EXPERIMENTAL TESTING OF A COMPOSITE GASKET WITH INTERNAL PRESSURE, BENDING, AND THERMAL LOADS

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SUMMARY

Gaskets are an integral part of the sealing process provided by bolted flange assemblies. Depending upon service conditions, a seal must prevent leakage in the presence of internal pressure, axial loads, and bending loads. In some conditions they must respond to high temperatures and corrosive environments. All of these factors work against the ability of the gasket to provide a reliable barrier to leakage.

Two separate experimental investigations were undertaken to test the capabilities of a composite gasket designed and manufactured by Pikotek. The first series of tests involved a 6-in Class 1500# flange assembly with a Pikotek gasket. The gasket was subjected to a hydrostatic pressure of 9,900 psi and a cyclic pressure test to 6,600 psi using nitrogen gas. Bending moments exceeding 130,000 ft-lbs were applied simultaneously with 5,000 psi internal pressure nitrogen gas. No leaks occurred during any of the above loading conditions. Even though the flanges were rated per ANSI B16.5 to only 3,705 psig, the gasket design permitted pressure containment far in excess of this value.

The second series of tests were conducted on a 2-in 2500# gasket and flange assembly. Experimentation involved gas test pressures up to 18,550 psig and cyclic temperature testing between 0°F and 300°F. The test procedure was based upon the methods outlined in API 6A, Appendix F. The gasket performed successfully during all phases of testing. As with the 6-in gasket, testing revealed that the gasket sealed at pressures much higher than those specified in ANSI B16.5.

The ability to qualify flange assemblies at pressures in excess of the specified ratings, using the appropriate analytical and testing procedures in compliance with industry standards, results in significant cost and weight savings for the end user. From an application standpoint, composite gaskets serve as an important alternative to current gasket designs and their associated limitations.

INTRODUCTION

Because of the differences in test procedures for the bending and thermal cycle tests, information relating to test methods and results are presented independently within the respective sections of this paper.

Bending Test

For a gasket to function properly in a variety of loading conditions, it must be designed to provide sealing pressure in the event of reduced bolt pre-load. The HP VCS gasket, manufactured by Pikotek, incorporates a spring-loaded Teflon seal in conjunction with an elastomeric O-ring to address this issue. **Figure 1** shows a schematic of this gasket design and the critical components. The Teflon ring is designed to provide the first pressure barrier and the O-ring supplies the secondary sealing mechanism in the event of an inner-ring leak.

A testing program was developed to test the Pikotek gasket considering internal pressure and bending loads. For testing purposes, a flange assembly was fabricated involving the following components,

- 6-in Class 1500# ASME B16.5 weld neck flanges with a bore diameter of 5.38-5.41 inches (A105)
- Studs and nuts for the respective flanges, Specifications A193 and A194, Grades B7 and 2H, respectively
- 6-in XXS end caps (A105)
- 6.625-in O.D. solid carbon steel bar joints (to prevent buckling) used in the bending test.

The flange assemblies were constructed by welding end caps to the flanges. Ports were drilled and tapped in the end caps to accommodate the necessary pressure equipment. Attached to each of the two end caps were solid bar segments for the purpose of applying bending to the flanges. Also welded to these solid bar joints were end caps to match the load frame for conducting the bend test. **Figure 2** provides a photograph of the assembly during the process of make-up.

Two of the twelve studs were fitted with gages to measure strain in the bolting during make-up and testing. These two studs were machined down to the minor diameter for ease in mounting the strain gages. **Figure 3** provides a photograph of this set-up.

Strain gages were also placed on one of the two solid bar segments. The purpose of the gages on the solid bar was to monitor strain during the bending process and to ensure that the correct level of bending was applied. Gages on the solid bar were installed at +90°, 0°, and -90° (or 12:00, 3:00 and 6:00 O-clock, respectively) relative to the neutral axis of the bar.

After installation of the strain gages, the make-up condition was established by torquing each bolt to 680 ft-lbs. This was done incrementally at 30, 65 and 100 percent of the specified maximum torque. A cris-cross pattern was used in the flange make-up. The desired bolt make-up stresses were verified by the strain gage readings.

Once the make-up process was complete, the flange assembly was placed in a 2.5 million lb. load frame. This load frame is capable of generating 2.5 million pounds in tension, 1 million in compression, and 150,000 ft-lbs. in bending. **Figure 4** is a photograph of flange assembly installed in the load frame. The sample was pressurized in the load frame. The load frame induced bending in the sample using two hydraulic cylinders (one located above and the other positioned beneath the sample).

The amount of bending moment applied to the flange assembly was controlled using the load from the hydraulic cylinders. The strain gages on the bar were monitored to ensure that the proper amount of bending was being applied. The gages were zeroed prior to application of load to remove the effects of weight.

Thermal Cycle Test

The desired objective of testing the 2-in 2500# gasket was to determine the response of this specific design to cyclic thermal and pressure loads. The testing objective involved testing for adequate sealing as well as verifying the structural integrity of the gasket itself. In relation to the latter objective, the blind flanges were disassembled between cycles for visual inspection of the gasket. **Figure 5** provides the basic configuration of the gasket (drawn to scale). As noted in this drawing, the smaller 2-in gasket design does not have the O-ring located outside of the Teflon seal as does the larger 6-in gasket shown in **Figure 1**.

The principal hardware components used in the thermal cycle test program are shown in **Figures 6** and **7** and are listed as follows,

- Set of 2-in 2500# blind flanges with appropriate bolting (eight 1-in 8UN studs)
- Electric oven to heat flanges to 300°F
- Liquid nitrogen system to cool flanges to 0°F
- Nitrogen gas pressure system
- Thermocouple and pressure transducer to monitor flange temperature and pressure
- Computer system to record data during testing.

The blind flanges were made up to axial bolt stresses of 40 ksi, which for the 1-in 8UN bolts corresponds to a bolt torque of 361 ft-lbs (refer to Appendix D, *Recommended Flange Bolt Torque*, of API Specification 6A). After make-up of the flanges, the test assembly was placed in the electric oven and the necessary hardware was attached.

A computer data acquisition system recorded the temperature and internal pressure of the flanges. Two controllers were used to govern the heating and cooling of the flanges. Readings were taken approximately every 30 seconds during the duration of the testing. Based upon the test set-up, it took approximately 2 hours for the heating and 2 hours for the cooling processes to induce the desired temperatures in the flange assembly. Upon average, an eight-hour period was required to complete one cycle of the testing program.

TESTING PROCEDURES

Two different test procedures were employed for the bending and thermal cycle tests.

Bending Test

The bending test procedure was as follows,

1. Install strain gages on the studs, flanges and bending bar.
2. Torque bolting with a cris-cross pattern to 680 ft-lbs.
3. Conduct a preliminary hydrostatic test to 3,300 psi to shakedown the strain gages and equipment.
4. Conduct a hydrostatic test to 9,900 psi and hold for 60 minutes
5. Conduct a cyclic nitrogen test, ambient to 6,600 psi, and hold for 30 minutes, with a total of 10 cycles
6. Apply 50,000 ft-lbs. bending moment and repeat step #3 for 5 cycles with 10 minute holds.
7. Disassemble the flanges and inspect the gasket.

A second bending test was conducted to determine the bending moment at which leakage would occur. This is reported later in the paper.

Thermal Cycle Test

The thermal cycle test procedure was as follows,

1. Weldable thermocouples should be attached to the flanges (either before make-up or after the hydrotest).
2. Make-up the flanges and test gasket, tightening the bolts in a cris-cross pattern to 361 ft-lbs. (40 ksi)
3. Perform a hydrotest to 27,825 psi. Hold pressure for 30 minutes and determine change in pressure over this time period (record pressure every minute).
4. After conducting the hydrotest, place assembly in the oven for pressure and thermal cycling.
5. Connect the necessary hardware to control the thermal cycles (thermocouples, computer, nitrogen system, etc.). The data acquisition system needs to be able to monitor pressure and temperature as a function of time.

Thermal cycles (apply a total of three times to flange assembly):

1.1 Fill test cavity with nitrogen. Turn on heater and raise the inside temperature of the test unit to 300°F. Monitor the thermocouples and determine when the temperature has stabilized.

1.2 Apply the rated pressure (18,550 psi) to the fixture and hold for one hour. Release pressure at the end of this time period.

1.3 Cool the unit with the external nitrogen system until temperature of the assembly stabilizes at 0°F.

1.4 Apply the rated pressure (18,550 psi) to the fixture and hold for one hour. Release pressure at the end of this time period.

1.5 Use heater to return assembly to room temperature.

RESULTS

Results are presented for both the bending and thermal cycle tests.

Bending Test

Results are presented for the hydrostatic and nitrogen cycle testing as well as the bending moment testing.

Hydrostatic Testing

A preliminary hydrostatic shakedown test was conducted to ensure that all of the strain gages, pressure transducers, and data acquisition system were calibrated and working properly. Data is not reported for this phase of the testing. All indications showed that the measuring devices were working correctly. On the day of testing, the ambient laboratory temperature was approximately 70°F.

The hydrostatic test was conducted by pressurizing the flange assembly to 9,900 psi using water. The pressure was held at this value for 60 minutes. During the hydrostatic testing, the pressure decreased from 9,924 to 9,834 psig during the 60 minute hold. This pressure drop is less than the 500 psig drop permitted by the Pikotek Testing Procedure (and API Specification 6A).

Nitrogen Cycle Pressure Testing

At the completion of the hydrotesting, water was drained from the flange assembly and the nitrogen gas pressure system was attached. The process for pressurizing with nitrogen gas followed these steps,

- Pressurize sample to 6,600 psi
- Hold pressure for 30 minutes
- Reduce pressure to ambient and repeat for a total of 10 cycles.

During the nitrogen testing, the largest pressure decrease occurred on cycle #6 and was from 6,648 to 6,602 psig. This pressure drop of 46 psig is less than the 330 psig drop permitted by the Pikotek Testing Procedure (and API Specification 6A). During this same time interval, the temperature decreased from 68.5 to 62.9°F.

Bending Test with Nitrogen Pressure

The primary objective in subjecting the flange assembly to the bending load was to determine if under the imposed test conditions the gasket would seal the flanges.

Two sets of bending tests were conducted. The first was in conjunction with the hydrostatic test and nitrogen cycle testing. The basic steps in this testing were,

- Apply a bending moment of 50,000 ft-lbs to the sample (verified using ram pressure load readings and strain gages located on pipe)
- Apply pressure of 6,600 psi to sample using nitrogen gas
- Hold pressure for 10 minutes
- Reduce pressure to ambient
- Repeat above steps for a total of 5 pressure cycles.

No leaks occurred during this test. Because of the success of this test, another bending test was conducted. The objective of the second round of testing was to determine the bending moment at which leakage would occur.

Prior to conducting the second bending test, the following pressure tests were performed,

- Hydrostatic test to 7,500 psig (15 minute hold)
- Nitrogen gas test to 5,000 psi (30 minute hold).

No leaks were detected during either of the above pressure tests. The bending test was conducted with an internal pressure of 5,000 psi nitrogen gas. The steps in the second bending test were,

- Maintain internal pressure of 5,000 psi
- Apply a bending moment to assembly in increments of 1,000 ft-lbs every 30 seconds
- Continue bending application until leakage is detected or plastic deformation of assembly occurs.

The strain in the piping attached to the flanges was monitored during the testing to ensure that the correct level of bending moment was being applied. As with any member placed in pure bending, the exterior surfaces have strains due to bending, one in compression and one in tension, while there is a neutral axis along which there is no bending strain. **Figure 8** shows strain in the attached piping and bolts on the top and bottom of the flange.

There are several important observations in studying **Figure 8**,

- The strains in the bolting change as a direct result of the bending moment. The strain in the top bolt at 0° increased while the strain in the bolt at 180° decreased.
- The changes in strain between the nitrogen pressure testing alone and the flange assembly subjected to the 130,000 ft-lbs. bending moment are summarized,

<u>Axial Strain in Top Bolt</u>	1079 $\mu\epsilon$ (N_2 pressure)
	1489 $\mu\epsilon$ (Bending)
	Net change of +410 $\mu\epsilon$
<u>Axial Strain in Bottom Bolt</u>	1054 $\mu\epsilon$ (N_2 pressure)
	748 $\mu\epsilon$ (Bending)
	Net change of -306 $\mu\epsilon$

Figure 8 data is plotted for bending moments exceeding 130,000 ft-lbs. It should be noted that plastic flow initiated in the bars (solid) attached to the flanges at this bending moment. The exact value of the bending moment could not be measured accurately. Deflection continued with additional stroke of the hydraulic cylinders at a nearly constant pressure. The onset of plastic deformation is theoretically validated by rearranging the bending stress calculation as shown below,

$$M = \frac{\sigma_{flow} I}{c} = \frac{51{,}000\,psi \cdot 94.6\,in^4}{3.3125\,inches} = 121{,}373\ ft\text{-}lbs$$

where:

M	Bending moment (in-lbs)
c	Outer radius (inches)
I	Moment of Inertia for solid bar with a diameter of 6.625 inches (in^4)
σ_{flow}	Flow stress of bar material (average of minimum yield and tensile strengths).

For grade X42 material flow stress is computed to be,

$$\sigma_{flow} = \frac{\sigma_{yield} + \sigma_{UTS}}{2} = \frac{42\,ksi + 60\,ksi}{2} = 51\ ksi$$

The calculated bending moment is relatively close to the experimental value of 130,000 ft-lbs. At this bending moment there was a noticeable change in the loading characteristics of the flange assembly.

Thermal Cycle Test

Presentation of the thermal test results involves a tabulated summary of the key points during the cyclic testing as well as plotted data obtained with the computer data acquisition system. As stated previously, the maximum decrease in pressure during any of the six hold periods was 386 psi, or 2.09 percent.

Table 1 provides a summary of test results for Cycle #1 through Cycle #3 of the test program.

Figure 9, **Figure 10** and **Figure 11** provide plots of the data collected during Cycle #1, #2 and #3, respectively. In these plots, pressure and temperature are plotted as functions of elapsed time. As with the data listed in **Table 1**, the plots show minimal decrease in pressure during the one hour hold periods.

As stated previously, the flanges were disassembled between the cycles to assess the structural integrity of the gasket. Other than the markings left by the raised face on the inner region of the gasket, no significant abrasions or deformities were developed during the cycling process. **Table 2** provides a summary of measurements taken from the gasket before and after testing. One dimension worth noting is the inner diameter of the gasket after testing. Due to differences in thermal expansion coefficients, the inner diameter of the composite material contracted to the point where it was permanently less than the inner diameter of the stainless steel portion of the gasket.

DISCUSSION OF RESULTS

Bending Test

Before conducting the bending tests, calculations were performed to predict the bending moment required to cause leakage in the flange assembly. There are multiple methods for addressing this issue; however, for simplicity it was assumed that leakage would occur when the make-up force of the bolting was overcome by the applied bending moment and pressure end load.

Summing forces in the longitudinal axis of the flanges gives,

$$F_{bolt} = F_{bending} + F_{pressure}$$

which can be re-written as,

$$\sigma_{bolt} \cdot A_{bolts} = T_{effective} + P_{internal} \cdot \frac{\pi}{4} D_i^2$$

The effective tension, $T_{effective}$, is a representative force for the bending moment applied to the flange assembly. It is derived using shell theory assuming a bending moment is applied to one edge of a cylinder where the other end is fixed. Summation of forces, in response to the applied bending moment, along the constrained end yields the following expression,

$$T_{effective} = \frac{4\ M}{D}$$

where D is the diameter of the shell. In the case of a flange it is the bolt circle. If the internal pressure, bolt make-up loading, and various geometries are known, the force equation is re-written to yield the moment required to overcome the bolt pre-load,

$$M = \frac{D}{4} \left(\sigma_{bolt} \cdot A_{bolts} - P_{internal} \cdot \frac{\pi}{4} D_i^2\right)$$

where :

σ_{bolts}	Make-up stress in bolts (34,254 psi)
A_{bolts}	Cross-sectional bolt area (13.86 in^2)
$P_{internal}$	Internal pressure (5,000 psi)
D_i	Inner diameter of gasket (ASME Code *G* value, for the given gasket it is 6.565 inches).
D	Bolt circle diameter (12.5 inches)

Substituting the appropriate terms into the above equation yields a moment of 79,599 ft-lbs. Theoretically, this is the moment required to overcome the make-up stress in the bolting. As observed in the testing of this flange, a bending moment exceeding this value (130,000 ft-lbs) induced plastic flow in a solid bar having the same nominal diameter as the 6-in 1500# ANSI flange; however, no apparent damage to the flange or gasket was imparted with this bending load. The reason that the experimental moment required to induce leakage exceeds the calculated moment is that the gasket acts as a compliant sealing mechanism even though the flange faces theoretically separate. The current design is not unique in that many gasket designs provide this type of sealing behavior.

The critical issue in assessing the test results is that the gasket did not leak when a moment was applied that would have certainly induced extensive yielding if applied to a heavy-wall 6-inch nominal pipe. For example, a 6.625-in x 0.5-in grade X52 pipe will begin to yield with a bending moment of 59,000 ft-lbs and would be expected to buckle with a moment close to this value. This is well below the experimentally determined moment of 130,000 ft-lbs measured in the testing of this flange and gasket combination.

Thermal Cycle Testing

The permitted drop in pressure per F107.3c of API Specification 6A is written as follows,

> *The hydrostatic or gas test at high or low temperatures shall be acceptable if the pressure change observed on the pressure measuring device is less than 5% of the testing pressure or 500 psi, whichever is less.*

Considering that the maximum pressure drop was 386 psi (2.09 percent), the gasket met the imposed requirements listed above.

Another issue to address is the level of flange rotation associated with the blind flanges used in this test program. Two axisymmetric finite element models were constructed to address the differences in flange rotations between the 2-in 2500# blind flange and weld neck flange. Based upon the results of the analyses, the rotation of the blind flanges is greater than the rotation for the weld neck flanges. This being the case, the test conditions (in terms of rotation) associated with the experimental fixture are more rigorous than would be expected in a weld neck field application. In other words, the favorable test results presented herein exceed the requirements affiliated with weld neck flanges.

CONCLUSIONS

The objective of the bending tests conducted on the 6-in 1500# HP VCS gasket was to determine if under an extremely high bending moment the gasket would seal the flanges properly. The testing consisted of hydrotesting to 9,900 psi, nitrogen pressure testing to 6,600 psi, and a bending test to 130,000 ft-lbs bending moment. While maintaining the respective internal pressure and bending loads, leakage was not detected under any of these conditions. The gasket sealed the flanges, even to the point where plastic flow occurred in the solid bars of the flange assembly.

Within this paper, results have been presented relating to thermal and pressure cycling of the Pikotek 2-in 2500# gasket. Test data show that in the presence of pressure, thermal and make-up cycles the gasket retained pressures sufficient to pass the 500 psi maximum pressure drop as outlined in API Specification 6A. Visual inspection between cycles and after testing indicated that the structural integrity of the gasket was sound.

Having conducted both the bending and thermal tests, it is apparent that the gasket design discussed in this paper works with the flange assemblies to contain pressures in excess of the pressure limits set forth in ASME B16.5. The ability to qualify flange assemblies at pressures exceeding the specified ratings, using the appropriate analytical and experimental test procedures in compliance with industry standards, results in significant weight and cost savings for the end user.

ACKNOWLEDGMENTS

The authors would like to thank Mr. Richard C. Biel and Dr. Joe R. Fowler of Stress Engineering Services, Inc. for their time and effort in reviewing this work. Calculations of the flange stresses were performed by Mr. John Fowler using the BASIC FLANGE program which was modified to accommodate the Pikotek gasket.

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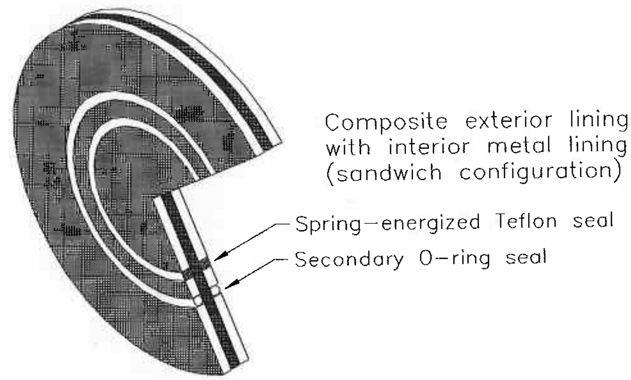

Figure 1 Components of the Pikotek gasket design

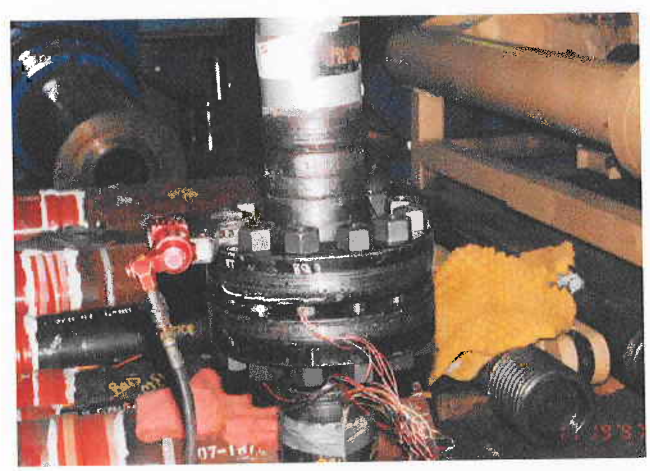

Figure 2 Flange assembly during makeup

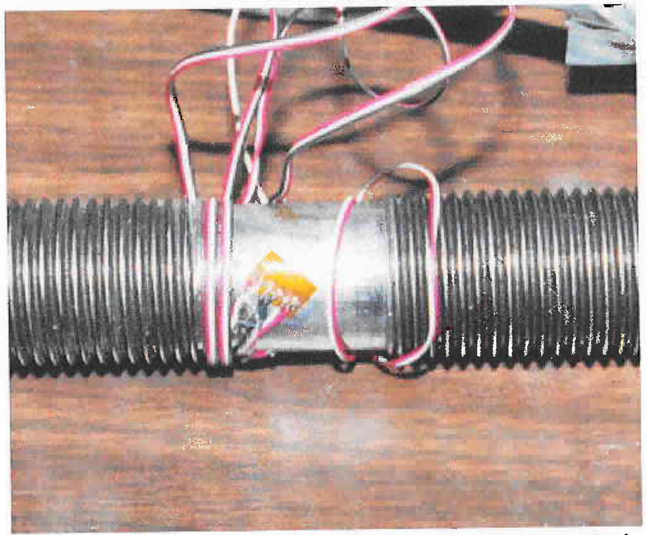

Figure 3 Photograph of strain gage installed on stud

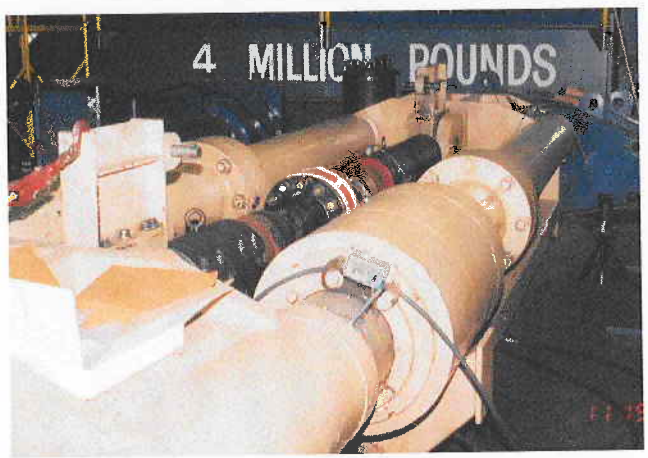

Figure 4 Flange assembly installed in SES load frame

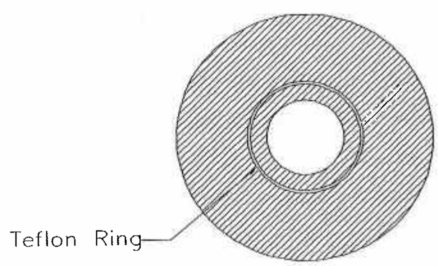

Figure 5 Drawing of the Pikotek 2-in 2500# Gasket

Figure 7 Flange assembly located in heating and cooling chamber

Figure 6 Raised face of blind flange prior to initial make-up

STRAIN AS A FUNCTION OF APPLIED BENDING MOMENT

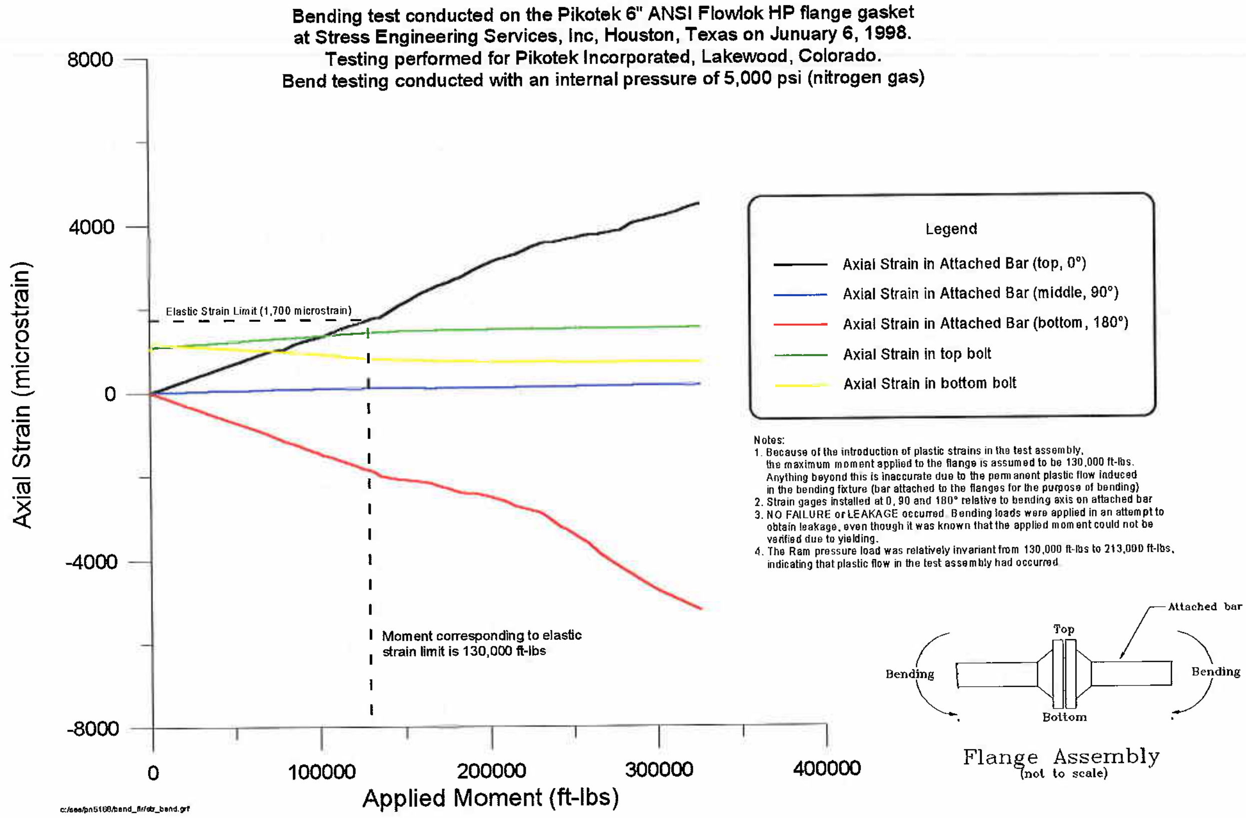

Figure 8 Strain gage results for bending load tests

INTERNAL PRESSURE AND FLANGE TEMPERATURE FOR 2"-2500# BLIND FLANGES AND PIKOTEK GASKET (0 to 18,500 psi AND 0°F to 300°F) CYCLE #1

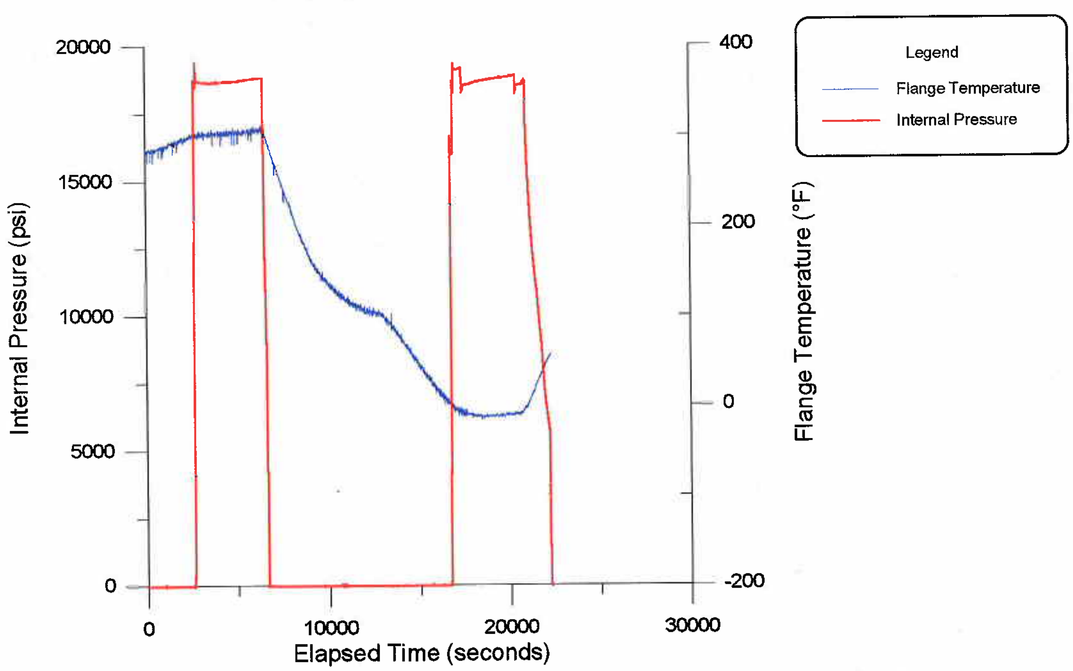

Figure 9 Temperature and pressure variations during Cycle #1

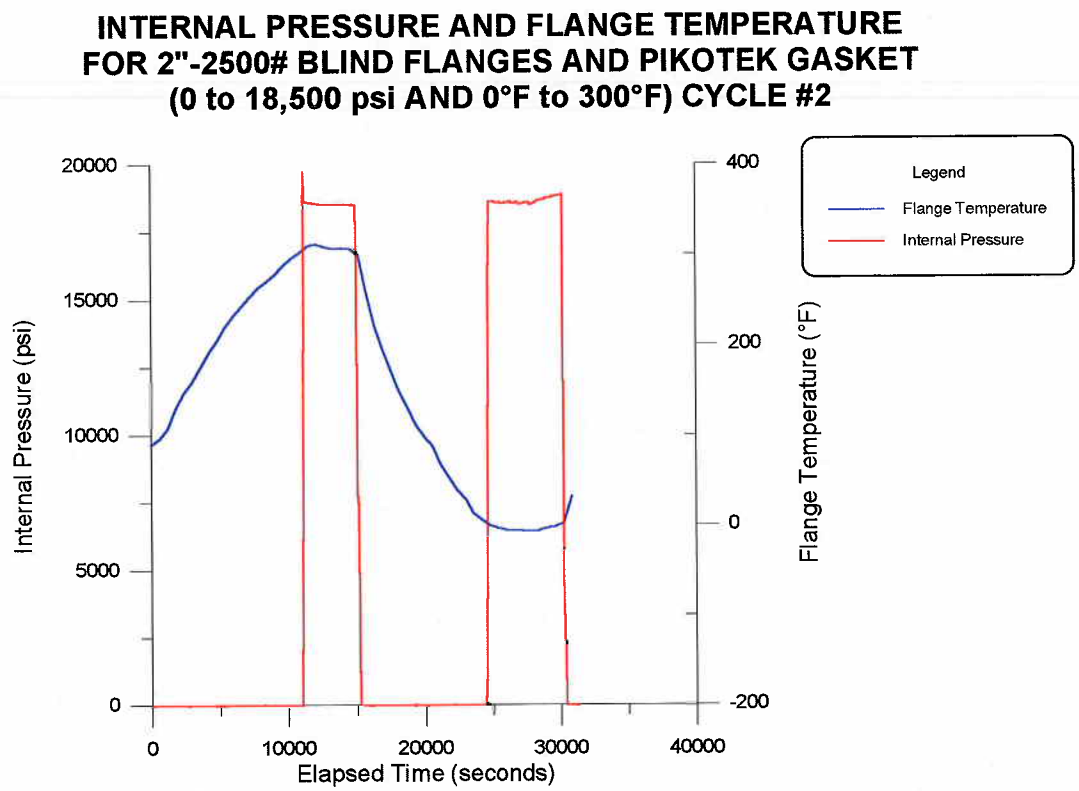

Figure 10 Temperature and pressure variations during Cycle #2

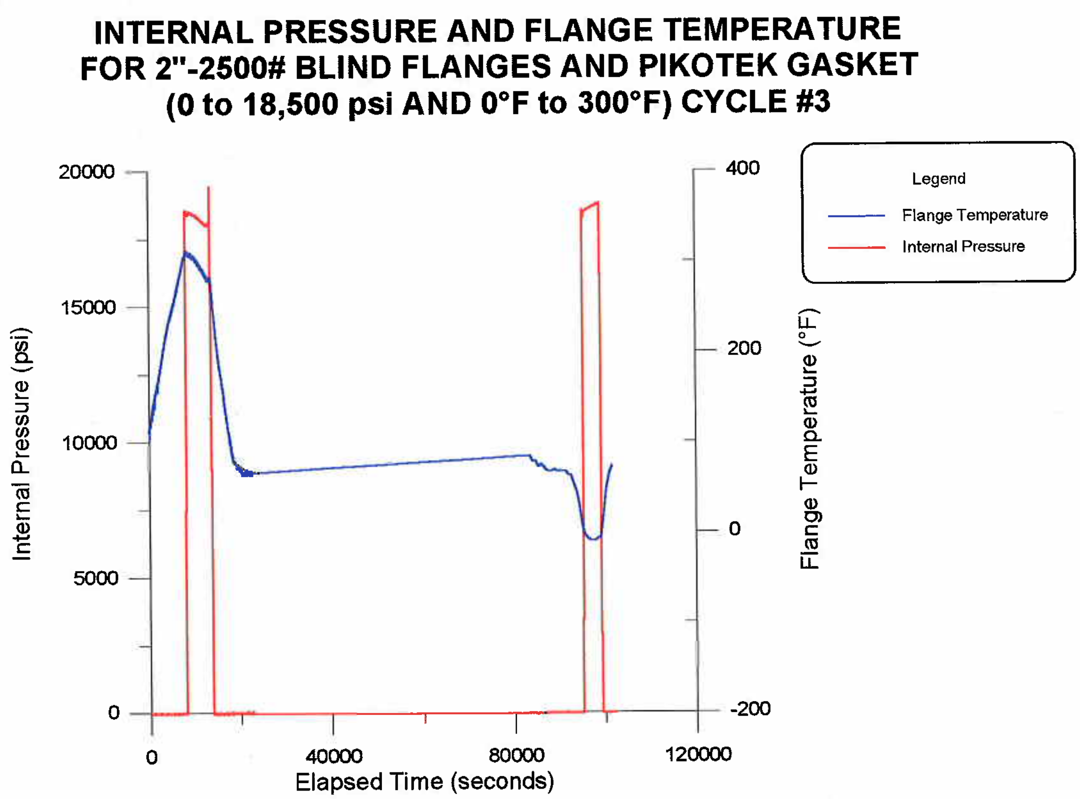

Figure 11 Temperature and pressure variations during Cycle #3

Table 1 Summary of Test Results

Elapsed Time (hours)	Flange Temperature (°F)	Internal Pressure (psi)	Stage Description (values in parentheses correspond to changes in pressure)
Cycle #1			
0:00	302	0	Oven preheated to 300°F
0:05	300	19,200	Start one hour hold
1:06	310	18,820	End one hour hold, start cool period (1.98 percent decrease)
4:02	-5	18,570	End cool down, start one hour hold
5:06	-3	18,690	End one hour hold (0.65% decrease)
Cycle #2			
0:00	306	0	Oven preheated to 300°F
0:02	306	18,660	Start one hour hold
1:04	304	18,550	End one hour hold, start cool period (0.59 percent decrease)
3:47	-3	18,690	End cool down, start one hour hold
5:17	0	18,930	End one hour hold (1.28 percent increase)
Cycle #3			
0:00	80	0	Start heat-up
2:18	309	18,500	End heat-up, start one hour hold
3:27	289	18,114	End one hour hold, start cool period (2.09 percent decrease)
6:54	-3.5	18,500	End cool down, start one hour hold
7:54	-0.9	18,800	End one hour hold (1.62 percent increase)

Table 2 Measurements Taken Before and After Testing

Gasket O.D.	5.630 inches *(initially 5.625 inches)*
Gasket I.D.	1.966 inches (composite material I.D. smaller than stainless steel I.D.) *(initially 2.0 inches)*
Gasket thickness	0.240 inches (0.233 inches on inside of raised face) *(initially 0.240 inches)*
Teflon ring thickness	0.238 inches (basically flush with surface of gasket) *(initially 0.278 inches)*
Teflon ring I.D.	2.856 inches *(initially 2.856 inches)*