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SPIRAL WOUND GASKET COMPRESSIBILITY AND PRESSURE RATINGS

John McCarthy
Dominion Generation
Chesapeake, VA U.S.A.
john.d.mccarthy@dom.com

A. Fitzgerald Waterland, III (Jerry)
VSP Technologies, Inc.
Prince George, VA U.S.A.
jerry.waterland@vsptechnologies.com

Dan Reid
VSP Technologies, Inc.
Houston, TX U.S.A.
dan.reid@vsptechnologies.com

ABSTRACT

The industry practice of using pressure rating nomenclature to describe spiral wound gasket compressibility can contribute to some level of confusion regarding the proper design and selection of these gaskets. This situation can result in a misconception that a more easily compressed "soft" gasket (for example, "0-999 psi" rating) cannot be used in a higher pressure application. This is not necessarily true, and in many cases a softer (less dense) gasket construction can actually be beneficial in both high and low pressure applications. This article addresses both the terminology used to describe spiral wound gasket compressibility and the design characteristics of these gaskets in an effort to improve the understanding of this subject.

INTRODUCTION

Most manufacturers rate the compressibility of their spiral wound pipe flange gaskets as "400#", "600#", "900#", etc., to match the ASME B16.5 pressure class of the piping system in which they are used. Windings-only spiral wound gaskets are typically assigned a compressibility rating using pressure units of "0-999 psi", "1,000-3,000 psi", etc. While in some cases the gasket compressibility rating is directly related to the operating pressure of the application, in many other cases there is no such relationship. This difference has to do primarily with the design of the gasket and the gasket seating area.

GASKET DENSITY

An important function of any gasket is to be soft enough to compress and conform to the seating surface imperfections, and at the same time be stiff enough to resist the applied forces without damage. The ability of a gasket to deform as desired under a given load is described as its "compressibility". In the case of spiral wound gaskets "compressibility" represents a measure of the gasket density.

The images below illustrate how the gasket sealing element (windings and filler) may be manufactured to different densities (stiffness) to support the compressive forces applied either from bolting, or from internal pressure if that internal pressure is used to compress the gasket.

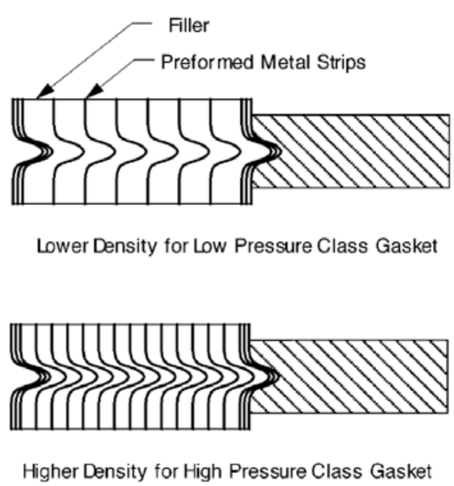

Figure 1 Cross sectional view of spiral wound gaskets with an outer centering / compression-stop ring [1]

GASKET SEATING SURFACE DESIGNS

The following images depict a few of the many types of flange face arrangements that are used in piping systems and equipment design. Gasketed joints that rely on metal-to-metal contact to avoid gasket over-compression are referred to as “hard” joints. Joints that rely on gasket stiffness alone to resist over-compression are called “soft” joints. According to EPRI, hard joints are an “industry-recommended practice” for reliable performance [1].

The first flange type, shown in Figure 2, is identified as “Male and Female”. A version of this design is often used for equipment covers and manway doors. In the case of boiler manway doors the force used to compress the gasket is typically not generated by bolting, but is supplied by internal pressure acting on the surface of the door itself. In this application, it is often impractical to incorporate a compression-stop ring, so the gasket density alone must support the force applied by internal pressure. In addition, the gasket must be soft enough to deform and seal under the available load. In these cases, it becomes critical to use a low density gasket in low pressure applications (0-999 psi), and a high density gasket in higher pressure applications (1,000 – 3,000 psi). This is a typical “soft” joint design.

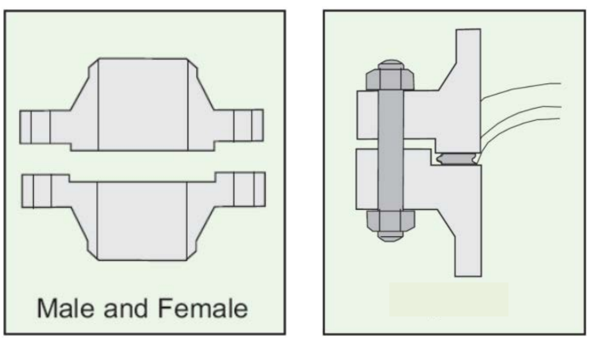

Figure 2 “Soft” joint male and female flanges with a windings-only spiral wound gasket [2]

In the second flange type, shown in Figure 3 and described as “Raised Face”, the amount of force applied to compress the gasket is determined strictly by the amount of bolting available. If a windings-only gasket were to be used in this application, the gasket density would need to be adequate to resist the bolting load, which is defined by the pressure class (400, 600, 900, etc.). However, the more conventional approach for this application is to use a gasket design that incorporates an outer metallic “compression-stop” ring, shown in the image on the right of Figure 3. The use of a properly selected and assembled gasket of this style, compressed down to the outer ring, results in a “hard” joint design that prevents over-compression of the gasket, and allows the use of a lower density (softer) gasket.

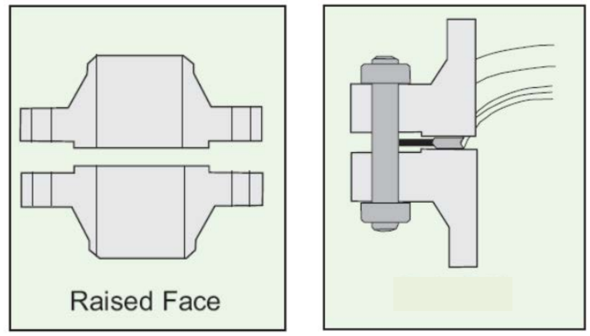

Figure 3 “Hard” joint, raised face flanges with a spiral wound gasket equipped with an outer compression-stop ring [2]

The third flange type shown in Figure 4 below, “Flat Face to Recess”, is a “hard” joint design. It provides the gasket with protection against over-compression because the gasket sits in a protected recessed area. This eliminates the need for either an outer or inner ring gasket compression-stop, or a higher density construction of the gasket sealing element. In this case, a low density gasket construction is often the best choice. A higher density gasket may be used, but only if there is adequate bolting available to properly compress the gasket.

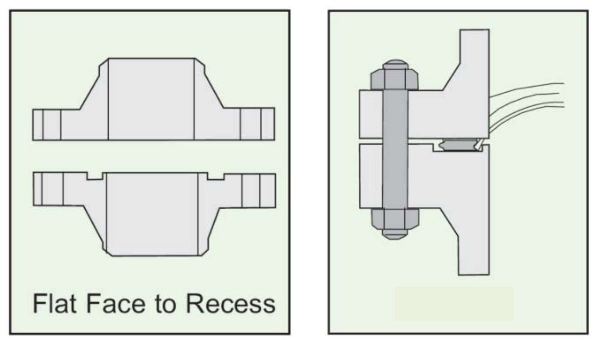

Figure 4 “Hard” joint, flat face to recess flanges with a windings-only spiral wound gasket [2]

OPERATING CHARACTERISTICS OF SPIRAL WOUND GASKETS

Figure 5 on the following page shows a schematic cross-section of a spiral wound gasket contained between two pipe flanges. The gasket windings and filler are shown in pink color. The gasket is fitted with reinforcement on the inside diameter (ID), and an outer “compression-stop” ring on the outside diameter (OD), both shown in green color. Over-compression of the gasket is prevented by the compression-stop ring, which also serves to contain the gasket and oppose internal system pressure. The bolting serves to keep the flanges in contact with the gasket and oppose the internal pressure that is trying to separate the flanges.

One of the unique features inherent in the design of this spiral wound gasket is that the windings and filler will deform to provide a balanced resistance to increasing internal pressure. As internal system pressure acts against the ID of the gasket, the windings are radially compressed, which leads to the filler being squeezed outward and against the flange faces [3]. Due to the concave (chevron) shape and orientation of the metal windings facing the internal pressure, the more the internal pressure increases, the more sealing force is generated against the flanges. This assumes of course that the bolting tension is sufficient to keep the flange faces in contact with the gasket. A photo of this gasket style is shown in Figure 6.

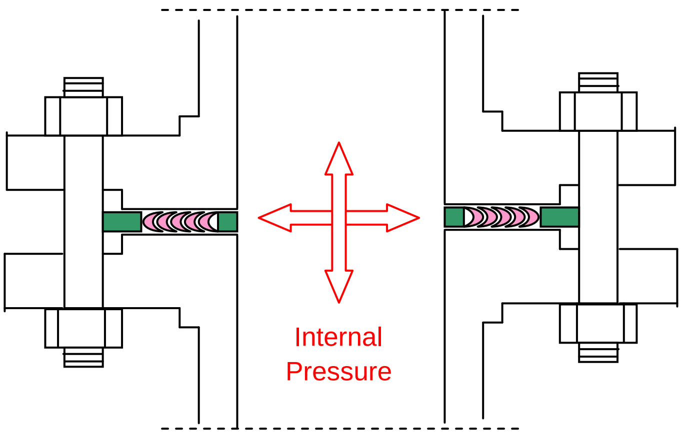

Figure 5 Cross section of a spiral wound gasket with inner and outer rings installed between raised face flanges

Figure 6 Photo of a typical pipe flange spiral wound gasket with inner and outer metal rings

In many spiral wound gasket applications, gasket over-compression is prevented by the use of a compression-stop ring, or, if a windings-only gasket is used, by installing the gasket in a protective recessed seating area (Figure 4). Extrusion of the gasket towards the OD is prevented either by a compression-stop ring, or by confining a windings-only gasket in a recessed seating area. Therefore, for a properly designed spiral wound gasketed joint, the internal pressure rating of the application will have little bearing on the choice of gasket density. Under these conditions, a softer gasket density can be beneficial, both to promote flowability of the filler (which produces maximum contact between the filler and the seating surfaces), and to reduce the amount of bolting required to compress the gasket.

GASKET COMPRESSIBILITY RATINGS – PIPE FLANGE GASKETS

Spiral wound gasket manufacturers have typically supplied their gaskets with different densities to match the bolting loads of the application. In the case of pipe flange gaskets, these densities are often identified using the nomenclature of the associated piping pressure class (for example, “400#”, “600#”, etc.). However, as discussed previously, these different gasket densities are not intended to resist design operating pressures, but are instead presumably selected to accommodate the bolting loads developed in each of these pressure classes.

The graph shown in Figure 7 illustrates how the ratio of bolt tensile stress area to gasket seating area varies for NPS (Nominal Pipe Size) 4 ASME B16.5 weld neck flanges in classes 150 through 1500. The inside diameter of the raised face seating area has been chosen based on typical pipe schedules for each flange pressure class. This trend may be used to compare the relative amount of bolting force available per square inch of gasket seating area across the full range of pressure classes.

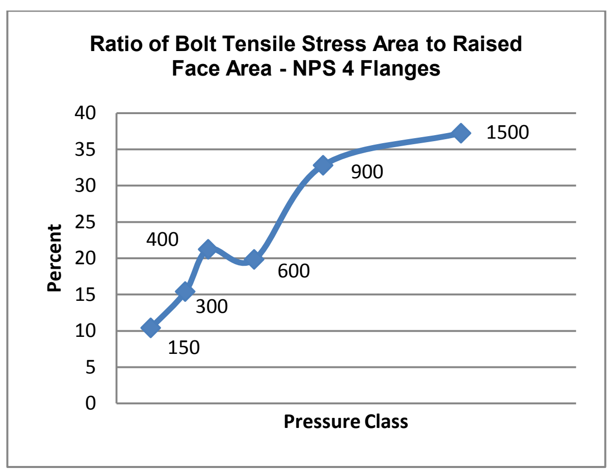

Figure 7 Ratio of bolt tensile stress area to raised face area NPS 4 flanges

The trend shown by this graph is similar to the trends for other flange sizes, such as NPS 2, 8, and 12, which all indicate a generally increasing available bolt load per square inch of seating area along with increasing flange pressure class. If manufacturers are designing their spiral wound gaskets to correlate with this increasing compressive force per unit area, then one would expect to see corresponding changes in gasket modulus of elasticity (gasket stiffness) along with increasing gasket pressure class ratings.

The graphs in Figures 8 and 9 show plots of compressive modulus of elasticity for gaskets supplied by two international spiral wound gasket manufacturers for pressure classes 150 through 1500. These plots were derived from 1995 ROTT test data, shown in Table 1 on page 8, performed on a random sample of two spiral wound gaskets (averaged) for each of the indicated pressure classes for each manufacturer. The ROTT testing was performed by an independent laboratory and was used to identify and quantify stress/leakage and stress/deflection characteristics of these standard ASME B16.20 spiral wound gaskets.

While Figure 7 on the previous page shows an increase in available compressive force per unit area of about 270% from class 150 to class 1500, Figures 8 and 9 show a maximum increase in gasket modulus of elasticity of only about 66% over this same range. In addition, the shapes of the trend lines for the two gasket manufacturers seem to exhibit little similarity where modulus of elasticity is concerned.

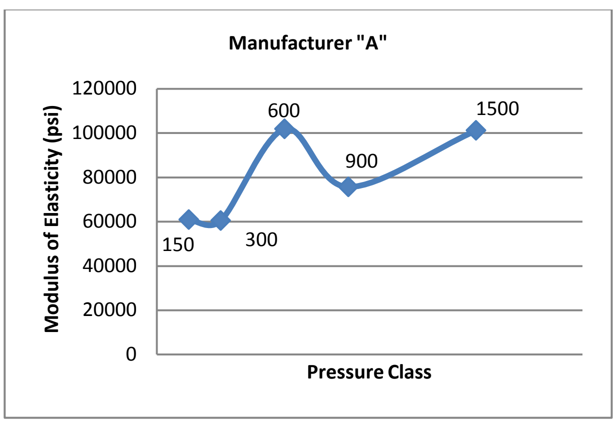

Figure 8 Variation in gasket modulus of elasticity with pressure class rating - Manufacturer "A"

Since leak rate is a primary criterion for judging gasket performance, these varying approaches to gasket design may lead one to question how differing gasket moduli of elasticity may affect leak rate. The two graphs in Figures 10 and 11, for the same manufacturers "A" and "B", show trends of measured leak rate versus compressive modulus of elasticity. These leak rates were measured during the 1995 ROTT test at 400 psi internal pressure. Except for an unexplained anomaly involving two class 150 gaskets from manufacturer "B", all measured leak rates were below 0.0001 mg/sec.

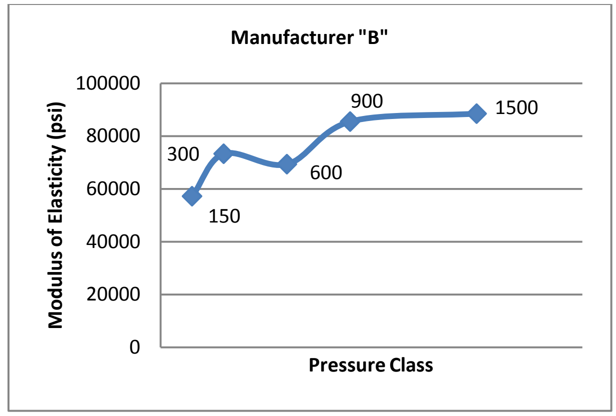

Figure 9 Variation in gasket modulus of elasticity with pressure class rating - Manufacturer "B"

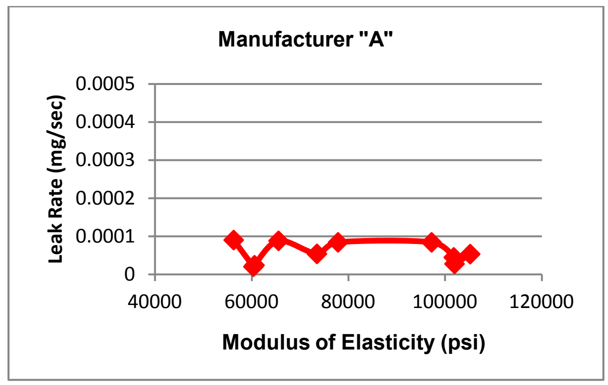

Figure 10 Leak rate versus modulus of elasticity - Manufacturer "A"

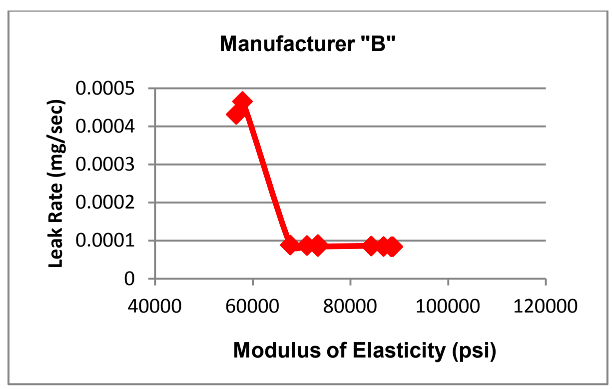

Figure 11 Leak rate versus modulus of elasticity - Manufacturer "B"

This data suggests that within the range of gasket moduli of elasticity used by these manufacturers, and within the parameters of this ROTT test, modulus of elasticity seems to have no significant, defined effect on spiral wound gasket leak rate.

As there are no enforced standards[1] for specific pressure/compression construction of spiral wound gaskets, there is no standardization in filler density for various pressure classes between manufacturers. For the same pressure class gasket, it is not unusual to see extreme variation in the filler density in gaskets supplied by different manufacturers [3].

The bar graph shown in Figure 12 below illustrates the amount of variation that may be found in spiral wound gasket construction among manufacturers, and even within a single manufacturer, for the same NPS gasket and pressure class. In this example, calibrated calipers were used to measure sealing element cross-sectional width, and a magnifying comparator was used to count the number of windings. Winding density varied between 27 and 38 windings-per-inch, or 29% across the range of gaskets tested.

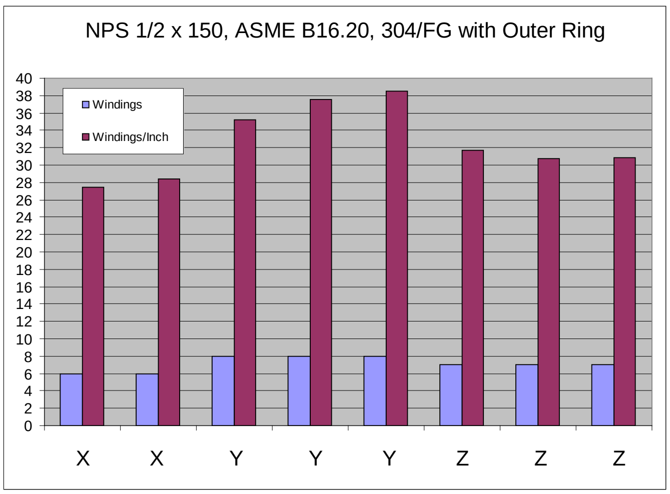

Figure 12 Spiral wound gasket construction for manufacturers “X”, “Y”, and “Z”, size 1/2”, pressure class 150

The photos in Figure 13 provide a visual example of a variation in winding density for two comparable spiral wound gaskets from two different manufacturers, each supposedly constructed per ASME B16.20 manufacturing requirements. The photos in Figure 14 show another example of variation in winding density, but this time using gaskets of the same size and pressure class from the same manufacturer. The samples were chosen to show extremes of variation in commercially available spiral wound gaskets. The samples were taken from typical warehouse inventories, which included a nuclear generating facility, and they are expected to be used in production equipment.

[1] ASME standard B16.20A, paragraph 3.2.6, describes compressibility requirements for spiral wound gaskets. However, these limits are routinely exceeded by many gasket manufacturers. According to a year 2000 EPRI report, “less than half of the gaskets tested reached full compression at 30 ksi bolt stress” (as required by the ASME standard) “but… most gaskets had reached full compression at a 60 ksi bolt stress” [4].

Figure 13 Variation in winding density for the same size and pressure class spiral wound gasket - NPS 1 x 150 Manufacturers “A” (top) and “Z” (bottom)

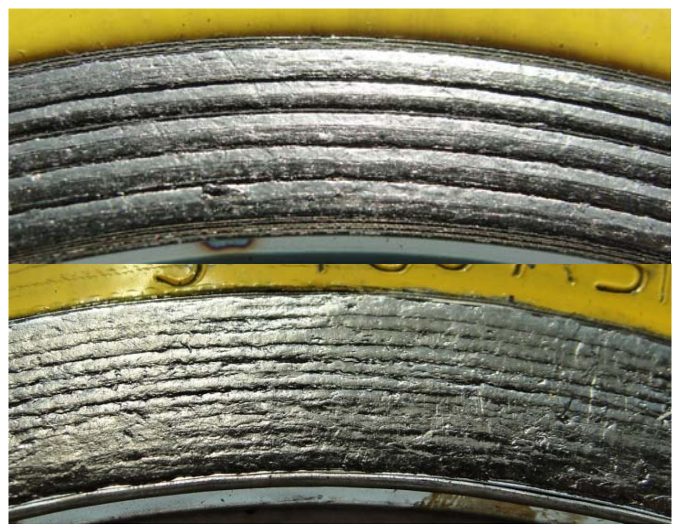

Figure 14 Variation in winding density for the same manufacturer, size, and pressure class spiral wound gasket, NPS 3 x 150, manufactured in Korea (top photo) and Mexico (bottom photo)

One would expect that spiral wound gaskets of the same size and pressure class should be identical regardless of the manufacturer or batch in which they were produced. This lack of standardization and manufacturing control can result in differing compression characteristics and unpredictable sealing performance for the end user.

The inconsistencies in spiral wound gasket construction have influenced the authors of this paper to work towards development of gasket standards which exceed the requirements of ASME B16.20, Metallic Gaskets for Pipe Flanges – Ring-Joint, Spiral-Wound, and Jacketed. These more aggressive standards require non-buckling, controlled-density design of windings and filler, and low-stress-to-seal construction for all ASME flange pressure classes. These standards are currently in use throughout one electric utility. In contrast to the use of varying gasket densities described previously, the resulting alternative design spiral wound gasket, shown in Figure 15, incorporates a single, low density construction for all pressure classes from class 150 through class 2500.

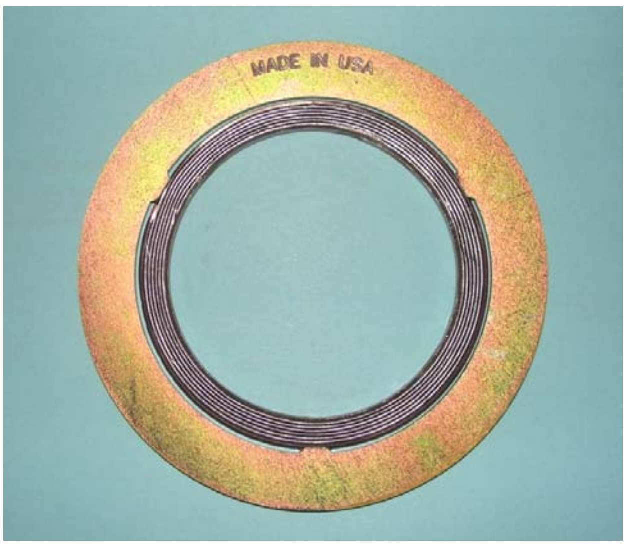

Figure 15 Spiral wound gasket with anti-buckling, controlled-density windings and filler, and low-stress-to-seal construction (Note: no inner ring required)

GASKET COMPRESSIBILITY RATINGS – WINDINGS-ONLY GASKETS

The ratings used to identify windings-only gaskets follow a convention similar to that used for pipe flange gaskets. In windings-only gasket applications, which involve primarily equipment joints (as opposed to pipe flanges), gasket densities are described in relation to the design pressure of the application. Typical identifiers are “0-999 psi” and “1,000-3,000 psi”. However, as with pipe flange gaskets, the choice of gasket density is more directly related to the compressive load applied to the gasket by the bolting. One exception to this rule is unconfined boiler manway or handhole door gaskets, which do rely on gasket density to resist forces applied by internal system pressure. This is one of the few applications where one must ensure that the correct winding density is specified and received.

DISCUSSION

The question is, will a low-density construction gasket perform acceptably in a high pressure application? The answer appears to be yes, when constructed with an inner or outer ring, or if used in a properly sized groove; both of which limit compression and prevent over-compression. If the gasket is used in a boiler drum door or handhole application, which is sealed by internal pressure and lacks similar protections, the answer is no. The high internal pressure acting against the hydrostatic area of the drum door or handhole cover will develop a very high compressive stress on the gasket. If the gasket is not designed to resist this stress, it can over-compress and over-stress the gasket resulting in failure.

The photo in Figure 16 shows an example of different density boiler manway gaskets. These “windings only” gaskets are constructed using alternating layers of stainless steel windings and compressible flexible graphite filler material. The higher density construction gasket on the left is designed to resist higher internal boiler pressures, which would tend to over-compress and crush the lower density gasket shown on the right. The photos in Figure 17 on the following page show a failed boiler manway door gasket. This failure was due to the use of an incorrect “low density” (0-999 psi rating) spiral wound gasket as opposed to a required “high density” (1,000 – 3,000 psi rating) gasket in a high pressure boiler drum application (2,400 psig).

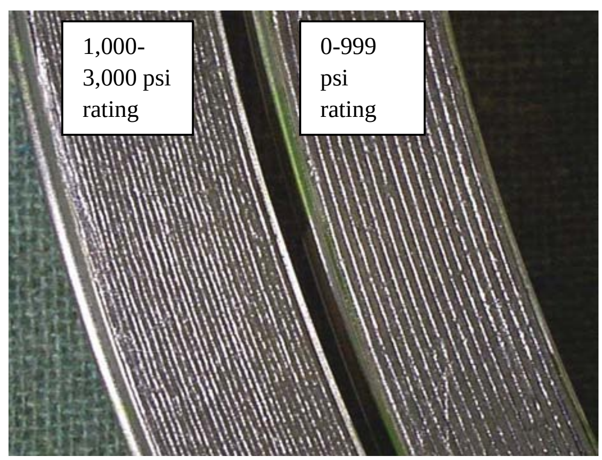

Figure 16 Comparison of high-density (left) and low-density (right) boiler manway spiral wound gasket constructions

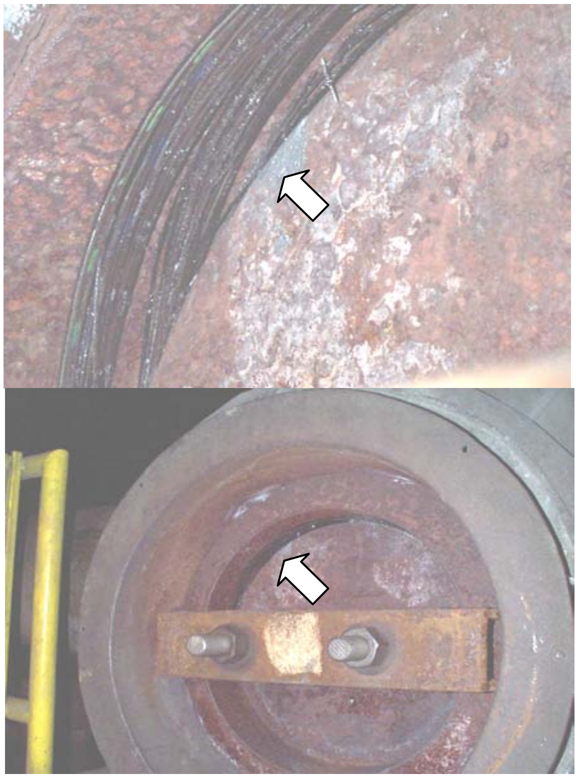

Figure 17 Failed manway gasket on a boiler drum door. Operating conditions: 2400 psig @ 600F.

Why are windings-only spiral wound gaskets usually specified as either “0-999 psi” or “1,000-3,000 psi”, even if they are to be used in designs that protect the gasket against over-compression? One reason is to protect against the inadvertent use of a high density gasket in a low bolt-load application, where bolt load is the only means of compressing the gasket. Low pressure applications typically involve bolting that would be inadequate to properly compress the harder, high-density gaskets. By using low density gaskets (“0-999 psi” construction) in low pressure applications, and high density gaskets (“1,000 – 3,000 psi” construction) in high pressure applications, this problem may be avoided.

Why haven’t more manufacturers of pipe flange spiral wound gaskets standardized on the use of low density construction windings and filler for all pressure classes, along with other optional design improvements? To answer this question, a review of the history of spiral wound gaskets may be helpful.

Flexitallic is credited with developing the first industrial spiral wound gasket approximately 100 years ago, in 1912. The basic design of the spiral wound gaskets produced by most manufacturers today has not changed significantly since the mid-1900s, when the outer compression-stop ring was first introduced. This track record is a testament to the robust nature of this impressively simple design.

The photo in Figure 18 shows an example of how spiral wound gaskets were manufactured prior to introduction of the compression-stop ring. The extended loops allowed support of the gasket by the bolting and centering of the gasket on the raised-face piping flanges. In the absence of a compression-stop ring, the sealing element density had to be designed to support the entire compressive force of the bolting.

With the introduction of the compression-stop ring, it became less important to design the sealing element to support the bolting load. However, it appears that the conventional approach still used by most manufacturers is to vary the gasket density in general relation to the available bolt load.

Figure 18 “Loop Winding” spiral wound gasket [5]

CONCLUSION

While spiral wound gasket density is often defined simply by describing the number of windings per inch, there are also additional factors to consider. These include filler thickness, winding assembly pressure, number of windings without filler, number of spot welds, and how the sealing element is constrained between the inner and outer rings, if used. All of these factors contribute to the compressibility of a gasket under a load, which is demonstrated in Figure 19 on the following page. They also determine how well a gasket will perform in simulated laboratory testing and, more importantly, under actual field conditions.

The use of pressure rating nomenclature alone to describe spiral wound gasket performance can also introduce confusion about the true design characteristics of these gaskets. A similar ambiguity may be found in the conventional terminology used to describe pressure classes, which are often referred to as “150 pound”, “300 pound”, etc. instead of the correct class 150, class 300, etc. Just as a “150 pound” descriptor may incorrectly imply a maximum design pressure of 150 psi, so too can a “0-999 psi” gasket rating incorrectly suggest a maximum design pressure of 999 psi. An understanding of these distinctions will help the end-user to better select spiral wound gaskets for individual applications and to properly assess root causes during investigations of gasketed joint failures.

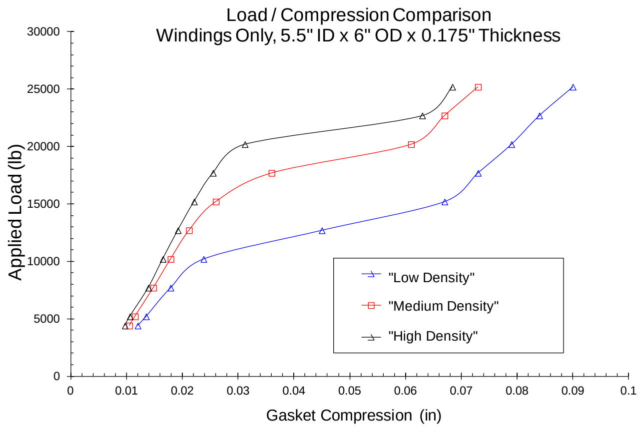

Figure 19 Compressibility performance of "Low", "Medium", and "High" density construction spiral wound gaskets

Highest Part A Stress, Leak Rate @ 400 psi						Calculations	
Manuf.	Pressure Class	Gasket Deflection (inches)	Original Gasket Thickness (inches)	Gasket Stress (psi)	Leak Rate (mg/sec)	Gasket Compressed Thickness (inches)	Compressive Modulus of Elasticity Stress/Strain (psi)
A	150	0.032	0.179	10,061	0.0000903	0.147	56,279
A	150	0.031	0.179	11,354	0.0000883	0.148	65,560
A	300	0.044	0.176	15,148	0.0000241	0.132	60,592
A	300	0.044	0.176	15,091	0.0000204	0.132	60,364
A	600	0.027	0.182	15,128	0.0000277	0.155	101,974
A	600	0.027	0.182	15,104	0.0000448	0.155	101,812
A	900	0.035	0.18	15,142	0.0000843	0.145	77,873
A	900	0.037	0.18	15,110	0.0000540	0.143	73,508
A	1500	0.028	0.18	15,124	0.0000622	0.152	97,226
A	1500	0.026	0.18	15,197	0.0000534	0.154	105,210
B	150	0.037	0.183	11,450	0.0004320	0.146	56,631
B	150	0.036	0.183	11,394	0.0004660	0.147	57,920
B	300	0.037	0.18	15,078	0.0000900	0.143	73,352
B	300	0.037	0.18	15,078	0.0000847	0.143	73,352
B	600	0.041	0.183	15,167	0.0000887	0.142	67,697
B	600	0.039	0.183	15,157	0.0000874	0.144	71,121
B	900	0.033	0.183	15,198	0.0000861	0.150	84,280
B	900	0.032	0.183	15,177	0.0000848	0.151	86,793
B	1500	0.031	0.181	15,135	0.0000841	0.150	88,369
B	1500	0.031	0.181	15,194	0.0000844	0.150	88,713

Table 1- 1995 ROTT (**RO**om **T**emperature **T**ightness) test data for NPS 4, 304/FG inner ring spiral wound gaskets [6]

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[2] Flexitallic, 2011, Gasket Design Criteria, Section I – Gasket Selection, page 33

[3] Waterland, A.F., 2009, "Analysis of the Compression Behavior of Spiral Wound Gaskets, ASME Pressure Vessels and Piping Conference, PVP2009-7742, p.1

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[5] VSP Technologies, 2012, Loop Winding Gasket Photo

[6] Bouzid, H., Derenne, M., and Payne, J.R. (JPAC, Inc.), 1995, Tightness Testing and Research Laboratory (TTRL) ROTT and HOTT Testing of SW Gasket Styles