DESIGN AND ANALYSIS OF LID CLOSURE BOLTS FOR PACKAGES USED TO TRANSPORT RADIOACTIVE MATERIALS*

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ABSTRACT

The design criterion recommended by the U.S. Department of Energy for Category I radioactive packaging is found in Section III, Division 1, of the ASME Boiler and Pressure Vessel Code. This criterion provides material specifications and allowable stress limits for bolts used to secure lids of containment vessels. This paper describes the design requirements for Category I containment vessel lid closure bolts, and provides an example of a bolting stress analysis. The lid-closure bolting stress analysis compares calculations based on handbook formulas with an analysis performed with a finite-element computer code. The results show that the simple handbook calculations can be sufficiently accurate to evaluate the bolt stresses that occur in rotationally rigid lid flanges designed for metal-to-metal contact.

INTRODUCTION

The U.S. Department of Energy (DOE) requires that a transport package for high-level radioactive material be designed and constructed in compliance with the structural requirements of DOE Order 5480.3[1] and Title 10 of the Code of Federal Regulations, Part 71 (10 CFR 71).[2] These regulations specify approval standards, structural performance criteria, and package integrity requirements that must be met during transport.

A package can be qualified to these requirements by testing or analysis. Qualification by analysis requires that the package be designed to criteria suitable for the environment and structural loadings unique to high-level,

Category I, radioactive materials transport packagings. At present, no radioactive materials packaging-specific design criteria exist. However, both the DOE[3] and the U.S. Nuclear Regulatory Commission (NRC)[4] recommend Section III, Division 1, Subsection NB of the ASME Boiler and Pressure Vessel Code[5] as an acceptable source for Category I design criteria.

For containment vessel lid closure bolting, the Code provides material qualification requirements and design data for acceptable bolting materials. The Code also offers guidance to select the minimum bolt size needed to seat typical gasketed joints, but does not provide guidance for initial bolt tightening nor for detailed stress analysis of the bolted joint.

The purpose of this paper is to describe the ASME Code requirements for the design and analysis of the lid-closure bolts for a Category I containment vessel, and provide an example of a typical stress analysis. The example compares calculations based on handbook formulas with an analysis performed with a finite-element computer code.

DESIGN CRITERIA

The ASME Section III, Subsection NB Code design criteria for vessel bolting consists of materials qualification requirements, materials specifications, allowable stress limits, and guidance for determining a minimum bolt cross-sectional area to seat typical gasket types and materials. This criteria does not provide guidance for initial bolt tightening nor for detailed stress analysis of the bolted joint.

The mechanical loadings on Category I containment vessel bolts are identified by the Code as the Design Loadings, the Level A Service Limits, and the Level D Service Limits. For transport packagings, the Level A Service Limits correspond to the Normal Conditions of Transport (NCT) and the Level D Service Limits to the Hypothetical Accident Conditions (HAC).

The Design Loading allowable stress, S_m, is limited to one-third the specified minimum yield strength at temperature and is used only for the initial bolt sizing purposes using the design pressure and gasket reaction forces. The actual service stresses allowed for bolts are listed in Table 1. The magnitude of these stresses are multiples of S_m and depend on the mechanical loadings during NCT and HAC. The maximum allowable stress for the NCT is equal to the yield strength when the bolts are tightened by hand using a torque wrench. For HAC, the maximum allowable stress from tension plus bending is the ultimate tensile strength. All calculated stresses in a Code design are based on stress intensity, which is defined as twice the maximum shear stress and is equal to the largest algebraic difference between any two of the three principal stresses.

MATERIALS

The qualification requirements for the bolting materials used in containment vessels to transport Category I high-level radioactive materials are provided in Article NB-2000 of the Code. The bolting materials recommended for these containment vessels are listed in Section II, Part D, Subpart 1, Table 4.[6] Non-Code materials may be acceptable for bolting if they are qualified by criteria equivalent to that applied to Code materials. These criteria are:

1. Procurement to an authoritative material specification such as ASTM, AMS, MIL, or SAE.
2. Quantitative proof of the material's suitability for both the maximum and minimum service temperatures.
3. Certification of materials and fabrication equivalent to the requirements given in Section III, Subsection NB, Article NB-4000 of the ASME Code.
4. Non-destructive examination equivalent to the requirements given in Section III, Subsection NB, Article NB-2000 of the ASME Code.

In addition, both Code and non-Code bolting materials should satisfy the following requisites to assure quality:

1. Procurement of the material or finished bolts from a vendor qualified in accordance with a quality assurance plan.
2. Quantitative proof that the bolts are not counterfeit, including confirmation of the chemical and mechanical properties.

STRESS ANALYSIS

The Code Design Condition analysis considers only internal pressure and the gasket seating forces which provides an initial estimate of the total bolt cross-sectional area required. The mechanical loadings encountered during NCT and HAC must be evaluated within the stress limits allowed for the Level A and D Service Limits.

The Design Condition stress analysis can be accomplished by following the guidance provided in Section III, Division 1, Appendix E of the Code. For gaskets and flange facing configurations not considered in the Code, manufacturers data for gasket reactions can be used.

For NCT, governed by the Level A Service Limit allowable stresses, the analysis should evaluate the following loadings:

1. Internal pressure.
2. External mechanical loads.
3. Initial bolt preload.
4. Flange rotations.
5. Differential thermal expansion.
6. Fatigue loadings.

This analysis can be accomplished by computer code modeling or hand calculations. Certain simplifying assumptions, when justified, such as no flange rotations and the concentration of all the differential thermal

expansion stresses in the bolts make the hand calculations manageable.*

In addition, Category I containment vessels do not usually experience external mechanical impact loadings on the closure flange bolting during NCT. Thus, the loading conditions that generally govern this analysis are the $3S_m$ (yield strength) limit for maximum stress intensity in the bolts. If the bolts are tightened by hand with a torque wrench, this stress limit must consider the torsional shear stress along with the axial preload and differential thermal expansion stresses in the evaluation. The amount of applied torque that actually induces axial and shear stresses in the bolts is not easily quantified. Bickford suggests that 90% of the applied torque is lost in nut and thread friction and only 10% transmitted to the bolt shank.[7] However, a recent experimental study using strain-gauged 1-inch diameter bolts with fine and coarse threads has shown different results.[8] The results of this study show that approximately 36 to 58% of the applied torque is transmitted to the shank of fine thread bolts and 36 to 45% for coarse thread bolts, depending on the thread friction during tightening. The lower values of transmitted torque are for bolts with dry threads, and the higher value for bolts with threads lubricated with oil. In addition, this study shows the bolt friction factor, K, is approximately 0.33 for dry fine threads and 0.22 for lubricated fine threads. The value of K for bolts with coarse threads was found to be approximately 0.26 for both the dry and lubricated test condition. Consequently, in the absence of experimental evidence for the specific bolt and joint configuration under consideration, a reasonable assumption would be to consider 50% of the torsional forces acting to induce shear stress on the bolts. Further, it also appears reasonable and conservative to use values of K equal to 0.20 for dry threads and 0.15 for lubricated threads.[9]

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* Recent studies of containment vessel design analysis methods have shown that the structural deformations at the region of the closure flange can be significant and not amenable to hand calculations during thermal events such as the NCT heat-up and HAC fire test.[10] Therefore simplifying ·assumptions such as negligible flange rotations may lead to nonconservative results.

The loads due to the interaction between the bolts and gasket, or bolts and flange under internal pressure should also be included in the stress analysis of bolts. This interaction will increase the bolt stresses due to gasket or flange relief under pressure. The maximum increase in total bolt load is half the pressure load when the stiffness of these components is equal. When the stiffness of the gasket or flange is much greater than the bolts, the contribution of the pressure load to the bolt stress becomes insignificant. The relationship between the bolt and flange or gasket stiffness is shown in Fig. 1. However, for the case where the gasket physically separates the flanges and the gasket is very soft, the contribution of the pressure load approaches one-hundred percent.

Another source of bolt stress is due to the bending deformation of the closure lid. A relatively thin closure lid will deform under pressure and induce additional axial plus bending loads on the bolts. For example, a thin flat lid will try to deform to a dome-shape and thus 'pry' the bolts away from their tightly clamped position. A method to estimate and account for these loads for flat closure lids is provided in Ref. 11.

The HAC stress analysis must consider the effects of the mechanical impact and thermal loadings that result from the qualification tests specified in 10 CFR 71. For a Code design, the allowable stresses are governed by the rules for the Level D Service Limits. These stresses may approach the ultimate tensile strength of the bolting material and the containment vessel may be damaged, but the release rate of the contents must be less than the allowable value of an A_2 quantity of radioactive material per week[2].

TYPICAL BOLTING ANALYSIS

A schematic of an idealized Category I containment vessel used for an example is shown if Fig. 2. This vessel is made from ASME SA-240 Type 316 stainless steel with corrosion and heat resistant steel bolts made to the Aerospace Material Specification (AMS) 5726B. This UNS designation for

this material is S66286, and is also known as A-286 stainless steel. The flat lid is sealed with a self-energizing O-ring gasket.

This example will consider only the case for the NCT where the bolt stresses are governed by the Level A Service Limits in the ASME Code. For this transport condition, the internal pressure is assumed to be 120 psig at a maximum temperature of 220 °F. The mechanical and physical properties of the bolt and vessel materials are given in Table 2. The bolts are assumed to be lubricated and hand-torqued to a value of 45 ft-lb prior to shipping. Further, it is assumed that the flat closure lid is stiff enough so that under all transport conditions, the 'prying' loads to the bolts are negligible.

Analysis Using Handbook Formulas   The handbook formulas used to determine the bolt stresses are listed in Appendix A. These formulas can be obtained from Refs. 12-14. Table 3 provides a comparison between the stresses calculated from handbook formulas and the stresses allowed by the ASME Code. This Table shows that the average stress due to preload, pressure, and thermal expansion is almost equal to the allowable $2S_m$ stress, and the maximum stress for the same loads but includes the torsional shear stress is only 81 % of the allowable $3S_m$ stress.

The largest contributor to the bolt stress is the 109.4 ksi axial stress due to the preload torque. This preload can be reduced if the bolts were found to be overstressed by reducing the bolt-up torque. This is easily accomplished because the stress is linear with preload torque. Therefore, a 10% reduction in preload stress can be achieved by a 10% reduction in torque. However, care must be exercised if the preload is reduced because the clamping force generated may be necessary to properly seat the gasket and/or prevent movement of the lid during transport.

Another way to reduce the stresses in the bolts is to increase their number. However, one must consider the man-effort necessary to install, tighten, and verify the bolt torque for a vessel containing highly radioactive material. Consequently, the prudent approach to reducing the bolt stresses

is to reduce the bolt torque, increase the bolt diameter, or change the bolt material to one with increased tensile strength.

The simplified containment vessel geometry analyzed in the present paper was chosen primarily to compare the bolt stresses determined by handbook calculations with results from a finite-element analysis. An actual Category I containment vessel would not typically be designed with the simple flat lid shown in Fig. 2. These containment vessels are usually designed with a recessed lid to reduce side movements due to impact loads or thermal transients that may occur during transport.

Analysis by a Finite-element Model   A detailed two-material, three-dimensional ANSYS finite-element model was developed for this analysis.[16] By taking advantage of axial symmetry, only one half of the bolt and a radial sector of the vessel flange and lid was modeled, i.e., a sector of an opening equal to $(360/8)/2=22.5°$. Two views of this model are shown in Fig. 3. The bolt was extend through the flange thickness for modeling simplicity. No bolt stem threads nor flange bolt hole threads were modeled. Rather, displacement continuity was enforced across the surfaces between the bolt stem and flange bolt hole.

The contact between surfaces of the flange, lid, and bolt head was controlled with contact elements, a feature available in ANSYS code. These contact elements effectively prevent penetration of designated surfaces by keeping track of mutual position of their respective nodes. The friction coefficient was taken to be zero in this analysis. The O-ring gasket was not incorporated in the model, but its presence was reflected by the fact that the internal pressure boundary on the lid was limited by the O-ring diameter.

The preload torque axial force in the bolt was achieved by pre-straining the bolt to a level such that when put in place, the bolt retains this force. After this stage, the model is subjected to uniform temperature

change from 70°F to 220°F, and then to an internal pressure of 120 psig. The shear stress due to the preload torque was applied after the finite-element modeling by using the formula for S_{qv} given in the Appendix. The results of finite element model are also presented in Table 3.

<u>Comparison of Bolt Stress Calculations</u> The data shown in Table 3 indicate that there is virtually no difference in the bolt stresses calculated by handbook formulas and the finite-element model. There are two reasons for this result. First, the finite-element model used pre-strained bolts to account for the axial preload torque, so this large axial bolt stress was fixed before the calculations began. Second, the model did not consider the shear stress in the bolts due to the preload torque. This shear stress was calculated by the formula in the Appendix, and added to the results of the finite-element analysis in the equation for the maximum stress intensity. Since the preload torque on the bolts overwhelms the other mechanical loads during the NCT, these two constraints effectively assure that the results from both analyses will be nearly identical.

A comparison of the individual contributions from preload, pressure, and thermal expansion to the total axial stress in the bolts is given in Table 4. The near equality in the preload stresses of 109.4 and 108.2 ksi for the handbook calculation and the finite-element model are due to the modeling constraints discussed above. The only significant difference between these methods of analysis are for the pressure stresses. Because the flange-to-bolt stiffness for this vessel is large (90:1), the fraction of pressure load added to the bolt preload for the handbook calculation is small (≈ 0.01) as indicated in Fig. 1. Consequently, the handbook calculated pressure stress on the bolts is low compared to the same stress calculated by the finite-element model. However, both of these methods of calculation result in bolt stresses that are much less than the value obtained from a strength-of-materials calculation (≈ 15 ksi.) that assumes only the pressure load is applied to the bolts.

CONCLUSIONS

The purpose of this paper was to describe the ASME Code requirements for the design and analysis of the lid-closure bolts for a Category I containment vessel and provide an example of a typical stress analysis. The example compares calculations based on handbook formulas with an analysis performed with a two-material, three-dimensional ANSYS finite-element model analysis.

The ASME Code requirements for a Category I containment vessel are based on the rules given in Section III, Subsection NB. These rules include acceptable materials, design criteria, and material qualification requirements. The Code design criteria contains stress limits allowed in the bolts that depend upon the mechanical loadings in service. These service loadings are identified in the Code as the Design Loadings, the Level A Service Limits, and the Level D Service Limits. The Design Loadings and Level A Service Limits correspond to the Normal Conditions of Transport (NCT) and the Level D Service Limits to the Hypothetical Accident Conditions (HAC).

A comparison of results of the different stress analyses for the NCT shows that the handbook analysis predicts bolt stresses that are about equal to those predicted by a finite-element model. The reason for this similarity of results is because the bolt preload torque is much greater than any of the other mechanical loads imposed on the bolts during the NCT. This preload is responsible for more than 97% of the axial load in the bolts. Consequently, when the internal pressure and differential thermal expansion is small relative to the initial bolt preload, differences between a simple analysis using handbook formulas and a finite-element analysis are insignificant.

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APPENDIX A – BOLT STRESS EQUATIONS

Axial Stress

1. Internal Pressure (with fraction of preload applied to pressure load – flange faces in contact)

$$S_{pa} = \frac{0.785\, p\, G^2}{A_b} \left[\frac{1}{1 + \frac{A_f E_f / t_f}{A_b E_b / L_b}} \right]$$

2. Differential Thermal Expansion

$$S_{ta} = \Delta T\, E_b (\alpha_f - \alpha_b)$$

3. Preload Torque

$$S_{qa} = \frac{T}{k\, d_n \dfrac{A_b}{n}}$$

4. Total Axial Stress

$$S_a = S_{pa} + S_{ta} + S_{qa}$$

Shear Stress

1. Preload Torque

$$S_{qv} = \frac{16\, T^*}{\pi\, d_r^3}$$

Maximum Stress Intensity

$$S_{13} = 2\sqrt{\left(\frac{S_a}{2}\right)^2 + S_{qv}^2}$$

Nomenclature

α_b Coefficient of thermal expansion, bolt material (coefficient 'B' from Ref. 6, Table TE, pp 638-649) [1/°F]

α_f Coefficient of thermal expansion, flange material (coefficient 'B' from Ref. 6, Table TE, pp 638-649) [1/°F]

A_b Total bolt area provided (value from any handbook table listing 'tensile stress area') [in^2]

A_f Flange contact area [in^2]

d_n Nominal diameter of bolt [in]

d_r Root diameter of bolt threads [in]

E_b Elastic modulus of bolt material at maximum temperature (from Ref. 6, Table TM, pp 664-667) [lb/in^2]

E_f Elastic modulus of flange material at maximum temperature (from Ref. 6, Table TM, pp 664-667) [lb/in^2]

G   Diameter to center of gasket, [in]

K   Bolt thread friction factor, use 0.15 for lubricated threads and nut face, 0.20 for non-lubricated case (Ref. 11).

L_b Effective length of bolt, [in]

n   Number of bolts

p   Internal pressure [lb/in^2]

t_f Thickness of flange [in]

T   Applied torque [in lb]

T^* Applied torque, reduced for losses due to thread and nut friction, $T^* = 0.50\ T$ [in lb]

ΔT Temperature difference between loading and maximum during transport [°F]

Table 1.   Allowable Stresses in Containment Vessel Bolting from the
ASME Code, Section III, Subsection NB, Paragraph NB-3230

Level/Code Ref. ¶	Loading	Stress Limit[a]
Design Conditions/ NB-3231	design pressure & gasket reactions	$S_a < S_m$
Level A Service Limits (Normal Conditions of Transport)/ NB-3232	pressure, preload, & thermal expansion. *average stress* *maximum stress* [b]	$S_a < 2\,S_m$ $S_{max} < 3\,S_m = S_y$
Level D Service Limits (Hypothetical Accident Conditions)/ NB-3235	accident *average stress* *maximum stress*[c] *shear stress* *combined stress*	$S_a <$ the smaller of $0.7\,S_u$ or S_y $S_b < S_u$ $S_v < 0.42\,S_u$ $(S_a/0.7\,S_u)^2 + (S_v/0.42\,S_u)^2 \le 1$

[a] The calculated axial, bending, and shear stresses are S_a, S_b, and S_v respectively. S_{max} is the maximum calculated stress intensity. Stress intensity is defined as twice the maximum shear stress and is equal to the largest algebraic difference between any two of the three principal stresses. S_y and S_u are the tensile yield and ultimate strength, and S_m is $1/3\,S_y$.

[b] From paragraph NB-3232.2 "Stress intensity, rather than maximum stress, shall be limited to this value when the bolts are tightened by methods other than heaters, stretchers, or other means which minimize residual torsion."

[c] For bolt materials with an ultimate tensile strength equal to or greater than 100 ksi at temperature.

Table 2.   Mechanical and Physical Properties of Materials

| | Temperature ($^\circ$F) | | |
	-40	RT	220
AMS 5726B (A-286/UNS S66286) Bolts			
Tensile strength, S_u (ksi)	213.0[b]	200.0[a]	182.2[b]
Yield strength, $S_y = 3S_m$ (ksi)	187.7[b]	180.0[a]	169.4[b]
Stress Intensity, S_m (ksi)[c]	62.6	60.0	56.5
$2S_m$ (ksi)	125.1	120.0	112.9
Elastic Modulus, E x10^3 (ksi)[d]	29.8	29.2	28.4
Cf. Therm. Exp, α x10^{-6} (in/in$^\circ$F)[e]	8.22	-	8.41
ASME SA-240 Type 316 Stainless Steel Flange and Vessel			
Tensile strength, S_u (ksi)[a]	75.0	75.0	74.7
Yield strength, S_y (ksi)[b]	30.0	30.0	25.3
Stress Intensity, S_m (ksi)[c]	20.0	20.0	20.0
Elastic Modulus, E x10^3 (ksi)[d]	28.9	28.3	27.5
Cf. Therm. Exp, α x10^{-6} (in/in$^\circ$F)[e]	8.46	-	8.81

[a] From AMS 5726B.

[b] Extrapolated from data in Ref. 15, p. 20.

[c] $S_m = (1/3)S_y$ from Ref. 6, Appendix 2, p. 755.

[d] From Ref. 6, Tbl. TM-1, Mtl. Gp. F, p. 664.

[e] From Ref. 6, Tbl. TE-1, Coef. A & B, 26 Ni-15Cr-2Ti Mtl., p. 640.

All data from Ref. 6.

    [a] Tbl. U, p. 490.

    [b] Tbl. Y-1, p. 562.

    [c] Tbl. 2A, p. 342.

    [d] Tbl. TM-1, Mtl. Gp. G, p. 664.

    [e] Tbl. TE-1, Coef. A @ -40 & B @ max temp., 18-8 Mtl., p. 640.

Table 3   Comparison of calculated and allowable bolt stresses for
Normal Conditions of Transport governed by the Level A
Service Limits in Section III, Subsection NB of the ASME Code

Case	Code Allowable Stress, ksi	Calculated Stress, ksi	
		Handbook Formulas	ANSYS Analysis
Preload, Pressure & Thermal Expansion			
Average Stress	112.9	111.3	111.7
Maximum Stress	169.4	137.6	137.9

Table 4   Comparison of bolt stresses for Normal Conditions of Transport
calculated by handbook formulas and finite-element analysis

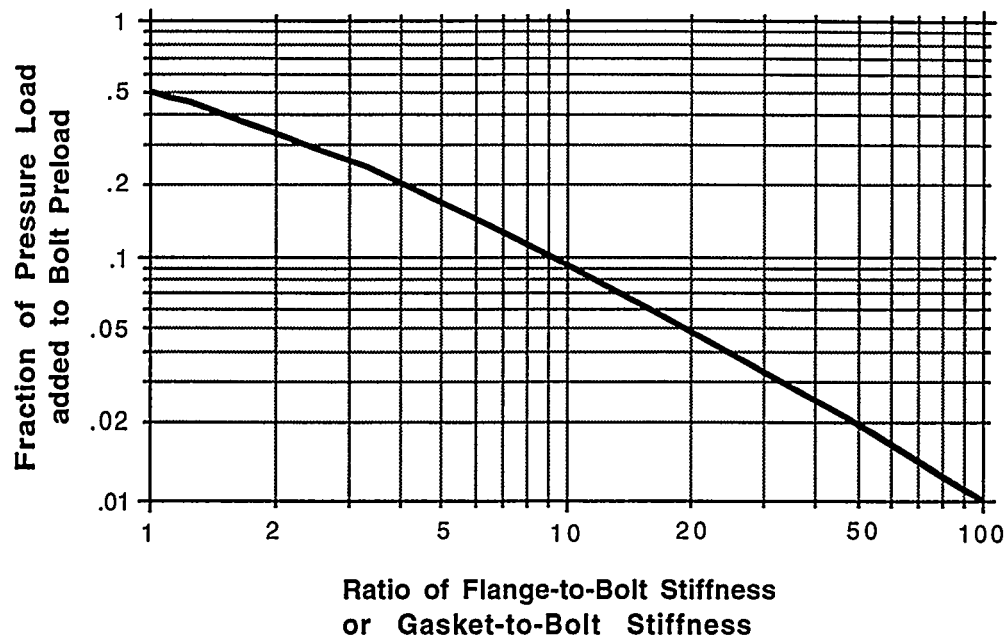| Loading	Calculated Stress, ksi	
	Handbook Formulas	ANSYS Analysis
Preload	109.4	108.2
Pressure	0.16	2.32
Thermal Expansion	1.70	1.20

Fig. 1 Relationship between stiffness of the flange or gasket and the closure bolts. As the ratio of flange-to-bolt or gasket-to-bolt stiffness increases, the fraction of the pressure load added to the bolt preload decreases.
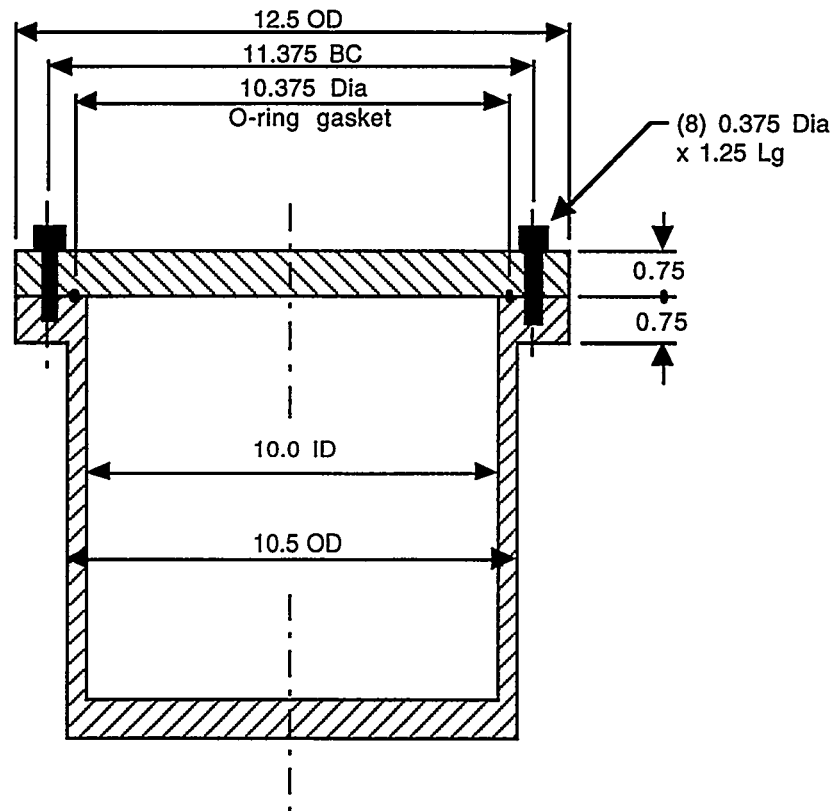

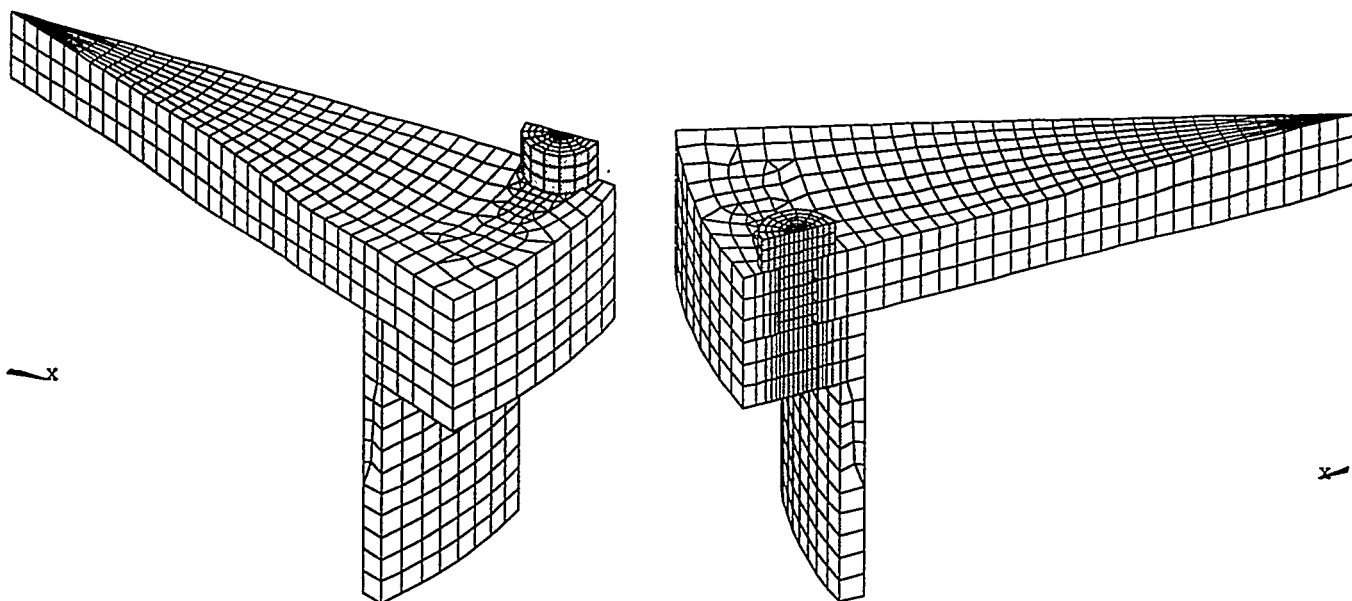Fig. 2 Cross-section of an idealized Category I containment vessel used as an example for calculating the stresses in the bolting.

Fig. 3. Two views of the finite element model.