IMPROVEMENT OF A HIGH TEMPERATURE BELLOWS SEAL

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ABSTRACT

An Inconel 718 metal bellows seal has been developed to meet the demand for high temperature applications in corrosive media such as high sulphur bearing crude oil. Finite element analysis was used to develop concepts having low seal face distortion at high temperatures and to minimize bellows stress. Laboratory testing at temperatures to 700°F demonstrates that the final seal design exhibits both low leakage and low wear.

A thick plate bellows construction provides high resistance to corrosion, fatigue, and pressure stress. The Inconel 718 bellows was evaluated for resistance to shear and fatigue failure in dry running tests using silicon carbide *vs* silicon carbide faces. This resulted in heavy damage to the silicon carbide faces, but in every test the Inconel bellows were intact and helium leak tight afterwards. The seal has a sufficient number of diaphragms to accommodate seal setting tolerance and shaft axial growth during thermal warm up of a pump.

INTRODUCTION

For many years AM-350 was the standard bellows material for high temperature bellows seals. AM-350 has high strength at elevated temperatures, but low corrosion resistance. The increasing production of sour crude oil, containing H_2S, requires alloys that are resistant to sulfide stress corrosion. Seal manufacturers have introduced Hastelloy C and Inconel 718 bellows to handle hot hydrocarbon services. Both of these materials have excellent corrosion resistance and good performance histories in high temperature bellows seals. Many users are specifying only Inconel 718, per API Standard 682 [1], for high temperature applications due to its high mechanical strength and resistance to stress corrosion at elevated temperatures. Inconel 718 is more expensive and requires strict cleaning and handling procedures to avoid oxide contamination during welding.

In addition to specifying Inconel 718 bellows, seal users are demanding longer seal life under a wider range of application conditions. Also, local and national government regulations require strict limits on leakage of volatile organic compounds.

A seal manufacturer's approach to upgrading a high temperature metal bellows seal is presented. First, the application window (temperature, pressure, etc.) and dimensional criteria are established. Second, is a review of alloys used in high temperature bellows. Next, bellows design is considered. Then, face assemblies that result in low distortion at high temperatures are designed. Finally, laboratory tests at temperatures to 700°F are run to demonstrate the final seal design exhibits low wear and low leakage. Also, dry running tests using SiC *vs* SiC faces are used to show that the Inconel bellows are resistant to fatigue failure.

APPLICATION AND DESIGN CONSIDERATIONS

Before a mechanical seal product line can be upgraded and redesigned in a product improvement program, certain basic information must be collected to allow specification of performance criteria and hardware dimensional constraints.

Performance

The bellows seal, which is the subject of this discussion was developed primarily for high temperature services in a refinery environment. The seal must meet the specifications of API 610 [2] and API Seal Standard 682 [1] for high temperature seals. API 682 also requires a specific performance test to qualify the overall seal design. The qualification test includes dynamic, static, and cyclic phases.

Temperature

Based on input from customers and application histories in the refining industry, the upper operating limit for a high temperature seal should be placed at 800°F. This limit represents the likely highest temperature that petroleum products are pumped in a refining process. Although in most applications the seal chamber temperature will be lower than the pumpage (due to flush cooling and/or pump water jacket), the seal design must be capable of handling the maximum expected temperature in the pump stream.

An arbitrary lower temperature limit was chosen at -100°F. The seal metal and gasket materials must be capable of reliable service at this temperature.

Pressure Range

Experience in high temperature sealing plus input from user personnel indicate that few high temperature API applications exceed 300 psi. Most installations have a seal chamber pressure between 50 and 150 psi. Based on this input, a high temperature seal should provide good performance to a pressure of at least 300 psi at maximum operating temperature.

Speed

Seals with stationary (nonrotating) spring systems (pusher or bellows type) are generally rated for higher speed limits than seals with rotating spring systems. The stationary bellows seal face can readily deflect to a fixed position to compensate for out of square seal chamber face. A rotating bellows arrangement, on the other hand, must flex once per revolution to accommodate out of perpendicularity of the flange mounted stationary face. Seals with rotating flexible elements are generally limited to surface speeds less than 75 ft/sec, while seals with stationary flexible elements are often rated above 150 ft/sec.

A stationary bellows seal with a maximum surface speed of 150 ft/sec should cover nearly all high temperature services. For instance, a 5.0 in seal turning at 5,200 rpm has a surface speed of 113 ft/sec. Also, a stationary bellows readily accepts a steam baffle for anticoking protection, whereas a rotating bellows does not.

Materials

Selection of materials to be offered in a mechanical seal product line are influenced by industry standards, customer requests, and seal manufacturer R&D programs. For instance, API 682 [1] specifies Inconel 718 bellows for refinery applications above 300°F.

Materials selected for the high temperature seal discussed in this paper are:

- Stationary face: Specific grade of carbon graphite (or silicon carbide for abrasive services).
- Rotating face: Reaction bonded silicon carbide.
- Bellows: Inconel 718.
- Bellows flange: Low expansion alloy. This assures shrink fit retention of the stationary face to temperatures well above 800°F.
- Balance of metal seal parts: 316 stainless steel.
- Gaskets: Flexible graphite.

Field experience shows these materials work well in high temperature applications.

Process Liquids

High temperature seals should provide good sealing on hydrocarbons, heat transfer fluids, and cryogenics. Fluids should be nonflashing (vapor pressure lower than one atmosphere at pumping temperature). The seal must have sufficient corrosion resistance to handle high sulphur crude, or streams with traces of chlorides, H_2S, and naphthenic acids.

Service Window

The operating window for the upgraded seal is contained in Table 1.

Dimensional Requirements

The original seal configuration with the Hastelloy bellows is easily fitted to most overhung process and multistage double

Table 1. Seal Service Window.

Temperature	-100 to 800°F
Pressure	0 to 300 psi
Max surface speed	150 ft/sec
Fluids	Hydrocarbons, heat transfer fluids, trace (3%) H_2S

ended high temperature pumps. It was designed for economy of space in all areas: bellows span and length; length of bellows end terminals; length, diameter, and cross section of face assemblies. Component studies show that further reductions in cross sections or lengths results in degraded performance. The new ANSI [3] and API 682 [1] standards specify seal chambers with larger radial cross sections. These standards, however, do not provide for any increase in axial length of the seal chamber.

A review of installation data revealed that there are over a thousand pumps using the original Hastelloy bellows high temperature seal. Since most of these existing pumps have limited radial space, it was decided that the upgraded seal must be fully interchangeable with the original seal. Thus, the axial length and radial cross section for both the stationary and rotating assemblies were designed to fit within the original seal dimensional window.

BELLOWS MATERIALS

Three materials commonly used for high temperature bellows are: AM-350, Hastelloy C-276, and Inconel 718.

AM-350

AM-350 has high tensile (185,000 psi) and yield (150,000 psi) strengths, comparable to Inconel 718. AM-350, however, is susceptible to stress corrosion cracking in services containing trace amounts of H_2S. Furthermore, AM-350 is relatively brittle, with an elongation to fracture of about 9.5 percent at 800°F. Over pressurization of the bellows, or stick slip operation in marginal lubrication conditions, can cause cracking of the diaphragms.

Hastelloy C-276

Hastelloy C-276 has excellent corrosion resistance, but has a low yield strength of only about 33,000 psi at 800°F. It has good ductility, with an elongation to failure of about 60 percent. This makes Hastelloy C-276 difficult to break, as it bends and relieves internal stresses to a large extent before cracking. Hastelloy C is a good choice in applications where stick slip may occur, such as in flashing fluid services or when product lubricity is low. Its low yield strength at elevated temperature may result in shortening of the bellows free length of about 15 percent, and a corresponding reduction of spring loading. This compression set effect can be compensated for by designing a low spring rate bellows and using a high installed deflection to insure adequate face loading.

Inconel 718

API 682 [1] specifies that metal bellows shall be Inconel 718 for refinery services above 300°F. Inconel 718 is a nickel chromium alloy with good oxidation resistance up to 1800°F and good corrosion resistance in a wide range of environments. It has high tensile and yield strengths, 180,000 and 150,000 psi, respectively. For mechanical seal bellows, it is normally heat treated per AMS 5596G [4]. Elongation is about 17 percent using this heat treatment.

NACE MR-0175 [5] also approves Inconel 718 bellows for use in sour crude services when it is solution annealed and aged to a maximum hardness of 40 HRC. AMS 5596G heat treatment specifies a minimum hardness of 36 HRC. Due to variations in the alloy composition and its response to the heat treatment cycle, however, actual hardness is typically 45 to 47 HRC. Thus, AMS 5596G heat treatment is likely to result in a hardness higher than the maximum of 40 HRC specified in NACE MR-0175.

Development of the high temperature Inconel 718 bellows seal required modification of the standard heat treat procedure to reduce hardness levels for compliance with NACE. A high temperature solution anneal and single step aging process reduced hardness to 40 HRC. The modified heat treat also increased ductility, by increasing elongation to 21 percent, for higher toughness and resistance to stress corrosion cracking.

Successful welding of Inconel 718 is relatively difficult, since many precautions must be taken to prevent the formation of oxide tails in the weld zones. First the material must be cleaned in a nickel cleaning solution containing hydrofluoric and nitric acids. The diaphragms must then be neutralized and protected from contamination and formation of oxides by storing them in a sealed bag containing an inert gas. Throughout the welding process, all material which may contact the weld areas must be extremely clean. Inert gas shielding is used during welding to exclude oxygen and contaminants from the weld area. Finally, before a production lot is done, a section analysis must pass metallurgical examination for weld quality. A destructive pull test is also performed to ensure weld integrity. A tightly controlled process results in near zero scrap rate.

Manufacturing costs are somewhat higher with Inconel 718, which may be reflected in a higher selling price to the user.

BELLOWS DESIGN CONSIDERATIONS

Performance of a bellows is dependent on material properties, plate thickness, radial span, pitch, number of diaphragms, and plate geometry (ripple pattern). These factors affect bellows spring rate, seal balance ratio, stresses due to pressure and installed deflection, axial compression for face loading, and amount of overtravel to solid height.

While a detailed model of bellows response to pressure, deflection, and temperature requires the use of finite element analysis, simple equations based on beam theory have proven practical in designing bellows for mechanical seal applications.

Balance Diameter

Seal balance diameter for bellows seals is approximately equal to the mean diameter of the diaphragms. Pressure, however, deflects adjacent diaphragms towards each other (Figure 1). Diaphragm to diaphragm contact may occur at a moderate level of pressure, depending on bellows pitch. OD pressurization acts to pinch diaphragms together near the OD and to separate them at the ID. The radial extent of contact increases with pressure. This effect results in a higher closing force on the seal face. Hence, balance ratio for bellows seals tends to increase with pressure. This effect is termed "balance shift."

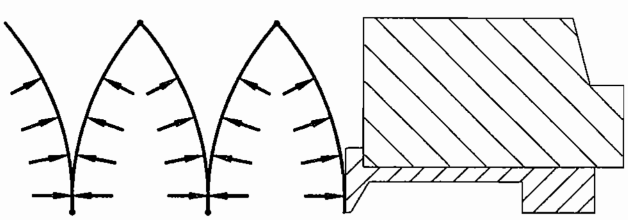

Figure 1. Diaphragm Contact May Occur at High Pressures.

Spring Rate

A low spring rate is desirable since it allows the bellows to maintain desired face loading in spite of assembly tolerances or axial thermal growth of the shaft. Bellows spring force is approximately given in Equation (1),

$$F_s = \pi D_{avg} E \left(\frac{\Delta y}{N}\right) \left(\frac{t}{s}\right)^3 \quad (1)$$

Load tests show very good agreement with this equation. Note that a small change in thickness (t) or radial span (s) has a large effect on spring force. Loading is directly proportional to installed deflection (Δy) and inversely proportional to the number of diaphragms (N).

Deflection Stress

Axial compression of the bellows induces bending stresses in the diaphragms. The highest stress is near the ID weld, a slightly lower stress occurs at the OD weld. Beam theory results in the following Equation (2) for maximum stress,

$$\sigma_d = \left(\frac{3Et}{s^2}\right)\left(\frac{\Delta y}{N}\right) \quad (2)$$

Deflection stress is proportional to thickness and inversely proportional to radial span squared. Excessive deflection can lead to yielding, near the welds, of low yield strength materials.

Pressure Stress

Pressure stresses are highest near the ID and OD welds. Provided diaphragm to diaphragm contact has not occurred, beam theory yields Equation (3) for pressure stress,

$$\sigma_p = 0.5\,(\Delta p)\left(\frac{s}{t}\right)^2 \quad (3)$$

Pressure stress is directly proportional to pressure, proportional to radial span squared, and inversely proportional to thickness squared. For an OD pressurized seal, pressure and deflection stresses bend the plates in the same direction near the OD and in opposite directions near the ID weld. Thus σ_d and σ_p are additive at the plate OD region and subtractive at the ID.

High pressures can result in yielding of low yield strength materials. In many cases, however, stresses are kept to moderate levels by allowing pressure induced contact between diaphragms. This shortens the effective radial span and can significantly reduce pressure stress that is proportional to the square of the span. Preliminary work modelling these effects using finite element analysis with a contact element feature supports these conclusions. It is planned to present this work in a future publication.

Overtravel

Overtravel is an important design factor for high temperature sealing. During warmup, the shaft may expand axially more rapidly than the pump case, causing an increased compression of the bellows. This problem is exaggerated for long shafts of double ended multistage pumps. The bellows design must compensate for this to ensure that the diaphragms are not damaged. This is done by using sufficient number of diaphragms and diaphragm pitch to allow for adequate travel from bellows working length to maximum compressed length. Excessive diaphragm pitch, however, may lead to dynamic instability of the bellows.

Slave Plates

A special support diaphragm, called a "slave plate," is welded to the OD of the first and last diaphragms (Figure 2). Slave plates provide sufficient stiffness to the first and last diaphragms to allow full nesting of the bellows. Slave plates also reduce stress in the first and last diaphragms arising from radial expansion at high temperature. Note that radial thermal growth of the first diaphragm is resisted by the low expansion alloy face flange, which has a thermal expansion coefficient less than half that for the bellows plates. Thermal stress is lower at the other end of the bellows due to a smaller difference in thermal expansion coefficients of the 316 stainless steel bellows end terminal and Inconel 718.

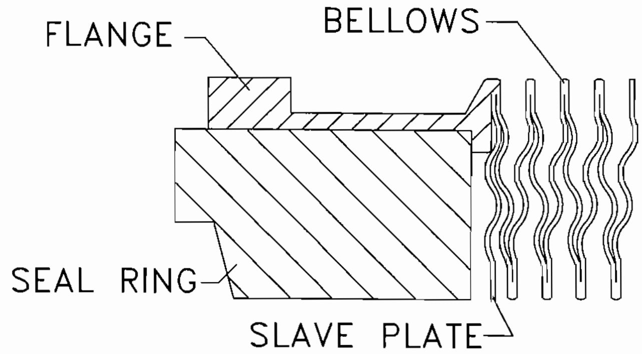

Figure 2. Original Flange, Seal Ring, and Partial Bellows Assembly. Note slave plate attached to first diaphragm and flange.

FACE DESIGNS

Rotating Face Design

The rotating face assembly was redesigned, tested, and introduced to the market in 1991. The rotating face, typically silicon carbide, seats on a flexible graphite gasket (Figure 3) and does not make mechanical contact with the 316 stainless steel holder. The face is retained with a weld band as shown.

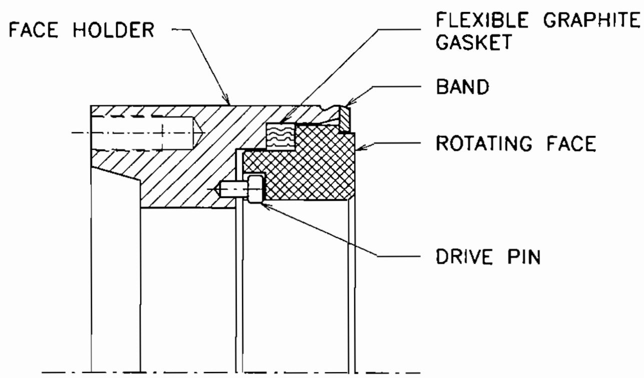

Figure 3. Rotating Face Assembly.

The flexible graphite gasket located on the tail of the silicon carbide face is precision die molded to assure close dimensional tolerances. Axial and radial compression of the gasket prevents leakage between the gasket and face. Radial contact loading between the gasket and face is low, to minimize the effect such forces might have on face distortion. A special installation procedure is used to maintain face flatness and squareness after inserting the face and welding on the retaining band.

Face flatness of this design at elevated temperatures was evaluated by heating different rotating assemblies and an optical flat to 500°F in an oven. There was no more than one lightband variation in flatness between room temperature and 500°F. All assemblies were helium leak tight.

Stationary Face Design

One of the key areas that had to be addressed in the design process arose out of initial testing of the new Inconel 718 bellows using the previous seal face assembly. A cross section is shown in Figure 2 of the original face flange, seal face, and partial bellows elements. The flange is manufactured from a low-expansion alloy that has a coefficient of thermal expansion of approximately 2.6

$\times 10^{-6}$ in/in°F. The Inconel bellows material has a much higher expansion coefficient of approximately 7.1×10^{-6} in/in°F.

Because of the difference in coefficients of thermal expansion for the low-expansion alloy and Inconel 718, excessive distortion of the carbon face was being generated at elevated temperatures due to the higher radial growth of the bellows diaphragms. This distortion caused the carbon face to contact at the OD, as illustrated in Figure 4. Test results on a nominal 3.5 in diameter seal at 250 psi and 500°F showed a worn-in distortion ranging between 800 to 1000 μin convex (more OD wear). This effect is greater for the Inconel bellows than for the original Hastelloy bellows, since the Inconel bellows is constructed with thicker plates and a higher yield strength material.

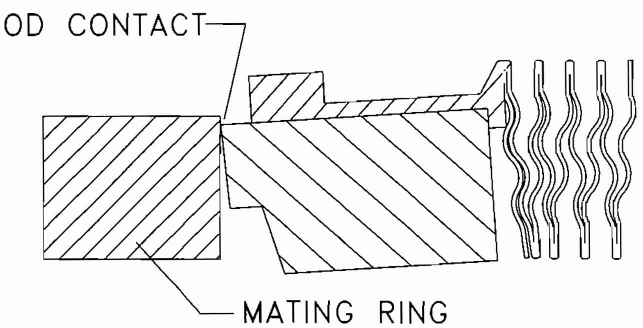

Figure 4. Bellows Radial Thermal Growth Results in OD Face Contact For Old Seal Design.

For an OD pressurized seal, this type of distortion produces very low leakage, but at the expense of extremely high face wear. The other disadvantage occurs at shutdown when the seal may be subjected to a much lower temperature. Bellows radial expansion is reduced, causing the distorted face to recover proportionally. The result is a highly convergent gap (Figure 5) which may result in a large opening force and excessive leakage.

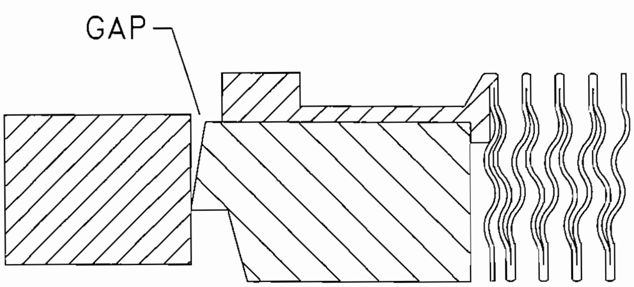

Figure 5. OD Wear May Result in High Leakage When Temperature is Decreased.

A commercial finite element analysis program [6] was used to investigate design concepts that exhibit low face distortion at high temperature. This structural analysis computer program includes a contact element which models interactions between surfaces, such as the interference fit between the face flange and face. The analyzed configurations fit within the diametral and axial envelope of the previous Hastelloy bellows seal.

One technique commonly used to reduce high temperature distortion is to increase the flange crossection, giving it additional strength and resistance to bending. This is shown schematically in Figure 6. While this is an effective means to control bending, it adds more mass to the face end of the bellows. Also, by increasing the resistance to bending in the flange, it was found that high stresses are developed, due to differential rates of expansion, in the first diaphragm ID and OD welds. The maximum stress occurs at the first ID weld. Under conditions of dry-running operation, it has been observed in other bellows assemblies that failure occurs at the first ID weld. As a result, it would be advantageous to concentrate on a design that minimizes stress in these weld regions.

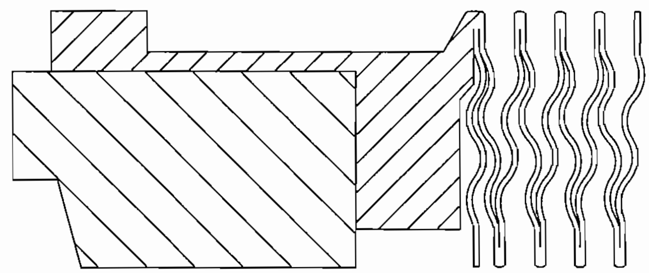

Figure 6. Thick Flange Resists Radial Thermal Growth of Diaphragms, but Adds Mass and Results in Higher Bellows Stress.

The design illustrated in Figure 7 minimizes additional mass at the face end and by increasing the length of the flange in the specified manner, incorporates a flexible region. The flexing region relieves some of the stress, due to bellows radial expansion, on the first diaphragm ID and OD welds.

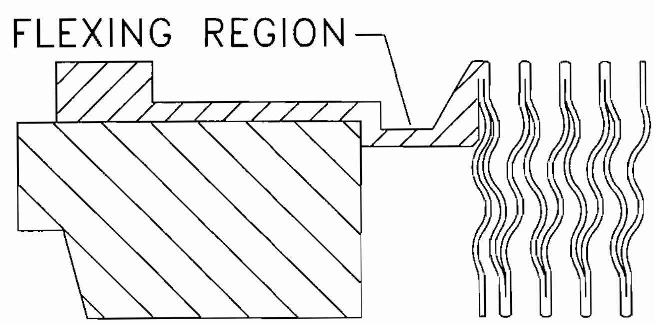

Figure 7. New Flange Design to Minimize High Temperature Face Distortion. Bellows radial thermal growth absorbed in flexible region.

Finite element analysis of the original design and the new configuration (Figure 7) was compared against actual test results. Test conditions were 500°F and 250 psi for a minimum of 44 hr. This comparison, which includes the computed effects of pressure caused distortion, is shown in Table 2 with thermal distortion due

Table 2. Stationary Face Distortion: Comparison of Calculated and Experimentally Measured Values. Operating Conditions: 250 psi, 500°F, 3600 rpm.

Seal Config.	Finite Element Analysis				Measured Distortion (μin.)
	Pressure Distortion (μin.)	Face Heated Thermal Distortion (μin.)	Bellows Expansion Face Distortion (μin.)	Calculated Net Face Distortion (μin.)	
Original with Inconel bellows	-312	+309	-485	-488	-800 to -1000
New	-89	+212	-204	-81	-25 to -68

to friction heating at the face, and bellows radial expansion caused face distortion. Measured distortion is determined from the post test profilometer traces.

The values shown are measured distortion in microinches across the sealing face. This measurement is illustrated in Figure 8. A positive value denotes convergent coning (as shown), while a negative value represents face distortion resulting in OD contact. Comparison between calculation and experiment shows that distortion is underpredicted for the original configuration, but is in fair agreement for the new design.

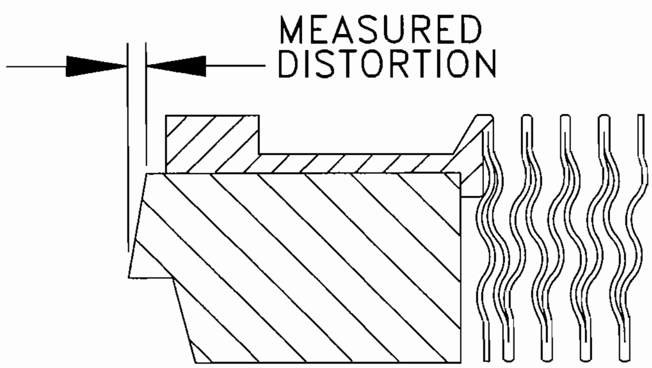

Figure 8. Measured Face Distortion. Convergent (in direction of leakage) coning shown here.

Finite element analysis has proven to be a useful tool in predicting distortion trends of different configurations, even though calculated face deflections may be off by a factor of two or so. There are several factors that affect distortion that are not known with high precision. These include the amount of frictional heat generation, heat transfer coefficients, the sliding coefficient of friction between the shrink fit carbon face and its holder, and variations in carbon material properties.

Face distortion due to friction heat generation was calculated assuming the faces were worn in and parallel, and the friction factor between the SiC and carbon faces was 0.1. Heat transfer coefficients were obtained from Lebeck [7]. Small variations in friction factor have a relatively large effect on total predicted face deflection.

Variations in material properties of carbon graphite are another source of uncertainty in the analysis. Furthermore, it has been reported [8, 9] that the stress-strain relationship of carbon graphite materials is not linear and that plastic deformation may be introduced at very low values of stress.

Regardless of the material property variation, it should be emphasized that the finite element model correctly predicted the trend in distortion resulting from the change in design. Reductions in all areas of distortion were achieved leading to a desirable smaller total-net distortion and substantial improvement in seal performance.

Comparison of Old and New Stationary Assemblies

The additional axial length required in the flange and the increase from 20 to 30 diaphragms required a reduction in the end terminal length, so that the new assembly fits in the existing seal envelope. The old and new stationary assemblies are compared in Figure 9. Note that the new and original stationary assemblies have identical total lengths and maximum diameters.

LABORATORY TESTS

Initial testing of design concepts was performed on an indoor seal tester at temperatures ranging from 110° to 150°F. Configu-

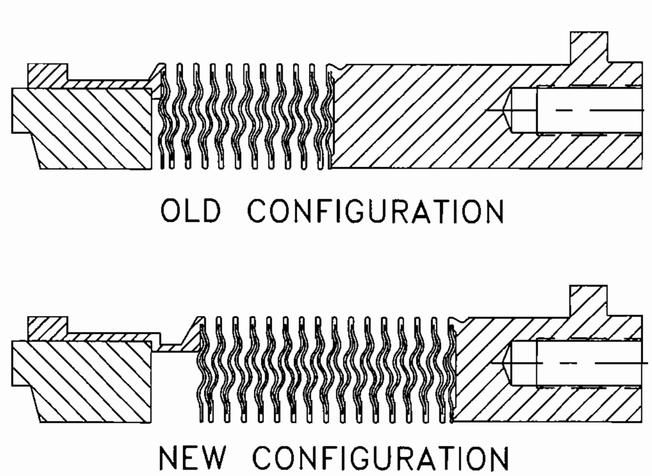

Figure 9. Old and New Stationary Face Assemblies.

rations that showed low leakage and wear were then evaluated on an outdoor tester at temperatures to 700°F.

Low Temperature Testing

Two seals were installed in the tester housing in a back to back configuration. The bore of the outboard seal was open for visual inspection to allow observation of bellows dynamics and identification of any leak pathways.

Seal flush is provided by means of a circulating pump, with flow passing through a heat exchanger (cooler) and a heater. Flush flow rate is measured with a rotameter. Any leakage is routed into a graduated cylinder. Automatic controls keep pressure within ± 5.0 psi and temperature within ± 2°F of set values. Pressure and temperature sensors are hard wired into a data acquisition computer. Automatic shutdown of the tester occurs if recorded parameters are outside of specified ranges. This system can be run around the clock unattended.

Low temperature testing was done using a 10 weight hydraulic oil. Two seal sizes were evaluated using the conditions contained in Table 3. All tests were conducted with the "new" rotating face assembly discussed earlier.

Various stationary face/flange configurations were tested, as well as different seal balance ratios. Bellows pitch and spring

Table 3. Low Temperature Test Program.

Seal sizes	2.125 & 3.500 in
Rotating face	RB SiC
Stationary face	Carbon or RB SiC
Temperature	110 - 150°F
Pressure	75 - 300 psig
Shaft speed	3600 rpm
Fluid	10 wt. hydraulic oil
Flush flow rate	3 gpm/seal
Test Duration	9 - 24 hours

loading variations were also evaluated. Contact pattern (ID or OD) was revealed with a post test profilometer trace of each face. The original Hastelloy bellows seal was tested to provide a bench mark comparison. Seal designs showing both low leakage and low wear were next tested in the high temperature tester described below.

High Temperature Testing

Seal configurations that performed well on low temperature testing were then evaluated in the high temperature tester shown in Figure 10. A centrifugal pump circulates heat transfer fluid through a loop consisting of an electrical heater, water cooled heat exchangers, and the test chamber. This system can handle seal sizes to 5.0 in, temperatures to 750°F, pressures to 300 psi, and shaft speeds ranging from 1,200 to 6,000 rpm.

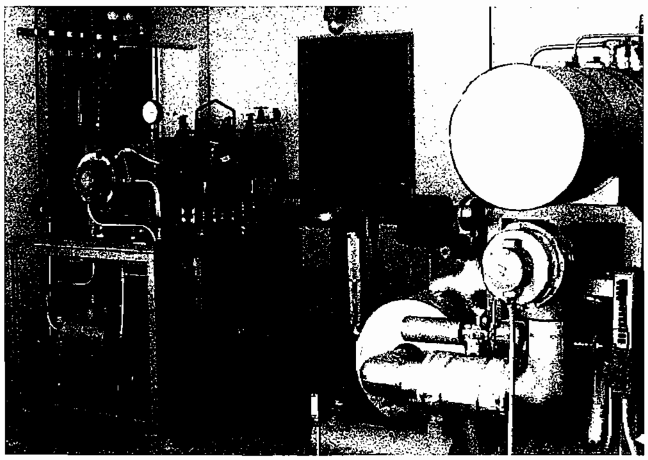

Figure 10. High Temperature Test Loop.

Programmable logic controllers are used to maintain temperature and pressure settings and allow unattended operation 24 hr/day. Various temperature and pressure transducers are wired into a computer for data collection and to monitor tester performance (e.g., bearing temperature). The test unit is automatically shut down if certain temperatures or pressures are outside of specified limits, or if seal leakage is excessive. Flush flowrate is measured using an orifice and a pressure difference gauge. The instrumentation is calibrated with traceability to NIST.

A synthetic heat transfer fluid, modified terphenyl, was used in this phase of testing. Recommended maximum bulk temperature is 650°F, but the fluid can be run at 700°F for short periods. It is suitable for low pressure operation as its vapor pressure is only 15 psia at 650°F. This fluid, however, has low viscosity (0.64 cSt at 500°F), and thus does not provide a thick lubricating film between the faces when used at high temperatures.

A diagram of the seal installation is shown in Figure 11. Note that the internal surfaces of the outboard seal, can be observed through a glass window (Figure 12). This view allows identification of leak pathways, either between the faces or past the flexible graphite gaskets. The window also prevents the escape of leaking vapors to the atmosphere. A nitrogen quench is provided to the low pressure side of the inboard seal. Leakage of vapors or liquid from both the inboard and outboard seals passes through stainless steel drain tubes and into graduated cylinders shown in Figure 13. Vapors are condensed upon contact with the drain tubes which are at ambient temperature. Liquid level readings are periodically noted and the fluid then drained off from the graduated cylinders. If leakage is excessive, it overflows to a tank (Figure 13) and then activates a float switch to shut down the tester.

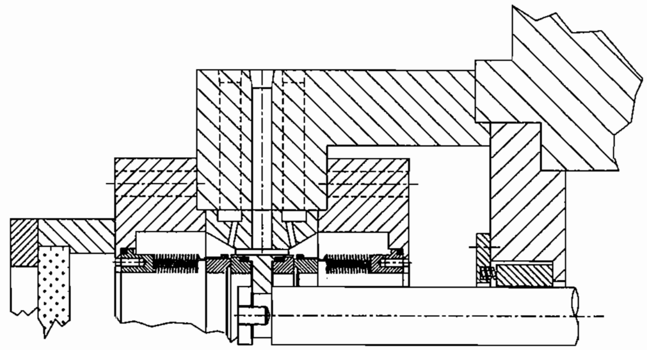

Figure 11. High Temperature Seal Test Installation. Note glass window on left.

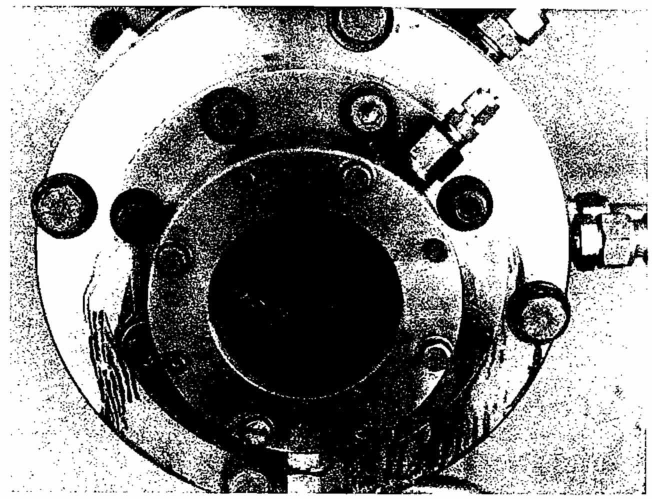

Figure 12. Window Allows Observation of Internal Surfaces of Test Seal.

The window was removed during portions of some tests to determine if there was visible leakage. Testing at 500°F and above revealed that condensed vapor leakrates less than about 10 cc/hr were not visible. Condensing vapors could be detected at slightly higher leakage rates. These observations were for the outboard seal which was not supplied with a nitrogen quench.

Screening Tests

The performance of various stationary bellows assembly configurations was initially evaluated using a two day screening test (Table 4). All tests used the "new" rotating assembly (Figure 3) which had been proven in earlier tests. Subsequent to dynamic testing some seals were evaluated for static leakage under the following conditions:

- 500°F bulk fluid temperature; pressure lowered from 250 to 50 psig, and
- 250 psi pressure; temperature lowered from 500°F to 230°F.

Some seal configurations experienced low leakage (< 3.0 cc/hr) on the dynamic phase of testing, then opened up and leaked heavily during the static test. The carbon face of these seals showed severe OD wear, caused by a large amount of either thermal and/or pressure distortion of the face. The test program was continued until a seal configuration was developed that had low leakage during both dynamic and static operation, and a low wear rate.

The measures of seal performance were dynamic and static leakrates

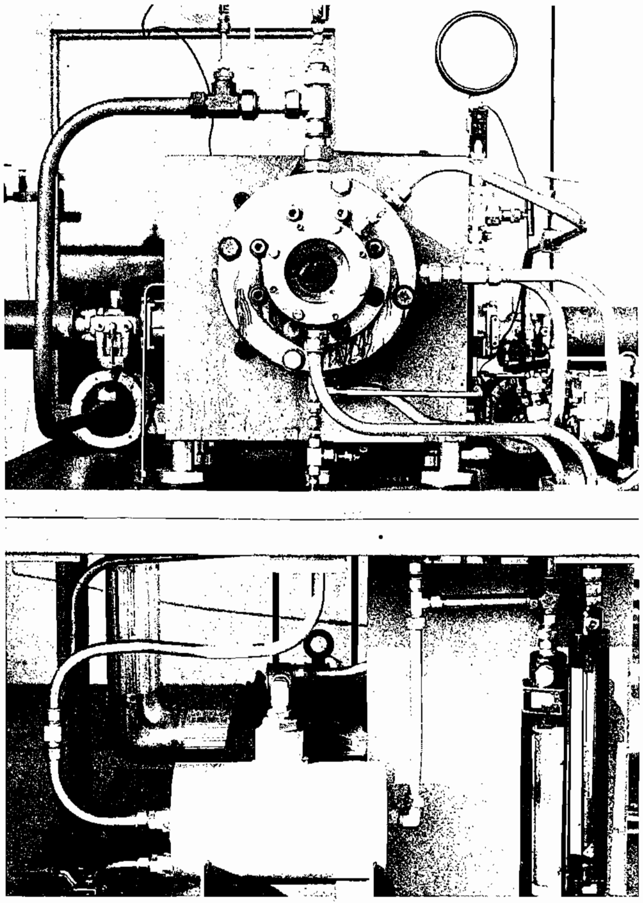

Figure 13. Graduated Cylinders (lower right) for Leakage Collection.

and post test face condition. The glass window provided a means to quickly determine if a seal was experiencing either very low or high wear rates. Well performing seals did not shed carbon and their ID surfaces remained clean and shiny. On the other hand, the internal surfaces of rapidly wearing seals were found to be coated with black carbon powder within a few hours after startup.

Existing Seal

The existing seal with a Hastelloy-C bellows (Figure 9) was tested to baseline the results. As noted earlier, the "new" rotating assembly was used in all tests. Evaluation of one 2.125 in and five 3.500 in Hastelloy seals, all pulled from inventory, revealed average condensed vapor leakrates from 1.0 to 3.0 cc/hr (liquid). In each of these tests the leakage was black, indicating excessive carbon wear. Also, seal ID surfaces were coated with carbon powder within 12 hours after startup. Post test inspection showed that the carbon faces wore about 600 to 750 μin more at the OD than the ID. As noted earlier, the heavy OD wear is due to face distortion caused by radial thermal expansion of the Hastelloy bellows being greater than that of the seal face flange and face. Total carbon wear was less than 0.001 in (micrometer resolution) for the 2.125 in seal, and an average of 0.006 in for the 3.500 in seal after the 44 hr tests. Inspection of the SiC faces showed a wear track depth of 5.0 to 15 μin (0.5 to 1.5 lightbands).

In field applications, this seal generally runs on fluids having a viscosity somewhat higher than that for the low viscosity test fluid. Those field fluids provide a lubricating film between the faces. Users report an average service life for the existing Hastelloy seal of over four years, while generally showing no visible leakage. The low viscosity fluid used in the lab test program does not provide a thick lubricating film at elevated temperatures, which results in somewhat accelerated wear rates. This allowed more rapid evaluation of relative performance of different seal designs.

New Seal

Several different stationary face/flange geometries were evaluated along with different values of bellows pitch, bellows spring loading, and seal balance ratio. Each test provided valuable guidance in achieving the final configuration of the "new" stationary assembly shown in Figure 9. Due to reduced distortion the new design requires a higher balance ratio to minimize leakage.

Screening tests (Table 4) of the final design revealed a dynamic leakrate of about 0.1 cc/hour for the 2.125 in seal and 2.2 cc/hour for the 3.500 in seal. All condensed vapor leakage was clear. Post test inspection revealed a good appearance of all faces. Surface traces of the carbon faces showed 100 μin convergent (OD) wear of the 2.125 in seal, and 25 to 68 μin convergent wear of the 3.500 in carbon face. Note that the existing stationary faces experienced 600 to 750 μin OD wear under the same test conditions. Total wear of the carbon faces in all tests of the new assemblies was less than the micrometer resolution. There was no appearance of carbon dust as had been the case with the original seal. In some cases, there was a trace amount of coking on the ID side of the carbon (note the test seals were not quenched). The SiC faces had wear tracks of 0 to 20 μin depth (0 - 2 light bands).

Table 4. Hot Loop Screening Test.

Seal sizes	2.125 & 3.500 in
Rotating face	RB SiC
Stationary face	Carbon or RB SiC
Temperature	500°F
Pressure	250 psig
Shaft speed	3600 rpm
Fluid	Modified terphenyl
Flush flow rate	2.2 & 3.5 gpm
Duration	44 hours

Additional testing of the new design was performed at 700°F and 250 psig. The 2.125 in seal leaked 0.3 cc/hr and the 3.500 inch seal leaked 1.0 cc/hr of condensed vapor. This leakage was clear. Post test inspection revealed a good appearance of the faces. Further testing was done at pressures of 75 and 100 psi, and temperatures of 500 and 600°F. Leakage was about 0.1 cc/hr for the smaller seal and 1.0 cc/hr for the larger. Again, the leakage was clear and the faces had a good post test appearance.

The next phase of lab testing was to run the two seal sizes on the API 682 seal qualification program.

API 682 Qualification Testing

API Standard 682 [1] specifies each seal/system offered for refinery services be shall be suitably tested prior to its market availability. The intent is not to test every size, but to qualify the overall design. Since approximately 90 percent of seal applications fall between 1.5 and 4.5 in, testing of 2.0 in and 4.0 in seals is considered representative of the size range. The test sequence consists of three phases: dynamic, static, and cyclic. The three phases are run consecutively without removing the seals. The operating conditions for each phase are:

- *Dynamic* phase is conducted for a minimum of 100 hr at constant temperature, pressure, and speed. This is defined as the "base point." The base point for high temperature seal applications is 500°F, 115 psig, and 3,600 rpm.

- *Static* phase follows the dynamic phase, and uses the basepoint temperature and pressure, but zero rpm. A minimum of four hours is required without shaft rotation.

- *Cyclic* phase includes varying temperatures and pressures, loss of flush, and shutdowns and startups. Pressure is dropped from 115 to 15 psig and then back to 115 psig. Seal chamber temperature is dropped from 500 to 300°F and returned to 500°F. Pressure is increased from 115 to 250 psig, then back to 115 psig. Seal flush is then turned off for at least one minute. Finally, shutdown (0 rpm) at 115 psig and 500°F for at least 10 minutes. Measurements are to be taken during each part of the cycle. Five complete cycles are run.

All tests are to be performed at the *maximum* allowable angular and radial misalignment as specified in API 682 paragraphs 2.3.5.2 and 2.3.8 [1]. Seal chamber register fit surface shall be concentric to the shaft, with a total indicated runout of not more than 0.005 in. Seal chamber face runout shall not exceed 0.0005 in/in of seal chamber bore.

The 2.125 in seal was installed with a concentricity of 0.003 in and seal chamber face runout of 0.0005 in/in. Condensed vapor leakrate was a steady 0.1 cc/hour throughout 103 hours of dynamic testing. The static phase (500°F, 115 psig, 0 rpm) lasted for 11 hours with a constant leakrate of 0.8 cc/hr. Cyclic phase leakrate averaged 0.5 cc/hr over the 4.75 hrs required to complete that phase of testing. All leakage was clear and the seal interior surfaces were clean at the conclusion of this test. The surface trace of the carbon face (Figure 14) showed 25 μin convergent coning wear (more wear at the OD). Net carbon wear was less than the micrometer resolution. The silicon carbide rotating face was very smooth, with a wear track less than one μin (Figure 15).

The 3.500 in seal was installed with 0.003 in concentricity and chamber face runout of 0.0003 in/in. Dynamic leakrate varied from 0.6 to 0.8 cc/hr (condensed vapor) through 103 hr of dynamic

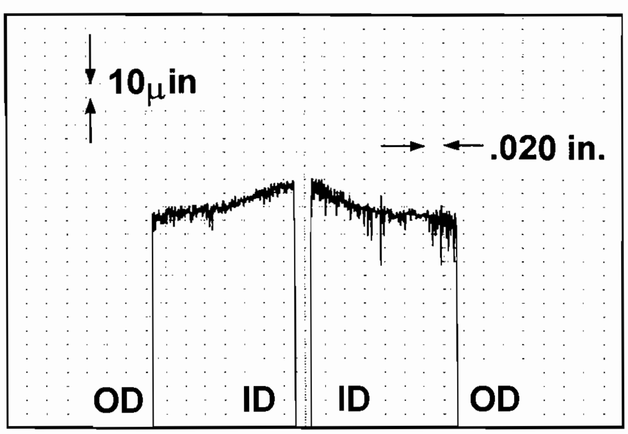

Figure 14. Surface Trace of Carbon Stationary Face After API 682 Qualification Test.

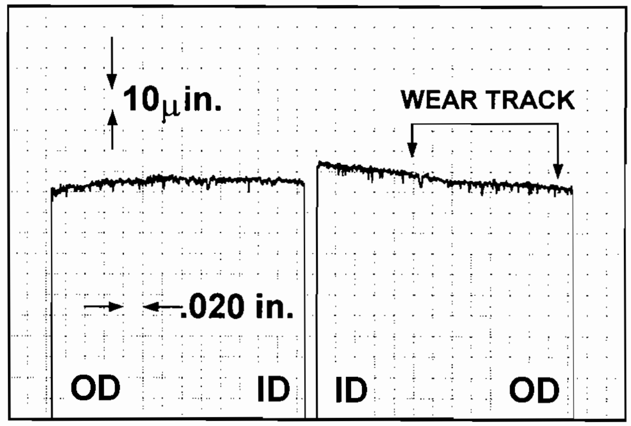

Figure 15. Surface Trace of Silicon Carbide Rotating Face After API 682 Qualification Test.

testing. Static phase leakage averaged 0.5 cc/hr during 11 hr in the shutdown mode (500°F, 115 psig, 0 rpm). Cyclic phase leakage averaged 1.1 cc/hr over 5.2 hr. All leakage was clear. The seal did not shed carbon dust. Stationary face coning wear measured 110 μin, and net carbon wear was less than the resolution of the micrometer. Wear groove in the silicon carbide rotating face was about 5 μin (0.5 light band).

Dry Running Tests

Part of the overall test program was to verify the integrity of the new bellows design. Experience has shown that a bellows can fail during operation that becomes unstable due to insufficient liquid film at the sealing interface. This condition of poor lubrication, known as "stick-slip," can induce forms of torsional and axial vibration that may fatigue the bellows. Fatigue failure of a bellows may occur within minutes or the bellows may survive many months, depending on bellows construction, face materials, and severity of fatigue inducement.

Stick-slip generally results from starting the pump dry, improper venting, or pump cavitation. Extreme pump vibration or high amplitude pressure pulsations may also cause bellows fatigue. If one of the faces is carbon graphite, the seal may survive an extended period without bellows failure. If both faces are of silicon carbide and the seal is running dry, then fatigue failure may occur within minutes. To quickly evaluate bellows integrity under stick-slip conditions, a series of tests were conducted in which seals were run in dry air with SiC *vs* SiC faces.

A total of 10 bench tests were performed where conditions were at ambient pressure and temperature and seal rotational speed was either 850 or 1750 rpm. All of the tests, except one, were run with a SiC rotating face against a SiC stationary face. The exception was a carbon running against SiC for one test. The carbon/SiC combination was not used beyond one test, because bellows vibration was not great enough to cause failure in a reasonable length of time. If the bellows were going to fail, it was the objective of the test program to find out within a few hours of operation so that both close observation and safety could be maintained.

Comparison testing was done between the old bellows design which consisted of a 20 plate, 0.007 in thick Hastelloy C bellows and the new 30 plate, 0.008 in thick Inconel 718 bellows. After bellows instability had been reached during testing, the vibrations were monitored using a strobe scope. It was observed for both bellows designs that the resultant instability encompassed multiple harmonics ranging from a low frequency wave traveling from the end terminal to the face end, to individual high frequency diaphragm vibration. Face temperature measurements at the ID of the SiC stationary face rose to as much as 800°F on the 3.5 in seal and 430°F on the 2.125 in seal.

Failure of the Hastelloy C bellows was attained after 15 to 17 minutes of dry running, resulting in shear of the first ID weld from the end terminal. The new Inconel 718 bellows had dry run times from 56 minutes to 2 hours and 55 minutes of continuous vibration without bellows failure. Those tests were stopped due to a failure of the faces, not the bellows. Severe axial vibrations caused fracturing of the rotating faces in some cases. In other tests, shearing of the rotating drive pin allowed the rotating face to stop turning, while its holder and flexible graphite gasket continued to rotate. In several tests, the stationary face spun due to excessive heat allowing the shrink fit to open up. In every test, the bellows was helium leak tight afterward.

Two additional dry running tests were also performed on the Inconel 718 bellows using the hot-loop tester. The procedure consisted of running the seal until equilibrium operating conditions (600°F, 250 psi and 3600 rpm) had been achieved. The seal chamber was then blocked off to eliminate circulation and all fluid drained from the seal cavity. The seal was operated under dry running conditions until some type of failure occurred. In both cases, the test was stopped after excessive vibration had caused the rotating face to chip out at the drive notch and allowed the face to spin in its holder. Again, the bellows was helium leak checked and found to be leakfree and undamaged.

Running SiC vs SiC faces in dry air proved to be a severe operating condition for the Hastelloy C bellows. The Hastelloy bellows sheared at the first ID weld (from the adapter end) in about 15 minutes of dry running. The thicker Inconel bellows did not fail (in runs of two to three hours), even upon seizure and destruction of the seal faces. Although the test time was relatively short in terms of seal life, the new Inconel bellows proved to be significantly stronger than the Hastelloy bellows. Note that the Hastelloy bellows seal provides reliable performance in field installations.

CONCLUSION

A high temperature seal has been developed that employs Inconel 718 bellows for corrosion resistant performance on services such as high sulphur crude oil, or streams with traces of chlorides, H_2S, and naphthenic acids. In addition, the thick plate bellows construction provides high resistance to corrosion, fatigue failure, and pressure stress.

Finite element analysis was used in the design phase to predict face distortion and bellows stress levels. Configurations computed to have low distortion and low stresses were evaluated in a high temperature tester at temperatures to 700°F. The final configuration was also evaluated per API 682 qualification test for high temperature seals. The two sizes of seals tested showed both low leakage and low wear.

Post test measurements show that high temperature distortion (coning) of the shrink fit retained stationary face has been reduced by a factor of 10, relative to the old design. This results in less wear and reduced leakage during temperature and pressure changes.

The Inconel 718 bellows was evaluated for resistance to shear and fatigue failure in dry running tests using SiC *vs* SiC faces. This resulted in heavy damage to the silicon carbide faces, but in every test the Inconel bellows was intact and helium leak tight afterwards.

The new bellows is constructed with 50 percent more diaphragms than the old. The diaphragms are also thicker. This construction allows the Inconel bellows to accommodate a larger amount of shaft axial growth during thermal warm up of a pump.

The original Hastelloy bellows seal continues to perform well in high temperature applications. Laboratory tests at temperatures to 700°F show that the upgraded seal with an Inconel 718 bellows and a redesigned stationary face exhibits superior performance (low leakage, lower wear, stronger bellows, and improved corrosion resistance).

NOMENCLATURE

D_{avg}	Mean diameter of bellows diaphragm
E	Young's modulus
F_s	Spring force
N	Number of bellows diaphragms
s	Diaphragm radial span between ID and OD welds
t	Diaphragm thickness
Δp	Pressure difference across seal
Δy	Axial compression of bellows
σ_d	Bending stress due to bellows axial deflection (compression)
σ_p	Bending stress due to pressure difference across bellows

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