FLANGED JOINT ANALYSIS USING PARAMETRICALLY CONTROLLED FINITE ELEMENT APPROACH

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ABSTRACT

The results of Finite Element Analysis (FEA) analysis of an ANSI 24 inch ANSI Class 150 flanged joint is presented. The use of 3-D FEA allows the engineer to more accurately evaluate a flange assembly subjected to internal pressure, external forces and moments for flange stresses, gasket contact stresses, and address leak tightness. In a critical process piping system, the integrity of flanged joints is of great importance to the safety of operating facilities. To facilitate the finite element analysis of a flanged joint, a parametric-driven program was developed to aid the engineer in investigating flanged joints within the time and expense parameters associated with the normal design process in the refinery and chemical industry. The ability to predict the leak tightness of a flanged joint is discussed by the authors. The authors present recommendations for assuring that the flanged joint will be suitable for the intended service.

DEFINITION OF PROBLEM

The investigation of flanged joints for ASME Boiler and Pressure Vessel Code (BPVC) Section VIII (1) allowable stress compliance, flange rigidity and additional investigation for gasket contact stress and deflections under all loading conditions is necessary to confirm the leak tightness of the assembly.

The complexity of movements and stresses is not fully addressed by the BPVC Section VIII flange design rules. The ongoing work by Pressure Vessel Research Council (PVRC) to characterize gasket parameters, as reported by Bickford 1997(2), is attempting to clarify the many design parameters necessary to assure a suitable tight joint. Additional investigation appears necessary to establish interacting design parameters which will address the actual flange and gasket situation under all operating conditions. The use of modern FEA software is facilitating these investigations. The use of parametrics driven modeling and pre and post processing is allowing increased analysis resulting in a better understanding of the flanged joint conditions. The FEA reported flange contact surface deflections and apparent variation of gasket contact stress during operation gives greater understanding to conditions that may be necessary to maintain a leak tight joint.

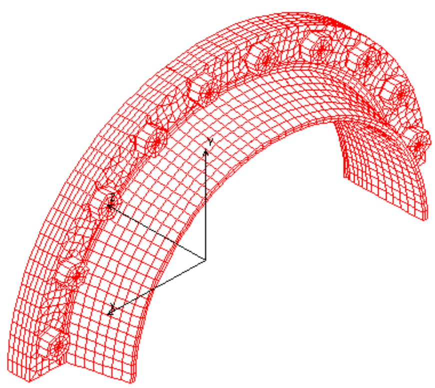
The gasket materials nonlinear compression and rebound properties are critical to maintaining a leak tight joint, but these material properties are not yet well defined and are not generally reported by the gasket manufactures.

This paper presents FEA generated results of a 24 inch ANSI class 150 flange set, under boltup and operating combined loadings of pressure and applied moment, using two types of gaskets. The characteristics of the gaskets are given in Table 1 and Figures 2 and 3 below.

FEA MODEL

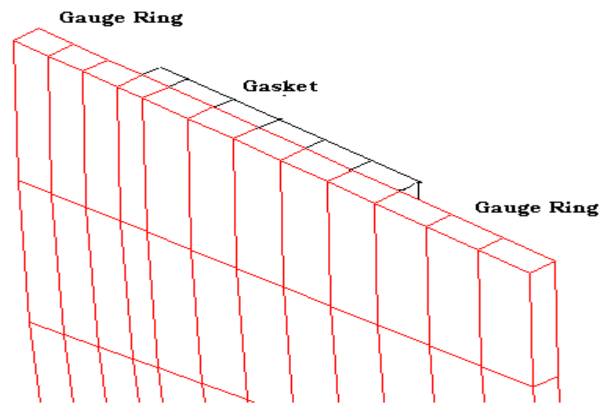
3-D solid elements were used to construct the flange body, gasket, and gauge rings. Bolts are modeled by 3-D beam elements. The modeling is controlled by a parametric approach which allows the user to input the dimensions of the flange assembly, to specify gasket/gauge ring configurations, and to apply internal pressure, loads/moments. The linear analysis was conducted using Cosmos/M software.

The bolt-up condition bolt load was applied evenly at the end of each beam element which is an appropriate simulation of bolts equipped with load control washers. The elongated beam element will then induce an equivalent tension stress of 22,500 psi in the bolts for the initial bolt-up. Moment was converted to the bolt axial forces as a function of the moment arm of each bolt. This results in a varying bolt load from a maximum bolt load at the longest moment arm to the essentially zero bolt load at the shortest moment arm. This loading approach is not displacing the flange in exactly the same manner as if the moment loading was applied at the end of the flange hub. The authors recognize that this is not a true representation of the flange displacements and stresses. The authors would recommend the use of a flexible connected spider arrangement attached to the end of the hub for obtaining greater accuracy, but it appears that there only a small flange body displacement difference for this particular analysis based on the authors previous work. The method utilized in this paper is probably indicating slightly less displacement at the ID of the flange than actually occurs. The model utilized for this analysis is indicated in Figure 1.



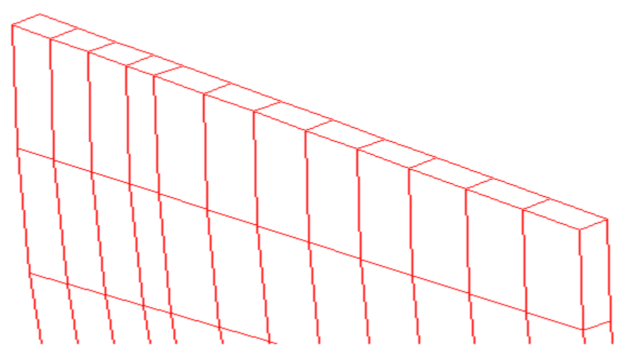
Finite Element Model of Flange
Figure 1

For the spiral wound gasket with metal rings, gap elements were used only at the selected area where the outer metal ring would come into contact with the flange raised face during the original bolt-up activity. It should be noted that for a typical spiral wound gasket with an outer gauge ring, it is standard practice to select a gasket with a design compression stress and sealing surface width such that the bolting will compress the gasket to the height of the outer gauge ring utilizing the design bolt stress. The inner gauge ring was not connected to the flange face as the analysis indicated that this ring would not be in contact with the flange gasket surface. A rigid link bar was used between the interface of the gasket and the flange raised face. Due to the large difference of the flange and gasket modulus of elasticity this did not substantially effect on the flange body stress and corresponding displacements. The model of the spiral wound gasket is indicated in Figure 2, note this is one half of the gasket as it is symmetrical about the thickness centerline of the gasket.



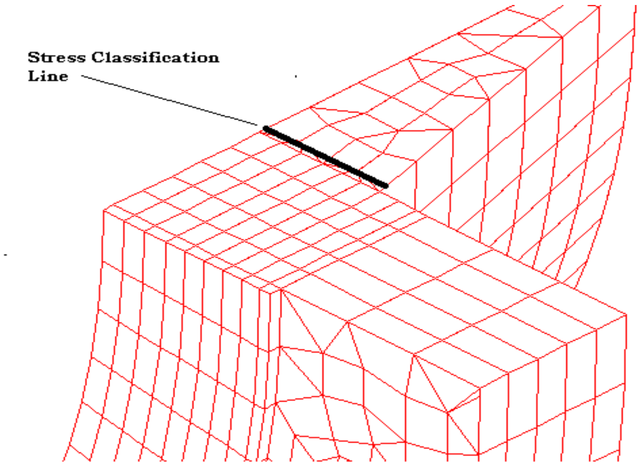
Spiral Wound Gasket With Gauge Rings
Figure 2

For the composite gasket without metal rings, no gap element was used in the model. The composite gasket extended beyond the ID and OD of the flange raised gasket contact surface face. The composite gasket is indicated in Figure 3.



Composite Gasket (no gauge rings)
Figure 3

The FEA reported flange stresses were linearized using the procedures described by Hsieh et al 1999 (8) to obtain membrane plus bending ($P_L + P_b$) stresses for the allowable stress criteria. The stress classification line is indicated in Figure 4.



Stress Classification Line Location
Figure 4

Two computer runs were made for each model with refined mesh so that the reported stresses from the FEA can be assessed for convergence. It was found that the reported stresses were within 10% variance, therefore the reported stresses are considered reasonably valid.

DISCUSSION OF RESULTS

The FEA analysis gives insight into the flange stress and associated strain deflection during bolt-up and imposed loadings. The FEA results are compared in Table 2 to the BPVC Section VIII (1) stress calculations and the ASME course "Design of Bolted Flange Joints" presented by W. Koves 1998 (4).

The flange gasket contact surface rotation is reported with respect to the BPVC Section VIII (1) code rigidity calculation methodology and acceptance criteria which is based on limitation of 0.3 degrees rotation. Different gasket configurations effect the joint stresses, deflections and rotations.

The gasket contact stress is a function of the bolt stress, the flange strain deflection due to various loadings, the gasket configuration, and gasket material properties as reported by Bibel and Gronhovd 1999 (3). The gasket contact stress displays a significant circumferential variation under the various load conditions.

Flange ASME and FEA Indicated Stress Comparison

Table 2 contains the resulting radial and tangential stresses for a 24" 150# raised face flange for the two different gasket types. These two gasket flange assemblies were further investigated for two design conditions; 1) a 285 psi design pressure only and 2) design pressure plus an external pipe moment of 3,025,000 in-lbs. The table also includes a comparison of the Code allowable flange face rotation of 0.3 degrees, ASME Section VIII, Div.1, Appendix S, Par. S-2 (1) to the rotation reported by the FEA models.

The FEA reported slightly increased stresses than reported by Hsieh et al 1999 (8) for the same conditions, see Table 2. The authors consider the current model to be a better indicator of the actual stresses as the model included such improvements as the use gap elements. For Case 1 the radial stresses remain close to those reported from the ASME Design method as presented in the ASME Flange Design Course presented by Koves 1998 (4) and the Tangential stresses remain significantly higher than the ASME design, although these stresses are less than the BPVC Section VIII (1) allowables. However it is noted that the stress difference between Case 1 and Case 2 is almost identical to the difference presented last year.

The composite gasket results in higher stresses than the spiral wound results, likely due to changing fulcrum point about which the flange is bending. Under Case 4 loading, the Code calculated radial stress exceeds the "old" Code allowable stress. A note on Code allowable stresses, the authors have elected to continue to use the allowables in effect prior to the 1999 Addenda for two reasons. The first is to provide continuity with work done in prior years and the second is that the authors are not using the allowable stress as the acceptance criteria for these designs but as a benchmark to indicate whether the resulting stresses would be acceptable.

Although the design gasket contact stress shown in Table 1 is identical for both types of gaskets, the different construction and dimensional differences produce quite different analysis results. The composite gasket produces a varying fulcrum point for flange bending, which is further form the bolt circle, and which produces the higher stress differences between Cases 3 & 4 than that shown between Cases 1 & 2 for the spiral wound gasket. It is noted that the outer gauge ring becomes a stationary fulcrum point that the flange rotates about.

Flange Rotation

The flange rotation acceptance criteria of 0.3 degree noted in table 2 is the basis for the flange rigidity calculations in BPVC Section VIII (1). The flange rotation developed from the FEA results is simply the displacement as shown in the respective figures for the length of the surface displaced expressed in degrees.

The spiral wound gasket application rotates around the outer gauge ring and the fulcrum for the rotation is the gauge ring. It evident that for the applied moment case the outer gauge ring restricts the additional compression of the gasket for the closing side of the flange. The applied moment causes the flange to open on the opposite side and this movement causes the flange rotation to increase as is apparent in the respective displacement figures and Table 2.

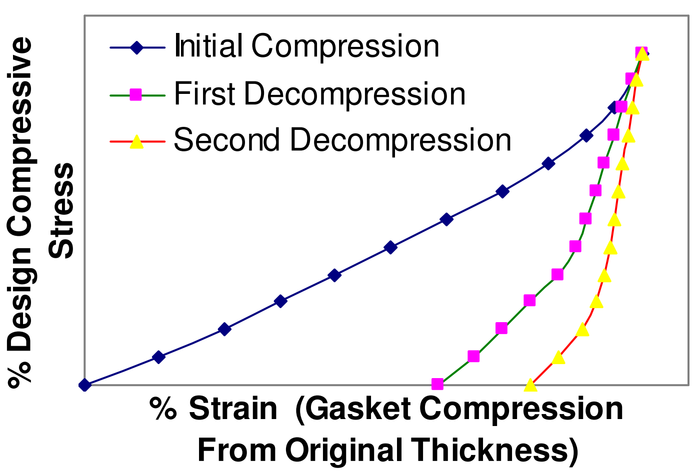
The composite gasket application does not provide a fixed fulcrum for any portion of the flange and this results in a changing bending structure with in the flange set. When the moment is applied the closing side of the flange further compresses the gasket and the opening side movement combines to significantly increase the flange rotation.

It is not evident that the apparent flange rotation is a characteristic that can be used to assess the leak tightness of one gasket type versus another type. The rotation of the flange, or the corresponding rigidity of the flange, maybe a characteristic that can be evaluated for comparison of various flange loading for the same gasket type. By inspection of the flange rotations shown in Table 2 it is apparent that the same flange with the same loadings has significantly different movements due to type of gasket utilized. It is the authors opinion that the restoration properties of the gasket will have a much greater impact on the leak tightness of the flanged joint than using the ASME flange rigidity calculation basis.

<u>Gasket Compression and Contact Stress</u>

As stated in the FEA MODEL description above certain concessions were made in the modeling of the joint structure but the concessions are not considered to have a significant impact on the results. The gasket displacements and contact stresses are presented in the included figures.

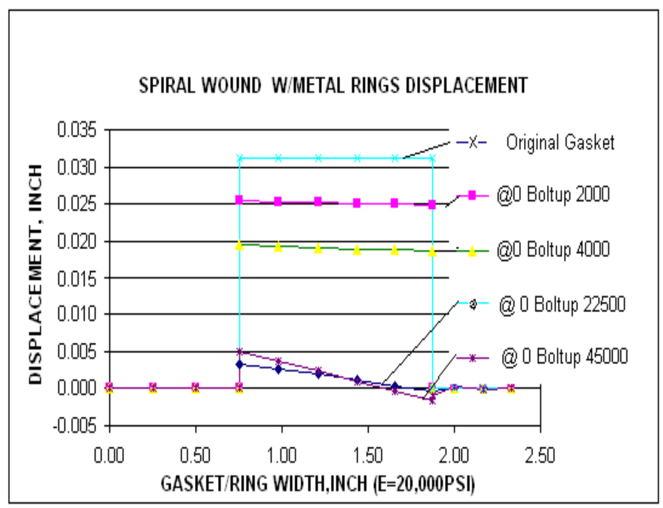
The distribution of gasket contact stress changes as the joint loading changes from boltup to operating conditions of pressure and imposed moment. The variation of the gasket compressive stress remains a concern as reported by Bouzid et al 1998 (5) 1999 (6) and Hsieh et al in 1998 (7) and 1999 (8). It is obvious that gaskets do not have the same stress-strain relationship for restoration as compression as reported by Shoji et al 1999 (9), and Tuckmantel 1991(10) and indicated in figure 5.



Typical Gasket Stress Strain Graph
Figure 5

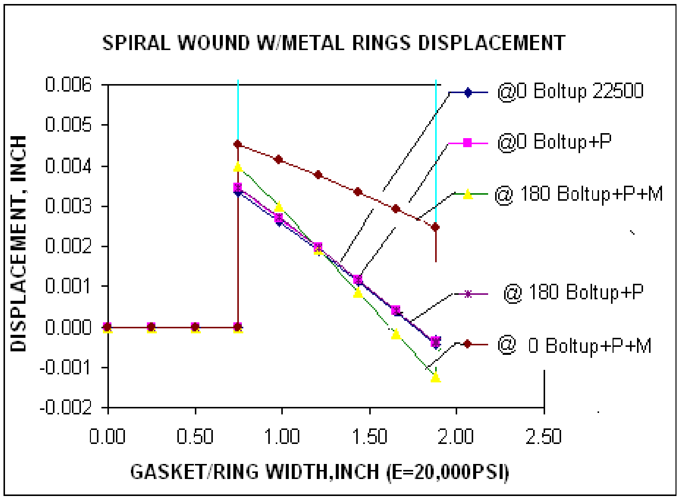
The same linear gasket compressive and restoration modulus of elasticity was used for this FEA investigation. Therefor the compressive stresses indicated for the gasket are not correct. For example it is reported that while a typical spiral wound gasket is compressed 0.030" from original thickness, until the flange contacts the outer gauge ring, the gasket will not rebound in thickness over 0.005" before it looses it's ability to maintain any significant compressive stress on the flange face.

A typical low seating stress spiral wound gasket joint, with outer and inner gauge rings is addressed in figures 6 through 10. It must be noted that the FEA model is symmetrical about the gasket and displacement figures in this paper are based on one half of the total gasket, therefor the indicated displacements are one half of the total gasket movements. Figure 6 indicates that the displacement of the flange/gasket interface during boltup results in flange rotation as the bolt stress increases. The rotation increases when the flange contacts the outer gauge ring. Only about 1/3 of the gasket width does not need to significantly rebound as the bolt stress is increased from the 22,500 psi design stress to twice design stress or 45,000 psi. The gasket would need to rebound about 0.003" at the ID as the bolt stress increases from 22,500 to 45,000 psi. The rebound could be expected to reduce the gasket contact stress to well less than one half of the design contact stress there by significantly reducing the sealing ability of the flange.



Displacement of Gasket During Boltup
Figure 6

Figure 7 indicates the displacement of the flange/gasket interface with 22,500 bolt stress when pressure and a bending moment is applied. The joint movement on the opening side (noted as @ 180) of the applied moment, indicates that the gasket would need to restore approximately 0.003" and this would result in less than one half of the gasket design contact stress. This would indicate that the joint would remain relatively leak tight. Any additional opening of the joint would increase the required gasket restoration and the joint could be expected to have significant leakage.

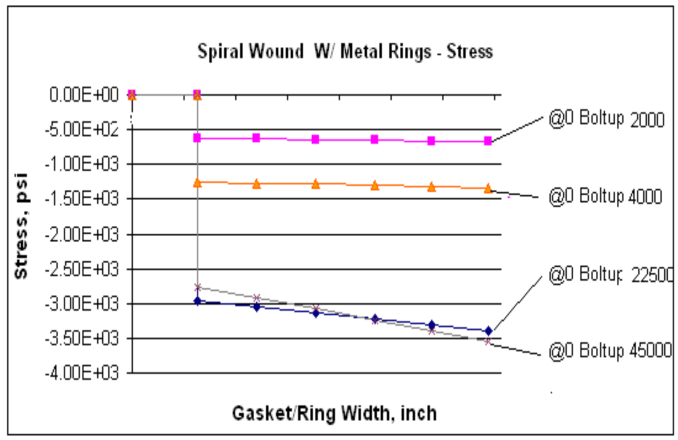


Displacement of Gasket – Boltup Plus Moment
Figure 7

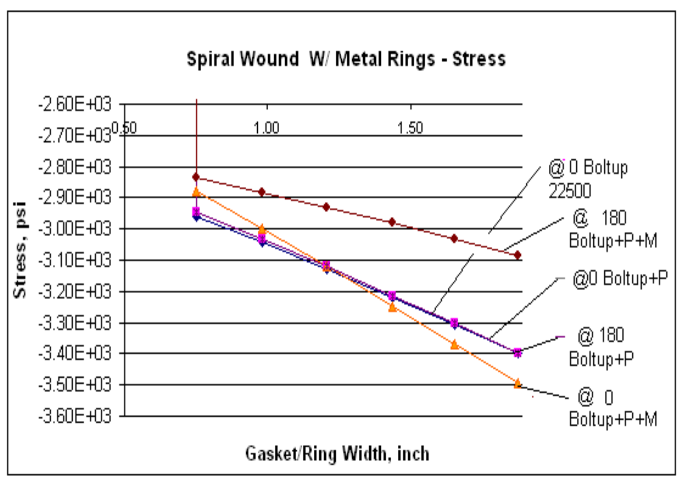
The displacement figures clearly indicate that a significant amount of the gasket contact surface must restore in thickness to maintain contact. The FEA assumed that the gasket properties were linear, but in reality there is significant non-linear characteristics in gasket materials. The gasket material non-linear stress-strain characteristics indicated in figure 1 give even greater concern for the ability of the gasket to maintain suitable sealing ability when a moment is applied as shown on the 180 degree side of the joint. It is reported that as little as 0.005" lifting of the flange/gasket interface from maximum compression achieved during the boltup is considered the practical limit of most spiral wound type gasket to maintain any sealing ability.

The gasket compressive stress for various bolt stresses that would occur during original assembly of the joint with the spiral wound gasket with gauge rings is presented in figure 9. As the modulus of elasticity properties of the gasket were assumed to be linear, the compressive stress on the gasket contact surface increases in direct proportion to the bolt stress until the gauge ring contacts the flange raised face. The gasket will compress fairly evenly as the flange rotates due to the contact force on the gasket. When the flange contacts the outer gauge ring the flange rotates about the outer gauge ring resulting in the ID of the gasket having to rebound as the bolt stress is increased.

Inspection of the contact stress information presented in Figure 8 and 9, indicates that the compressive gasket stress varies significantly from the OD to the ID of the gasket when pressure and bending moments are applied to the joint. The maximum compressive contract stress is achieved at the OD of the gasket and the compressive stress decreases significantly towards the ID. When pressure is applied to the joint the compressive stress is lessened at the ID and will become less at the OD as the flange contact force on the gauge ring is overcome. When a bending moment is applied to the joint the flange strain deflection adds to the gasket compression on one side of the joint (shown as 0 degree location) and reduces the compressive stress on the other side (shown as 180 degree location). As discussed earlier, the gasket contact stress indicated in these figures is considerably overstated as here the gasket is attempting to restore it self in thickness due to the non-linear gasket material properties.



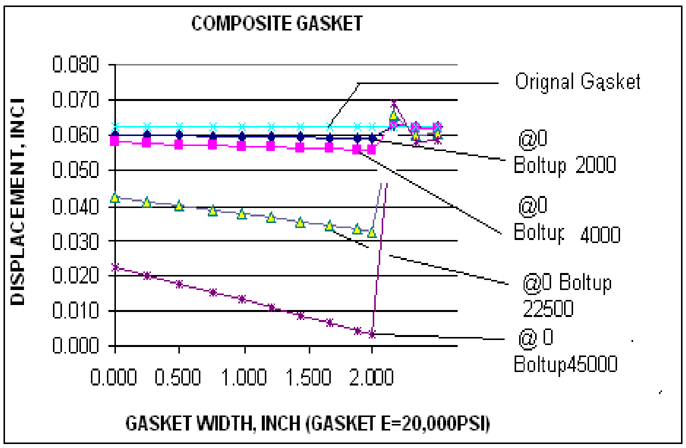
Stress on Gasket Sealing Surface During Bolt-up
Figure 8



Stress on Gasket Sealing Surface – Boltup Plus Moment
Figure 9

The compressive stress on the gauge rings has been eliminated from figure 8 and 9 for clarity. The compressive stress on the outer gauge ring was indicated to exceed yield (over 70,000 psi) with design bolt stress of 22,500 psi. The displacement FEA results do not indicate the inner gauge ring contacts the flange raised face. When the bolt stress is increased to 45,000 psi, twice the design stress, the indicated outer gauge ring compressive stress increases significantly. The compressive stress is well over yield and would result in some permanent distortion of the gauge ring and flange contacting surface.

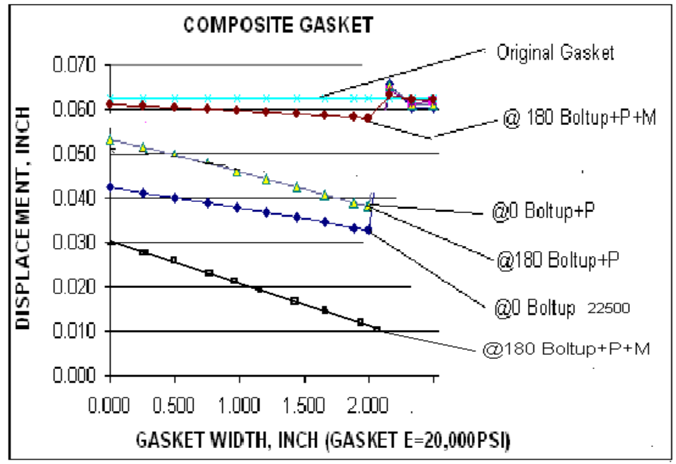
Inspection of the composite gasket displacement information in figure 10 indicates significant flange rotation during bolt-up.



Displacement of Gasket During Boltup
Figure 10

As the bolts are tightened the gasket is compressed about 0.030" at the OD or about one half the gasket original thickness at a bolt stress of 22,500 psi. For the 44,500 psi bolt stress the gasket is compressed to less than 0.010" thick at the OD , or 1/6 the original gasket thickness, and probably over compressed. As the gasket material properties are non-linear it is anticipated that the gasket displacement is overstated.

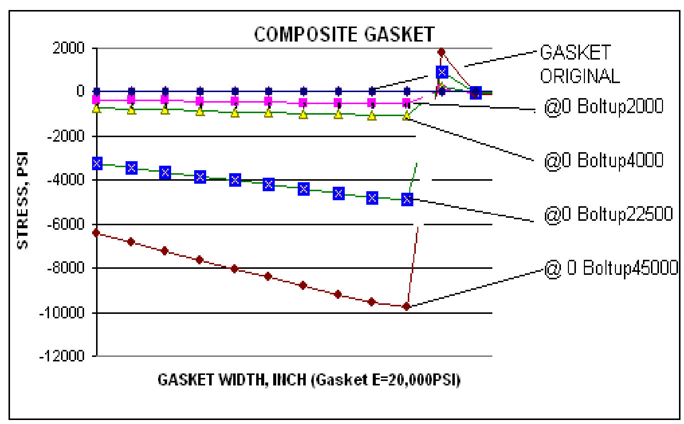
Figure 11 indicates the composite gasket displacement due to pressure and applied moment with an initial boltup 22,500 bolt stress. The indicated displacements and associated flange rotation for boltup and pressurization tend to make the flange rotate but close more at the OD and appears to be consistent with information presented by Bibel and Gronhovd 1999 (3). The moment causes the flange to rotate further such that under the combined pressure and moment loading the gasket is unloaded significantly on the opening side of the flange. This indicates that the joint will not be leak tight. On the opening side the gasket must restore it's thickness to within approximately 0.010" of its' original thickness. Due to the gasket non linear modulus of elasticity this could indicate that the gasket will have very little sealing ability across it's radial width.



Displacement of Gasket – Boltup Plus Moment
Figure 11

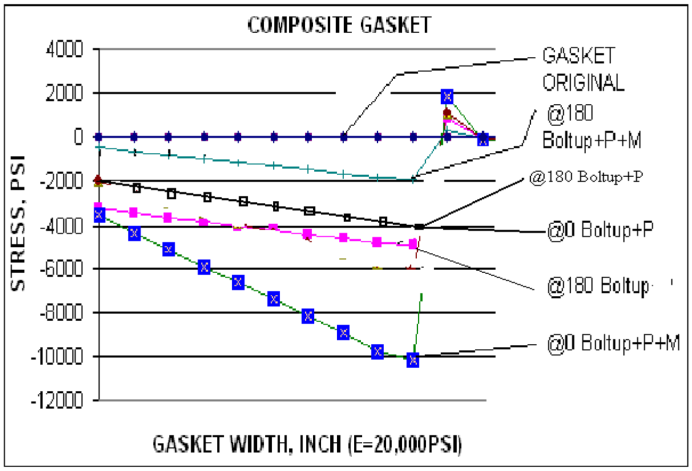
Figures 12 and 13 provide composite gasket contact stress information for boltup and the application of pressure and a moment on the composite gasketed joint. Inspection of these figures indicates the compressive stress on the opening side (@ 180 degrees) of the gasket reduces to approximately 20% of the gasket OD stress at the closing side (@ 0 degrees). The gasket material must be able to restore it self with sufficient residual stress or a leak will develop.

The composite gasket compressive stress information indicates that the stress on the gasket is reduced approximately one half from the bolt up condition when the applied loadings occur due to the flange strain displacement.



Stress on Gasket Sealing Surface During Bolt-up
Figure 12

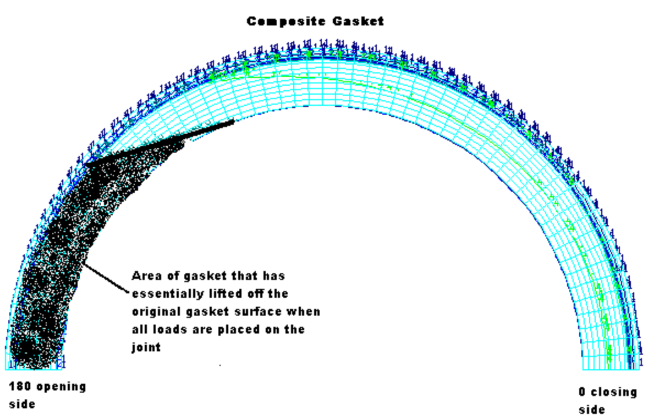
It should be noted that the authors have omitted a temperature profile from the FEA analysis as a uniform temperature gradient with a higher temperature on the ID than the OD introduces a uniform deflection of the flange and tends to increase the gasket contact stress, bolt stress and rotation with only small influences on leak tightness of the joint Hsieh et al 1998 (7).



Stress on Gasket Sealing Surface – Boltup Plus Moment
Figure 13

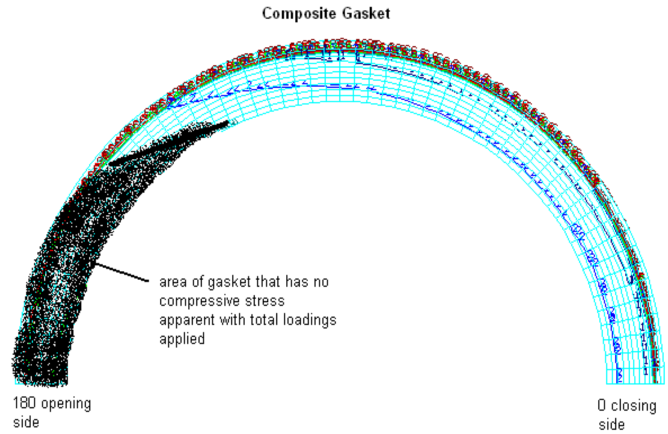
<u>Non Uniform Bolt Stress</u>

The composite gasket model was utilized to study the effect of an uneven boltup condition. The flange contains 20 bolts and 6 adjacent bolts were set to have one half of the original bolt up stress as the remaining 14 bolts. The low stress bolts were placed at the 180 position to accent the gasket restoration requirement. The gasket displacement is indicated in figure 14 and contact stress is indicated in figure 15.



Displacement of Gasket – Boltup Plus Moment
With Non Uniform Bolt Strss
Figure 14

The figures represent Table 2 load case 4 conditions. Inspection of figures 11 and 13 versus figures 14 and 15 indicates that the non uniform bolt stress results in significant less compressive stress on the opening side. It is apparent from the figures that non uniform bolt stress significantly increases total flange displacement or rotation.



Stress on Gasket Sealing Surface – Boltup Plus Moment
Figure 15

The figure 15 indicates that the compressive stress on the gasket does not continue into the low bolt-up stressed bolts. The gasket is totally unloaded and would be expected to present a leaking condition. The gasket restoration ability is obviously not sufficient to fully restore to its' original thickness for a considerable length of the gasket perimeter and the non linear properties will have the effect of further reducing the gaskets' sealing ability as shown in Figure 5.

CONCLUSIONS

The leak tightness parameters being established by the PVRC work should be applicable to determining the leak tightness of this joint if the gasket non-linear modulus of elasticity properties were known as indicated in Figure 5, thereby allowing prediction of the leak rate at the actual gasket compressive stress conditions. As there is little information published by the gasket manufactures pertaining to the non-linear modulus of elasticity and the change in these properties during successive loading cycles, the ability to predict the joints tightness and corresponding leak rate is hampered.

The application of typical pressure and piping moments to the flange results in significant changes in the flange rotation. The gasket displacement is significantly different for a gasket with an OD gauge ring than a gasket without a gauge ring. The OD gauge ring acts as the rotation fulcrum and stops the over compression of the gasket due to applied moments. A composite type gasket without an OD gauge ring may be significantly over compressed due to imposed moments.

The ability to predict the flange stresses and movements is a practical engineering effort with modern FEA software and desk top computers. The ability to confirm a flanged joint leak tightness and leak rate is limited by the lack of sufficient non-linear gasket material properties definition. If these properties were available and incorporated into a parametrics controlled FEA of the joint it would seem that the tightness and leak rate investigation would be far more successful than the procedures utilized today.

RECOMMENDATIONS

The authors recommend that the non-linear gasket material properties be fully investigated and published by the gasket manufactures to facilitate and enhance the engineering communities ability to successfully predict the serviceability of the flanged joint in typical piping systems.

The authors would suggest the following criteria be utilized in addition to ASME Leak Tightness methodology for predicting a successful leak tight joint for a critical service (all design loadings considered and based on linear gasket material properties):

Spiral Wound Gasket With OD and ID Rings:

1. maximum flange rotation of 0.2 degree calculated at the opening side of the flange when resisting an applied moment
2. maximum movement of 0.003" above the OD gauge ring measured at the ID of the gasket
3. flange and OD gauge ring to remain in contact and gasket rebound at outer 1/3 of the gasket nearest the OD gauge ring is less than 0.001" under all conditions

Composite Gasket With Out Gauge Rings.

1. maximum flange rotation of 0.3 degree calculated at the opening side of the flange when resisting an applied moment
2. maximum all load movement of 33% of the predicted compressed gasket thickness at boltup over the outer one half of the gasket. This is to be true for restoration or additional compression (restoration is considered the limiting gasket property).
3. maximum indicated compression limited to one half the original gasket thickness under all loading conditions.

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Table 1
GASKET AND FLANGE INFORMATION TABLE

Gasket	Design Contact Stress psi	Young's Modulus (linear)	OD/ID metal rings/material	Flange & Bolt material ASTM	Pressure and Temperature Applied (1)	Moment Applied Inlb (2)
Spiral Wound (3) soft seat stress	3,300	20,000	Yes Carbon Steel	A-105& B-7	285 psi Ambient	3,025,500
Composite (4)	3,300	20,000	No	A-105 & B-7	285 psi Ambient	3,025,500

1. Uniform Temperature Applied to flange assembly
2. Moment is equivalent to 18,680 psi stress in ½" wall 24"OD pipe
3. Spiral Wound gasket selected is a low seating stress type (original gasket surface is 0.030" above the 0.125" thick gauge rings for a total uninstalled gasket thickness of 0.185")
4. Composite gasket selected as a low seating stress type of elastomeric with fiber fill (original uninstalled gasket thickness of 0.125")

Table 2
ASME Methodology and FEA Methodology Stress Comparison

Gasket type and Load Conditions versus allowable stress	Code Calculated Radial stress	FEA Radial stress	Code Calculated Tang. stress	FEA Tang. Stress	Code Acceptable Rotation degrees	FEA Rotation degrees
Case 1 Spiral Wound Pressure + Bolt up (285 psi) (Pl) allowable 17,500 psi (Note 1)	10744	13177	3933	10816	0.3	0.196
Case 2 Spiral Wound Pressure + Bolt up + Moment (696 psi equil) (Note 2) (Pl+Pb) allow. 26,250 psi (Note 1)	26238	19460	8287	15027	0.3	0.267
Case 3 Composite Pressure + Bolt up (285 psi) (Pl) allowable 17,500 psi (Note 1)	11865	16423	3748	13605	0.3	0.27
Case 4 Composite Pressure + Bolt up Moment (696 psi equil) (note 2) (Pl+Pb) allowable 26,250 psi (Note 1)	28976	25734	9152	21269	0.3	0.446

1. Allowable stress per ASME Section VIII Division 1 1998 Addenda
2. The applied moment is changed to an equivalent pressure for the application of the ASME Flange Design Course methodology to calculate flange stresses, Koves 1998 (4)