normal pressure differential of only 25 psi. The test rig was modified to incorporate an adaptor to mount the seal directly off of the bearing frame. In addition, a modification was made to the mating ring. To improve the interface temperature of the inboard seal, a large chamfer was added to the inner back surface of the mating ring to expose more surface area to the cooler barrier fluid.

With the new seal chamber mounting, the housing squareness is 0.0027 in and the shaft runout is 0.0016 in. The installed seal chamber concentricity is 0.003 in. The modifications made to reduce the test rig runouts and the modifications to the mating ring made a significant difference on the leakage performance of the seal. The complete dual Type C seal was retested. In this test, the heat exchanger was removed from the barrier fluid circuit. This raised the average barrier temperature by almost 100°F, but lowered the temperature rise by a factor of two. Post test examination showed a full contact pattern on both faces. Also, there was no depth to the wear track on either mating ring. A retest of the inboard seal, as a single seal, had almost identical results. The leakage values were similar to the previous test with a slight decrease in horsepower being the only significant change. Again, the only noticeable face condition was a slight coke buildup on the mating ring face ID. A summary of the Type C test results, before and after test rig modifications is shown in Table 7.

DUAL GAS SEAL DESIGN

Dual pressurized gas seals using a spiral groove pattern on the mating rings to provide noncontacting operation were also tested to the standard's propane test procedure. These seals are back to back dual pressurized seals that use an inert gas instead of a liquid to pressurize the seals. These seals are not currently covered by the standard and do not conform to the Arrangement 3 configuration.

Table 7. Dual Pressurized Type C Hot Oil Qualification Test Result.

Seal Type	2" Type C Dual Pressure	2" Type C Sgl. Inboard	2" Type C Dual Pressure	2" Type C Sgl. Inboard
Flow Rate Inbd/Outbd (GPM)	0.7 / 0.5	1.3	1.3 / 1.6	1.3
Process ΔT (°F)	16.2	8.8	26.1	12
Barrier Fluid Temp/ ΔT (°F)	148 / 31	N/A	266 / 11	N/A
Average Inboard Leakage Dyn/Stat/Cycle (cc/hr)	3.5 / N/A / N/A	0.1 / 0.008 / 0.12	0 / 0 / 0	0.06 / 0 / 0.1
Average Outboard Leakage Dyn/Stat/Cycle (cc/hr)	1.0 / 1.2 / 0.8	N/A	0.014 / 0 / 0.01	N/A
Average Power (Hp)	2.4	0.62	2.6	0.57
Comments	Excessive Runout	Pre Rig Modifications	After Rig Modifications	After Rig Modifications

The test arrangement used is shown in Figure 5. The inboard seal on the test rig was a single Type A LPD design, while the dual gas seals tested were located on the outboard end of the test rig. Pressure and thermal rotations for these seals are critical in order to maintain a noncontacting mode of operation. The API 682 test pressures for propane exceed the carbon primary ring limitations. Therefore, the dual gas seal was put through a pseudo qualification test at slightly lower pressures and temperatures. Since these types of seals are not currently covered by API 682, it was not the intent to qualify the seal per the standard, but instead provide information for future consideration of dual gas seals into API standards. The design discussed above is patent protected.

Pressure Reversal Test

Prior to the qualification testing a customer specific test was conducted on a 2.0 in dual gas seal to ascertain how the seal performed when exposed to a loss of barrier pressure. The face materials were carbon for the primary ring *vs* a silicon carbide mating ring. In this test, the process was propane at 175 psig and 70°F with room temperature nitrogen at 200 psig. Shaft speed was 3600 rpm. After a 24 hr run at steady state conditions the nitrogen supply was shut off to the seal. In the first test sequence, the seal chamber was isolated, so that only five ft of 0.250 in tubing and the volume in the dual gas seal chamber was pressurized. It took only 5.0 min to bleed down the barrier pressure from 200 psig to 175 psig, matching the propane pressure. In the second sequence, a 500 cu in reservoir was teed into the line to provide additional gas volume. In this sequence, it took 79 min for the barrier pressure to bleed down to the propane pressure. The seal was disassembled and no face contact was observed, although the inboard seal was run in excess of 20 min with less than a 5.0 psi differential. In a subsequent test the same setup was run, but with propane at 250 psig and no reservoir. During this sequence, the barrier pressure was shutoff and allowed to drop over a period of one hour. Again, within five min the propane and nitrogen pressures equalized. The barrier pressure was allowed to drop over the remaining 55 min. At the end of one hour, the barrier pressure was 75 psi lower than the propane pressure of

250 psi. Upon restoring the nitrogen pressure, dynamic seal performance was unchanged. Examination of the faces did show contact, but only a slight scuffing was noticed. These tests show that the seal can withstand temporary losses of barrier pressure. They also show that there is a short period of time between loss of barrier pressure and equalization of pressures across the inboard seal. This time can be extended if the volume of barrier gas is increased by a suitable gas reservoir.

Dual Gas Seal Qualification Test

In this test, the propane pressure was set at 175 psig and the temperature was lowered to 70°F. This was done to maintain a suitable boiling point margin in the circulation loop. Shaft speed was 3600 rpm. The barrier gas was dry nitrogen set at 200 psig. The throat bushing was removed from the test chamber and propane was circulated through the center chamber at a rate of 2.0 gpm. This was done to simulate pump conditions especially during the cyclic phase of the test.

The dynamic steady state seal leakage of propane emissions was zero. The dual gas seal chamber temperature averaged 74°F during this period. The flowrate of nitrogen into the dual gas seal chamber was a constant 0.0098 scfm. Leakage was measured as flow going into the dual gas seal chamber and was corrected for pressure conditions. There was no leakage of propane by the seal during the static test phase. During the cycle test, the inboard seal was exposed to pressure reversals up to 100 psi. In order to vaporize the propane, block valves isolate the test rig and pressure is reduced to achieve a pressure-temperature relationship to completely vaporize the propane. When pressure and flow is reestablished to the test rig, momentary pressure spikes can occur. In this reverse pressure condition propane did leak into the dual gas seal chamber. When allowed to run in a steady state condition after being subjected to the pressure reversal, fugitive emission leakage dropped from 170 ppm to zero within 8.0 min. Throughout the cycle test, leakage did not exceed 500 ppm. Post test examination of the seal faces showed no measurable wear and only slight face scuffing.

USER CASE HISTORIES

The Type A seal and the dual gas seal have been in existence prior to the existence of API 682. The dual pressurized Type A seal is relatively new, especially in the configuration discussed herein. Some brief case histories of refinery applications follow where these technologies have been applied as solutions to difficult applications.

A southeastern refinery was having difficulty sealing a 20 percent methyl ethyl amine service in an overhung API pump. The conditions are a suction pressure of 10 psig, discharge of 240 psig, product temperature of 179°F, and a shaft speed of 3550 rpm. The previous seals installed in this service were both pusher and metal bellows seals from various manufacturers. Seal life was typically less than six months. A single Type A seal was installed using a chemically resistant carbon *vs* self sintered silicon carbide along with perfluoroelatomer secondary seals. A steam quench was to be installed in between the seal and floating bushing. The seals have been running in excess of two years without the steam quench, which was never hooked up. In total, six other pumps have been converted to the same arrangement.

In the same refinery an overhung API pump had sporadic service with a single pusher seal. The service conditions are a light straight run, a light hydrocarbon with a specific gravity of 0.48 at 326°F. The vapor pressure at pumping temperature is 210 psia. The remaining service conditions are a suction pressure of 240 psig, discharge pressure of 315 psig, and a shaft speed of 3500 rpm. A 3.500/3.250 in Type A dual unpressurized seal was installed with carbon *vs* self sintered silicon carbide using a Plan 21/52, where the product is cooled down to approximately 210°F. The seal has been in service for close to three years without interruption.

A western refinery had a double ended horizontal split case pump in an HF alkylation unit. The initial seals were tandem pusher seals using an Plan 54 alkylate flush. The flush arrangement on the suction side was sufficient, but the discharge end had problems as the seal chamber pressure was at a higher pressure. This sometimes allowed HF acid into the outer seal reservoir. The flush plan was changed to isobutane at the same time the seals were changed out to a 3.875/3.375 in Type A dual pressurized arrangement. The isobutane flush flows through the outer seal cavity and then through an orifice, check valve and rotometer into the inner seal chamber at 250 psi and 3.5 to 4.0 gpm, where the pressure drops to seal chamber pressure. In the same refinery, five overhung API pumps with single seals were converted to the same arrangement at about the same time. All of the pumps converted have been performing since late February or early March 1995 without any visible outboard leakage.

CONCLUSION

A number of single and dual seals have been successfully qualified per API 682 requirements. Prior to qualifying these designs, seal flush arrangements, face materials and seal face geometry evaluation tests were conducted. The results of these tests were used to optimize the designs. The modifications to the qualified designs were used to modify and update the line of Type A seals.

Type A seal designs were verified using finite element analysis. The input for this analysis was guided by comparisons to preliminary evaluation tests. Using this method, the analysis and the final test results had good correlation. This tool was also used to establish design integrity on intermediate sizes.

Three different single Type A seals, designed for progressively higher pressure services, all provided satisfactory performance in API 682 propane testing. The HPD design utilizing hydropads to enhance face lubrication provided significantly lower power consumption and wear than seals with plain face geometry. The leakage for the HPD design was slightly higher than the plain face geometry seals, but was well under current emission standards. The performance and integrity of the LPD design was demonstrated with stability testing of the design at an intermediate pressure after nearly 100 hr of dynamic and subsequent cycle testing.

Pressurized dual seals were designed and tested in both Type A and Type C configurations per API 682. The hydraulically retained mating ring in the Type A dual configuration ran well in a pressurized dual arrangement with face temperatures averaging only 30°F to 45°F higher than process temperature. When run as a single seal, the inboard seal ran almost as well as the single Type A design without using a distributed flush arrangement. The axially retained mating ring for the dual Type C seal was tested and proven to have limited distortion in the installation, and when subjected to 300 psig. The design integrity was confirmed by dynamic testing. Both Type A and Type C inboard seals tested showed that they are capable of running successfully when pressurized by either the product or barrier fluids. Also, both dual designs provided near zero emissions under steady state and cyclic conditions.

Finally, noncontacting dual gas seals were tested in pseudo API 682 testing in a flashing hydrocarbon. Based upon this test and a large number of successful field installations, this technology can be used to provide zero emissions, while also providing extended life over conventional liquid lubricated seals. The seal was also exposed to a loss of nitrogen barrier pressure and pressure reversals with fully recovery and no change in subsequent dynamic performance.

NOMENCLATURE

hp	horsepower
PV	Pressure-Velocity (psi x fpm)
μ	Coefficient of friction (dimensionless)
A_o	Face area (in^2)

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