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# THE INFLUENCE OF WINDING DENSITY IN THE SEALING BEHAVIOR OF SPIRAL WOUND GASKETS

**José C. Veiga**
**Carlos F. Cipolatti**
**Nelson Kavanagh**
Teadit Indústria e Comercio Ltda.
Rio de Janeiro, RJ, Brazil

**David Reeves**
Chevron Corporation
Chevron Richmond Refinery Tech Center
Richmond, CA, USA

## ABSTRACT

Spiral wound gaskets are used worldwide in piping and equipment flanges and can be manufactured in several combinations of materials, and in a wide range of dimensions, winding densities and shapes. This paper shows the sealability influence of winding densities, which are not specified by the current Spiral Wound B16.20 gasket standards, including flexible graphite filler thickness, height and number of windings. The effect of the flange rotation is also shown.

## INTRODUCTION

Most Spiral Wound Gaskets (SW) are produced according to ASME B16.20 – 2007 Metallic Gaskets for Pipe Flanges [1]. This standard indicates the dimensions and manufacturing tolerances for ASME B16.5 [2] and ASME 16.47 [3] flanges. Because SW Gaskets are widely used by industry in process piping and equipment they have been subjected to a wide range of research [4, 5, 6, 7, 8, 9].

The preferred design per ASME B16.20 for SW Gaskets consists of a sealing element with alternating plies of a metal and a soft filler spirally wound as shown in Figure 1, with an inner ring and a outer guide ring as shown in Figure 2. The nominal sealing element thickness is 4.45mm (0.175in), the inner and outer rings are 2.97mm to 3.33mm (0.117in to 0.131in).

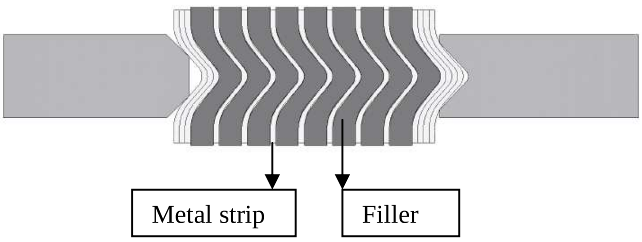


**FIGURE 1: SPIRAL WOUND GASKET WINDING**

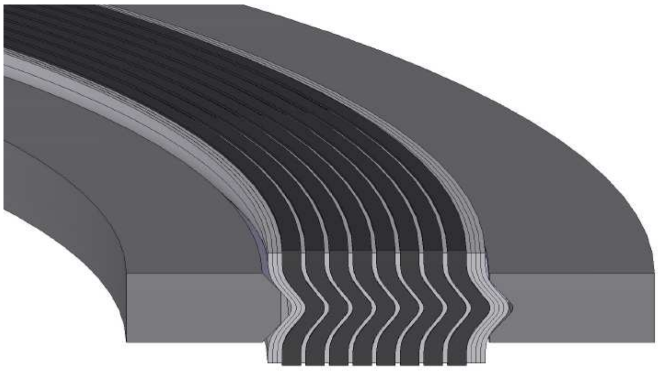

**FIGURE 2: SPIRAL WOUND GASKET**

For general service applications the winding metal is Stainless Steel or a Nickel Alloy, with Flexible Graphite or PTFE as filler. The ASME B16.20 standard specifies the metal strip nominal thickness as 0.19mm (0.0075in). There is no specification for the filler thickness. A compression test specification requires a thickness of 3.30mm +/- 0.13mm (0.130in +/- 0.005in) when the gasket is subjected to a compression force which varies according to the size and the flange pressure class. There is no sealability performance specification.

Failures due to the inward buckling of SW Gaskets are a known problem within the industry. Several reports, studies [10, 11, 12] and a US Patent [13] have linked these failures to the gasket construction. To prevent it the ASME B16.20 Standard was reviewed in 2007 to recommend that all SW Gaskets be fitted with an inner ring regardless of filler type. Previous editions required inner rings only for PTFE filled gaskets. For this paper all gaskets tested were with inner rings.

Studies have been performed with spiral wound gaskets showing their differences of compressibility according the winding density [14]. These studies showed large compressibility variations for the same gasket dimensions. SW Gaskets have been developed with "low stress capabilities" to address the lack of bolt load in Class 150 flanges [15, 16].

For this paper we were initially concerned with SW Gaskets for high pressure class flanges. There is a trend in the oil refining industry to use these gaskets instead of Ring Joint Gaskets (RTJ).

## TEST RIGS

All tests were performed in ASME B16.5 welding neck (WN), raised face (RF) flanges manufactured in ASTM 105 forged carbon steel [17]. Sealing surfaces per ASME (PCC-1 2010 Guidelines for Pressure Boundary Bolted Flange Joint Assembly [18]) for SW Gaskets it is 3.2 - 6.4 µm (125 - 250 µin). Figures 3 and 4 shows the 6 in Class 900 and the 3 in – Class 150 respectively.

**FIGURE 3: 6 in – CLASS 900 TEST RIG**

**FIGURE 4: 3 in – CLASS 150 TEST RIG**

All stud materials were ASTM SA-193-B7 [19] with machined ends to allow a precise bolt elongation measurement. The elongation was used to calculate the bolt load and gasket stress. All dimensions were measured at room temperature. Figure 5 shows the bolt elongation measurement.

**FIGURE 5: BOLT ELONGATION MEASUREMENT**

## TEST MEDIA PRESSURE AND LEAK DETECTION

Methane was the test media. It was chosen to establish a correlation with field surveys as mandatory by the US Environmental protection Agency (EPA) Fugitive Emission regulations. Measurements were performed using Thermo TVA 1000 Volatile Organic Compound Analyzer [20] with readings in parts per million (ppm).

Test Pressure was 20 bar (290 psi). All tests were at room temperature.

To reduce the effects of air currents in the laboratory, the flange edges were sealed with a tape with one orifice for the probe and another orifice opposite to the probe location as shown in Figure 6. This way the values show the Methane concentration in a constant flow. It is more severe that the EPA Method 21 [21], which verifies the concentration in the flange vicinity.

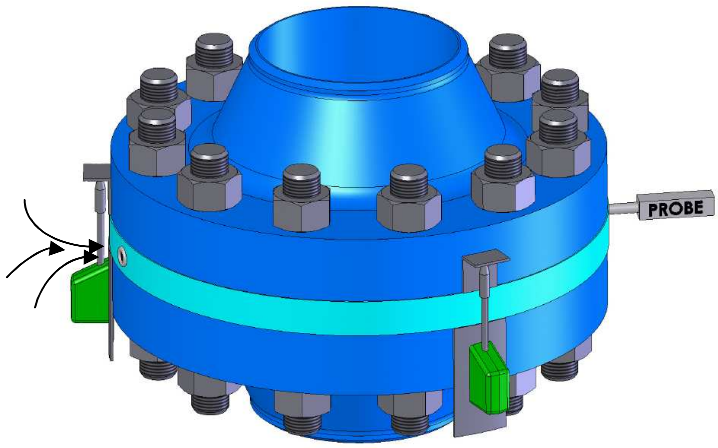


**FIGURE 6: LEAK DETECTION**

## GASKET DISPLACEMENT MEASUREMENT

To measure the gasket displacement, transducers were installed on the flange edge, 120 degrees apart, as shown in Figure 7.

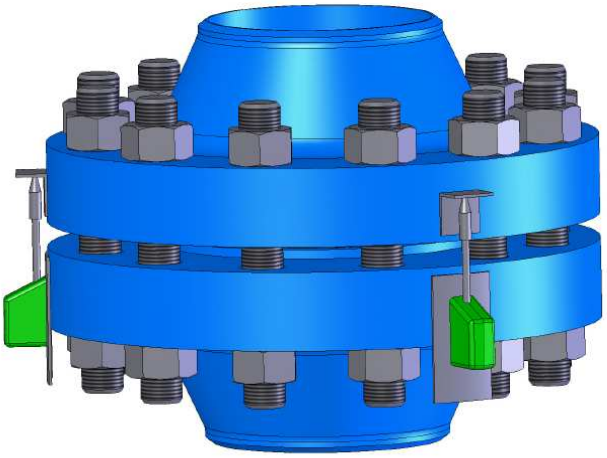

FIGURE 7: GASKET DISPLACEMENT TRANDUCERS

## TEST GASKETS

All gaskets were with Inner Rings in Stainless Steel type 304, Carbon Steel Guide Rings. Windings were in 304 Stainless Steel and Flexible Graphite Filler.

Several combinations of manufacturing winding force and filler thickness were tested. These combinations provide a number of filler sealing windings per mm (in) of gasket contact width, as shown in Table 1 and 2.

TABLE 1: TEST GASKETS 6 in – CLASS 900

| Density | | Sealing Windings per mm (in) |
|---|---|---|
| Low | A | 0.818 (20.77) |
| | B | 0.994 (25.25) |
| | C | 1.132 (28.75) |
| High | A | 1.509 (38.33) |
| | B | 1.698 (43.13) |
| | C | 1.824 (46.33) |

TABLE 2: TEST GASKETS 3 in – CLASS 150

| Density | | Sealing Windings per mm (in) |
|---|---|---|
| Low | A | 0.869 (22.07) |
| | B | 0.994 (25.25) |
| | C | 1.118 (28.40) |
| High | A | 1.491 (37.87) |
| | B | 1.615 (41.02) |
| | C | 1.863 (47.32) |

Gaskets which have more windings per gasket sealing width have more steel wraps, consequently, more density.

All gaskets tested were manufactured with high purity Flexible Graphite filler protruding approximately 0.2mm (0.008in) from the metal wraps as shown in Figure 8.

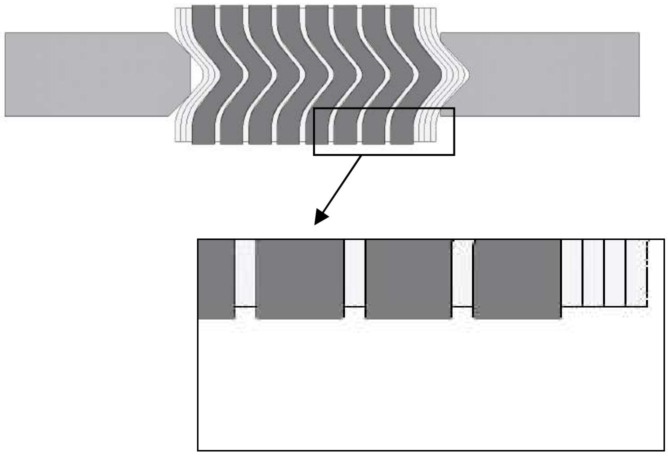

FIGURE 8: FILLER PROTRUSION

## TEST PROTOCOL

The Test Protocol was designed to reproduce field conditions of gasket installations. The ends of the studs were prepared to obtain elongation measurements with a micrometer. The stud stretch is used to calculate the gasket stress. Three displacement transducers were equally positioned around the flanges edges. The gasket seating stress was limited in the maximum yield strength of the studs and applied in steps.

A summary of the Test Protocol is as follows:

1 – Measure the thicknesses of the specimen before and after testing.
2 - Install flanges without gasket and record the initial displacement transducer value. This value is the zero displacement in charts.
2 - Install gasket and studs. Hand tighten nuts.
3 - Measure the initial stud lengths and record the value of the displacement transducer. The difference between this and the initial value is the gasket thickness.
4 - Tighten the studs to the required gasket stress using 3 cross pattern rounds, followed by two more rotational patterns.
5 - Measure and record stud length.
6 – Seal flange edges with tape between two opposite orifices.
7 – Pressurize with methane gas at 20 bar (290 psi);
8 – After 30 minute, measure the leakage in ppm with probe in orifice.
9- Repeat steps 4 to 8 for each tightness step.
10- Loosen the studs and record the displacement transducer value.
.

## TEST RESULTS

The following charts show test results for each representative gasket tested grouped by flange size and gasket density. The displacement is the average value for the three displacement transducers.

The initial displacement value is the winding thickness before tightening. The theoretical minimum displacement value is 3 mm (0.118 in), however due to the flange rotation, displacement has reached low values such us 2 mm (0.079 in). The theoretical minimum value is the thickness of both inner and outer rings, which are produced from solid metal plates and should function as gasket compression stops.

The difference between the last two values shows the gasket winding recovery plus the flange spring back.

Dotted lines show the Methane leak concentration in parts per million (ppm) for the corresponding gasket stress.

At least two tests were performed for each gasket size and density combination. The identification of each sample follows Tables 1 and 2.

## TEST RESULTS FOR 6 IN – 900#

The test results for the 6 in Class 900 are shown in Figures 9 to 14.

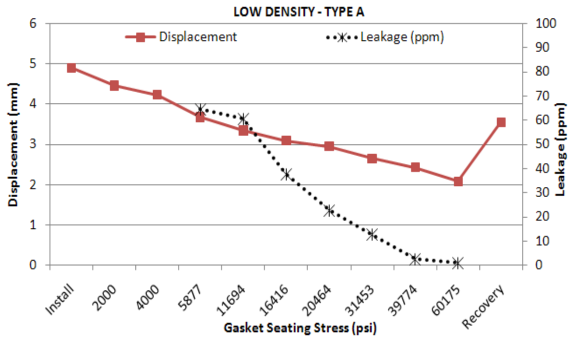


**FIGURE 9: LOW DENSITY – TYPE A**

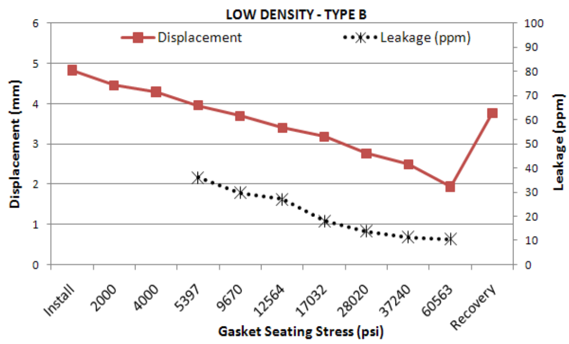


**FIGURE 10: LOW DENSITY – TYPE B**

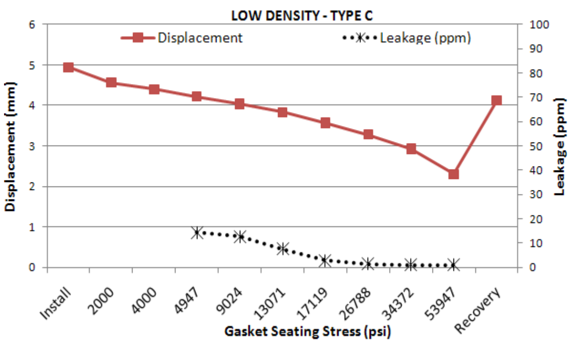


**FIGURE 11: LOW DENSITY – TYPE C**

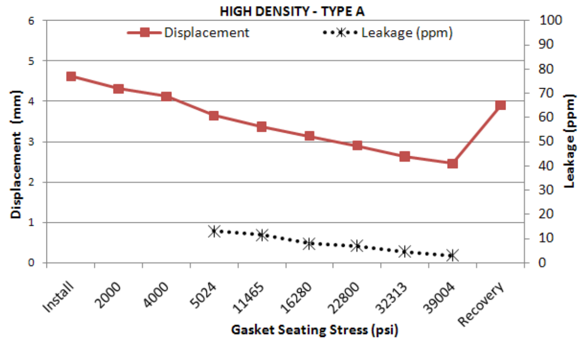


**FIGURE 12: HIGH DENSITY – TYPE A**

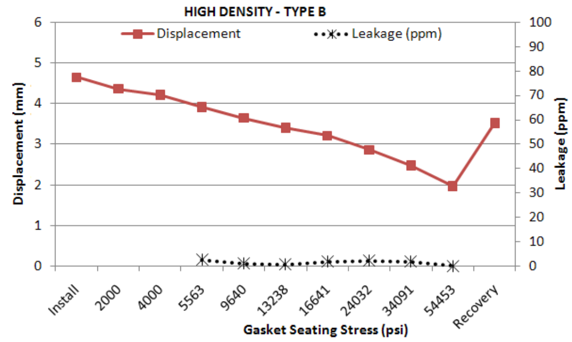


**FIGURE 13: HIGH DENSITY – TYPE B**

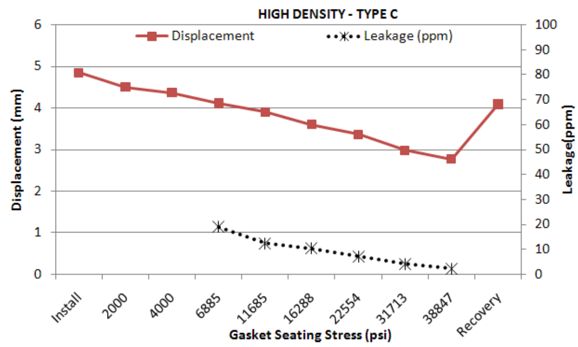


**FIGURE 14: HIGH DENSITY – TYPE C**

Test results show that the best sealability is achieved when the winding density is between 1.615/mm (41.02/in) and 1.863/mm (47.32/in).

The A and B low density gasket samples showed higher leak rates at the usual seating stress for piping flanges which is between 34.5 MPa (5,000 psi) and 207 MPa (30,000) psi. These lowest density gaskets showed leak concentrations in the range of 10 to 65 ppm for this sealing stress range. A visual analysis of the low density gasket samples after testing indicated that the raised face of the flanges were contacting the gasket guide ring as shown in Figure 15. For values less than 3 mm (0.118 in), which is the guide ring thickness, the displacement charts indicate that the flange's raised face edges have rotated and contacted the guide ring as showed in Figure 16. A confirmation of this effect is obtained by measuring the winding thickness on the inside and outside diameters as shown in Figure 17 and Table 3.

FIGURE 15: RAISED FACE GUIDE RING CONTACT

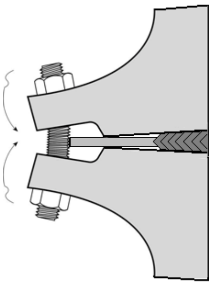

FIGURE 16: FLANGE ROTATION

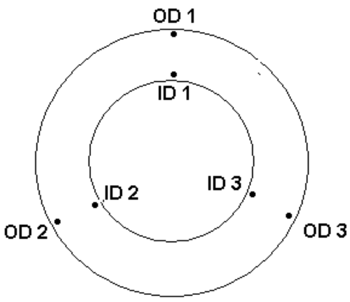


FIGURE 17: WINDING THICKNESS LOCATIONS

| Gasket Thickness (mm) | | | |
|---|---|---|---|
| Initial | | | |
| | ID | OD | Δ (ID-OD) |
| 1 | 4.688 | 4.767 | -0.079 |
| 2 | 4.709 | 4.805 | -0.096 |
| 3 | 4.721 | 4.785 | -0.064 |
| AFTER TEST | | | |
| 1 | 3.930 | 3.795 | 0.135 |
| 2 | 3.930 | 3.736 | 0.194 |
| 3 | 3.810 | 3.610 | 0.200 |
| Gasket Thickness (In) | | | |
| Initial | | | |
| | ID | OD | Δ (ID-OD) |
| 1 | 0.185 | 0.188 | -0.003 |
| 2 | 0.185 | 0.189 | -0.004 |
| 3 | 0.186 | 0.188 | -0.003 |
| AFTER TEST | | | |
| 1 | 0.155 | 0.149 | 0.005 |
| 2 | 0.155 | 0.147 | 0.008 |
| 3 | 0.150 | 0.142 | 0.008 |

TABLE 3: GASKET WINDING THICKNESS

Additional tests were performed to determine if the sealability was being provided by the guide ring instead of winding. To verify this effect a small groove was machined on both sides of guide rings as shown in Figure 18.

FIGURE 18: GUIDE RING GROOVES

The sealability for these sample gaskets are shown in Figure 19 for a Low Density gasket and in Figure 20 for a High Density gasket.

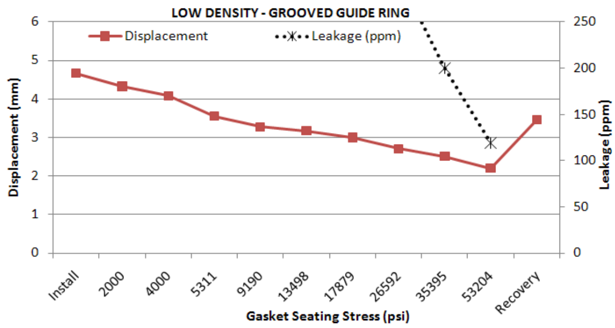


FIGURE 19: LOW DENSITY WITH GROOVES

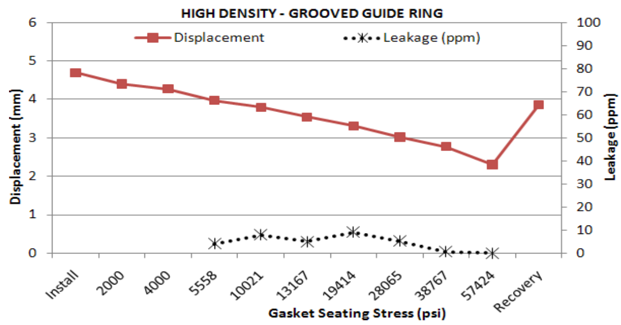


FIGURE 20: HIGH DENSITY WITH GROOVES

When the tape (figure 6) was removed and the VOC probe was moved around the flange OD, with readings of 3 ppm, but up to the vicinity of the groove, it increased to 100 ppm.

The behavior of the High Density gaskets was not affected by the guide ring groove, indicating that the seal was being provided by the windings.

It was not possible with a Low Density gasket to get a good seal even at high stress levels, indicating that the guide ring has been acting as a flat metal gasket. This is not considered to be a reliable seal because the small amount of thermally driven differential expansion between the flanges will cause the metal to metal seal to leak at some point. An example of this is shown in Figure 21, which shows the thermally induced movement between two 10”, 1500 class flanges over 7 days of operation. Field experience has shown that a leak can develop anywhere from a day after startup, to 15 years later after a major plant upset, and the severity of the leak can range anywhere from a PPM level violation of a consent decree to a major fire.

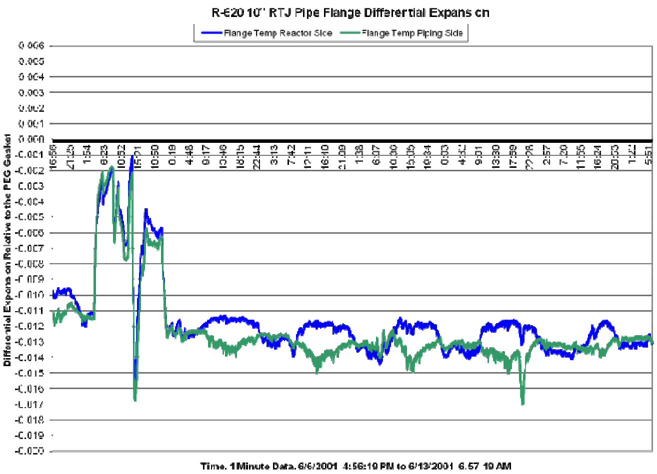


**FIGURE 21: Flange Movement Relative to the RTJ Ring**

ASME B16.20 requires that gaskets be designed so that a uniform bolt stress of 30,000 psi will compress the gasket to a thickness of 0.130 in ± 0.005 in (3.3 mm ± 0,127 mm).

For gaskets 6" 900# the stress x compressed thickness was determined as follows:

$$Gs = (Bs*Br*N)/Ga$$

were

Gs - Gasket Stress

Bs - Bolt Stress = 30000 psi

Br - Bolt Root Area = 0.7276 in.$^2$

N - Number of bolts = 12

Ga - Gasket Area = 16.28 in.$^2$

For this test a pin was inserted near the flange welding neck as shown in Figure 22, OD of the sealing area, to reduce the effect of the flange rotation. This way the measured displacement value is closer to the actual winding thickness at the corresponding stress.

The calculated Gs at uniform bolt stress of 30,000 psi is 16,089 psi (111MPa). Given the value of Gs, we draw a vertical line on the chart to determine the corresponding winding thickness. Figures 23 and 24 show the calculated gasket stress and the corresponding winding thickness.

**FIGURE 22: MEASURING PIN**

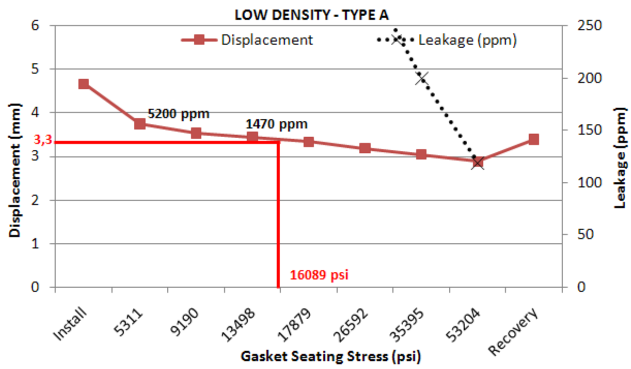


**FIGURE 23: LOW DENSITY STRESS X DISPLACEMENT**

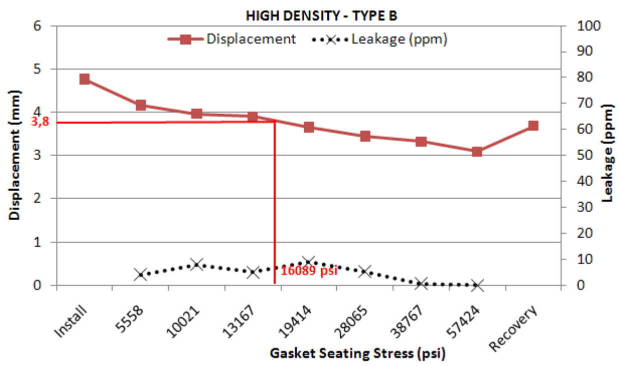


**FIGURE 24: HIGH DENSITY STRESS X DISPLACEMENT**

It can be observed in Figure 23 that a gasket that meets the ASME B16.20 winding crush requirement (3.3 mm, 0.130 in) at the required bolt stress of 30,000 psi will not seal properly.

On the other hand, the High Density gaskets that do not meet the ASME B16.20 crush requirements exhibit low leak concentrations across a very broad range of gasket stresses, as shown in Figure 24.

An additional ASME B16.20 requirement is that “the filler shall be essentially flush with, but not below, the metal winding on both contact faces of the gasket”. Gaskets with the filler flush with the metal winding as shown in Figure 25 were tested. A result is shown in Figure 26.

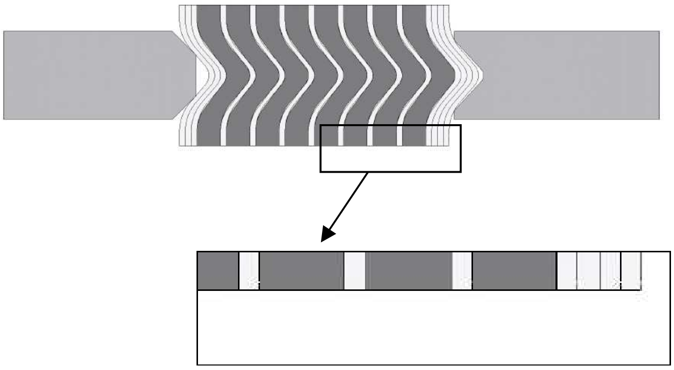

FIGURE 25: FILLER FLUSH WITH WINDING METAL

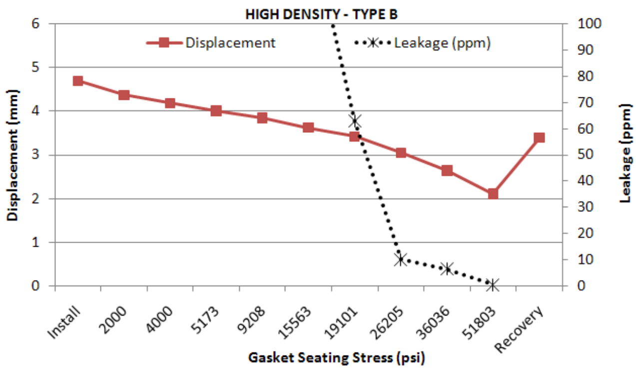


FIGURE 26: FILLER FLUSH WITH WINDING METAL

It can be observed that a High Density gasket that provided a good seal does not exhibit the same behavior until a much higher seating stress is achieved. This test result suggests that the filler has to be protruding from the metal windings in order for the graphite to achieve the density needed for optimum sealability. Gaskets that exhibited a good sealability were built with the filler protruding approximately 0.2mm (0.008in) from the metal windings as shown in Figure 8 above.

## TEST RESULTS FOR 3 IN – 150#

Test results for the 3 in Class 150 flanges are shown in Figures 27 to 32. This flange size is known for its limited maximum seating stress [22]. The objective was to have a gasket behavior comparison between a high stress flange, like the 6 in Class 900, with a lower stress one.

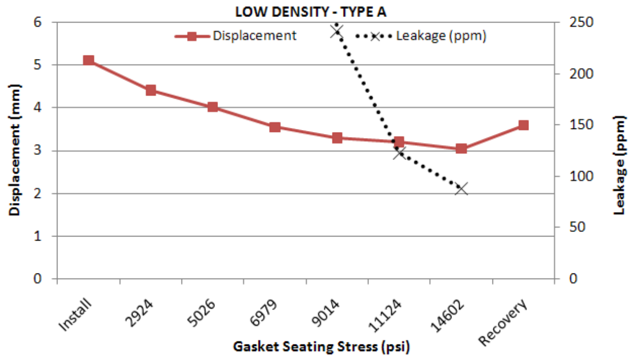


FIGURE 27: LOW DENSITY – TYPE A

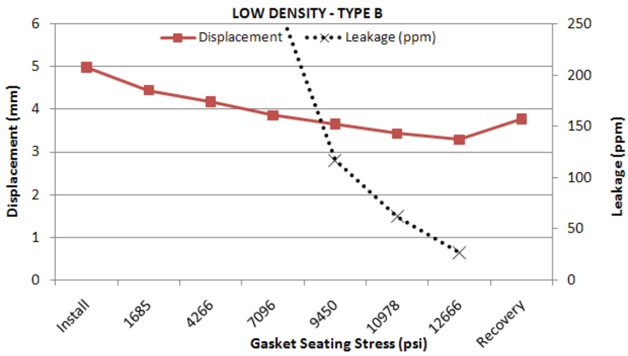


FIGURE 28: LOW DENSITY – TYPE B

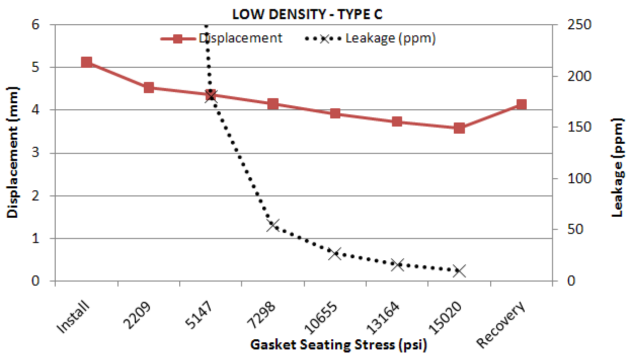


FIGURE 29: LOW DENSITY – TYPE C

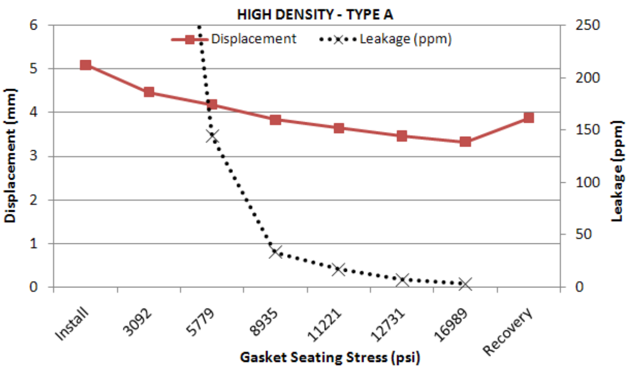


FIGURE 30: HIGH DENSITY – TYPE A

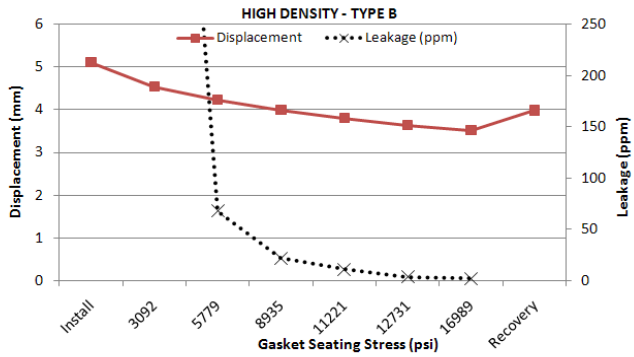


FIGURE 31: HIGH DENSITY – TYPE B

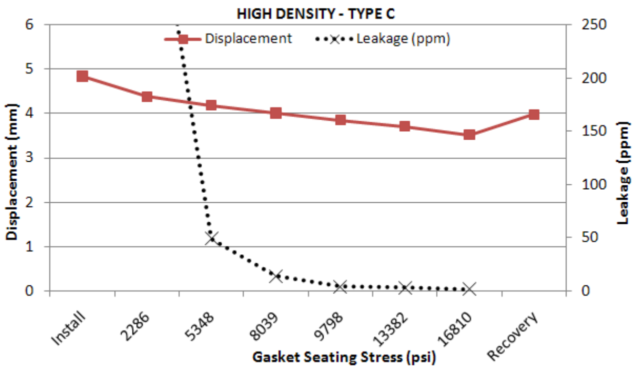


FIGURE 32: HIGH DENSITY – TYPE C

Test results show the same behavior, in spite of the limited bolt load available, gaskets with higher density seal better than lower density ones.

It was observed that the flange's raised face did not contact the guide ring. The charts give an indication of this since the displacement values are higher than the 3 mm (1/8 in) guide ring thickness. For this reason no test with a grooved guide ring was performed.

## TEST RESULTS - DIFFERENT MANUFACTURERS

A study was also conducted to evaluate the amount of winding crush on standard SW gaskets from multiple manufacturers. The objective was to determine if any of the manufactures met the B16.20 winding crush requirement. The gaskets were installed in a JJenco Flange Assembly Demonstration Unit (FADU) [23] which consists of a 4" 150 class flange, where each of the 5/8" studs contains a strain gage that allows the stud stress to be read directly. Testing was done at room temperature by loading the gasket to an average stud stress of 88,300 psi to obtain an 18,500 psi gasket stress, then removing the gasket and inspecting the guide ring and final winding height. Given that this stress is over 3 times higher than the 30,000 B16.20 requirement bolt stress for this flange class, all flanges should have contacted the outer guide rings.

Figure 33 contains the results. Using the first gasket labeled "1 No Inner Ring" as an example, we can see that the windings can be expected to rebound about 0.305 mm (0.012in) after the gasket had been removed. Both sides of this guide ring showed significant contact by the raised faces. This level of contact was also observed in gasket 5, but not in gasket 6, suggesting the density in the gasket 6 windings was high enough to prevent significant contact with the outer guide ring after initial loading.

Gasket 12 is the "High Density type B" shown above after being subjected to an 18,500 psi gasket stress in a 6" 900 class flange. Gasket 13 is the same gasket after being reinstalled and subjected to a calculated gasket stress of 57,000 psi.

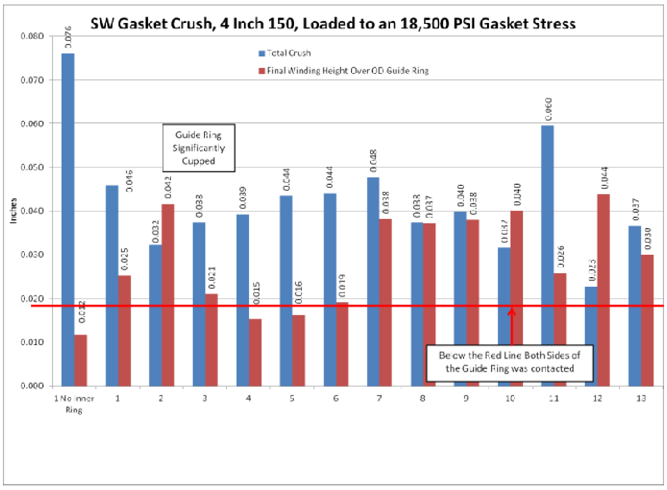


FIGURE 33: SW Winding Crush

## TEST RESULTS - REINSTALLED SW

It is interesting to note the leakage measurement for gasket 13 after it was reinstalled (right side), versus the leakage measurements after its initial installation (left side), which is shown in Figure 34. While we are not attempting to suggest that SW gasket can be reused, this result is consistent with field experience using several graphite covered metal core gasket designs.

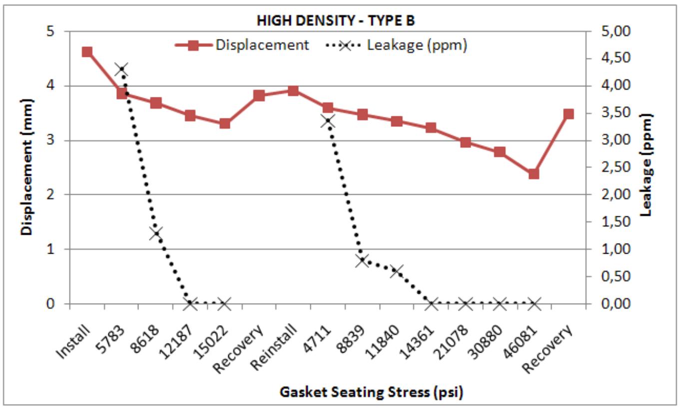


FIGURE 34: SW Gasket Sealability after Reinstallation

## CONCLUSIONS

As previous studies have indicated, the winding density is a fundamental SW Gasket parameter. In order to reliably meet current maximum fugitive emission levels, a seating leak test must be performed to assure that a specific gasket construction will perform as required. This test needs to ensure that the windings are providing the seal and not the less reliable guide rings, and the test much include thermal cycles to mimic plant startups, shutdowns and upsets. Chevron has already developed a test procedure.

Tests shown in this paper indicates that ASME B16.20 needs to address gasket sealability requirements in order to meet the Fugitive Emissions requirements mandated by the U.S. Environmental Protection Agency (EPA). The Hazardous Organic National Emission Standards for Hazardous Air Pollutants (also known as the HON rule) in 1995 was the first Federal rule to require regular monitoring of connectors and flanges in US refineries. Since then a number of local air quality management districts, as well as EPA consent decrees, have both increased inspection frequencies and lowered the acceptable level of fugitive emission. Given that it is much more likely that future regulations will continue to become more stringent than less, end-users with fugitive emissions requirements would significantly benefit if a sealability requirement was included in B16.20.

During testing a high dree of flange rotation was observed, even for a high pressure class forged steel flange. A study of this effect on the gasket's sealability is suggested as a future research.

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[1] ASME B16.20 – 2007 Metallic Gaskets for Pipe Flanges, American Society of Mechanical Engineers, 3 Park Avenue, New York, NY 2008

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