DESIGN OF FLANGED JOINTS SUBJECTED TO PRESSURE AND EXTERNAL LOADS

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ABSTRACT

The integrity of flanged joints is of great importance to the safety of operating facilities. This paper presents an analysis of a typical ANSI weld neck gasketed flanged joint. The analysis utilizes ASME Sect. VIII design rules plus design considerations from the ASME course "Design of Bolted Flange Joints" plus finite element methods to analyze flange design stresses and deflections. Comparisons of ASME Code vs. Finite Element Analysis (FEA) flange stresses as well as gasket contact stress distribution are presented in this study.

DEFINITION OF PROBLEM

The effect of piping system imposed moment loading on the typical ANSI flange joint is of concern to the design engineer. The engineer must determine the effect of actual loadings on the flange stress and on the joint's ability to remain leak tight. In a previous paper Hsieh et al (1998) a finite element analysis of a standard carbon steel 24" 150# raised face flange was presented which included the effects on the gasket contact stress of adding moments from attached piping to the pressure design considerations. That model will be utilized to compare FEA stress results to the flange stresses from the ASME Sect. VIII, Division 1 (1998) design method with modifications to include the imposed piping moment as presented in the ASME course "Design of Bolted Flange Joints", presented by W. Koves (1998). In addition the flange/gasket contact area will be investigated for deflection and contact stress.

FEA MODEL

The linear elastic model for the flange and the gasket assembly (see Figs. 1,2, &3) was formed using three dimensional solid element except for the bolts which are simulated by beam elements. The model is the same as the model in the previous paper presented by Hsieh et al (1998) with increased mesh density to ensure convergent results.

Two types of gaskets are considered in the analysis: spiral wound graphite filled with ID & OD metal gauge rings and spiral wound graphite filled without metal rings. The spiral wound graphite filled gasket with ID and OD rings was used to compare the ASME closed form calculation results with the FEA results. The spiral wound graphite filled gasket with out ID or OD gauge rings FEA results are reported for comparison to the FEA results from the gasket with rings for deflection and gasket contact stress. This paper continued the work reported by Hsieh et al (1998) to quantify flange stresses and further investigate the gasket behavior. The flange is subjected to 285 psi internal pressure, bolt-up load, and external bending moment. By refining the mesh density, convergent results were verified in the finite element analysis. The convergence was considered achieved when the stress variance across an element was less than 10%. In Hsieh et al (1998) the model was not fully converged as only qualification of flange deflection and flange/gasket contact stress was investigated.

Bolt-up load was applied to induce 22,500 psi of bolt stress. This bolt stress produced the required gasket seating stress of 10,000 psi. Moments applied to the flange set (1,317,000 in-lbs and 3,025,000 in-lbs.) were established to produce maximum ASME Section VIII Division 1 Appendix 2 methodology flange design stresses close to allowables of 17,500 psi and 26,250 psi. These moments resulted in nominal pipe bending stresses of 8,140 psi and 18,680 psi respectively for 24" diameter standard wall pipe. The moments were applied around the Y-axis resulting in closing at the 0 degree location and opening at the 180 degree location.

The Stress Classification Line (SCL) was established at the flange and hub junction (see Fig. 4 for location). This location was selected as similar to the location recommended in the PVRC work published by Hechmer and Hollinger (1997) for a nozzle to shell junction. It is anticipated that the selected SCL is a reasonable representation as described by Hechmer and Hollinger (1997). It was found that the maximum Tresca stress occurred at 0 degrees, therefore the SCL @ 0 degree location was chosen for the study. The linearized flange stresses at the SCL were obtained using the methods presented by Broyles (1997) and Bibel (1990). The results of stress linerization by the Broyles (1997) and Bibel (1990) methodology were found to be very comparable. Linearized radial, tangential, and longitudinal stress for the SCL at 0 degrees were calculated and moment 2 results presented in figures 5, 6, & 7. Linearized radial, tangential, and longitudinal stress for the SCL at 180 degrees for moment 2 were calculated and presented in figures 12, 13, & 14 for comparison. The stresses from the FEA results at 0 degrees are compared against ASME code calculations in Table 1. The FEA total stresses (peak) and Tresca stresses at the same 0 degree location were also extracted for comparison purpose and included in the table 1.

As in the previous paper by Hsieh et al (1998), the Young's modulus for the gasket material was considered to be the same for the gasket before and after original bolt up compression. This assumption is not true as the crushing of the gasket material due to the contact stress results in a significant change in the gaskets' ability to restore to it's original thickness. The ability of the gasket to restore it's thickness is critical as the gasket contact compressive stress becomes less during various flange loading conditions. The restoration ability of the gasket material is an important parameter in flanged joint design and little information is published about this parameter. The model does include the original gasket compression deflection during bolt up as it achieves the 10,000 psi initial compressive load but utilizes the bolt up condition as the zero basis for deflection comparison for applied moment investigations. Therefore the reported stress/deflection at this interface does not represent true condition when the gasket stress is in tension. Due to excessive number of elements and nodes in this model, it was impractical to impose non-linear contact elements (gap elements) at the interface of the gasket and the flange to run on a Pentium based computer. However, this indicates that the gasket and the flange are actually trying to separate where the tensile stress is indicated. The Young's modulus of the gasket is 0.7% of the flange material. It is therefore considered that lacking the gap elements in this model would not have significant effect on the flange stress results.

CALCULATION METHODS

The standard ASME Sect. VIII, Div. 1, Appendix 2 calculation method was used to arrive at the standard stresses for the 24" 150 series flange. The gasket reaction diameter is modified to reflect the results from the finite element analysis reported by Hsieh et al (1998). The results of this calculation is presented as Pressure + Boltup in row one of Table 1 and compared to the FEA results for the same condition.

The ASME Section VIII Division 1 Appendix 2 does not provide methodology to address piping loads imposed on the flange. The addition of the induced piping moment is accomplished by effectively increasing the design pressure utilizing the Equivalent Pressure Method modified by a Moment Correction Factor (F) as suggested by the ASME course presented by W.Koves (1998) as follows:

$$P_{eq} = \frac{16 * M * F}{(pi) * G^3} + \frac{4\,Fa}{(pi) * G^2}$$

where

$$F = \frac{1}{1 + \dfrac{J}{2\,(1 + v)\,I}}$$

Peq : "equivalent" additional pressure to add to the design pressure

M: Moment added by attached piping.

Fa: Axial End load added by attached piping

G: Gasket reaction diameter

J: Torsional moment of intertia of flange cross section

I: Moment of inertia of flange cross section

v: Poisson's ratio

This calculation results in equivalent total pressure of 464 psi for the 1,317,000 inlb moment case and 696 psi for the 3,025,000 inlb moment case. This information is presented in rows 2 and 3 respectively in Table 1.

COMPARISON OF RESULTS

Table 1 shows the resulting radial stresses for a 24" 150# raised face flange with a design pressure of 285 psi, and the spiral wound gasket with inside and outside gage rings. Two moment loadings were investigated.

The FEA SCL location was selected as a reasonable location for comparison to the closed form calculation method contained in the code with FEA obtained results. Although the direction of the stresses between the two methods can be compared, the location where these stresses are investigated may be different as the ASME Section VIII Division 1 Appendix 2 does not clarify the actual calculation location. In addition, some engineering assumptions used in classical flange design are not applicable when a full 3D FEA model is constructed. The most notable of these is that the flange cross-section is assumed to be axisymmetric in the ASME flange design calculation methodology. The FEA model includes bolts as well as the distinct loading at those locations. This is most apparent in the differences in the tangential stress outputs.

An interesting observation from Table 1 is that the numerical increase in tangential stress from moment 1 to moment 2 are fairly close for both methods, indicating that these two methods are in agreement in predicting the effects. This is not the case for the radial stress results. It would appear that the difference in the basic "pressure plus bolt up" case, and perhaps the radial stress results, is due to differences in how the bolts are addressed in the methodology.

The flange/gasket contact surface deflections and sealing surface contact stresses are presented in graphical format in figures 8,9,10, & 11. It should be noted that the gasket original deflection from zero loading to maximum loading under original bolt-up is not indicated in these graphs. Only the effect of flange distortion on the gasket surface is indicated. The spiral wound gasket with inner and outer gage rings still appears to experience compression only on the outer gage ring at the tension side of the moment load. Sealing at this location will only be provided subject to the restoration capability of the gasket material. Although the magnitude of the deflections at the gasket face has changed somewhat from the first FEA Model presented by Hsieh et al (1998), the conclusions from these results remain the same.

A check of the flange's rigidity per ASME Section VIII, Div. 1, Appendix S, para S-2 shows that the rigidity index parameter is acceptable to control leakage. This parameter may not be sufficiently conservative for all type of gaskets and does not address imposed piping loads.

The gasket without gage rings does display considerable compression in the outer 2/3 of the gasket. This would indicate adequate sealing at bolt-up. The applied moments have the effect of altering the contact stress by a +/- factor of approximately two and the deflection by approximately 0.004". Both of these conditions have a detrimental effect on sealing and appear excessive.

It should be noted that if the increased ASME Section VIII Division 1 allowable stresses, as defined by Code Case 2278, are applied to the flange design method, then the gasket face deflection and resulting potential leak problems created by this effect will be increased due to increased flange flexibility.

CONCLUSIONS

1. FEA stress results cannot be directly compared to ASME closed form calculated stress results due to differences in modeling assumptions:

   1.1 The ASME Code closed form calculated stresses when compared to Code allowables indicate the flange is not overstressed. This is also true for the equivalent pressure methodology stress calculation results.

   1.2 The FEA indicated radial stresses at the selected SCL are somewhat less than the closed form values but the opposite is true of tangential stresses. However, the flange is not overstressed for any of the loading conditions.

1.3 For this model the Code closed form stress calculations appear to result in more conservative flange design than the FEA results.

1.4 It is necessary that the point is identified where the ASME Code stresses are calculated so that a meaningful comparison with FEA can be done.

2. The differences between the ASME closed form developed stresses and the FEA developed stresses indicates that additional investigation should be undertaken to clarify the correctness of each. The authors anticipate that the FEA results are more representative of the true stresses than predicted by the closed form methodology.

3. Sealing ability of flanged joints subjected to external loads remains a concern:

3.1 The use of an outer gage ring gasket configuration in this flange application provides a fulcrum point for the flange to bend around. This significantly reduces the sealing ability of the gasket.

3.2 The gasket without gage rings is subjected to changing contact stress by a factor of approximately +/- two (Fig. 11) and a variation in gasket thickness of +/- 0.008" (0.004" in model times two to include the other flange in the pair, see Fig. 10)

3.3 The sealing ability of a flange cannot be judged by ASME Section VIII Division 1 Appendix 2 stress calculations or Appendix S rigidity calculation design alone. These methodologies do not address the variations in flange seating surface deflections due to piping imposed loadings or due to different types of gaskets.

3.4 The authors remain concerned that the flanged joint leak criteria that ASME is currently developing will not address the flange and gasket impacts from piping imposed loadings. The typical flanged joint has varying loading conditions including non-symmetrical loadings resulting in flange to gasket contact surface stress changes and flange deflection changes. The incorporation of a parameter predicting the gaskets' restoration ability to compensate for varying contact stress and flange deflection would appear to be necessary.

REFERENCES

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Broyles, R.K., "Finding Design Acceptability using Finite Element Analysis, PVP Vol. 353, ASME, New York, N.Y., 1997

Bibel, G.D., and Kovach, J., "Pinpointing Stress Out Designs: Interpreting FEA Results", Machine Design, Dec. 6, 1990.

Hechmer, J.L. and Hollinger, G.L., 1997, "3D Stress Criteria: Guidelines for Application", PVRC Grants 92-09 and 91-14, ASME, New York, N.Y.

Porter, M.A., Martens, D.H. and Hsieh, C.S., 1998, "Stress Evaluation of a Typical Vessel Nozzle using PVRC 3D Stress Criteria: Guidelines for Application", PVP Vol. 368, ASME, New York, N.Y., pp. 297-301.

Table 1
ASME Methodology and FEA Methodology Stress Comparison

Load Case (Gasket with ID and OD gauge rings)	Code Radial stress	FEA Radial stress (note 3)	Code Tang. stress	FEA Tang. Stress (note 3)	FEA Peak Stress (note 4)	FEA Tresca Stress (note 4)
Pressure + Bolt up (285 psi)	10744	10900	3933	8527	13120	7096
(Pl) allowable psi	17500*					
Pressure + Bolt up + Moment (note 1) (464 psi equil)	17492	13783	5525	10437	16600	9022
(Pl+Pb) allow. psi	26250*					
Pressure + Bolt up Moment (note 2) (696 psi equil)	26238	17523	8287	12906	21120	11520
(Pl+Pb) allow. psi	26250*					

* allowable stress per ASME Section VIII Division 1

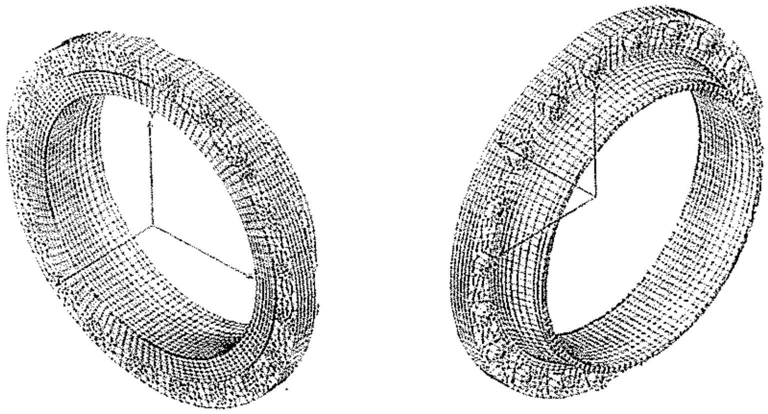
Notes:

(1) Moment = 1,317,000 in-lbs (= 8,140 psi stress in ½" wall 24" OD pipe)

(2) Moment = 3,025,000 in-lbs. (= 18,680 psi stress in ½" wall 24" OD pipe)

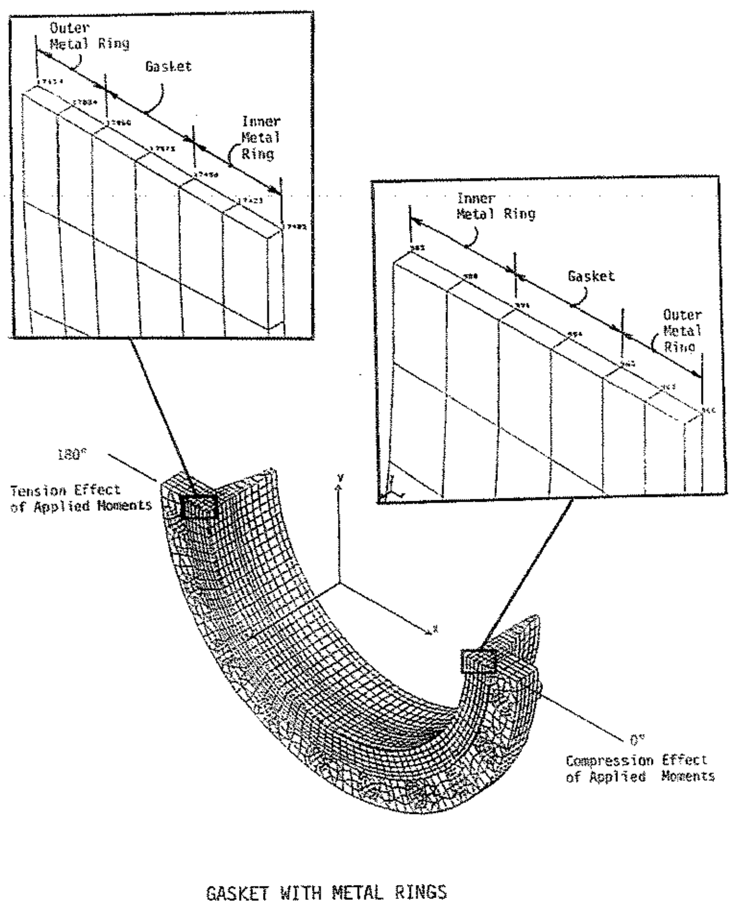
(3) Linearized stress @ SCL (located at 0 degrees)

(4) Stress @ SCL (located at 0 degrees)



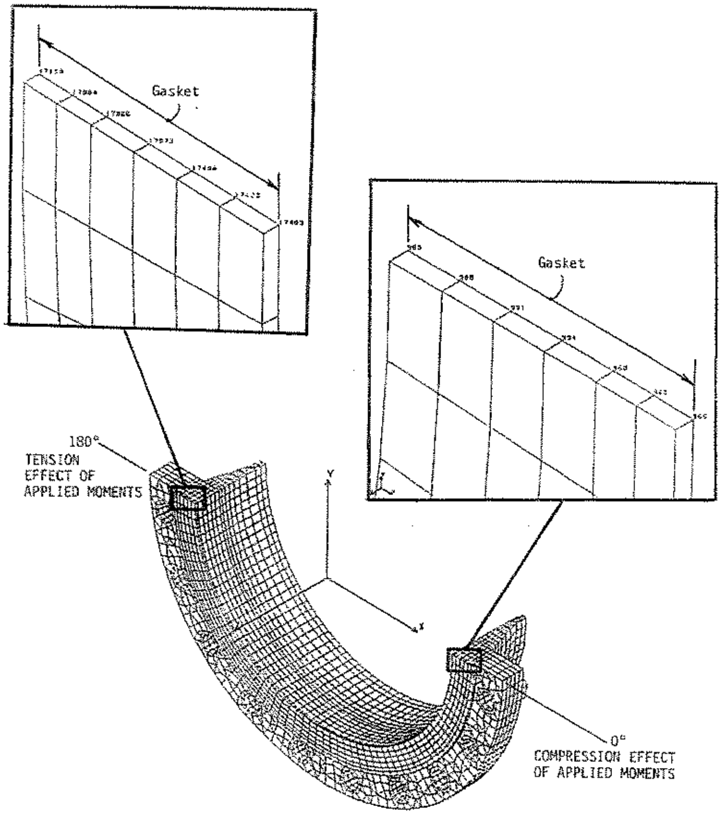
FINITE ELEMENT MODEL

FIGURE 1



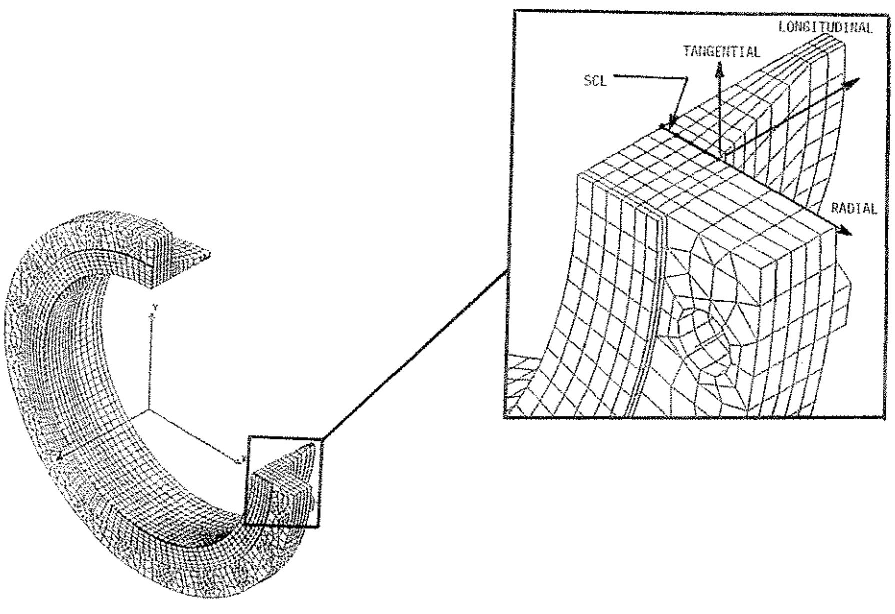
GASKET WITH METAL RINGS

FIGURE 2



GASKET WITHOUT METAL RING

FIGURE 3



STRESS CLASSIFICATION LINE

FIGURE 4

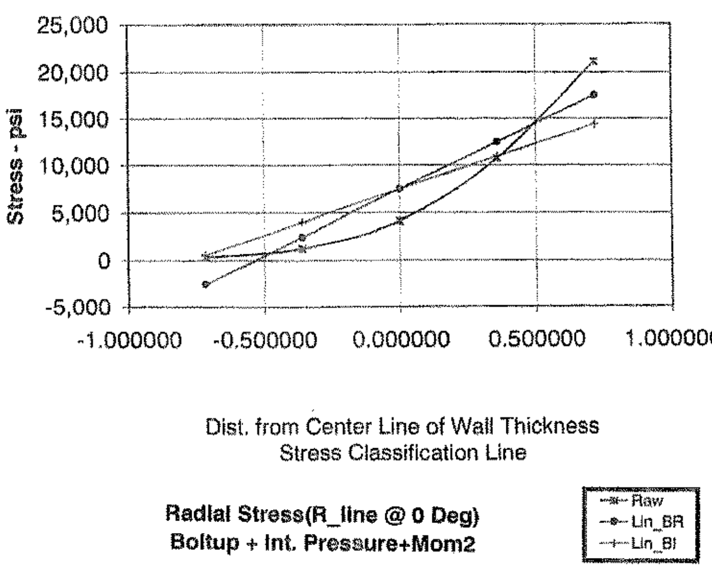


FIGURE 5

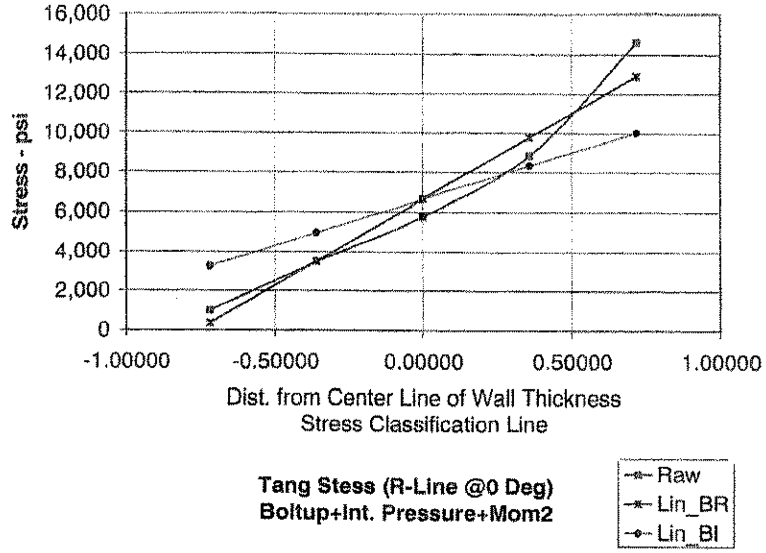


FIGURE 6

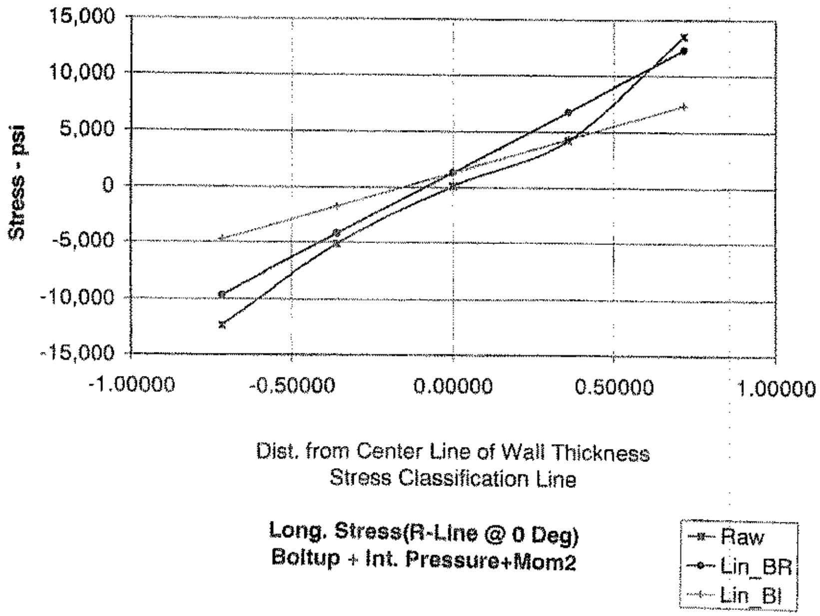


FIGURE 7

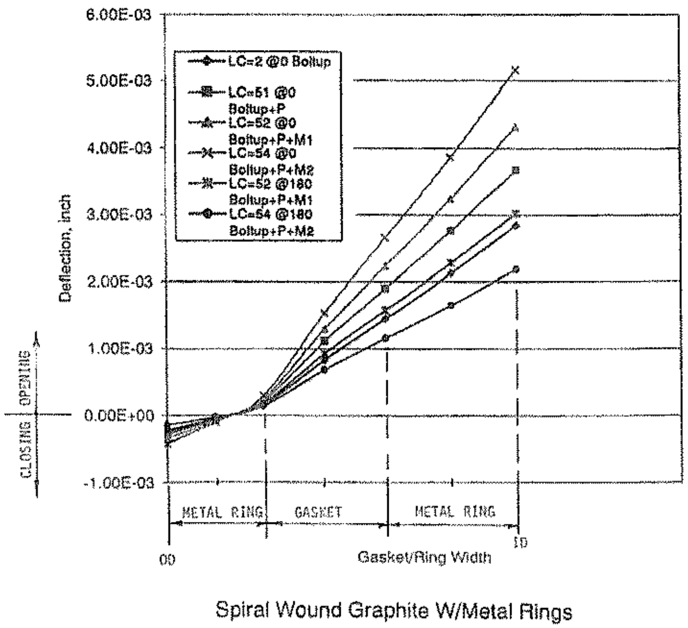
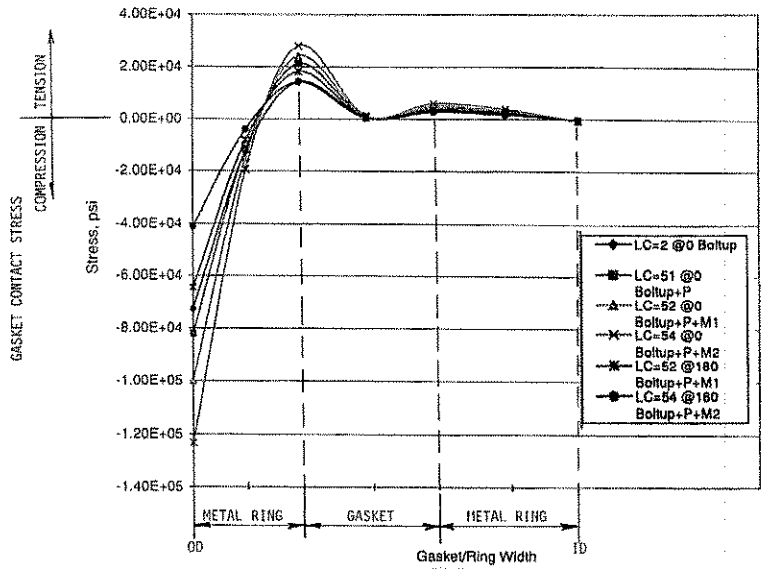
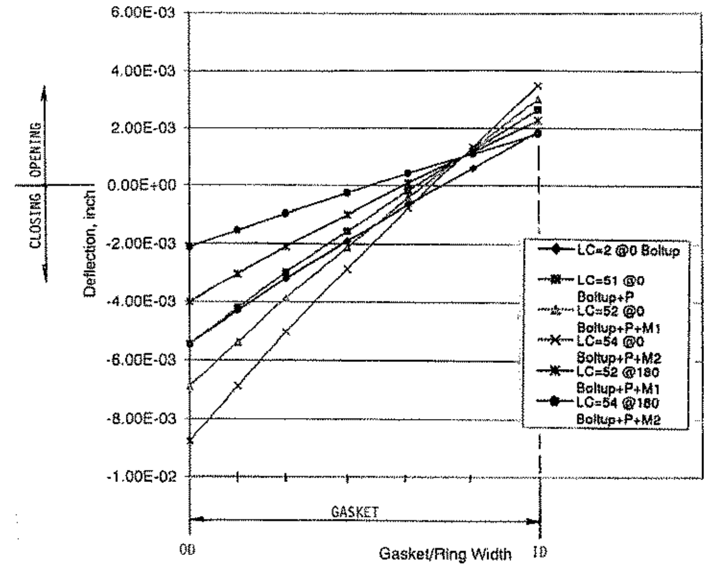


FIGURE 8



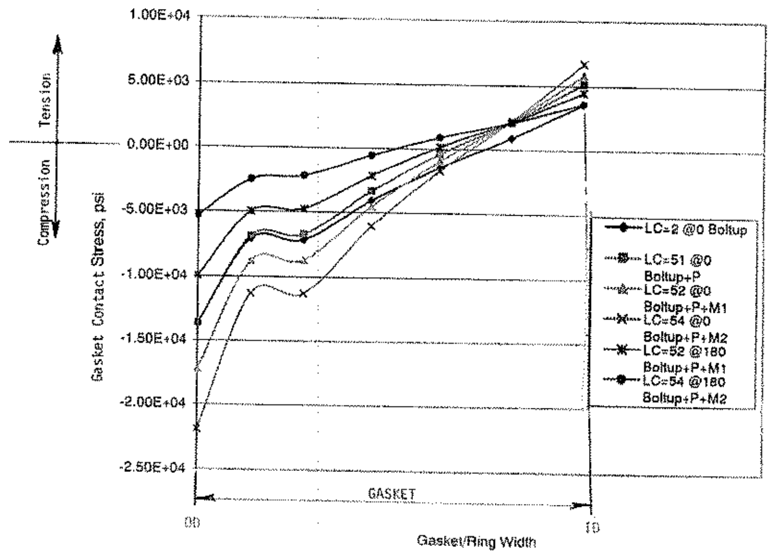
Spiral Wound Graphite W/Metal Rings

FIGURE 9



Spiral Wound Graphite W/O Metal Rings

FIGURE 10



Spiral Wound Graphite W/O Metal Rings

FIGURE 11

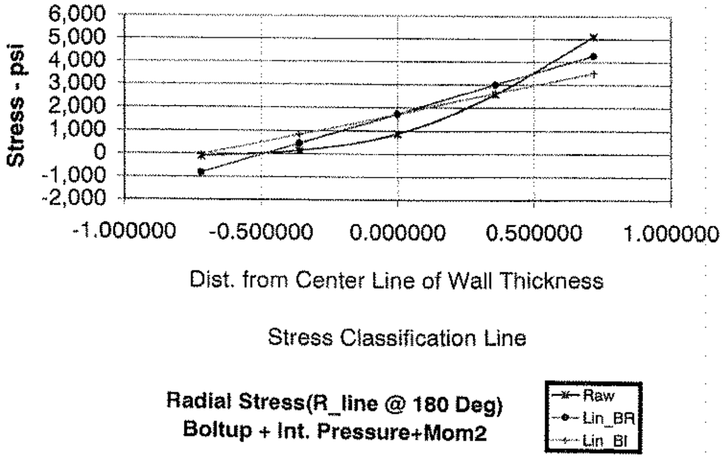


FIGURE 12

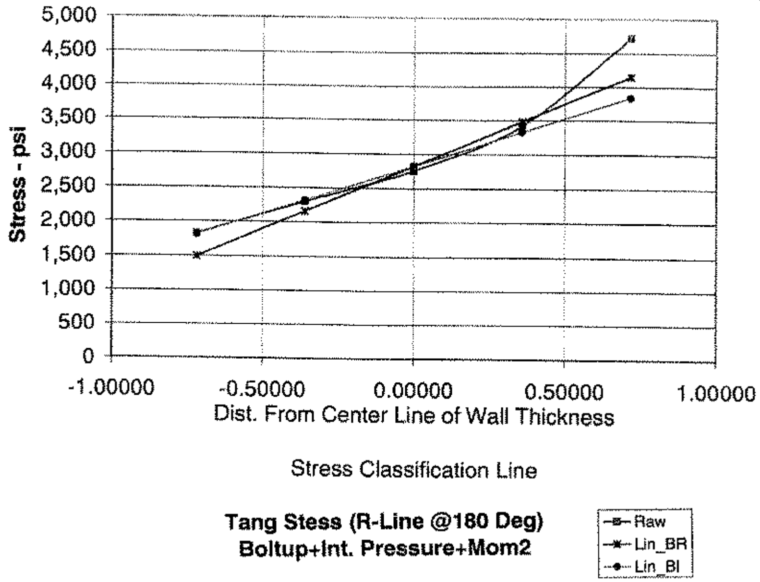


FIGURE 13

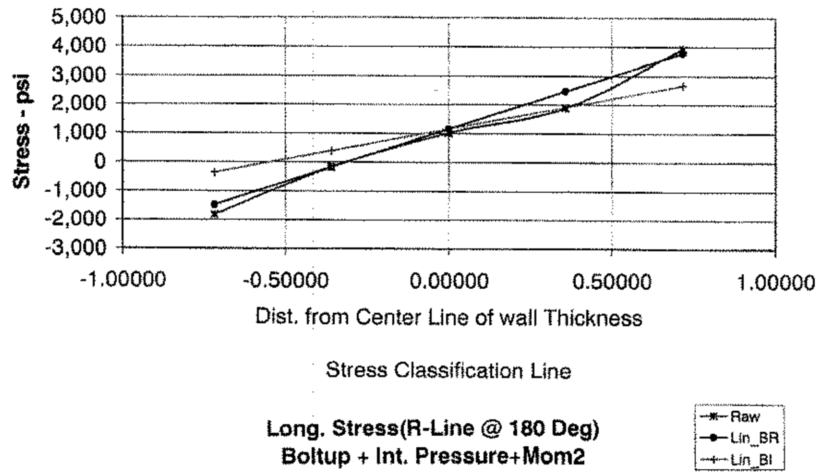
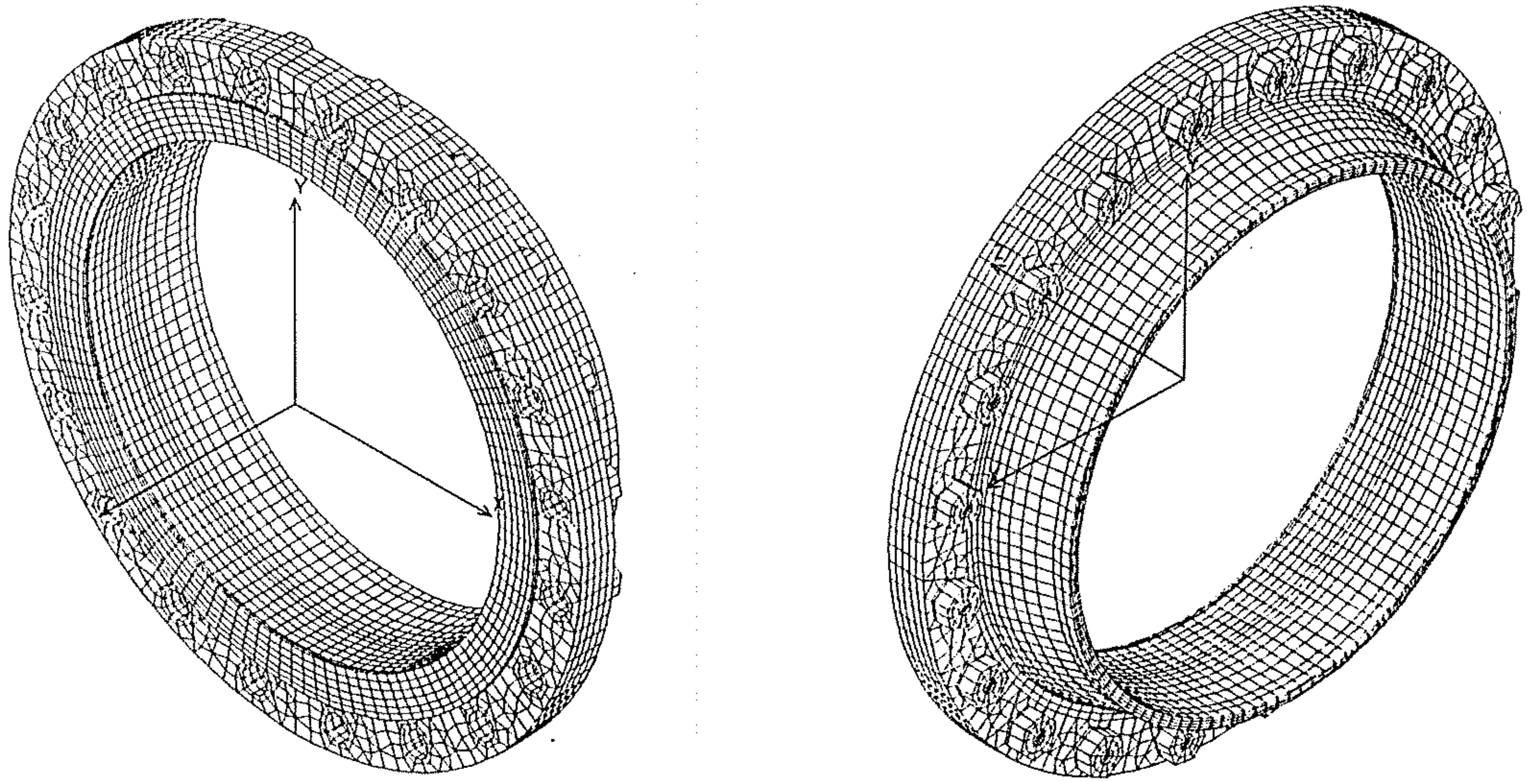
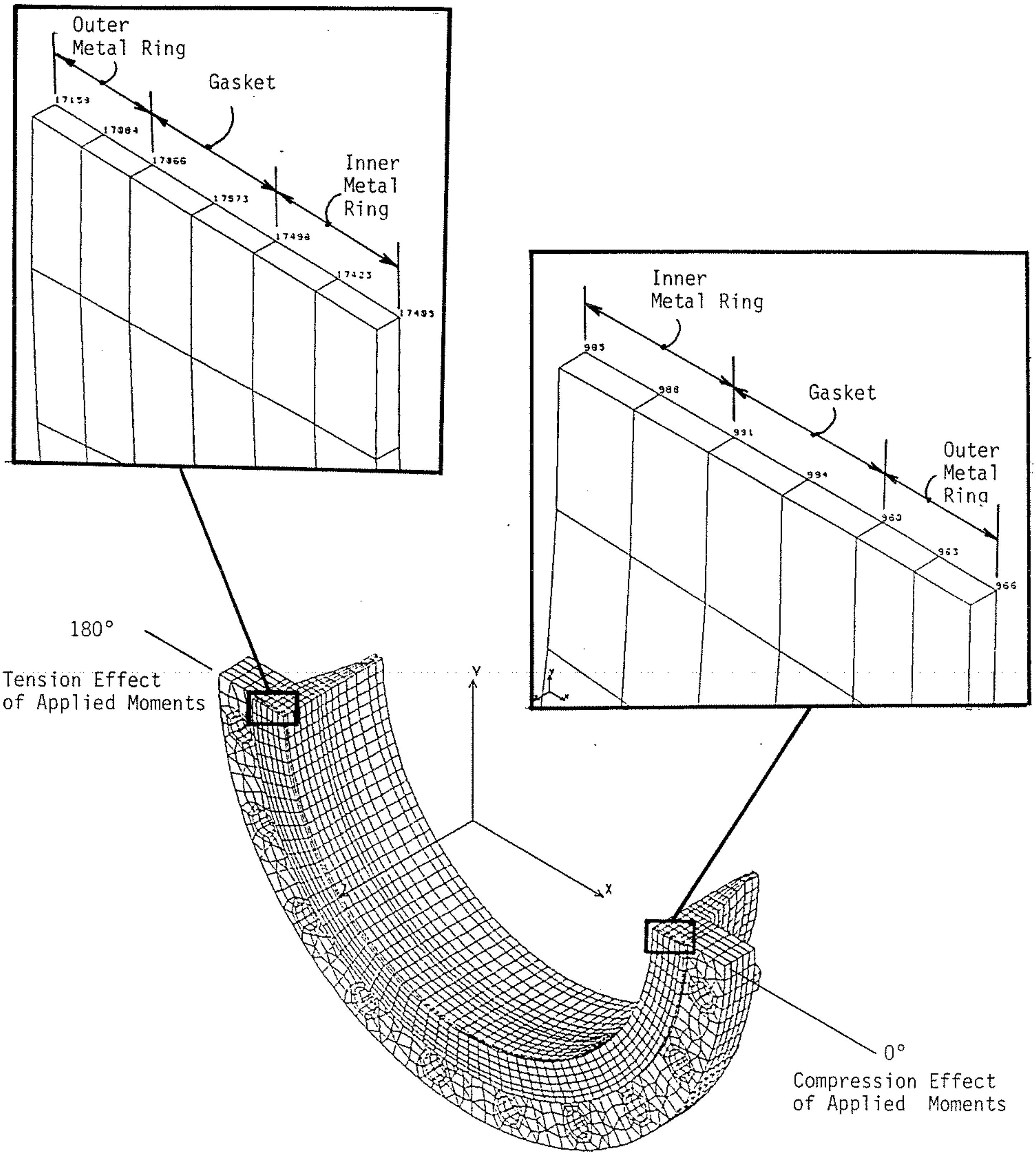


FIGURE 14

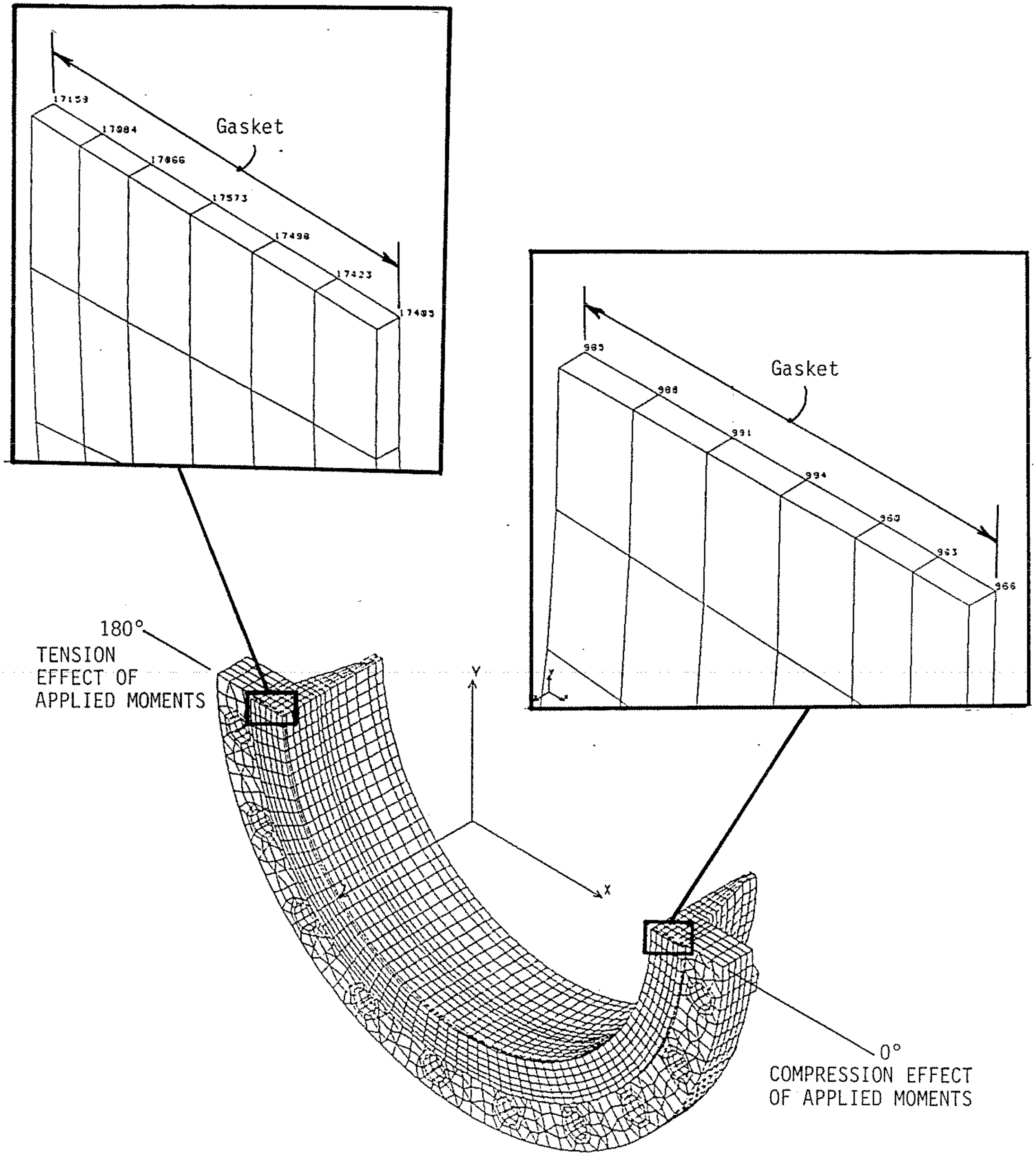


FINITE ELEMENT MODEL
FIGURE 1



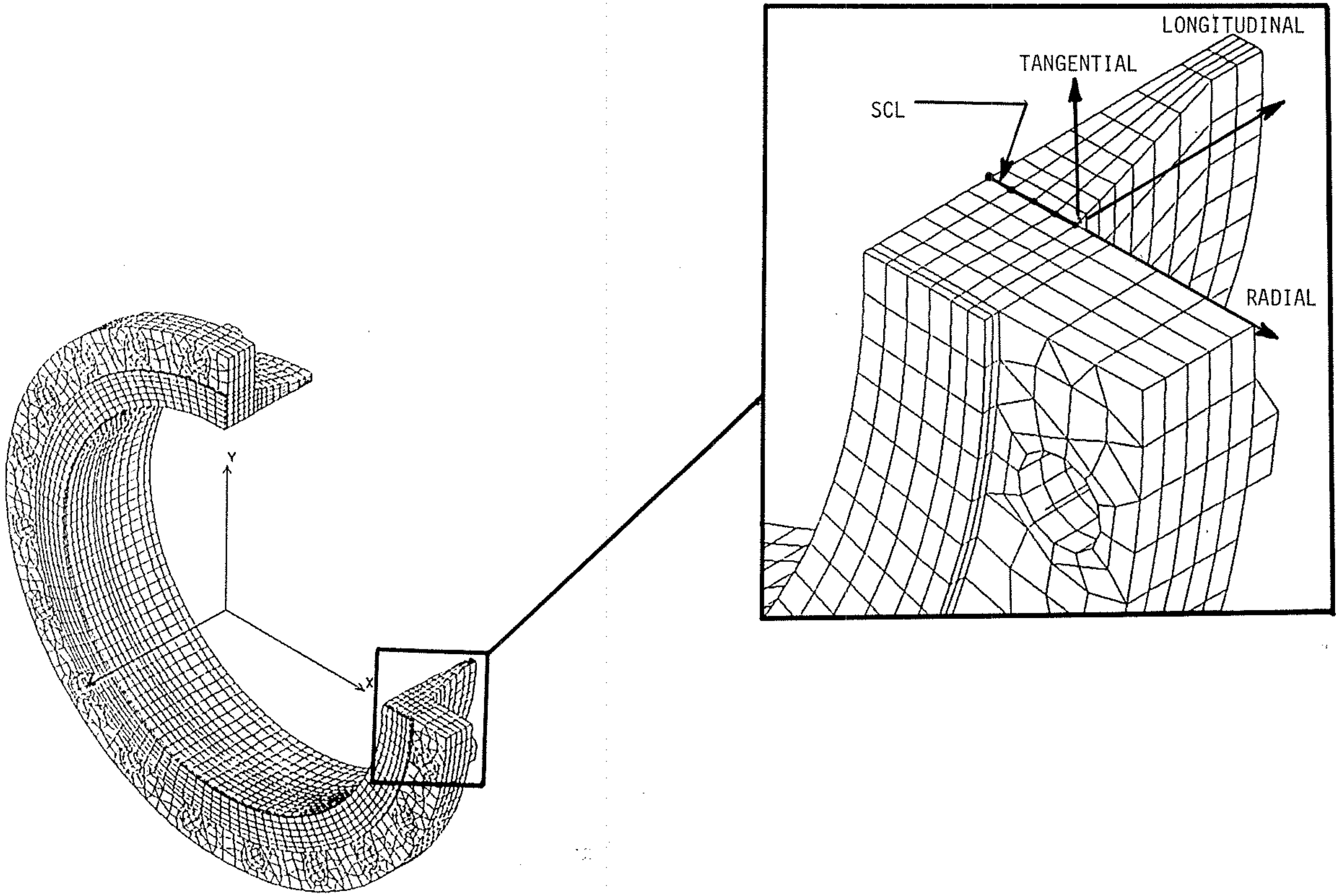
GASKET WITH METAL RINGS

FIGURE 2



GASKET WITHOUT METAL RING

FIGURE 3



STRESS CLASSIFICATION LINE

FIGURE 4

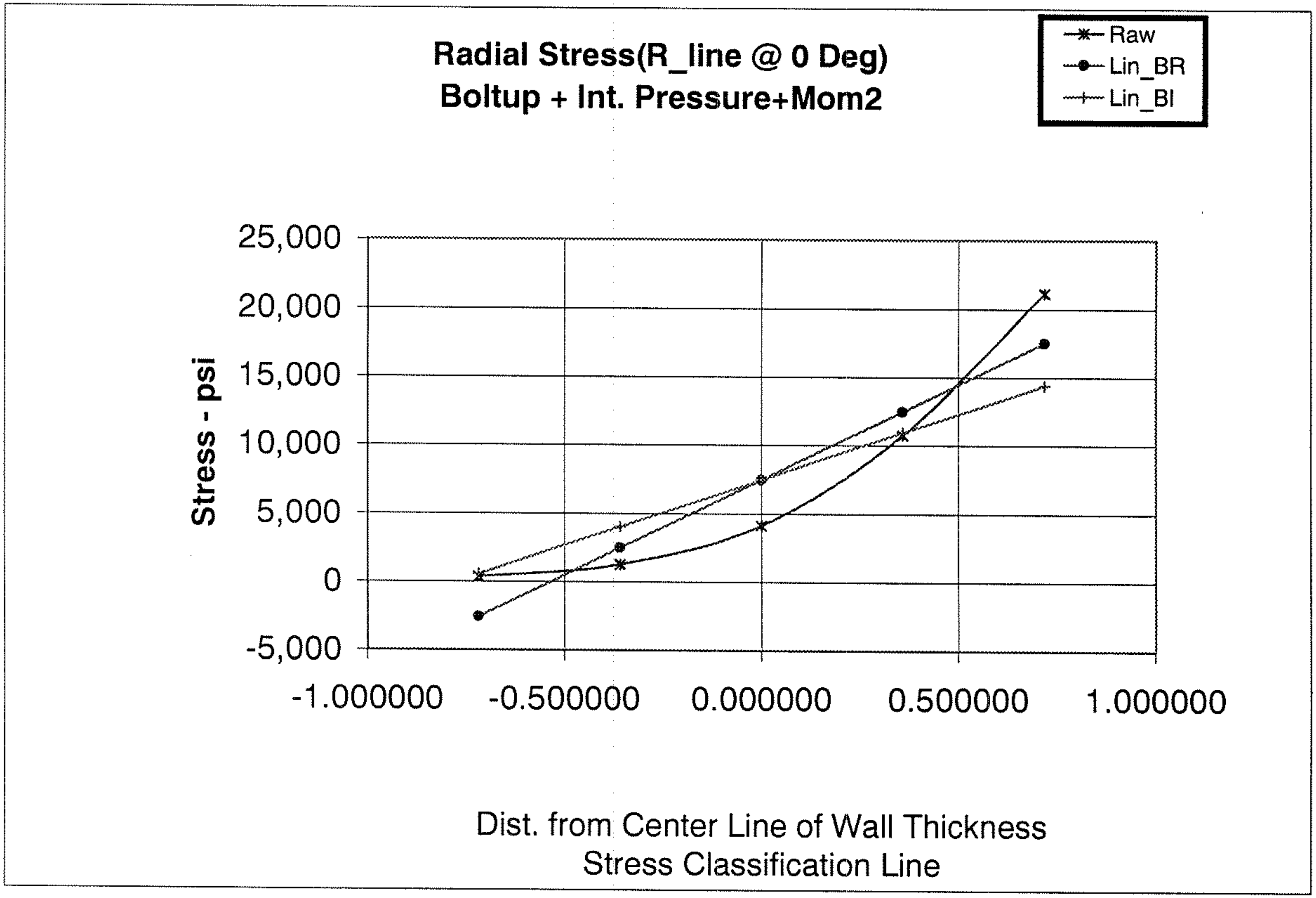

Radial Stress(R_line @ 0 Deg)
Boltup + Int. Pressure+Mom2
Raw
Lin_BR
Lin_BI
Stress - psi
25,000
20,000
15,000
10,000
5,000
0
-5,000
-1.000000
-0.500000
0.000000
0.500000
1.000000
Dist. from Center Line of Wall Thickness
Stress Classification Line

FIGURE 5

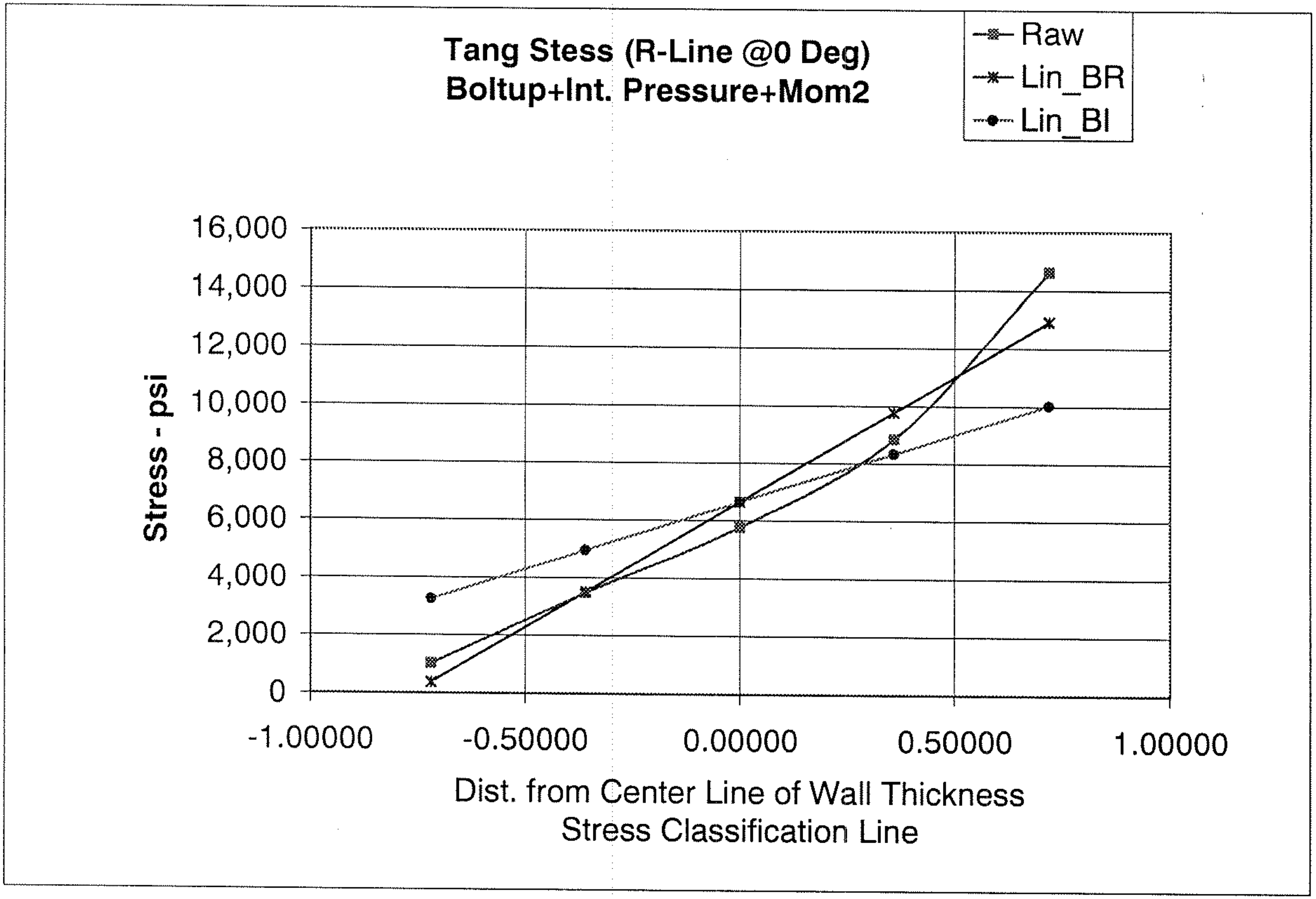

Tang Stess (R-Line @0 Deg)
Boltup+Int. Pressure+Mom2
Raw
Lin_BR
Lin_BI
16,000
14,000
12,000
10,000
8,000
6,000
4,000
2,000
0
Stress - psi
-1.00000
-0.50000
0.00000
0.50000
1.00000
Dist. from Center Line of Wall Thickness
Stress Classification Line

FIGURE 6

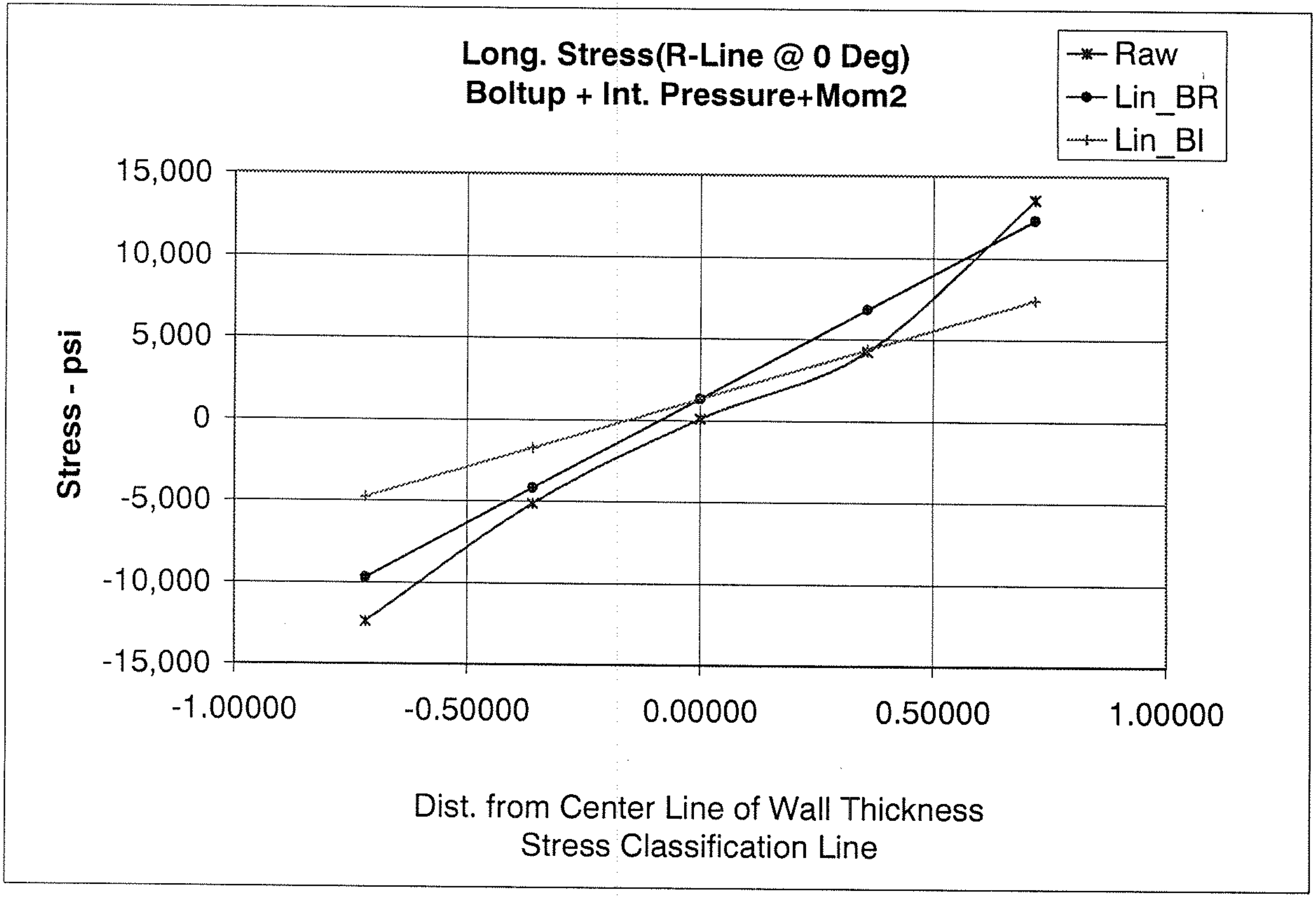
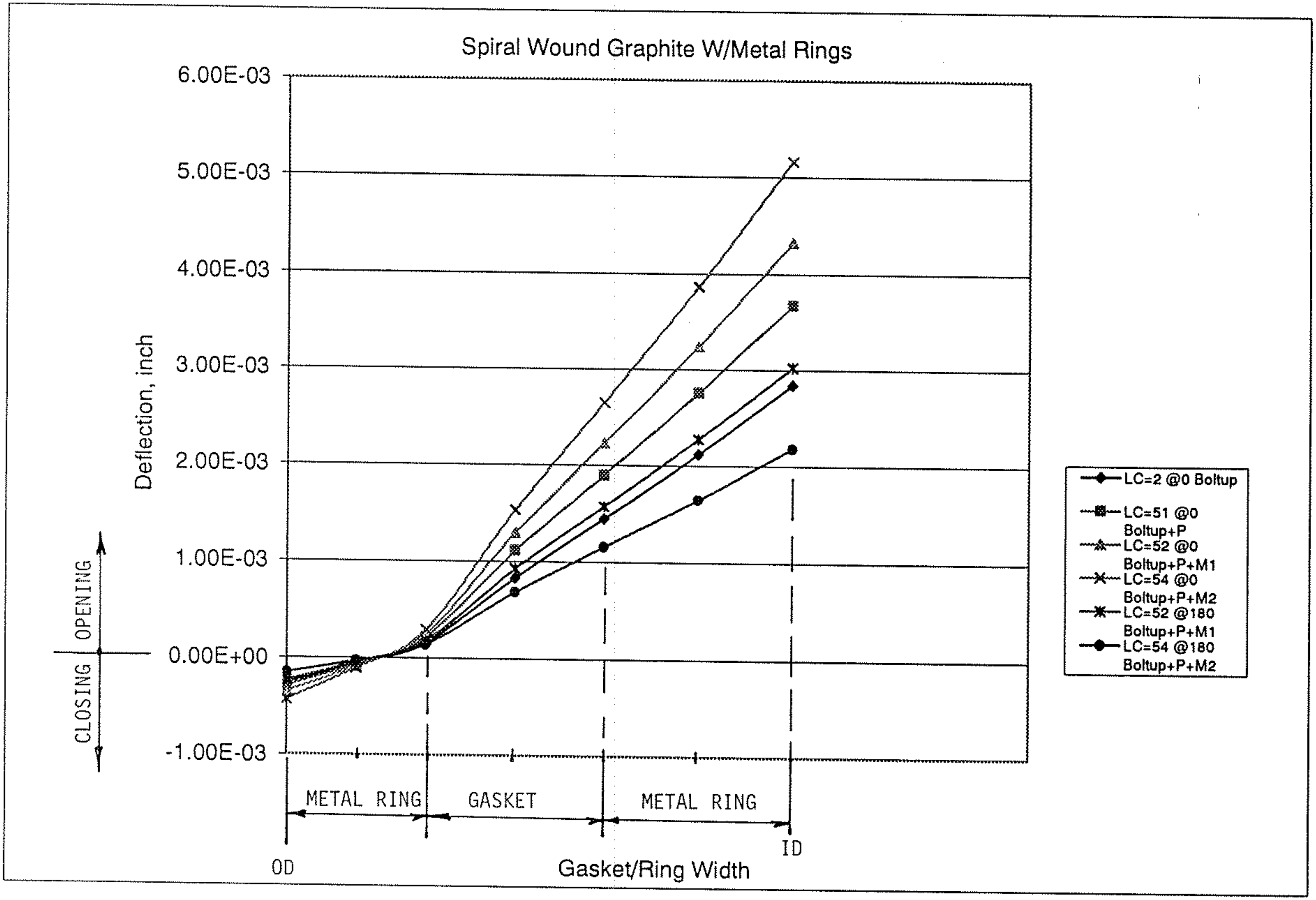
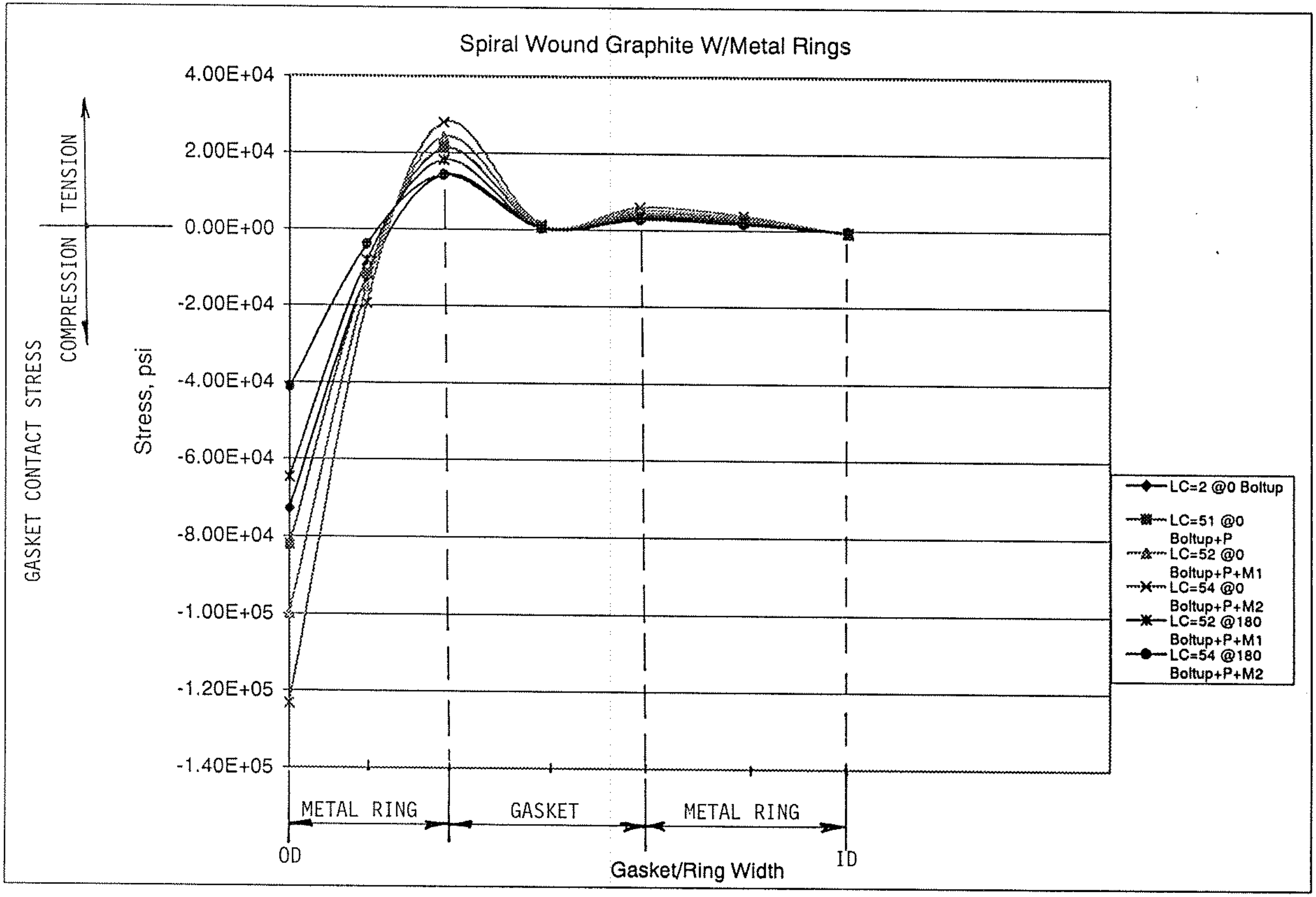

Long. Stress(R-Line @ 0 Deg)
Boltup + Int. Pressure+Mom2
Raw
Lin_BR
Lin_BI
15,000
10,000
5,000
0
-5,000
-10,000
-15,000
Stress - psi
-1.00000
-0.50000
0.00000
0.50000
1.00000
Dist. from Center Line of Wall Thickness
Stress Classification Line

FIGURE 7



Spiral Wound Graphite W/Metal Rings
6.00E-03
5.00E-03
4.00E-03
3.00E-03
2.00E-03
1.00E-03
0.00E+00
-1.00E-03
Deflection, inch
OPENING
CLOSING
METAL RING
GASKET
METAL RING
OD
ID
Gasket/Ring Width
LC=2 @0 Boltup
LC=51 @0 Boltup+P
LC=52 @0 Boltup+P+M1
LC=54 @0 Boltup+P+M2
LC=52 @180 Boltup+P+M1
LC=54 @180 Boltup+P+M2

FIGURE 8



Spiral Wound Graphite W/Metal Rings
GASKET CONTACT STRESS
TENSION
COMPRESSION
Stress, psi
4.00E+04
2.00E+04
0.00E+00
-2.00E+04
-4.00E+04
-6.00E+04
-8.00E+04
-1.00E+05
-1.20E+05
-1.40E+05
METAL RING
GASKET
METAL RING
OD
ID
Gasket/Ring Width
LC=2 @0 Boltup
LC=51 @0 Boltup+P
LC=52 @0 Boltup+P+M1
LC=54 @0 Boltup+P+M2
LC=52 @180 Boltup+P+M1
LC=54 @180 Boltup+P+M2

FIGURE 9

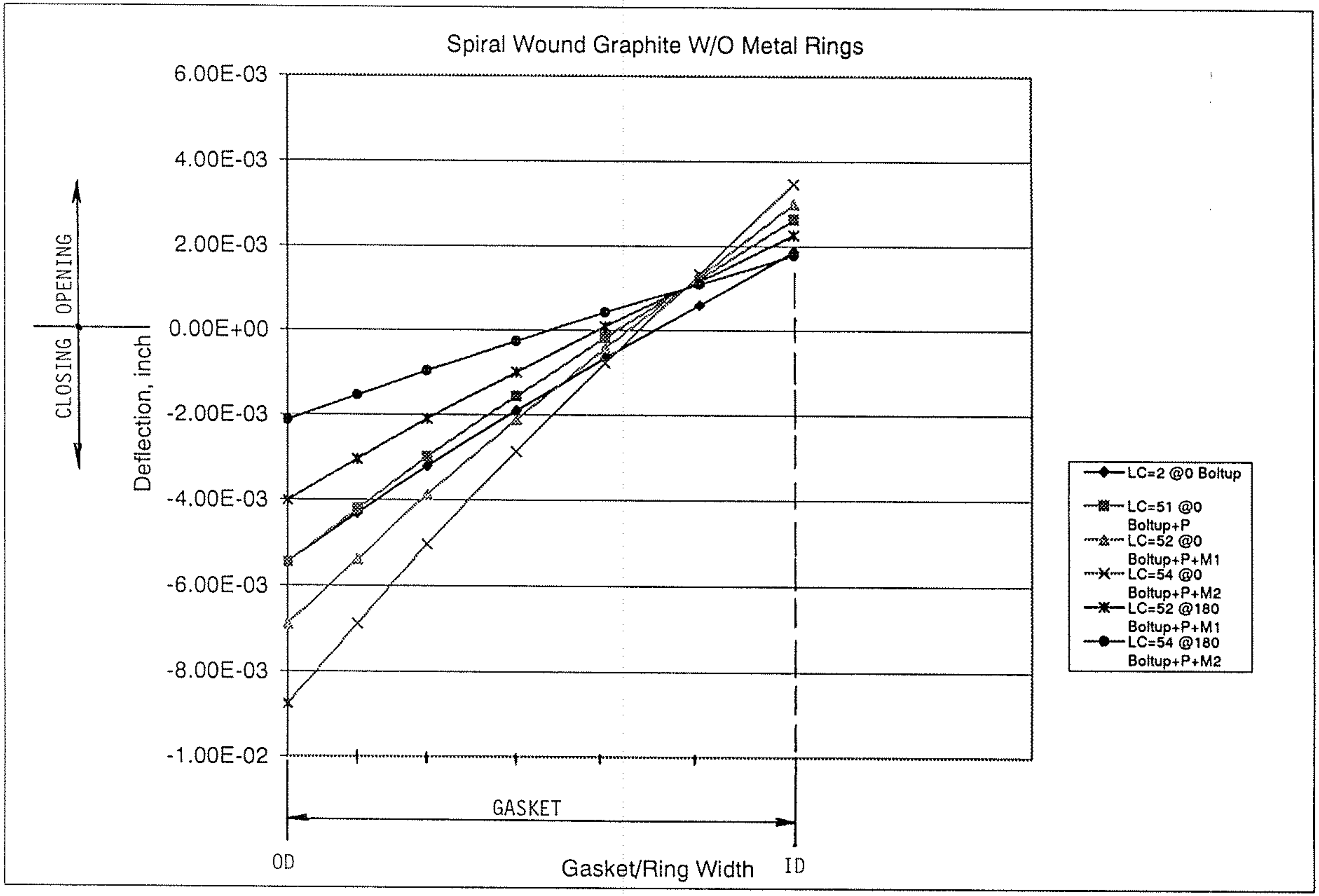
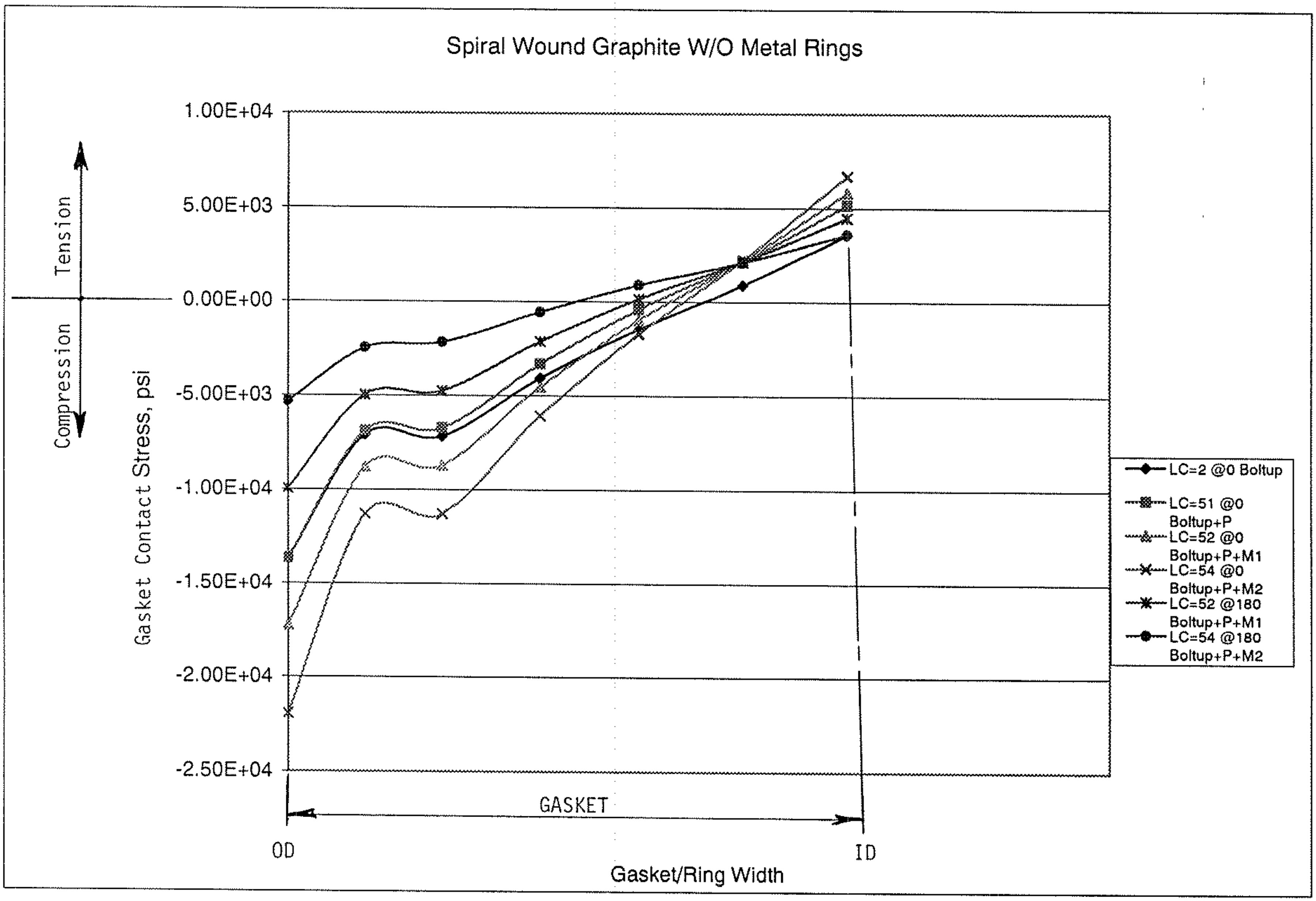

Spiral Wound Graphite W/O Metal Rings
6.00E-03
4.00E-03
2.00E-03
0.00E+00
-2.00E-03
-4.00E-03
-6.00E-03
-8.00E-03
-1.00E-02
Deflection, inch
OPENING
CLOSING
GASKET
OD
Gasket/Ring Width
ID
LC=2 @0 Boltup
LC=51 @0 Boltup+P
LC=52 @0 Boltup+P+M1
LC=54 @0 Boltup+P+M2
LC=52 @180 Boltup+P+M1
LC=54 @180 Boltup+P+M2

FIGURE 10



Spiral Wound Graphite W/O Metal Rings
1.00E+04
5.00E+03
0.00E+00
-5.00E+03
-1.00E+04
-1.50E+04
-2.00E+04
-2.50E+04
Tension
Compression
Gasket Contact Stress, psi
LC=2 @0 Boltup
LC=51 @0 Boltup+P
LC=52 @0 Boltup+P+M1
LC=54 @0 Boltup+P+M2
LC=52 @180 Boltup+P+M1
LC=54 @180 Boltup+P+M2
GASKET
OD
ID
Gasket/Ring Width

FIGURE 11

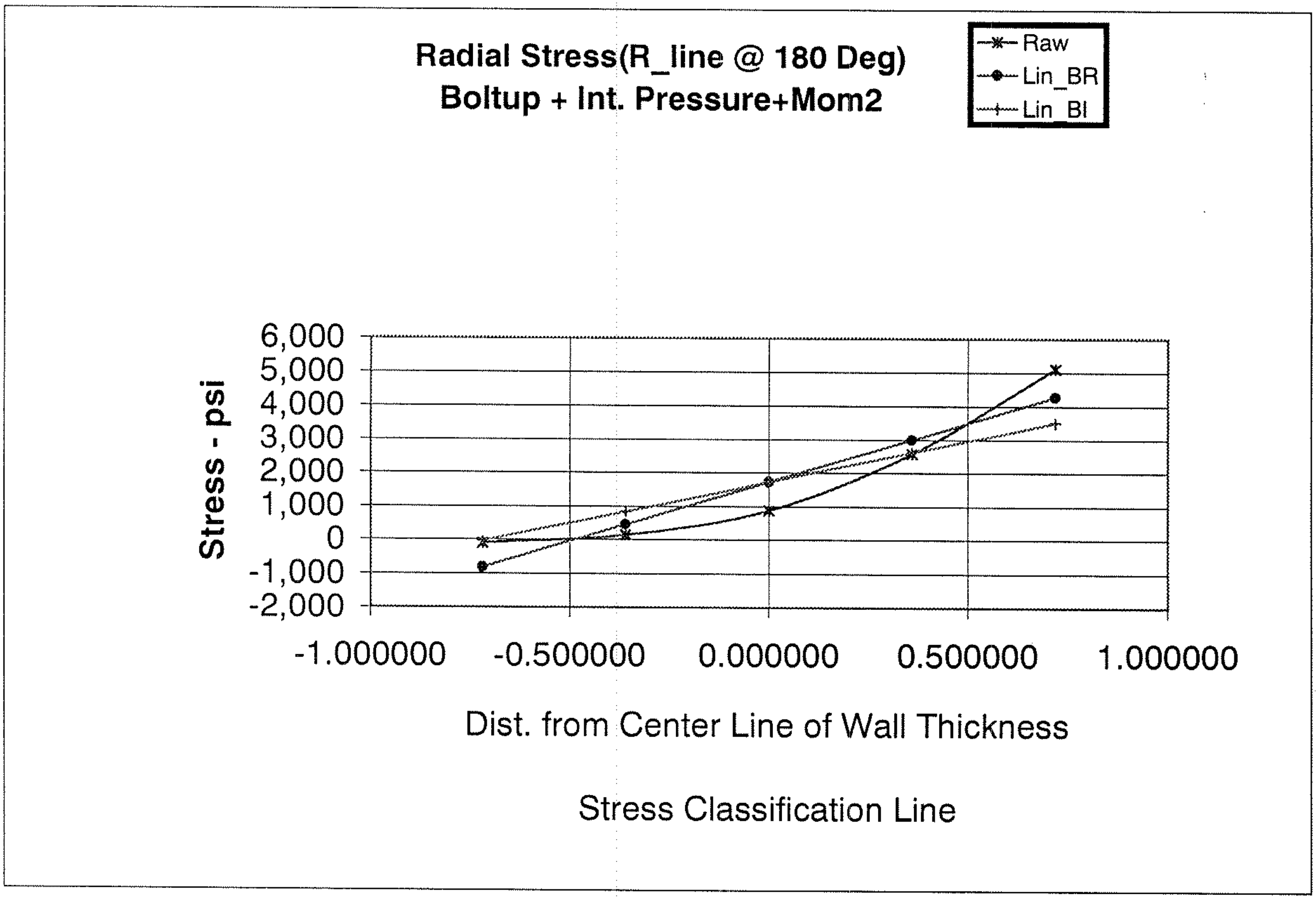
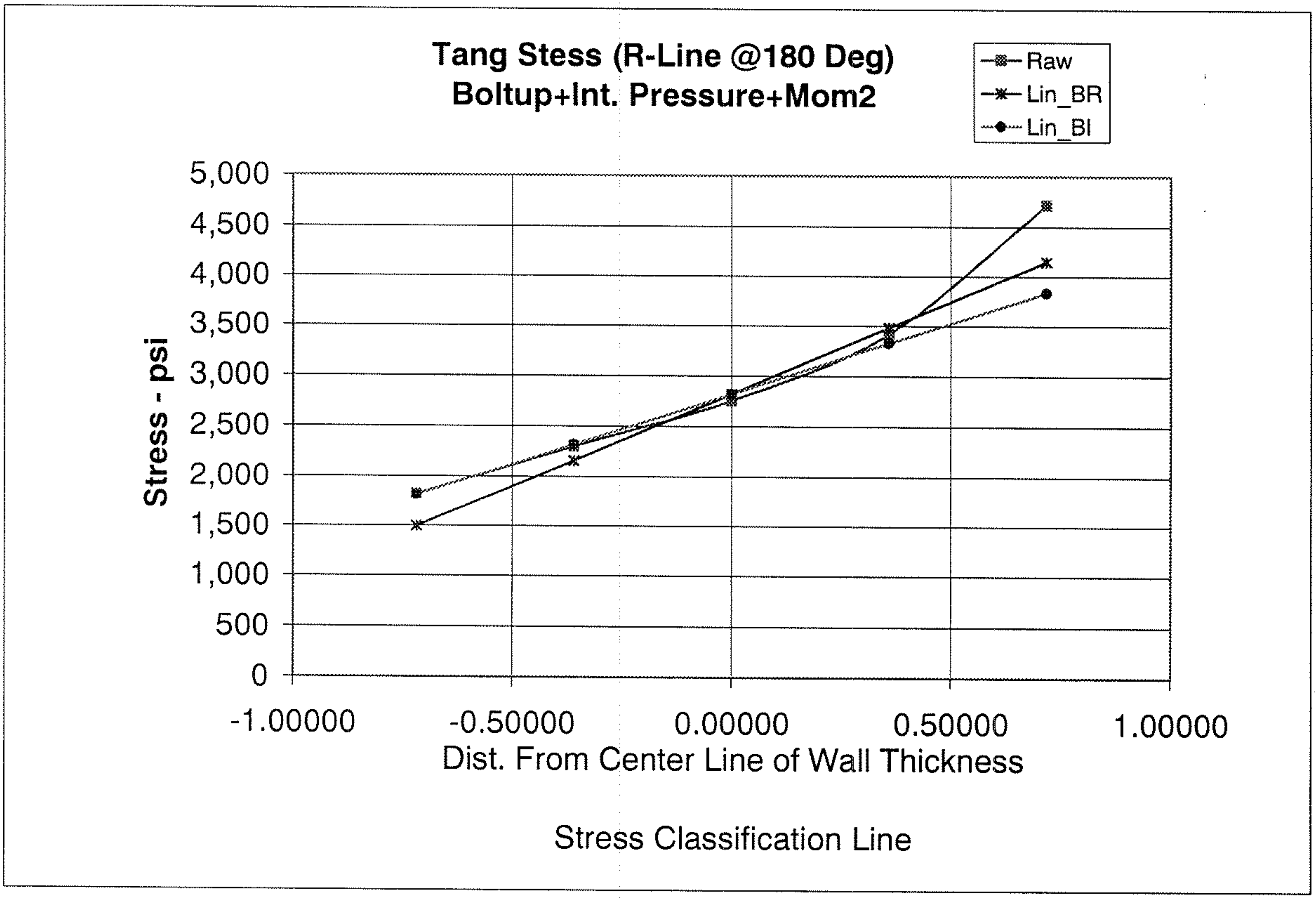

Radial Stress(R_line @ 180 Deg)
Boltup + Int. Pressure+Mom2
Raw
Lin_BR
Lin_BI
Stress - psi
6,000
5,000
4,000
3,000
2,000
1,000
0
-1,000
-2,000
-1.000000
-0.500000
0.000000
0.500000
1.000000
Dist. from Center Line of Wall Thickness
Stress Classification Line

FIGURE 12



Tang Stess (R-Line @180 Deg)
Boltup+Int. Pressure+Mom2
Raw
Lin_BR
Lin_BI
5,000
4,500
4,000
3,500
3,000
2,500
2,000
1,500
1,000
500
0
Stress - psi
-1.00000
-0.50000
0.00000
0.50000
1.00000
Dist. From Center Line of Wall Thickness
Stress Classification Line

FIGURE 13

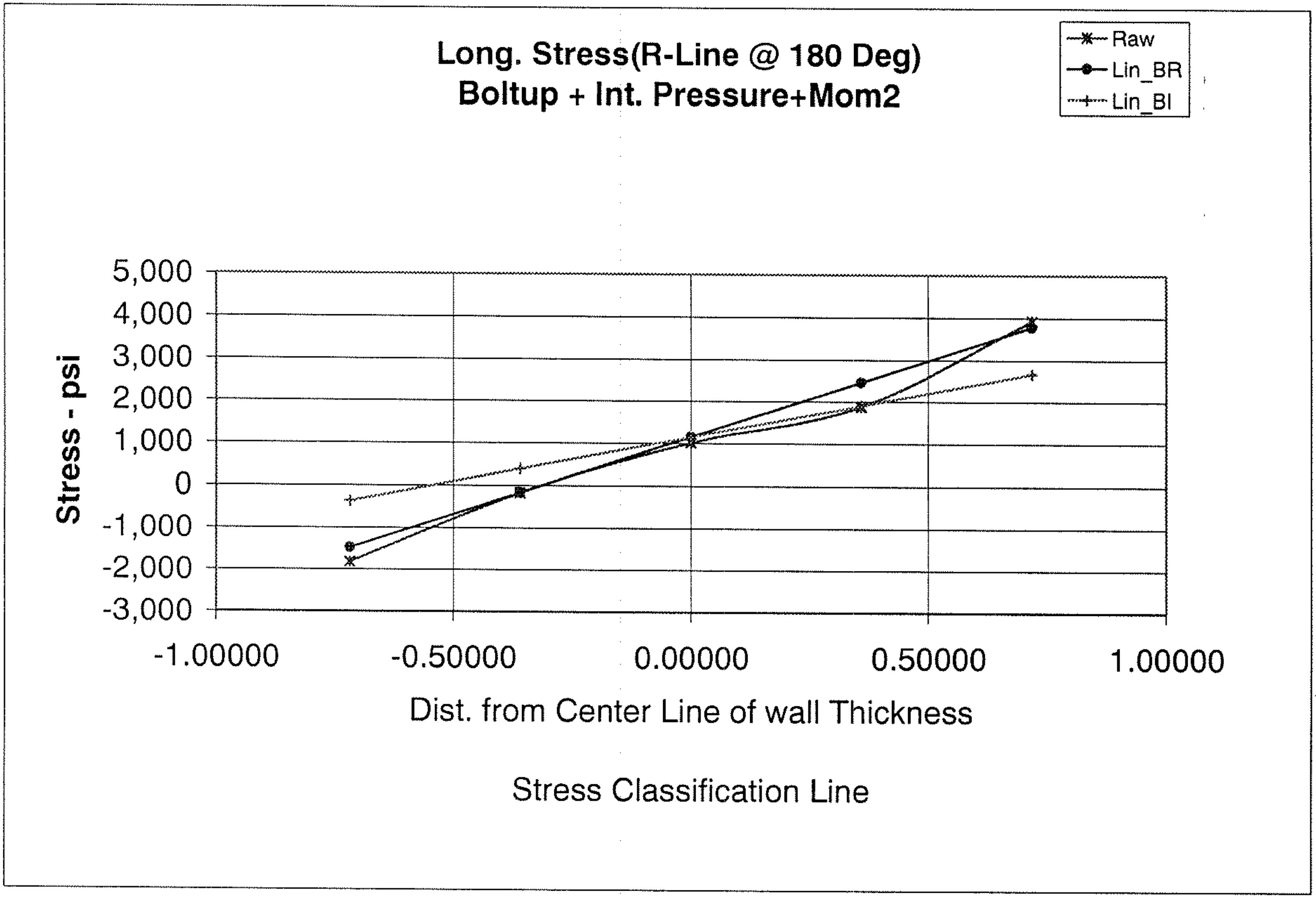

Long. Stress(R-Line @ 180 Deg)
Boltup + Int. Pressure+Mom2
Raw
Lin_BR
Lin_BI
Stress - psi
5,000
4,000
3,000
2,000
1,000
0
-1,000
-2,000
-3,000
-1.00000
-0.50000
0.00000
0.50000
1.00000
Dist. from Center Line of wall Thickness
Stress Classification Line

FIGURE 14