Appendix A.2.13.4
MP197HB Structural Analysis of the Shield Shell

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

Appendix A.2.13.4
MP197HB Structural Analysis of the Shield Shell

A.2.13.4.1 Introduction

This section presents the structural analysis of the shield shell of the MP197HB Transport package. The shield shell consists of a cylindrical shell section and closure plates at each end which connect the shell to the cask body. The shell is evaluated for Normal Condition of Transport (NCT) and Hypothetical accident condition (HAC) using ANSYS for both side and end drop events. These stresses are compared to the allowable stress limits in Chapter A.2 to assure that the design criteria are met.

A.2.13.4.2 Description

The shield shell is constructed from low-alloy carbon steel and is welded to the shield surface of the gamma shield shell. The cylindrical shell section is 0.375 in. thick and the closure plates are 0.50 inch thick. Pertinent dimensions are shown in Figure A.2.13.4-1 and Drawings in Appendix A.1.4.10.

A.2.13.4.3 Materials Properties and Stress Criteria

The shield shell cylindrical section and closure plates are SA-516 Gr 70. Material properties and allowable stresses for normal (NCT) and accident (HAC) drop analyses are based on 300°F which bound -40°F, -20°F, and 100°F ambient conditions. The material properties are taken from the ASME Code [2] at a temperature of 300 °F and are listed below.

MP197HB Non-Containment Structure Material Properties

Temp (°F)	E (10^3 ksi)	S_m (ksi)	S_y (ksi)	S_u (ksi)	α_{AVG} (°F^{-1} $\times 10^6$)	ρ (lb/in.^{3})
300	26.7	22.4	33.6	70.0	6.9	0.284

The stress criteria are in according to ASME Code Section III, Subsection NF and Appendix F [1] and are listed in Table A.12.13.4-1.

A.2.13.4.4 Finite Element Model Description

ANSYS Model

The geometry used in the ANSYS [3] finite element analysis of the neutron shield shell is shown in Figure A.2.13.4-1 (0.3125" instead of 0.375" is used in the ANSYS model to simulate the 5/16" longitudinal weld).

Cask and neutron shield shells were modeled using ANSYS SOLID45 (8 nodes having 3 translational DOF) elements. Partial penetration welds were simulated using couplings (all degrees of freedom) at the interface of cask to top and bottom end plates of shield shell and at the interface of longitudinal plate of shield shell to its top and bottom end plates (see Figure A.2.13.4-2).

CONTACT52 elements were used at the interface of resin and neutron shield structure and between resin and cask interface. This CONTACT52 gap element introduces the nonlinearity in the analysis depending whether they are open or closed. Additionally COMBIN14 weak spring (10 lbs/in stiff, conservative) elements were modeled along with CONTACT52 interface, which is useful for preventing rigid body motion that could occur in the analysis.

Boundary Conditions

Symmetric boundary conditions were applied at the cut face of the model. For the end drop simulation, all bottom surface nodes of the casks were constrained in vertical direction, whereas for the side drop, cask nodes on the shield diameter, above top end plate and below bottom end plates, were constrained up to 30° in the hoop direction. Figures A.2.13.4-2, A.2.13.4-3, and A.2.13.4-4 show the overall finite element ASNSY model, and boundary conditions for both side and end drop.

Vyal "B" Resin Density Calculation

The Vyal B resin density (0.0650 lb/in^3) is adjusted to account for (aluminum boxes, bearing blocks, tie bars and other components) that are not captured in the finite element model.

A.2.13.4.5 Applied Loads

The neutron shield shell structure is analyzed for both side and end drops to bound all the possible maximum stress cases resulting from normal and accident events. For side drops acceleration due to gravity, g-loads were applied on the model in the hoop direction and for end drops g-loads, were applied in the axial direction.

An internal pressure of 25 psig is applied on all the inner walls of the neutron shield shell and also at the interfaces of the partial penetration welds in the ANSYS model.

During the slap down drops the maximum g loads are at the top and bottom ends of the shield shell while the g loads in the majority of the shield shell are below the side drop g loads. Furthermore, there are enough safety margins in the side drop analysis to cover the peak g loads at the ends of the shield shell. Therefore no explicit analysis are performed for HAC slap down drop/corner drop, slap down drop/corner drop are considered to be bounded by the end drop and the side drop evaluations. Load cases analyzed are listed below.

Summarized of Load Cases Analyzed

Loading	Service Level	Load [1]	Analysis Method
25g Side Drop	NCT	25g	Finite Element Elastic Analysis
25g End Drop	NCT	25g	Finite Element Elastic Analysis
55g Side Drop	HAC	55g	Finite Element Elastic Analysis
55g End Drop	HAC	55g	Finite Element Elastic Analysis

[1] g-loads are taken from Appendix A.2.13.12 "MP197HB Transport Package Impact Limiter Analysis Using LS-DYNA".

A.2.13.4.6 Analysis Results

A. Normal Condition Side and End Drops

The resulting stress intensity distribution and displacements on the neutron shield shell for 25 g side and end drop are shown in Figures A.2.13.4-5 through A.2.13.4-10. It is seen that the maximum nodal stress intensity in the structure is 17.12 ksi (Figure A.2.13.4-5) for side drop and 11.47 ksi (Figure A.2.13.4-8) for end drop. All the normal condition stresses are below the allowable stress values. See Table A.2.13.4-2 for stress comparison

B. Accident Condition Side and End Drop

Accident condition side and end drop stress intensity and displacement plots are shown in Figures A.2.13.4-11 through A.2.13.4-16. It is seen that the maximum nodal stress intensity in the structure is 37.37 ksi (Figure A.2.13.4-11) and 25.90 ksi (Figure A.2.13.4-14) for 55 g side drop and end drops respectively. All the accident condition stresses are below the allowable stress values. See Table A.2.13.4-2 for stress comparison.

C. Weld Stress Calculations

All partial penetration welds in ANSYS were represented by couplings. For partial fillet weld penetration, stresses are calculated as,

$F_w = F_{\text{resultant}} / (L_{\text{tributary}})\ (T_{\text{weld}})$,

where,

$F_{\text{resultant}}$ = maximum resultant nodal force = $(F_x^2 + F_y^2 + F_z^2)^{1/2}$

$L_{\text{tributary}}$ = minimum tributary length associated with the nodes = $\pi R / (n - 1)$, where $R = 42.25$ (cask radius) and n is the maximum nodes used at weld interface locations

T_{weld} = appropriate weld throat or base metal dimension: The effective throat thickness is taken as 0.4375" (7/16") for welds at the cask to neutron shield shell top and bottom end plates, where as welds throat thickness at the interface of longitudinal plate to top and bottom end plates of shield shell is taken as 0.3125 (5/16").

The weld stress results are listed in Table A.2.13.4-2.

D. Summary of Results

The critical stresses are summarized in Table A.2.13.4-2. Based on the results of the structural analysis, it is concluded that the Neutron Shield Shell structure is adequate for the specified loads to use with NUHOMS® MP197HB Transport Package.

A.2.13.4.7 References

1. ASME Boiler and Pressure Vessel Code, Section III, Division 1, Subsection NF and Appendices, 2004 with 2006 Addenda.
2. ASME Boiler and Pressure Vessel Code, Section II, Part D, 2004 with 2006 Addenda.
3. ANSYS Computer Code and Users Manual, Release 8.0.

Table A.2.13.4-1
MP197HB Transport Cask Non-Containment Structure/Weld Allowable Stress

Stress Category	Structure Allowable Stresses	
	Normal Conditions	**Accident Conditions**
Primary Membrane General P_m Local P_L	S_m $1.5\ S_m$	Lesser of $2.4S_m$ or $0.7\ S_u$ (1) Lesser of $3.6S_m$ or S_u (1)
Primary Membrane + Bending (P_m or P_L) + P_b	$1.5\ S_m$	Lesser of $3.6S_m$ or S_u (1)
Range of Primary + Secondary (P_m or P_L) + P_b +Q	$3.0\ S_m$	Not applicable
Bearing Stress	S_y	Not applicable
Average Shear Stress	$0.6\ S_m$	$0.42\ S_u$
Fatigue	Not Applicable	Not Applicable
Weld Allowable		
NCT	Full Penetration	Same as base metal
NCT	Partial Grove/Fillet	Tension - 0.3 x S_u Shear – 0.4 x S_y
HAC	Full Penetration	Same as base metal
HAC	Partial Grove/Fillet	Normal condition allowables are increased by a factor: Smaller of 2 or $1.167S_u/S_y$ if $S_u > 1.2S_y$, or 1.4 if $S_u \leq 1.2S_y$

Note:

1. Classification and stress limits are as defined in ASME Code, Section III, and Subsection NF. When evaluating the results from the nonlinear elastic plastic analysis for the accident conditions, the general primary membrane stress intensity, Pm, shall not exceed greater of $0.7\ S_u$ or $S_y + 1/3\ (S_u - S_y)$ and the maximum primary stress intensity at any location (P_L or $P_L + P_b$) shall not exceed $0.9\ S_u$. These limits are in accordance with Appendix F of Section III of the Code [1].

Table A.2.13.4-2
Maximum Stress Summary–Neutron Shield Shell Structure

Drop Orientation	Stress Category	Maximum Stress (ksi)	Allowable Stress (ksi)
25g Side Drop Elastic Analysis (NCT)	P_m	13.15	22.4
	$P_m + P_b$	17.13	33.6
Partial Groove/Fillet Welds	Cask and shell interface top	2.84	13.44
	Cask and shell interface bottom	2.78	13.44
	Longitudinal shell and top plate interface	3.12	13.44
	Longitudinal shell and bottom plate interface	3.09	13.44
	Longitudinal plate	10.4	13.44
25g End Drop Elastic Analysis (NCT)	P_m	8.51	22.4
	$P_m + P_b$	11.48	33.6
Partial Groove/Fillet Welds	Cask and shell interface top	0.60	13.44
	Cask and shell interface bottom	5.94	13.44
	Longitudinal shell and top plate interface	1.70	13.44
	Longitudinal shell and bottom plate interface	1.03	13.44
	Longitudinal plate	9.31	13.44
55g Side Drop Elastic Analysis (HAC)	P_m	28.70	49
	$P_m + P_b$	37.37	70
Partial Groove/Fillet Welds	Cask and shell interface top	6.21	26.88
	Cask and shell interface bottom	6.06	26.88
	Longitudinal shell and top plate interface	6.83	26.88
	Longitudinal shell and bottom plate interface	6.73	26.88
	Longitudinal plate	22.55	26.88
55g End Drop Elastic Analysis (HAC)	P_m	18.70	49
	$P_m + P_b$	25.86	70
Partial Groove/Fillet Welds	Cask and shell interface top	1.41	26.88
	Cask and shell interface bottom	12.80	26.88
	Longitudinal shell and top plate interface	3.60	26.88
	Longitudinal shell and bottom plate interface	2.11	26.88
	Longitudinal plate	21.26	26.88

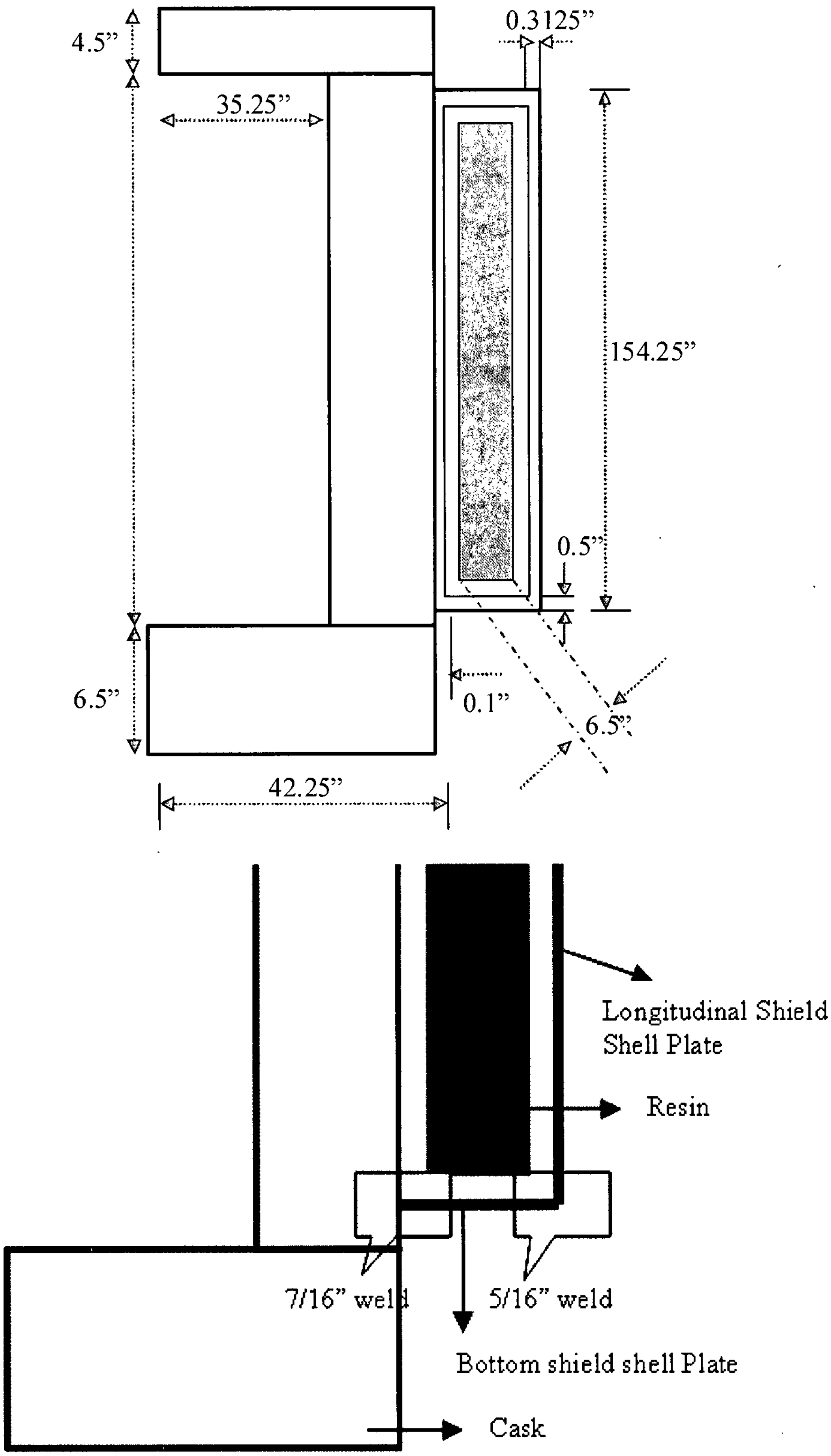

Figure A.2.13.4-1
Cask Shield Shell and Connection to Cask Body

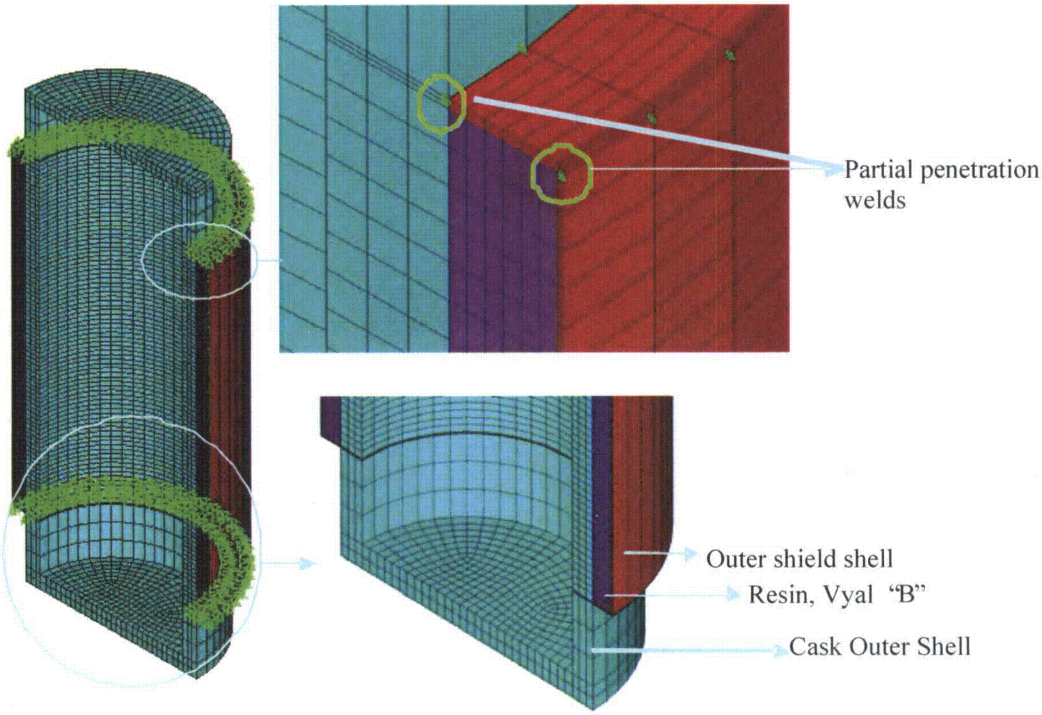

Figure A.2.13.4-2
Finite Element ANSYS Mesh for the Neutron Shield Shell

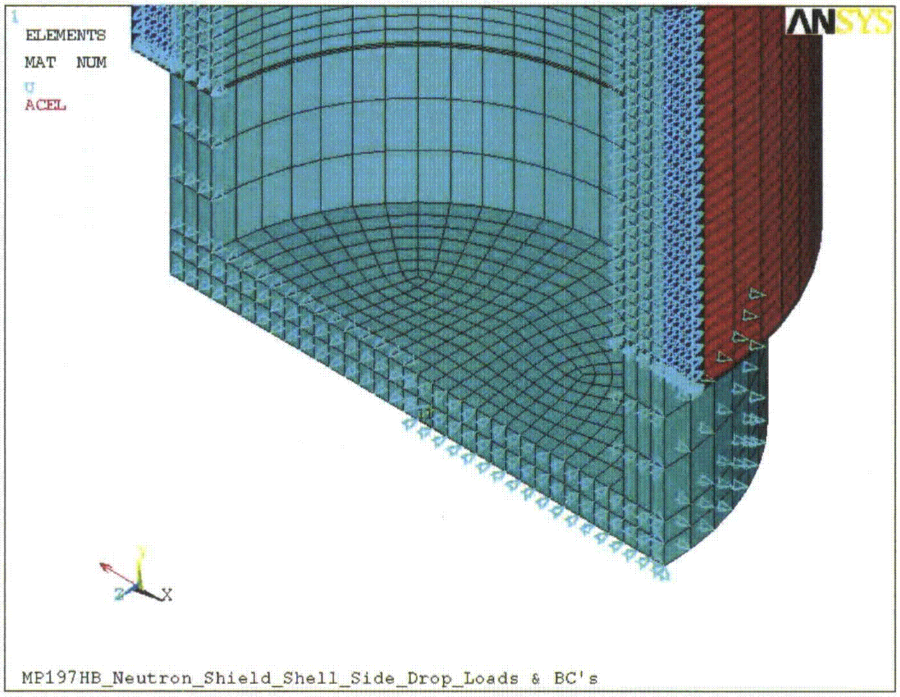

Figure A.2.13.4-3
MP197HB Neutron Shield Shell Loads and Boundary Condition

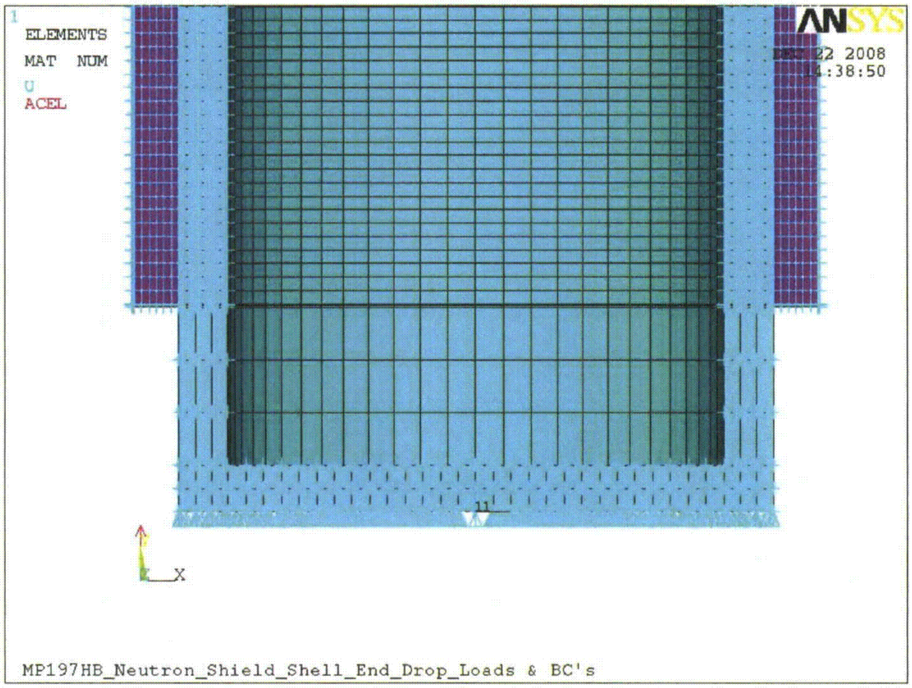

Figure A.2.13.4-4
Side and End Drop Boundary Conditions

Figure A.2.13.4-5
Normal Condition Side Drop 25 g Stress Intensity–Neutron Shield Shell

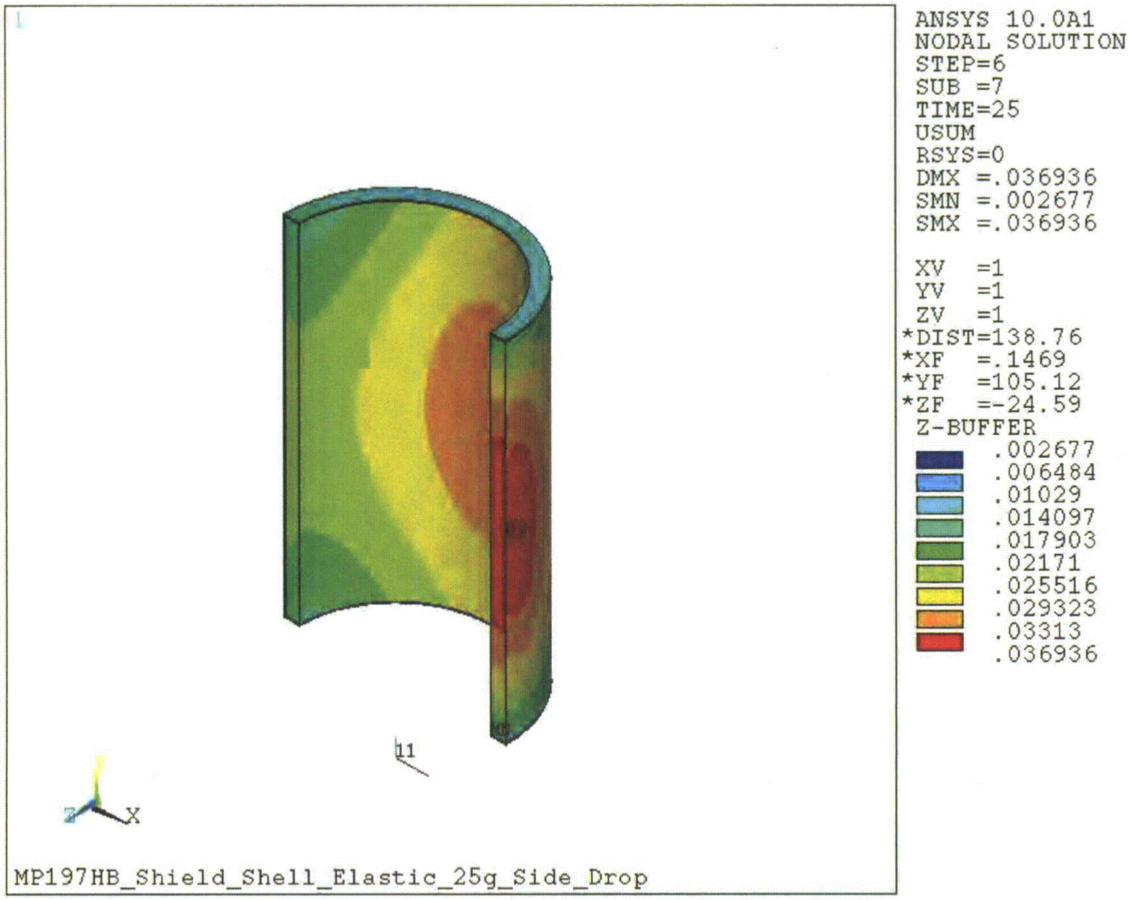

Figure A.2.13.4-6
Normal Condition Side Drop 25 g Maximum Displacement–Neutron Shield Shell

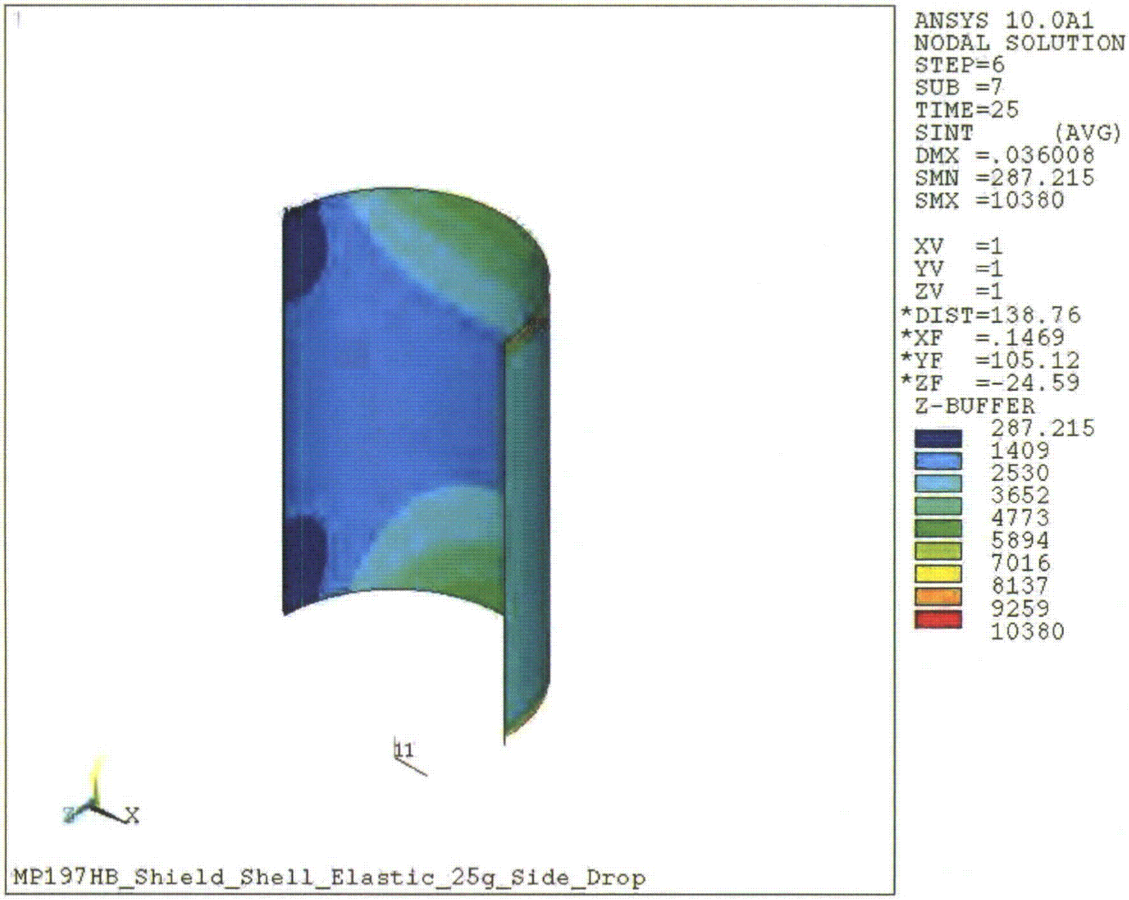

Figure A.2.13.4-7
Normal Condition Side Drop 25 g Stress Intensity–Longitudinal Plate

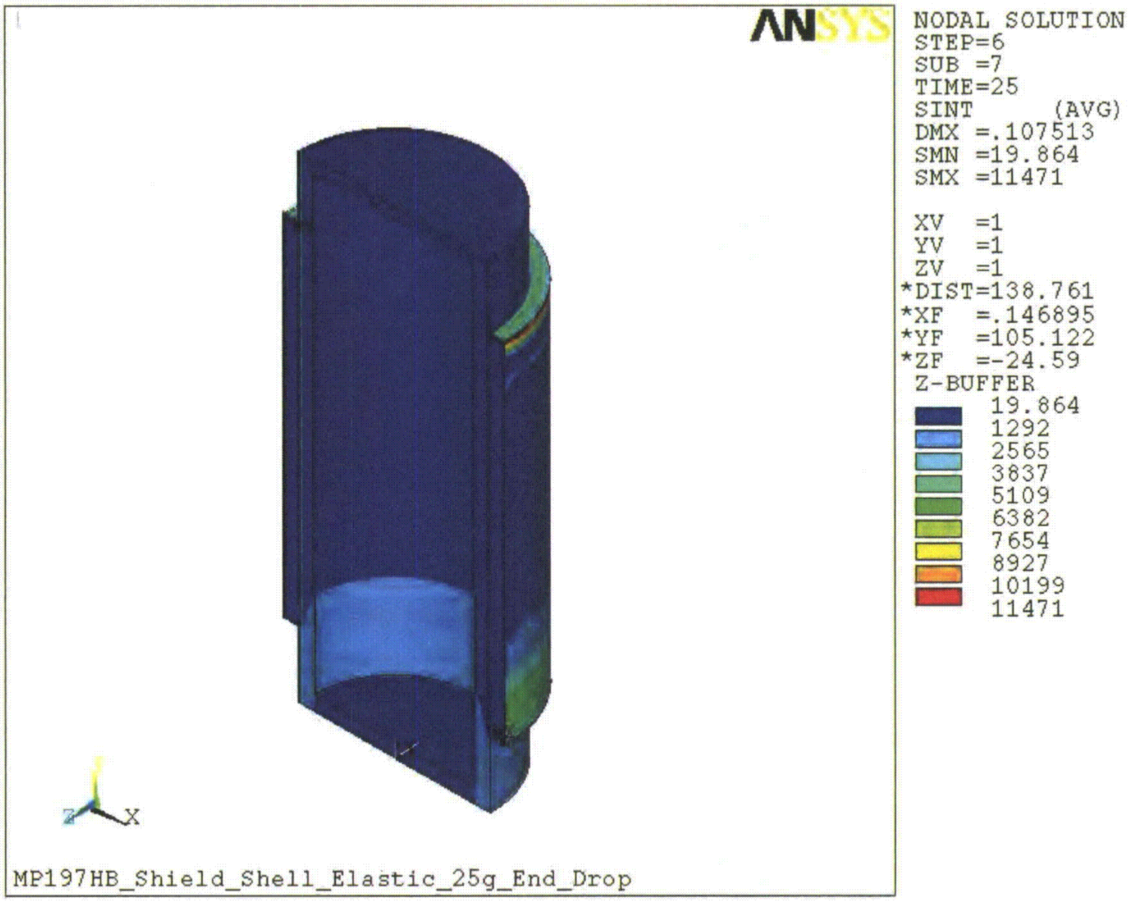

Figure A.2.13.4-8
Normal Condition End Drop 25 g Stress Intensity–Neutron Shield Shell

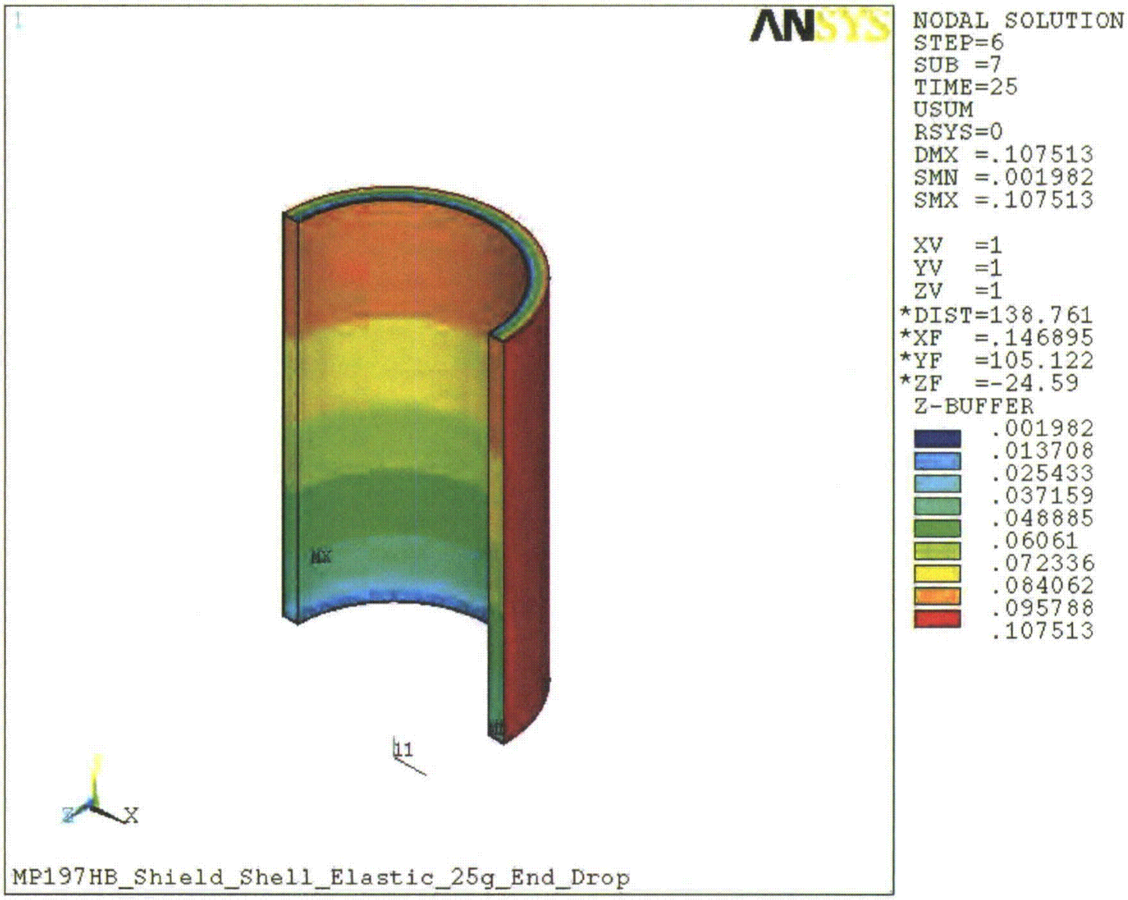

Figure A.2.13.4-9
Normal Condition End Drop 25 g Maximum Displacement–Neutron Shield Shell

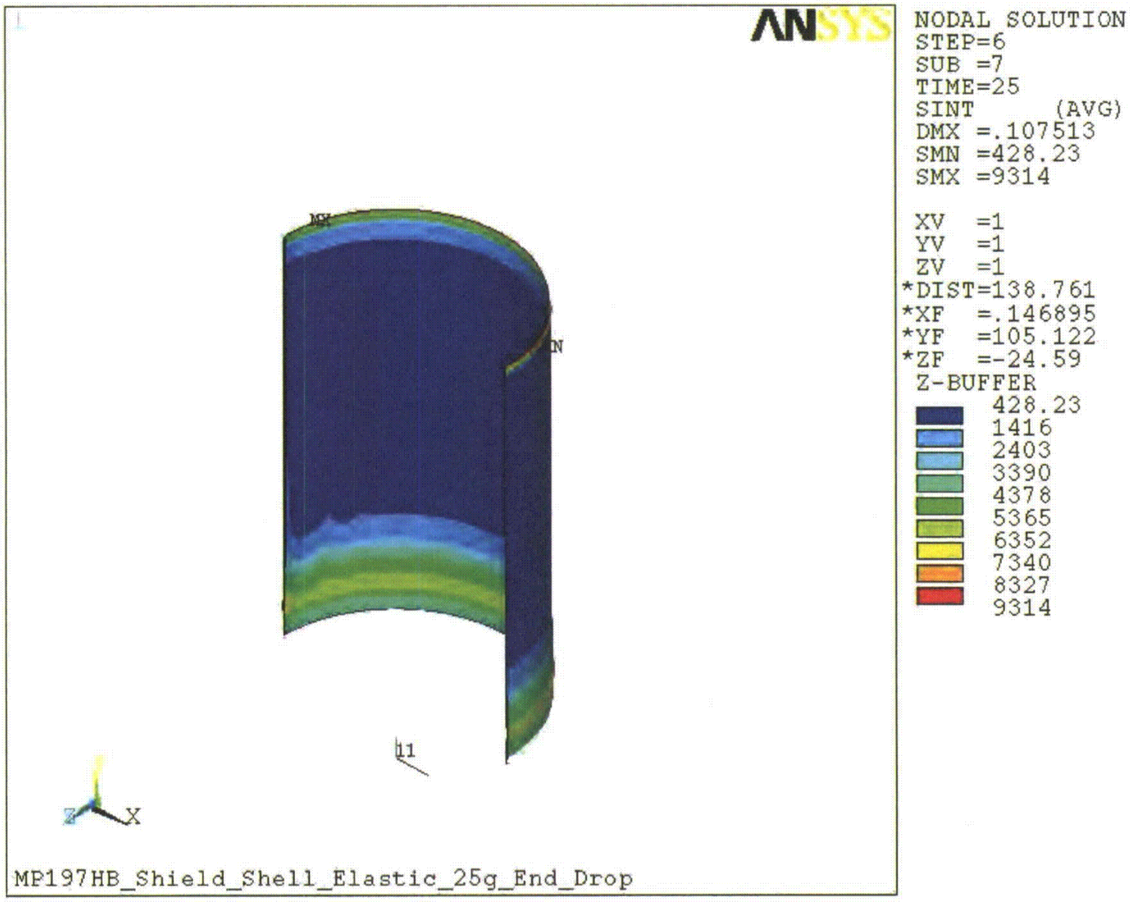

Figure A.2.13.4-10
Normal Condition End Drop 25 g Stress Intensity–Longitudinal Plate

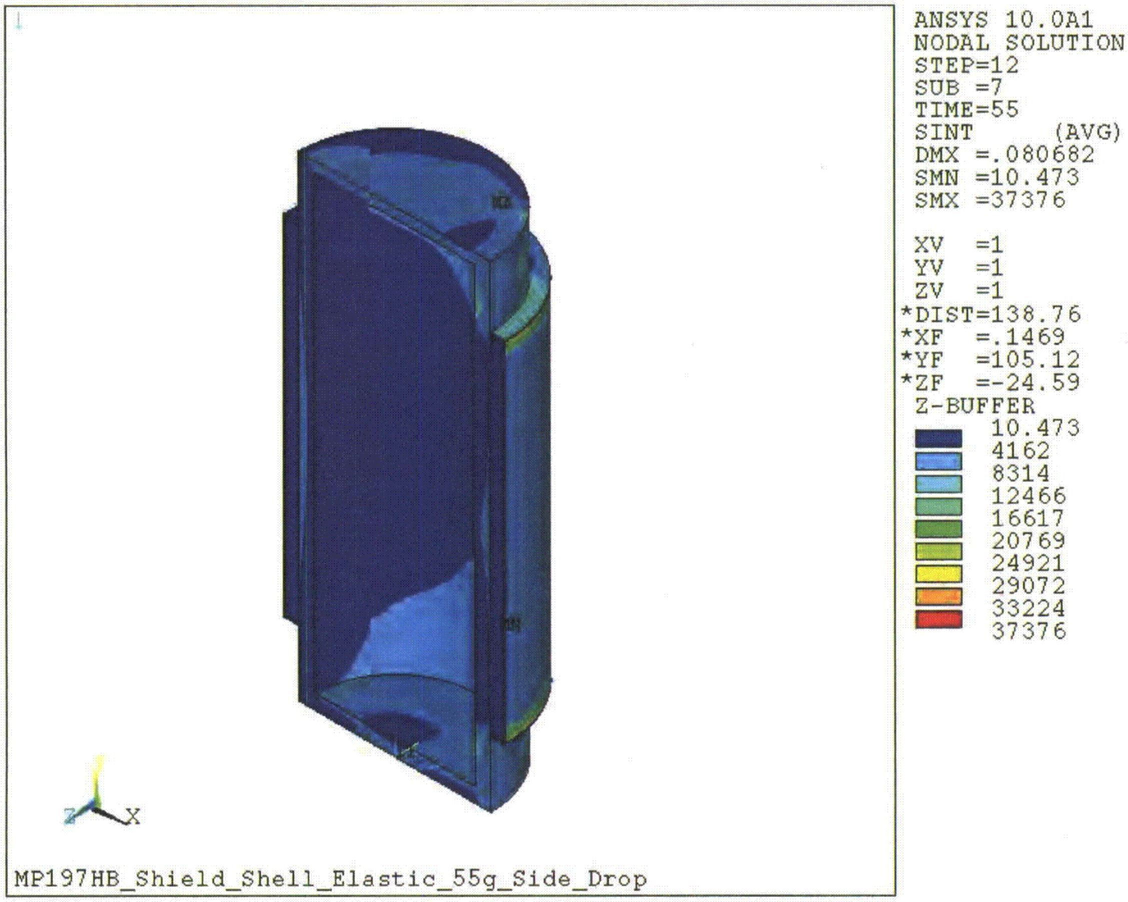

Figure A.2.13.4-11
Accident Condition Side Drop 55 g Stress Intensity–Neutron Shield Shell

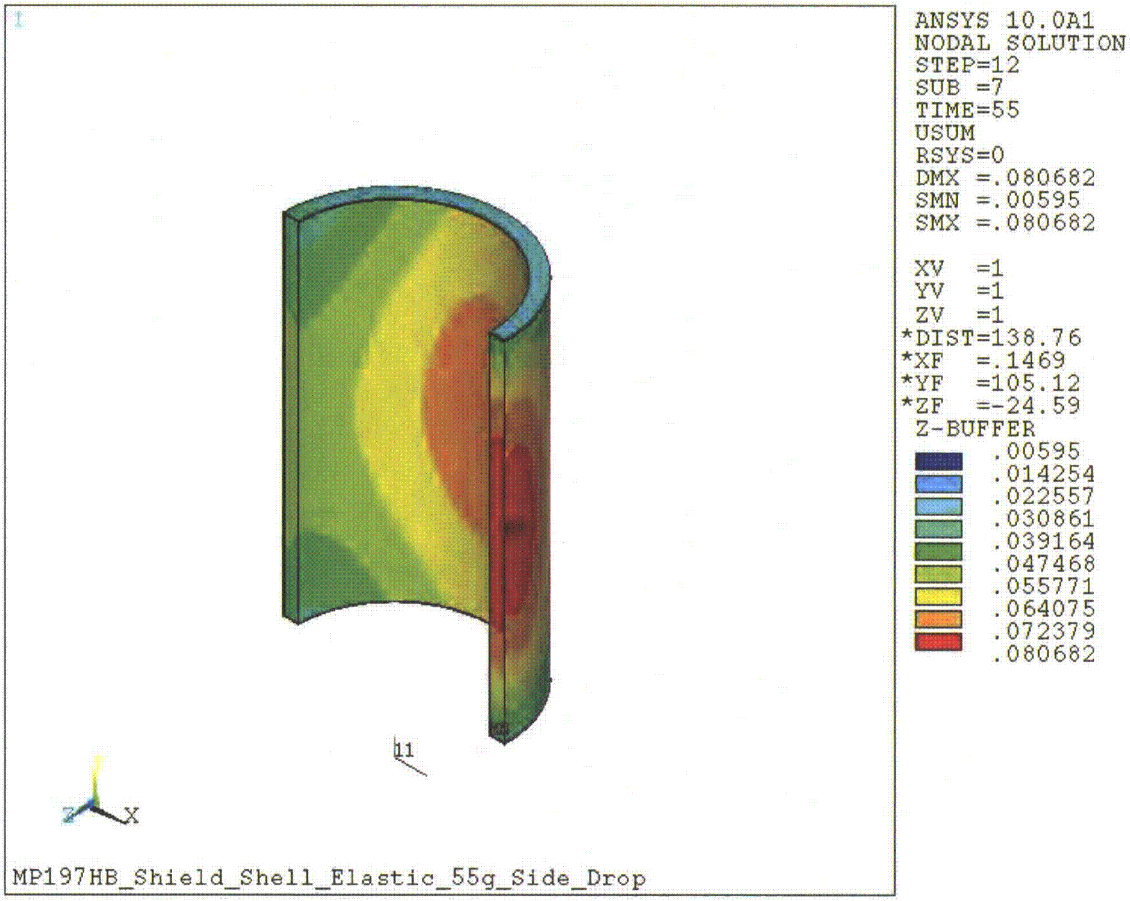

Figure A.2.13.4-12
Accident Condition Side Drop 55 g Maximum Displacement–Neutron Shield Shell

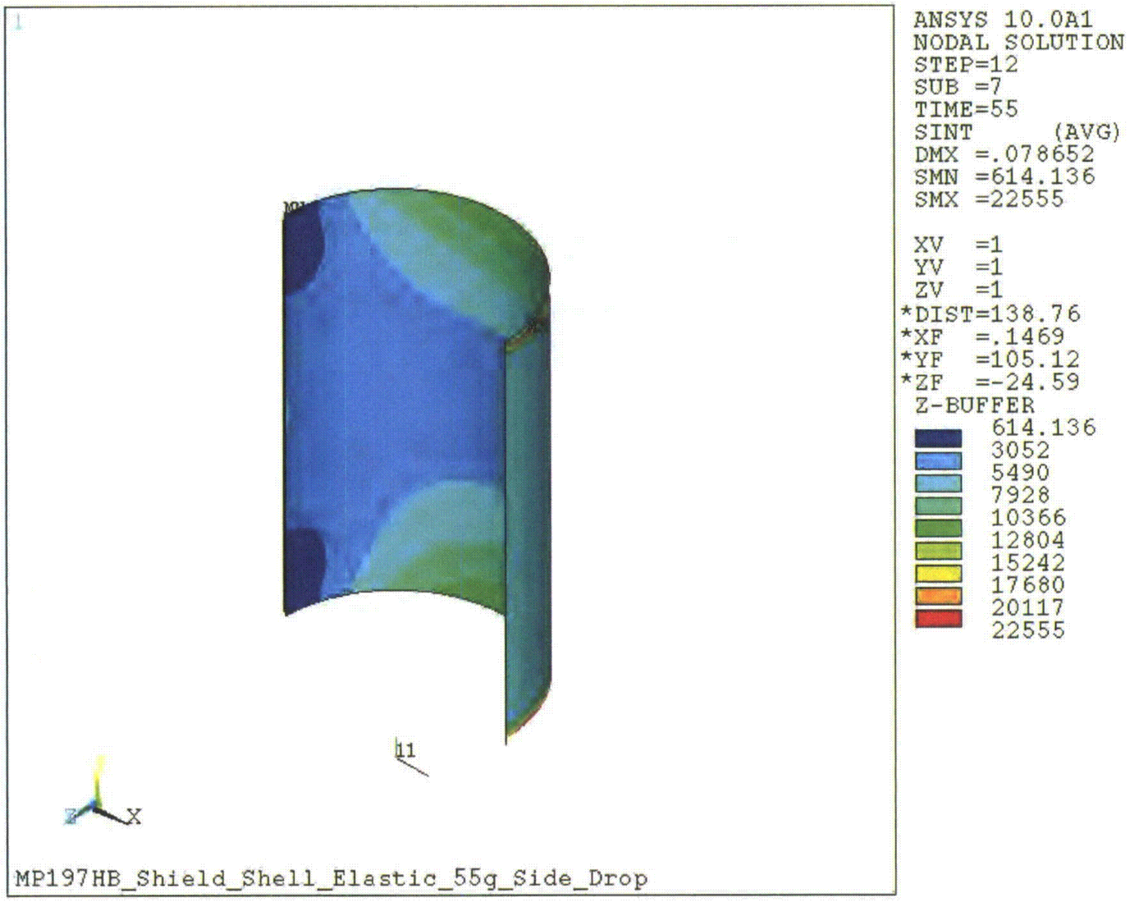

Figure A.2.13.4-13
Accident Condition Side Drop 55 g Stress Intensity–Longitudinal Plate

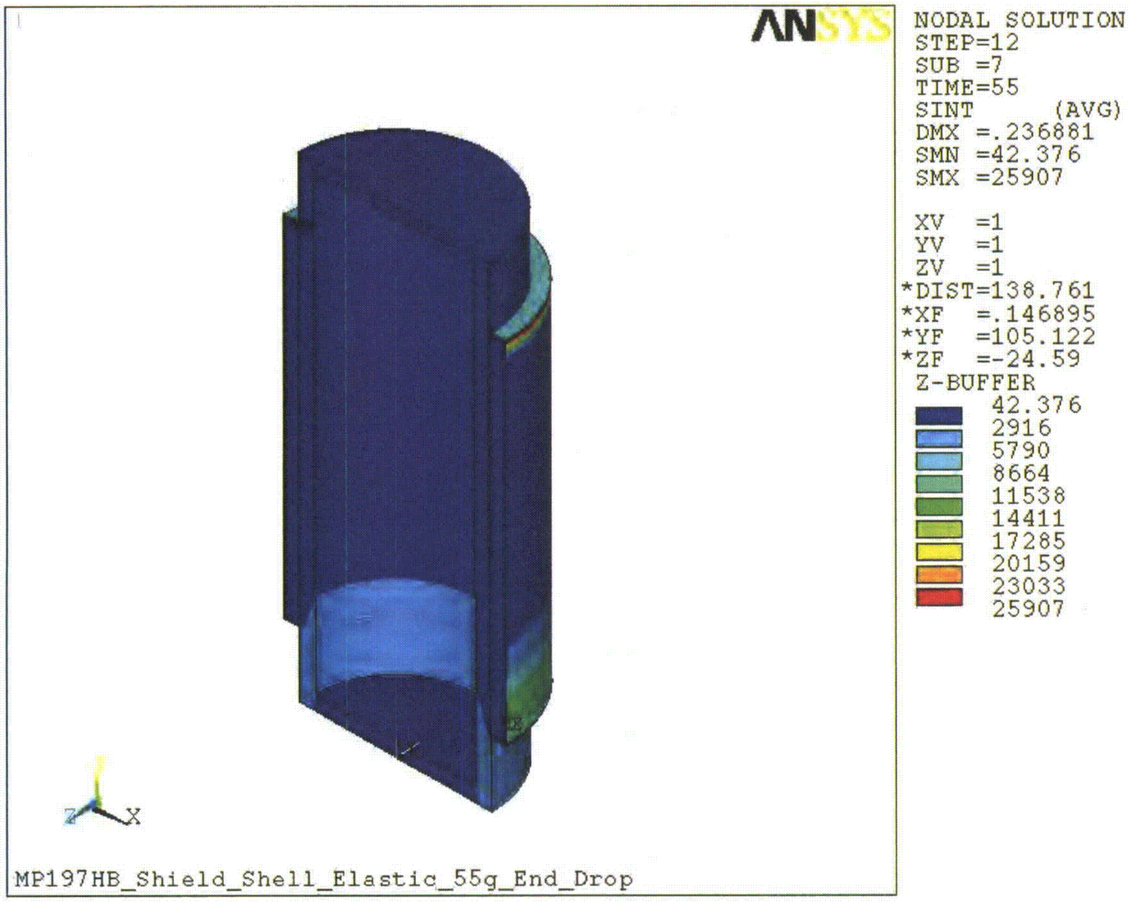

Figure A.2.13.4-14
Accident Condition End Drop 55 g Stress Intensity–Neutron Shield Shell

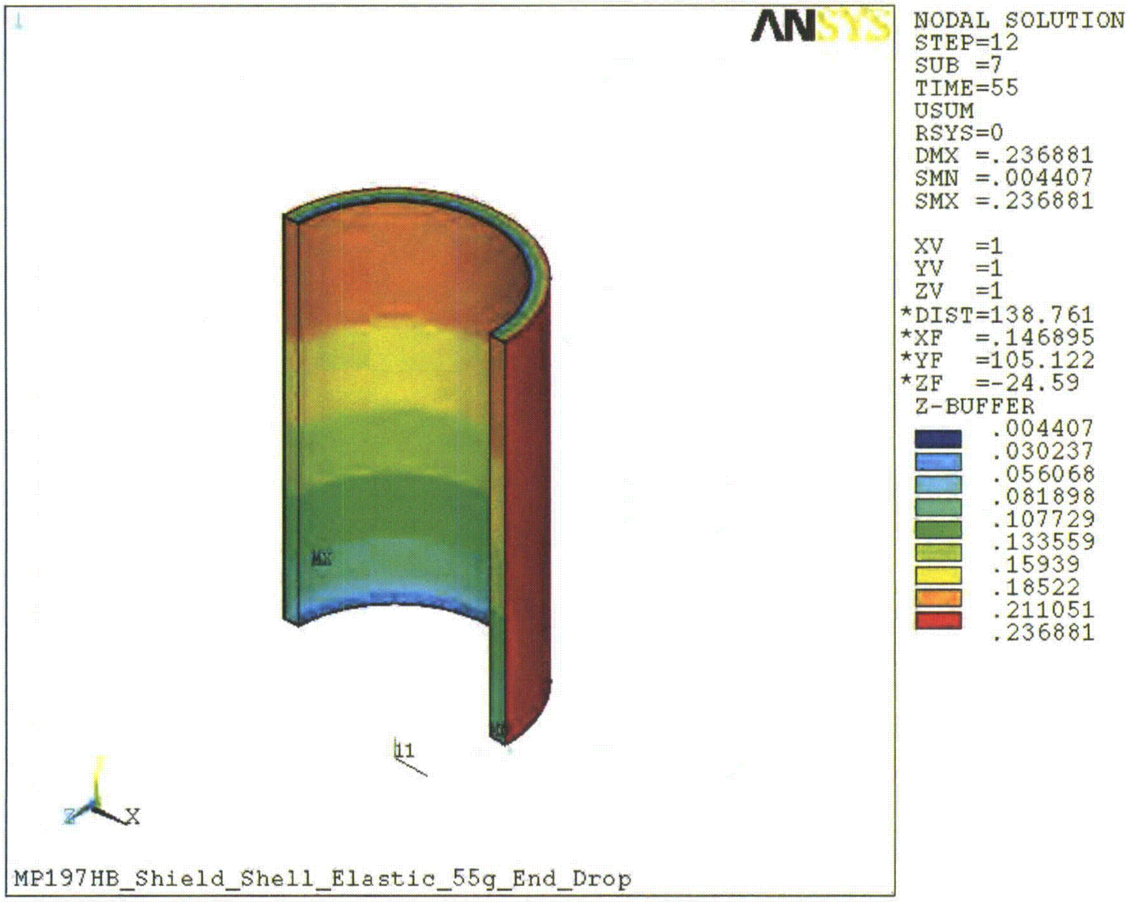

Figure A.2.13.4-15
Accident Condition End Drop 55 g Maximum Displacement–Neutron Shield Shell

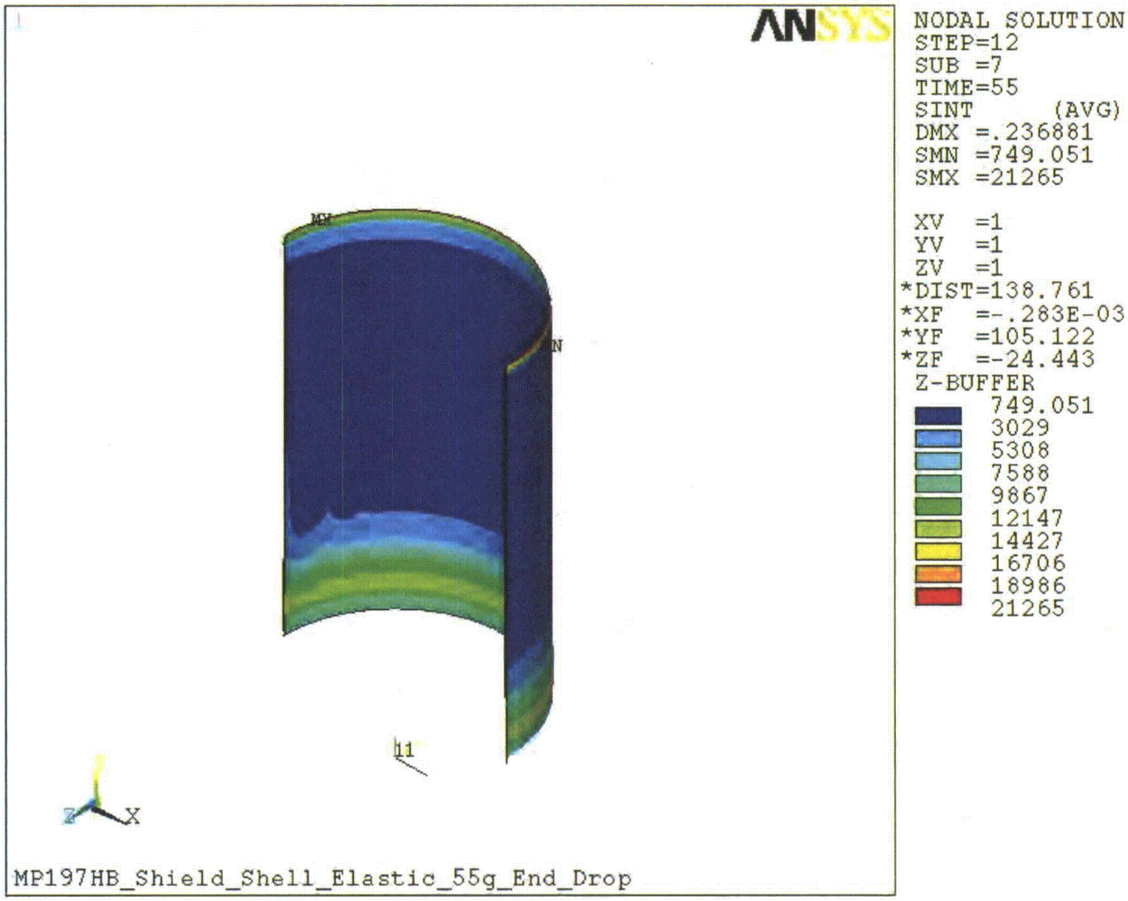

Figure A.2.13.4-16
Accident Condition End Drop 55 g Stress Intensity–Longitudinal Plate

Appendix A.2.13.5
MP197HB Cask Lifting and Tie-Down Devices Structural Evaluation

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

Appendix A.2.13.5
MP197HB Cask Lifting and Tie-Down Devices Structural Evaluation

NOTE: References in this Appendix are shown as [1], [2], etc. and refer to the reference list in Section A.2.13.5.5.

A.2.13.5.1 Purpose

A.2.13.5.1.1 Lifting Devices

The NUHOMS® - MP197HB transport cask is lifted by the upper two removable trunnions. The trunnion attachment blocks are welded to the cask structural shell and as such are considered a structural part of the package. The removable trunnions are evaluated to meet the requirements of 10CFR71.45 [4] and are designed and fabricated based on ANSI N14.6 [5].

10CFR71.45 (a) requires that a minimum factor of safety of three and five are needed against material yields and ultimate strengths, respectively, for all lifting attachments which are a structural part of the MP197HB transportation package for any lifting condition.

In addition, the package must be designed such that "failure of any lifting device under excessive load would not impair the ability of the package to meet the requirements" of 10CFR71 [4].

Section A.2.13.5.2 provides the analysis of the trunnions which are the only components used to lift the cask. Two sets of trunnions will be provided for the NUHOMS® - MP197HB transport package lifting. One set of trunnions has double shoulders (non single failure proof). The other set of trunnions has a single shoulder (single failure proof). Only one set of trunnions will be used depending on site and transfer operation requirements. Appendix A.2.13.1 provides an analysis of the global stresses in the cask walls due to the effects of the lifting loads on the trunnions. The global stress intensities from the ANSYS run at the stress reporting locations of the containment vessel and outer shell are presented in Table A.2.13.1-7.

The local stress intensities in the cask walls due to the 3 g (double shoulder trunnion) and 6 g (single shoulder trunnion) lifting loads are calculated below and presented in Tables A.2.13.5-5 and A.2.13.5-7. The maximum combined stress intensity for 3 g lifting is 20.6 ksi. The maximum combined stress intensity for 6 g lifting is 26.0 ksi. These stresses are less than the allowable stresses of the outer shell material (see Table A.2.13.5-1 for yield and ultimate stresses). Therefore, the requirements of 10CFR71.45 (a) are met. The stress analysis of the front trunnion and trunnion flange bolts are provided in the following sections.

A.2.13.5.1.2 Tie-Down Devices

10CFR71.45 (b) (1) requires that a system of tie-down devices that is a structural part of the package must be capable of withstanding, without generating stress in any material of the package in excess of its yield strength, a static force applied to the center of gravity of the package having a horizontal component along the direction in which the vehicle travels of 10 times the weight of the package with its contents.

The shear key bearing block and pad plate are parts of the cask structure designed to resist the 10 g longitudinal transportation load. The bearing block is a welded structure. The 36” × 37.20” × 1.5” pad plate is used to spread the longitudinal shear load over a large area of the cask structural shell to which it is welded, thus preventing the cask outer shell to be subjected to any bending moment resulting from the longitudinal load.

A.2.13.5.2 Trunnions Analysis

A.2.13.5.2.1 Double Shoulder Trunnions

Lifting Load

The NUHOMS® - MP197HB transport cask is lifted from the fuel pool vertically by its upper two removable trunnions using the fuel building crane. The weight of the cask used for trunnion structural evaluation is 290,000 lbs, this weight bounds the weight specified in Chapter A.2, Section A.2.1.3 of 278,600 lbs. During vertical lifting the impact limiters are not attached to the cask.

The maximum weight of the cask is W_L = 290,000 lb for the vertical lift from the fuel pool, distributed evenly between the two upper trunnions. Using a dynamic load factor of 1.1 and a lifting load of 3 g, the vertical design load (yield) for one trunnion is:

$$F_V = W_L \times DLF \times \frac{a_V}{N_{tr}} = 290{,}000 \times 1.1 \times \frac{3}{2} = 478{,}500 \text{ lb}.$$

A section of a double shoulder trunnion is shown on Figure A.2.13.5-1.

Trunnion Stresses

The stresses in various sections of the trunnion are shown in Figure A.2.13.5-2. The material properties are shown in Table A.2.13.5-1. The design parameters are shown in Tables A.2.13.5-2 and A.2.13.5-3. The stress results are shown in Table A.2.13.5-4.

Trunnion Bolt Evaluation

Load Due to Trunnion Moment

The trunnions are attached to the cask using 12 1 ¼”-7UNC bolts. The bolts are in tension because of the moment on the trunnion flange. The shear load is supported by the tight-fitting trunnion flange shoulder and a recess in the trunnion attachment block welded to the cask body. The radial clearance between the screw heads and shanks and trunnion flange holes is large enough so that the shear load is supported by the trunnion flange shoulder-to-block recess interface by bearing and not by the bolts.

The bending length is equal to:

$$L_1 + L_2 + Th_{flange} - Th_{tool} = 4.38 + 4.56 + 3.25 - 1.0 = 11.19 \text{ in.}$$

Therefore, the bending moment M_{D_D} is equal to F_V × (bending length), which is equal to:

$$M_{D\_D} = 478{,}500 \times 11.19 = 5{,}354{,}415 \text{ in.lb.}$$

According to [6], case 3, for bolt patterns symmetrical about the vertical axis and flange rotating about the bottom bolt, the maximum bolt force F_m due to the bending moment M_{D_D} is:

$$F_m = \frac{4}{3 \times D_{bolt}/2 \times N_b} M_{D\_D} = \frac{8 \times 5{,}354{,}415}{3 \times 21 \times 12} = 56{,}660 \text{ lb}.$$

Thermal Load

From [9], Table 4.4, the bolt force due to differential thermal expansion is calculated as follows:

$$F_{th} = 0.25 \times \pi \times D_b^2 \times E_b \times (\alpha_t \times \Delta T_t - \alpha_b \times \Delta T_b).$$

Where $\Delta T_t = \Delta T_b$ = Temperature Change = 300 – 70 =230°F.

Therefore,

$$F_{th} = 0.25 \times \pi \times 1.25^2 \times 26{,}700{,}000 \times 230 \times 10^{-6} \times (9.2 - 6.9) = 17{,}333 \text{ lb}.$$

Bolt Stresses

For a lifting load of 3 g, the total bolt force is equal to:

$$F_{max} = F_{th} + F_m = 17{,}333 \text{ lb} + 56{,}660 \text{ lb} = 73{,}993 \text{ lb.}$$

The maximum tensile stress σ_{max} in a bolt is:

$$\sigma_{max} = \frac{F_{max}}{S_{bolt}} = \frac{73{,}993}{0.969} = 76{,}360 \text{ psi}.$$

For a lifting load of 5 g, the total bolt force is equal to:

$$F_{max} = F_m + (5/3) \text{ x } F_t = 17{,}333 \text{ lb} + (5/3) \text{ x } 56{,}660 \text{ lb} = 111{,}766 \text{ lb.}$$

The maximum tensile stress σ_{max} in a bolt is:

$$\sigma_{max} = \frac{F_{max}}{S_{bolt}} = \frac{111{,}766}{0.969} = 115{,}342 \text{ psi}.$$

Minimum Engagement Length

The minimum engagement length L_e for the bolt and flange is (see Ref. [3], page 1490):

$$L_e = \frac{2 \times S_{bolt}}{3.1416 \times K_{n\max} \times \left[\frac{1}{2} + .57735 \times n \times \left(E_{s,\min} - K_{n\max}\right)\right]}.$$

$$L_e = \frac{2 \times 0.969}{3.1416 \times 1.1230 \times \left[\frac{1}{2} + .57735 \times 7 \times (1.1439 - 1.1230)\right]}.$$

$$L_e = 0.940 \text{ in}.$$

According to [3], page 1490:

$$J = \frac{A_s \times S_{ue}}{A_n \times S_{ui}}.$$

S_{ue} is the tensile strength of external thread material, equal to 165 ksi, and S_{ui} is the tensile strength of internal thread material, equal to 70 ksi.

A_s is the shear area of external threads:

$$A_s = 3.1416 \times n \times L_e \times K_{n\max} \times \left[\frac{1}{2n} + 0.57735 \times \left(E_{s\min} - K_{n\max}\right)\right].$$

A_n is the shear area of internal threads:

$$A_n = 3.1416 \times n \times L_e \times D_{s\min} \times \left[\frac{1}{2n} + 0.57735 \times \left(D_{s\min} - E_{n\max}\right)\right].$$

Therefore:

$$A_s = 3.1416 \times 7 \times 0.940 \times 1.1230 \times \left[\frac{1}{2 \times 7} + 0.57735 \times (1.1439 - 1.1230)\right].$$

$$A_s = 1.938 \text{ in}^2.$$

$$A_n = 3.1416 \times 7 \times 0.940 \times 1.2232 \times \left[\frac{1}{2 \times 7} + 0.57735 \times (1.2232 - 1.716)\right].$$

$$A_n = 2.559 \text{ in}^2.$$

So:

$$J = \frac{1.938 \times 165}{2.559 \times 70} = 1.785.$$

Therefore, the minimum required engagement length Q = J × L_e = 1.785 × 0.940 = 1.678 in.

Threaded inserts 1185-20CN-2500 are used. Their maximum length is 2.50 in., and they are used with bolts of maximum threaded length 2.5 in. = 3.5 in. (total bolt shank length) -3.25 in. (trunnion flange thickness) + 2.25 in. (counter bore depth). Both helicoils and bolts are cut to fit as necessary, no more than 0.25 in. below the minimum depth of the bolt holes, which is 1.93 in.

Therefore, the minimum threaded length is equal to:

$$1.93 \text{ in} - 0.25 = 1.68 \text{ in},$$

which is greater than the minimum required engagement length Q.

Trunnion Flange Stresses

The trunnion flange is shown in Figure A.2.13.5-3.

Stresses at section AA:
Length L_{f1} = 0.5 × (D_{bolt} - D_{flange}) = 10.5 – 8.5 = 2.0 in.

Length L_{f2} = 0.5 × [D_{bolt} × cos(30°) - D_{flange}] = 10.5 × cos(30°) – 8.5 = 0.593 in.

Flange length: $L = \sqrt{\left(D_{\max}^2 - D_{flange}^2\right)} = 2 \times \sqrt{\left(12.5^2 - 8.5^2\right)} = 18.33 \text{ in}.$

Flange thickness at AA: Th_{flange} = 3.25 in.

Maximum bolt load due to 3 g is F_{max}.

It is conservatively assumed that the two bolts on either side of the vertical axis support the same load F_{max}.

Bending moment at AA:

$$M = F_m \times L_{f1} + 2 \times F_m \times L_{f2} = 56{,}660 \times (2 + 2 \times 0.593) = 180{,}518 \text{ in.lb}.$$

The modulus of section at AA is equal to:

$$Z = \frac{L \times Th_{Flange}^2}{6} = \frac{18.33 \times 3.25^2}{6} = 32.27 \text{ in}^3.$$

The bending stress is equal to:

$$\frac{M}{Z} = \frac{180{,}518}{32.27} = 5{,}593\ \text{psi}.$$

The shear stress is equal to:

$$\frac{3 \times F_m}{L \times Th_{flange}} = \frac{3 \times 56{,}660}{18.33 \times 3.25} = 2{,}853\ \text{psi}.$$

The maximum stress intensity is equal to:

$$\sqrt{5{,}593^2 + 4 \times 2{,}853^2} = 7{,}989\ \text{psi}.$$

Trunnion Attachment Block and Cask Shell Weld

There is a 1.75" groove weld on the outer circumference of the attachment block and the cask shell (Ø D_{w_ext} = 27 in). On the inside of the attachment block, there is a 1.25" groove weld (Ø D_{w_int} = 17 in). The weld is subjected to a bending moment.

The outer diameter of the cask is 97.75 in. Since the inner radius of the cask is 42.25 in and the rear trunnion centerline is 1.18 in from the cask centerline, the maximum height of the block at a distance 13.5 in from its centerline is:

$$H_{\max} = \sqrt{\left(\frac{97.75}{2}\right)^2 - (1.18 + 13.5)^2} - \sqrt{42.25^2 - (1.18 + 13.5)^2} = 7.00\ \text{in}.$$

The minimum height of the block is:

$$H_{\min} = \sqrt{\left(\frac{97.75}{2}\right)^2 - (1.18 + 13.5)^2} - \sqrt{42.25^2 - \left(1.18 + \frac{17.04}{2}\right)^2} = 5.50\ \text{in}.$$

The average height H_{avg} is therefore 0.5 × (7.00 + 5.50) = 6.25 in.

The bending length is equal to:

$$L_1 + L_2 - Th_{tool} + H_{avg} + Th_{flange} - 2.79 = 14.65\ \text{in}.$$

Therefore, the weld bending moment is equal to F_V × (bending length), which is equal to:

$$M_w = 478{,}500 \times 14.65 = 7{,}010{,}025\ \text{in.lb}.$$

The footprint of the weld is conservatively assumed to be circular for calculating moment of inertia of weld metal.

The weld moment of inertia is:

$$I_{weld} = \frac{\pi}{64}\left[\left(D_{w\_ext} + 2\times Th_{w\_ext}\right)^4 - D_{w\_ext}^4 + \left(D_{w\_int} + 2\times Th_{w\_int}\right)^4 - D_{w\_int}^4\right]$$

$$I_{weld} = \frac{\pi}{64}\left[(27+2\times1.75)^4 - 27^4 + (17+2\times1.25)^4 - 17^4\right].$$

$$I_{weld} = 19{,}389\,\text{in}^4.$$

The bending stress σ_b is:

$$\sigma_b = \frac{M_w \times 0.5\times\left(D_{w\_ext} + 2Th_{w\_ext}\right)}{I_{weld}} = \frac{7{,}010{,}025\times0.5\times(27+2\times1.75)}{19{,}389}.$$

$$\sigma_b = 5{,}513\,\text{psi}.$$

Bolt Torque

The bolt torque is calculated so that the preload bolt tensile stress is equal to the maximum applied stress σ_{max} (76,360 psi), and there is lubrication on the threads.

That stress is induced by F_{max} = 73,993 lb.

The maximum torque required for this preload is $Q = K \times D_b \times F_{max}$.

$Q = 0.135 \times 1.25 \times 73{,}993 = 12{,}486$ in.lb = 1,040 ft.lb.

Local Stresses in Cask Outer Shell at Trunnion Attachment Block

Local stresses are calculated using the methodology [7] assuming a rectangular attachment of circumferential side length $2 \times c_1$ and longitudinal side length $2 \times c_2$.

The trunnion shear loads in the longitudinal and circumferential directions are respectively $V_L = F_V$ = 478,500 lb and V_C = 0 lb.

The external overturning moments supported by the intersection in the longitudinal and circumferential directions with respect to the shell are respectively $M_L = M_w$ = 7,010,025 in.lb and M_C = 0 in.lb.

The thickness of the outer shell is Th_{os} = 2.75 in. The cylinder mean radius is:

$$R_m = R_{weld} - 0.5 \times Th_{os} = 40.88\text{ in.}$$

The block circumferential side length is equal to $2 \times c_1$ = 27.00 in. Its equivalent longitudinal length $2 \times c_2$ is calculated based on its foot-print length L since the block shape is not fully rectangular:

$$L = \pi \times 13.5 + 17 \times 2 + 27 = 4 \times (c_1 + c_2) = 103.4 \text{ in}.$$

Therefore:

$$c_2 = \frac{\pi \times 13.5 + 17 \times 2 + 27}{4} - c_1 = 12.35 \text{ in}.$$

The geometric parameters are:

$$\gamma = \frac{R_m}{Th_{os}} = \frac{40.88}{2.75} = 14.9, \; \beta_1 = \frac{c_1}{R_m} = \frac{13.50}{40.88} = 0.33 \text{ and } \beta_2 = \frac{c_2}{R_m} = \frac{12.35}{40.88} = 0.30.$$

For a rectangular attachment subject to a longitudinal moment, the parameter β has different values:

- When considering membrane forces N_i: $\beta = \sqrt[3]{\beta_1 \times \beta_2^2} = 0.31$. The value found in figure 3B or 4B of [7] has to be multiplied by C_L from Table 8 of [7].
- When considering bending moments M_i: $\beta = K_L \times \sqrt[3]{\beta_1 \times \beta_2^2} = K_L \times 0.31$. The modified β is obtained by multiplying the original β by K_L from Table 8 of [7].

The above quantities are used in spreadsheet in Table A.2.13.5-5 to calculate the stresses in the outer shell of the cask.

The maximum stress intensity is 20,639 psi.

A.2.13.5.2.2 Single Shoulder Trunnions

Lifting Load

As discussed before, the maximum weight of the cask is W_L = 290,000 lb for the vertical lift from the fuel pool, distributed evenly between the two upper trunnions. Using a dynamic load factor of 1.1 and a lifting load of 6 g, the vertical design load (yield) for one trunnion is:

$$F_V = W_L \times DLF \times \frac{a_V}{N_{tr}} = 290{,}000 \times 1.1 \times \frac{6}{2} = 957{,}000 \text{ lb}.$$

A section of a trunnion is shown on Figure A.2.13.5-4.

Trunnion Stresses

The stresses in various sections of the trunnion shown in Figure A.2.13.5-5 are calculated in Table A.2.13.5-6.

Trunnion Bolt Stress Evaluation

Load Due to Trunnion Moment

The trunnions are attached to the cask using 12 1 ¼"-7UNC bolts. The bolts are in tension because of the moment on the trunnion flange. The shear load is supported by the tight-fitting trunnion flange shoulder and a recess in the trunnion attachment block welded to the cask body. The radial clearance between the screw heads (and shank) and trunnion flange holes is large enough so that the shear load is supported by the trunnion flange shoulder-to-block recess interface by bearing and not by the bolts.

The bending length is equal to:

$$L_3 + Th_{flange} - Th_{tool} = 4.00 + 3.25 - 1.0 = 6.25 \text{ in.}$$

Therefore, the bending moment M_{B_B} is equal to $F_V \times$ (bending length), which is equal to:

$$M_{B\_B} = 957{,}000 \times 6.25 = 5{,}981{,}250 \text{ in.lb.}$$

According to [6], case 3, for bolt patterns symmetrical about the vertical axis and flange rotating about the bottom bolt, the maximum bolt force F_m due to the bending moment M_{B_B} is:

$$F_m = \frac{4}{3 \times {}^{D_{bolt}}\!/_{2} \times N_b} M_{B-B} = \frac{8 \times 5{,}981{,}250}{3 \times 21 \times 12} = 63{,}300 \text{ lb}$$

Thermal Load

From [9], Table 4.4, the bolt force due to differential thermal expansion is calculated as follows:

$$F_{th} = 0.25 \times \pi \times D_b^2 \times E_b \times \left(\alpha_{ts} \times \Delta T_{ts} - \alpha_b \times \Delta T_b\right)$$

Where $\Delta T_t = \Delta T_b$ = Temperature Change = 300 – 70 = 230°F.

Therefore,

$$F_{th} = 0.25 \times \pi \times 1.25^2 \times 26{,}700{,}000 \times 230 \times 10^{-6} \times (6.3 - 6.9) = -4{,}522 \text{ lb}$$

Bolt Stresses

For a lifting load of 6 g, the total bolt force is equal to:

$$F_{max} = F_{th} + F_m = -4{,}522 \text{ lb} + 63{,}300 \text{ lb} = 58{,}778 \text{ lb.}$$

The maximum tensile stress σ_{max} in a bolt is:

$$\sigma_{max} = \frac{F_{max}}{S_{bolt}} = \frac{58{,}778}{0.969} = 60{,}658 \text{ psi}$$

For a lifting load of 10 g, the total bolt force is equal to:

$$F_{max} = F_{th} + (10/6) \times F_m = \mathit{-4{,}522} \text{ lb} + (10/6) \times \mathit{63{,}300} \text{ lb} = \mathit{100{,}978} \text{ lb.}$$

The maximum tensile stress σ_{max} in a bolt is:

$$\sigma_{max} = \frac{F_{max}}{S_{bolt}} = \frac{100{,}978}{0.969} = 104{,}208 \text{ psi}.$$

Trunnion Flange Stresses

The trunnion flange is shown in Figure A.2.13.5-3.

Stresses at section AA:
Maximum bolt load due to 6 g is F_{max}.

It is conservatively assumed that the two bolts on either side of the vertical axis support the same load F_{max}.

Bending moment at AA:

$$M = F_{max} \times L_{f1} + 2 \times F_{max} \times L_{f2} = 63{,}300 \times (2 + 2 \times 0.593) = 201{,}674 \text{ in.lb}.$$

The flange length is:

$$2 \times \sqrt{12.5^2 - 8.5^2} = 18.33 \text{ in}.$$

Therefore:

$$Z = 18.33 \times 3.25^2 = 32.27 \text{ in}^3.$$

The bending stress is equal to:

$$\frac{M}{Z} = \frac{201{,}674}{32.27} = 6{,}250 \text{ psi}.$$

The shear stress is equal to:

$$\frac{3 \times Fm}{L \times Th_{flange}} = \frac{3 \times 63{,}300}{18.33 \times 3.25} = 3{,}188 \text{ psi}.$$

The maximum stress intensity is equal to:

$$\sqrt{6{,}250^2 + 4 \times 3{,}188^2} = 8{,}928 \text{ psi}.$$

Trunnion Attachment Block and Cask Shell Weld

The bending length is now equal to:

$$L_3 - Th_{tool} + H_{avg} + Th_{flange} - 2.79 = \mathit{4.0} - 1.0 + 6.25 + 3.25 - 2.79 = \mathit{9.71} \text{ in.}$$

Therefore, the weld bending moment is equal to F_V × (bending length), which is equal to:

$$M_w = 957{,}000 \times \mathit{9.71} = \mathit{9{,}292{,}470} \text{ in.lb.}$$

I_{weld} = 19,389 in^4

The bending stress σ_b is:

$$\sigma_b = \frac{M_w \times 0.5 \times (D_{w\_ext} + 2Th_{w\_ext})}{I_{weld}} = \frac{9{,}292{,}470 \times 0.5 \times (27 + 2 \times 1.75)}{19{,}389}.$$

$$\sigma_b = 7{,}310 \text{ psi}.$$

Bolt Torque

The bolt torque is calculated so that the preload bolt tensile stress is equal to the maximum applied stress σ_{max} (*60,658* psi), and there is lubrication on the threads.

That stress is induced by F_{max} = *58,778* lb.

The maximum torque required for this preload is $Q = K \times D_b \times F_{max}$.

Q = 0.135 × 1.25 × *58,778* = *9,920* in.lb = *827* ft.lb.

Local Stresses in Cask Outer Shell at Trunnion Attachment Block

Local stresses are calculated using the methodology [7].

The trunnion shear loads in the longitudinal and circumferential directions are respectively $V_L = F_V$ = 957,000 lb and V_C = 0 lb.

The external overturning moments supported by the intersection in the longitudinal and circumferential directions with respect to the shell are respectively $M_L = M_w$ = *9,292,470* in.lb and M_C = 0 in.lb.

The same shell parameters as used in the analysis of the double shoulder trunnions are used.

The above quantities are used in Table A.2.13.5-7 to calculate the stresses in the outer shell of the cask.

The maximum stress intensity is *27,361* psi.

A.2.13.5.2.3 Shear Key Bearing Block Assembly

Horizontal load

The NUHOMS® - MP197HB transport cask is blocked in translation by its shear pin key. The weight of the package used in the analysis is 320,000 lbs. This weight bounds the weight of package specified in Chapter A.2, Section A.2.1.3 of 303,600 lbs.

The maximum weight of the cask is W_H = 320,000 lb for horizontal loads, concentrated on the shear key (10 g).

Bearing Stress Between the Shear Key and the Bearing Block

Using a dynamic load factor of 1.1, the horizontal design load (yield) is:

$$F_H = W_H \times DLF \times a_L = 320{,}000 \times 1.1 \times 10 = 3{,}520{,}000\,\text{lb}.$$

The bearing stress due to the 10 g longitudinal transportation load is calculated assuming the load is applied uniformly to one face of the bearing block.

The bearing area is divided in two areas (see Figure A.2.13.5-6): a trapezoidal area A_1, of average width $(L_1+L_2)/2$ and height Y, and an area A_2, which is a segment of solid circle. The bearing area is the sum of A_1 and A_2.

$$A_1 = \frac{L_1 + L_2}{2} \times Y.$$

L_1 is the width of the top of the shear key (including its chamfer):

$$L_1 = W_{sk} - 2 \times H_c \times \tan(\alpha_c) = 22.25 - 2 \times 5 \times \tan(12.5^\circ) = 20.03\text{ in.}$$

L_2 is the width of the shear key at the lowest lateral point of contact with the bearing block:

$$L_2 = L_1 + 2\left[H_{bb} - Th_p - g - \left(\sqrt{R_{weld}^2 + \frac{L_1^2}{4}} - R_{weld}\right)\right] \times \sin(\alpha_c).$$

$$L_2 = 20.03 + 2\left[6.625 - 0.375 - 0.5 - \left(\sqrt{42.25^2 + \frac{20.03^2}{4}} - 42.25\right)\right] \times \sin(12.5^\circ).$$

$$L_2 = 22.02\,\text{in}.$$

Y is the distance between the planes L_1 and L_2:

$$Y = \sqrt{(R_{weld} + H_{bb} - g)^2 - \left(\frac{L_2}{2}\right)^2} - (R_{weld} + Th_p)$$

$$Y = \sqrt{(42.25 + 6.625 - 0.5)^2 - \left(\frac{22.02}{2}\right)^2} - (42.25 + 0.375)$$

$$Y = 4.48 \text{ in .}$$

$$A_1 = \frac{L_1 + L_2}{2} \times Y = \frac{20.03 + 22.02}{2} \times 4.48 = 94.21 \text{ in}^2$$

According to [8], Table 1:

$$A_2 = \frac{1}{2}(R_{weld} + H_{bb} - g)^2 \times [2\alpha - \sin(2\alpha)]$$

$$\alpha = \sin^{-1}\left[\frac{L_2}{2 \times (R_{weld} + H_{bb} - g)}\right] = \sin^{-1}\left[\frac{22.02}{2 \times (42.25 + 6.625 - 0.5)}\right]$$

$$\alpha = 0.230 \text{ rad .}$$

$$A_2 = \frac{1}{2}(42.25 + 6.625 - 0.5)^2 \times [2 \times 0.230 - \sin(2 \times 0.230)]$$

$$A_2 = 18.67 \text{ in}^2$$

Therefore, A = 94.21 + 18.67 = 112.88 in^2.

The bearing stress is equal to:

$$\frac{F_H}{A} = \frac{3{,}520{,}000}{112.88} = 31{,}184 \text{ psi}$$

Stresses in the Bearing Block

The maximum bending length at the horizontal section A–A on the bearing block for the longitudinal load is e' = x – Th_{pp} (see Figure A.2.13.5-7):

$$e' = \frac{H_{bb} - g + Th_p}{2} - Th_{pp} = \frac{6.625 - 0.5 + 0.375}{2} - 1.5 = 1.75 \text{ in}$$

Therefore, the maximum bending length at the horizontal section A–A on the bearing block for the longitudinal load is:

$$F_H \times e' = 3{,}520{,}000 \times 1.75 = 6{,}160{,}000 \text{ in.lb}$$

The moment of inertia is:

$$I_{yy} = \frac{bd^3}{12} - \frac{(b - 2\times Th_{bb}) \times (d - 2\times Th_{bb})^3}{12} = \frac{26.3 \times 12.06^3}{12} - \frac{20.3 \times 6.06^3}{12}$$

$$I_{yy} = 3{,}468 \text{ in}^4$$

The bending stress is:

$$\frac{6{,}160{,}000 \times d/2}{I_{yy}} = \frac{6{,}160{,}000 \times 12.06/2}{3{,}468} = 10{,}711 \text{ psi}$$

The shear stress is:

$$\frac{F_H}{b \times d - (b - 2\times Th_{bb}) \times (d - 2\times Th_{bb})} = \frac{3{,}520{,}000}{26.3 \times 12.06 - 20.3 \times 6.06} = 18{,}129 \text{ psi}$$

The maximum stress intensity is:

$$\sqrt{10{,}711^2 + 4 \times 18{,}129^2} = 37{,}808 \text{ psi}$$

Weld Between the Bearing Block and the Pad Plate

The bearing block is welded to the 1.5"-thick pad plate with a full penetration weld and a ½" outside cover fillet weld f_w (see Figure A.2.13.5-8). The welds are loaded in bending, resulting from the offset e (see Figure A.2.13.5-8) of the 10 g longitudinal point to the center of the pad plate (the ½" outside cover fillet weld (f_w) is conservatively neglected for the calculation of the bending length of the applied moment).

The bending moment is applied at the middle of the bearing block bearing area, therefore at a distance x from the outer shell:

$$x = \frac{H_{bb} - (g + Th_p)}{2} + Th_p = \frac{H_{bb} - g + Th_p}{2}$$

The bending length is equal to e = x – 0.5 × Th_{pp}.

$$e = \frac{H_{bb} - g + Th_p}{2} - \frac{Th_{pp}}{2} = \frac{6.625 - 0.5 - 1.5}{2} = 2.5 \text{ in}$$

The bending moment M is therefore $F_H \times e = 3{,}520{,}000 \times 2.5 = 8{,}800{,}000$ in.lb.

The section modulus of the weld is computed by treating the weld as a line per unit thickness t_{eff} [2]:

$$S_w = \left(bd + \frac{d^2}{3}\right) \times t_{eff}$$

$$t_{eff} = Th_{pp} + \frac{\sqrt{2}}{2} f_w = 1.5 + \frac{\sqrt{2}}{2} 0.5 = 1.85 \text{ in}$$

$$S_w = \left(26.3 \times 12.06 + \frac{12.06^2}{3}\right) \times 1.85 = 677.77 \text{ in}^3$$

The bending stress is equal to:

$$\frac{M}{S_w} = \frac{8{,}747{,}750}{677.77} = 12{,}984 \text{ psi}$$

Weld Between the Pad Plate and the Outer Shell

The shear key pad plate is welded to the cask structure all around with a 1" partial penetration groove weld (g_w) and a 5/8" fillet weld (f_{wp}). The shear area in the base metal of the structural shell is:

$$b_{pp} \times d_{pp} - (b_{pp} - 2 \times g_w) \times (d_{pp} - 2 \times g_w) + 2 \times (b_{pp} + d_{pp}) \times f_{wp}$$

$$36 \times 37.2 - (36 - 2 \times 1) \times (37.2 - 2 \times 1) + 2 \times (36 + 37.2) \times \frac{5}{8} = 233.9 \text{ in}^2$$

The weld shear stress at the junction of the weld material and the cask structural shell is:

$$\frac{F_H}{233.9} = \frac{3{,}520{,}000}{233.9} = 15{,}049 \text{ psi}$$

A.2.13.5.3 Results

The margin of safety will be calculated as follows:

$$\text{Margin} = \frac{\text{Allowable stress}}{\text{Calculated stress}} - 1$$

A.2.13.5.3.1 Double Shoulder Trunnions

The stresses calculated for a load of 3 g are summarized in Table A.2.13.5-8 and compared with allowable values (S_y). The stresses for a load of 5 g are also indicated in Table A.2.13.5-8 (simple 5/3 ratio of the values calculated for 3 g) and compared with the allowable values (S_u).

The recommended bolt torque for the trunnion bolts is 1,040 ft.lb.

The minimum engagement length is 1.678 in. Threaded inserts 1185-20CN-2500 are used. The minimum threaded length is equal to 1.68 in, which is greater than the minimum required engagement length.

A.2.13.5.3.2 Single Shoulder Trunnions

The stresses calculated for a load of 6 g are summarized in Table A.2.13.5-9 and compared with allowable values (S_y). The stresses for a load of 10 g are also indicated in Table A.2.13.5-9 (simple 10/6 ratio of the values calculated for 6 g) and compared with the allowable values (S_u).

The recommended bolt torque for the trunnion bolts is *830* ft.lb.

A.2.13.5.3.3 Shear key assembly

The stresses calculated for a longitudinal load of 10 g are summarized in Table A.2.13.5-10 and compared with allowable values.

A.2.13.5.4 Conclusions

All of the stresses calculated above are less than the allowable stresses.

A.2.13.5.4.1 Trunnion assembly

Based on the above calculations, the design meets the requirements of 10CFR71.

A.2.13.5.4.2 Shear key assembly

Based on the above calculations, the design meets the requirements of 10CFR71.

A.2.13.5.5 References

1. NOT USED.
2. “Pressure Vessel Design Handbook,” Henry Bednar, 1981.
3. Machinery Handbook, 26th Edition, Industrial Press, 2000.
4. 10CFR Part 71, “Packaging and Transportation of Radioactive Materials.”
5. ANSI N14.6, “Special Lifting Devices for Shipping Containers Weighing 10,000 Pounds or More,” 1993.
6. Machine Design, August 17, 1967, “Eccentrically Loaded Joints,” Richard T. Burger.
7. “Local Stresses in Spherical and Cylindrical Shells Due to External Loading,” Welding Research Council (WRC), Bulletin 107 by Wichman, Hopper and Mershon.
8. “Formulas for Stress and Strain,” Roark, 4th Edition.
9. NUREG/CR-6007 "Stress Analysis of Closure Bolts for Shipping Casks", By Mok, Fischer, and Hsu, Lawrence Livermore National Laboratory, 1992.
10. ASME Code Section II, Part D, 2004 Edition Including 2006 Addenda.

Table A.2.13.5-1
Steel Structural Properties at 300°F (ksi)

Part	**Material**	S_y	S_u	**E**
Double shoulder trunnions	SA-182 F316N	28.5	77.0	N/A
Trunnions attachment blocks	SA-350 LF3	33.2	70.0	N/A
Outer shell	SA-203 Gr. E	35.4	70.0	N/A
Trunnion bolts	SA-540 Gr. B23 Cl. 1	140.3	165.0	26,700
Shear key bearing block *and single shoulder trunnions*	SA-182 F6NM	84.6	115.0	N/A
Pad plate	SA-516-70	33.6	70.0	N/A

Note: Material properties are taken from ASME Code [10]. Material properties and allowable stresses are based on 300 °F which bound -40 °F, -20 °F, and 100 °F ambient conditions.

Table A.2.13.5-2
Bolt Parameters

D_b	Bolt nominal diameter (in)	1.25
S_{bolt}	Bolt stress area (in^2)	0.969
n	Number of threads per inch	7
$K_{n\,max}$	Maximum minor diameter of internal threads (in)	1.1230
$E_{s\,min}$	Minimum pitch diameter of external threads (in)	1.1439
$D_{s\,min}$	Minimum major diameter of external threads (in)	1.2232
$E_{n\,max}$	Maximum pitch diameter of internal threads (in)	1.1716

Table A.2.13.5-3
Design Parameters

N_{tr}	Number of trunnions	2
a_V	Vertical acceleration (g)	3
a_L	Longitudinal acceleration (g)	10
Th_{os}	Outer shell thickness (in)	2.75
N_b	Number of bolts per trunnion	12
K	Nut factor	0.135
D_{bolt}	Bolt circle diameter (in)	21.00
R_{weld}	Radius of outer shell at attachment block (in)	42.25
Th_{tool}	Thickness of lifting tool (in)	1.00
L_{ef}	Length of extremity flange (in)	0.63
D_{cav}	Depth of inner trunnion cavity (in)	8.76
D_i	Diameter of trunnion cavity (in)	5.00
D_{ext1}	Minimum shoulder diameter - double shoulder trunnion (in)	9.84
D_{ext1s}	*Minimum shoulder diameter – single shoulder trunnion (in)*	*9.75*
L_1	Outer shoulder length - double shoulder trunnion (in)	4.38
D_{ext2}	Maximum shoulder diameter (in)	11.81
L_2	Inner shoulder length - double shoulder trunnion (in)	4.56
L_3	Shoulder length - single shoulder trunnion (in)	*4.00*
D_{max}	Maximum trunnion diameter (in)	25.00
D_{flange}	Diameter of trunnion flange (in)	17.00
Th_{flange}	Thickness of flange at closure bolt circle (in)	3.25
H_{max}	Maximum height of attachment block (in)	7.00
$Th_{w\ ext}$	Thickness of external weld of attachment block (in)	1.75
$Th_{w\ int}$	Thickness of internal weld of attachment block (in)	1.25
$D_{w\ int}$	Inner weld diameter (in)	17.00
$D_{w\ ext}$	Outer weld diameter (in)	27.00
W_{sk}	Shear key width (in)	22.25
H_c	Height of shear key chamfer (in)	5.00
α_c	Angle of shear key chamfer	12.5°
H_{bb}	Height of bearing block (in)	6.625
g	Thickness of bearing block closure plate groove (in)	0.50
Th_p	Thickness of protection plate (in)	0.375
Th_{pp}	Thickness of pad plate (in)	1.50
b	Width of the base of the bearing block (in)	26.30
d	Longitudinal dimension of the bearing block (in)	12.06
Th_{bb}	Thickness of bearing block wall (in)	3.00
b_{pp}	Longitudinal dimension of pad plate (in)	36.00
d_{pp}	Lateral dimension of pad plate (in)	37.20
α_t	Coefficient of thermal expansion of *double shoulder* trunnions and trunnion blocks at 300°F (in/in/°F)	9.2×10^{-6}
α_{ts}	*Coefficient of thermal expansion of single shoulder trunnions at 300°F (in/in/°F)*	*6.3×10^{-6}*
α_b	Coefficient of thermal expansion of trunnion bolts at 300°F (in/in/°F)	6.9×10^{-6}

Table A.2.13.5-4 Double Shoulder Trunnion Stress Calculation

Section	**A–A**	**B–B**	**C–C**
Stress area (in^2)	$S_{AA}=\frac{\pi}{4}\left(D_{ext1}^2-D_i^2\right)=$ $\frac{\pi}{4}\left(9.84^2-5.0^2\right)=56.41$	$S_{BB}=\frac{\pi}{4}\left(D_{ext2}^2-D_i^2\right)=$ $\frac{\pi}{4}\left(11.81^2-5.0^2\right)=89.91$	$S_{CC}=\frac{\pi}{4}\times D_{ext2}^2=$ $\frac{\pi}{4}\times 11.81^2=109.54$
Moment of inertia (in^4)	$I_{AA}=\frac{\pi}{64}\left(D_{ext1}^4-D_i^4\right)=$ $\frac{\pi}{64}\left(9.84^4-5.00^4\right)=429.5$	$I_{BB}=\frac{\pi}{64}\left(D_{ext2}^4-D_i^4\right)=$ $\frac{\pi}{64}\left(11.81^4-5.0^4\right)=924.2$	$I_{CC}=\frac{\pi}{64}\times D_{ext2}^4=$ $\frac{\pi}{64}\times 11.81^4=954.9$
Bending distance (in)	L_1-Th_{tool} $=4.38-1$ $=3.38=L_{AA}$	$L_1+L_2-(L_{ef}+L_1+L_2-D_{cav})-Th_{tool}$ $=D_{cav}-L_{ef}-Th_{tool}$ $=8.76-0.63-1$ $=7.13=L_{BB}$	$L_1+L_2-Th_{tool}$ $=4.38+4.56-1$ $=7.94=L_{CC}$
Bending moment (in.lb)	$M_{AA}=F_V\times L_{AA}$ $=478{,}500\times 3.38$ $=1{,}617{,}330$	$M_{BB}=F_V\times L_{BB}$ $=478{,}500\times 7.13$ $=3{,}411{,}705$	$M_{CC}=F_V\times L_{CC}$ $=478{,}500\times 7.94$ $=3{,}799{,}290$
Shear stress (ksi)	$\frac{F_V}{S_{AA}}=\frac{478{,}500}{56.41}$ $=8.5$	$\frac{F_V}{S_{BB}}=\frac{478{,}500}{89.81}$ $=5.3$	$\frac{F_V}{S_{CC}}=\frac{478{,}500}{109.54}$ $=4.4$
Bending stress (ksi)	$\frac{M_{AA}}{I_{AA}}\times\frac{D_{ext1}}{2}$ $=\frac{1{,}617{,}330}{429.5}\times\frac{9.84}{2}$ $=18.5$	$\frac{M_{BB}}{I_{BB}}\times\frac{D_{ext2}}{2}$ $=\frac{3{,}411{,}705}{924.5}\times\frac{11.81}{2}$ $=21.8$	$\frac{M_{CC}}{I_{CC}}\times\frac{D_{ext2}}{2}$ $=\frac{3{,}799{,}290}{954.9}\times\frac{11.81}{2}$ $=23.5$
Max. stress intensity (ksi)	$\sqrt{18.5^2+4\times 8.5^2}$ $=25.1$	$\sqrt{21.8^2+4\times 5.3^2}$ $=24.2$	$\sqrt{23.5^2+4\times 4.4^2}$ $=25.1$

Table A.2.13.5-5 Cask Outer Shell Stresses Calculations (Double Shoulder Trunnion)

From fig.	Read curves for		Mult.	Abs. stress values		Au	Al	Bu	Bl	Cu	Cl	Du	Dl
3C & 4C			0	0	0	0	0	0	0	0	0	0	0
1C & 2C–1			0	0	0	0	0	0	0	0	0	0	0
3A			0	0						0	0	0	0
1A			0	0						0	0	0	0
3B	1.465	β=0.31 C_L=0.89	4,901	7,182		-7,182	-7,182	7,182	7,182				
1B or **1B–1**	0.026	β=0.32	426,542	11,090		-11,090	11,090	11,090	-11,090				
	Σ(phi - circumferential stresses)					**-18,272**	**3,908**	**18,272**	**-3,908**	**0**	**0**	**0**	**0**
3C & 4C			0	0	0	0	0	0	0	0	0	0	0
1C–1 & 2C			0	0	0	0	0	0	0	0	0	0	0
4A			0	0						0	0	0	0
2A			0	0						0	0	0	0
4B	0.633	β=0.31 C_L=0.97	4,901	3,101		-3,101	-3,101	3,101	3,101				
2B or **2B–1**	0.043	β=0.33	407,853	17,538		-17,538	17,538	17,538	-17,538				
	Σ(X - longitudinal stresses)					**-20,639**	**14,437**	**20,639**	**-14,437**	**0**	**0**	**0**	**0**
	Shear stress due to torsion M_T				0	0	0	0	0	0	0	0	0
	Shear stress due to load V_C				0	0	0	0	0				
	Shear stress due to load V_L				3,522					3,522	3,522	-3,522	-3,522
			Σ(shear stresses τ)			0	0	0	0	3,522	3,522	-3,522	-3,522
			Stress intensities			**20,639**	**14,437**	**20,639**	**14,437**	**7,045**	**7,045**	**7,045**	**7,045**

Table A.2.13.5-6 Single Shoulder Trunnion Stress Calculation

Section	*A–A*
Stress area (in²)	$S_{AA} = \frac{\pi}{4} \times D_{ext1s}^2 = \frac{\pi}{4} \times 9.75^2 = 74.66$
Moment of inertia (in⁴)	$I_{AA} = \frac{\pi}{64} \times D_{ext1s}^4 = \frac{\pi}{64} \times 9.75^4 = 443.60$
Bending distance (in)	$L_3 - Th_{tool} = 4.00 - 1 = 3.00 = L_{AA}$
Bending moment (in.lb)	$M_{AA} = F_V \times L_{AA} = 957{,}000 \times 3.00 = 2{,}871{,}000$
Shear stress (ksi)	$\frac{F_V}{S_{AA}} = \frac{957{,}000}{74.66} = 12.8$
Bending stress (ksi)	$\frac{M_{AA}}{I_{AA}} \times \frac{D_{ext1s}}{2} = \frac{2{,}871{,}000}{443.60} \times \frac{9.75}{2} = 31.6$
Max. stress intensity (ksi)	$\sqrt{31.6^2 + 4 \times 12.8^2} = 40.7$

Table A.2.13.5-7 Cask Outer Shell Stresses Calculations (Single Shoulder Trunnion)

From fig.	Read curves for		Mult.	Abs. stress values		Au	Al	Bu	Bl	Cu	Cl	Du	Dl
3C & 4C			0	0	0	0	0	0	0	0	0	0	0
1C & 2C–1			0	0	0	0	0	0	0	0	0	0	0
3A			0	0						0	0	0	0
1A			0	0						0	0	0	0
3B	1.465	β=0.31 C_L=0.89	*6,497*	*9,521*		*-9,521*	*-9, 521*	*9, 521*	*9, 521*				
1B or **1B–1**	0.026	β=0.32	*565,472*	*14,702*		*-14,702*	*14,702*	*14,702*	*-14,702*				
	Σ(phi - circumferential stresses)					***-24,223***	***5,181***	***24,223***	***-5,181***	**0**	**0**	**0**	**0**
3C & 4C			0	0	0	0	0	0	0	0	0	0	0
1C–1 & 2C			0	0	0	0	0	0	0	0	0	0	0
4A			0	0						0	0	0	0
2A			0	0						0	0	0	0
4B	0.633	β=0.31 C_L=0.97	*6, 497*	*4,111*		*-4,111*	*-4,111*	*4,111*	*4,111*				
2B or **2B–1**	0.043	β=0.33	*540,696*	*23,250*		*-23,250*	*23,250*	*23,250*	*-23,250*				
	Σ(X - longitudinal stresses)					***-27,361***	***19,139***	***27,361***	***-19,139***	**0**	**0**	**0**	**0**
	Shear stress due to torsion M_T				0	0	0	0	0	0	0	0	0
	Shear stress due to load V_C				0	0	0	0	0				
	Shear stress due to load V_L				7,045					*7,043*	*7,043*	*-7,043*	*-7,043*
			Σ(shear stresses τ)			0	0	0	*7,045*	*7,043*	*7,043*	*-7,043*	*-7,043*
			Stress intensities			***27,361***	***19,139***	***27,361***	***19,139***	***14,086***	***14,086***	***14,086***	***14,086***

Table A.2.13.5-8
Summary of Lifting Stresses - Double Shoulder Trunnions (ksi)

	Calculated (3 g)	Allowable (S_y)	Margin	Calculated (5 g)	Allowable (S_u)	Margin
Stress intensity in section A-A	25.1	28.5	0.13	41.9	77.0	0.84
Stress intensity in section B-B	24.2		0.18	40.3		0.91
Stress intensity in section C-C	25.1		0.13	41.8		0.84
Bolt tensile stress	76.4	140.3	0.84	111.8	165.0	0.48
Stress intensity in trunnion flange	8.0	28.5	2.56	13.3	77.0	4.77
Weld bending stress	5.5	35.4	5.44	9.2	70.0	6.62
Outer cask shell stress	20.6	35.4	0.72	34.3	70.0	1.04

Table A.2.13.5-9
Summary of Lifting Stresses - Single Shoulder Trunnions (ksi)

	Calculated (6 g)	***Allowable (S_y)***	***Margin***	***Calculated (10 g)***	***Allowable (S_u)***	***Margin***
Stress intensity in section A-A	*40.7*	*84.6*	*1.08*	*67.8*	*115.0*	*0.70*
Bolt tensile stress	*61.0*	*140.3*	*1.30*	*102.0*	*165.0*	*0.62*
Stress intensity in trunnion flange	*8.9*	*28.5*	*2.20*	*14.8*	*77.0*	*4.19*
Weld bending stress	*7.3*	*35.4*	*3.85*	*12.2*	*70.0*	*4.75*
Outer cask shell stress	*27.4*	*35.4*	*0.29*	*45.7*	*70.0*	*0.53*

Table A.2.13.5-10
Summary of Longitudinal Stresses - Shear Key Assembly (ksi)

		Calculated (10 g)	**Allowable (S_y or $0.6\times S_y$)**	**Margin**
Bearing block	Bearing stress	31.2	84.6	1.71
	Bending stress	10.7	84.6	6.91
	Shear stress	18.1	50.8	1.81
	Maximum stress intensity	37.8	84.6	1.24
	Bending stress in the weld with pad plate	13.0	84.6	5.51
Weld between pad plate and cask outer shell	Shear stress in base metal	15.0	21.2	0.41

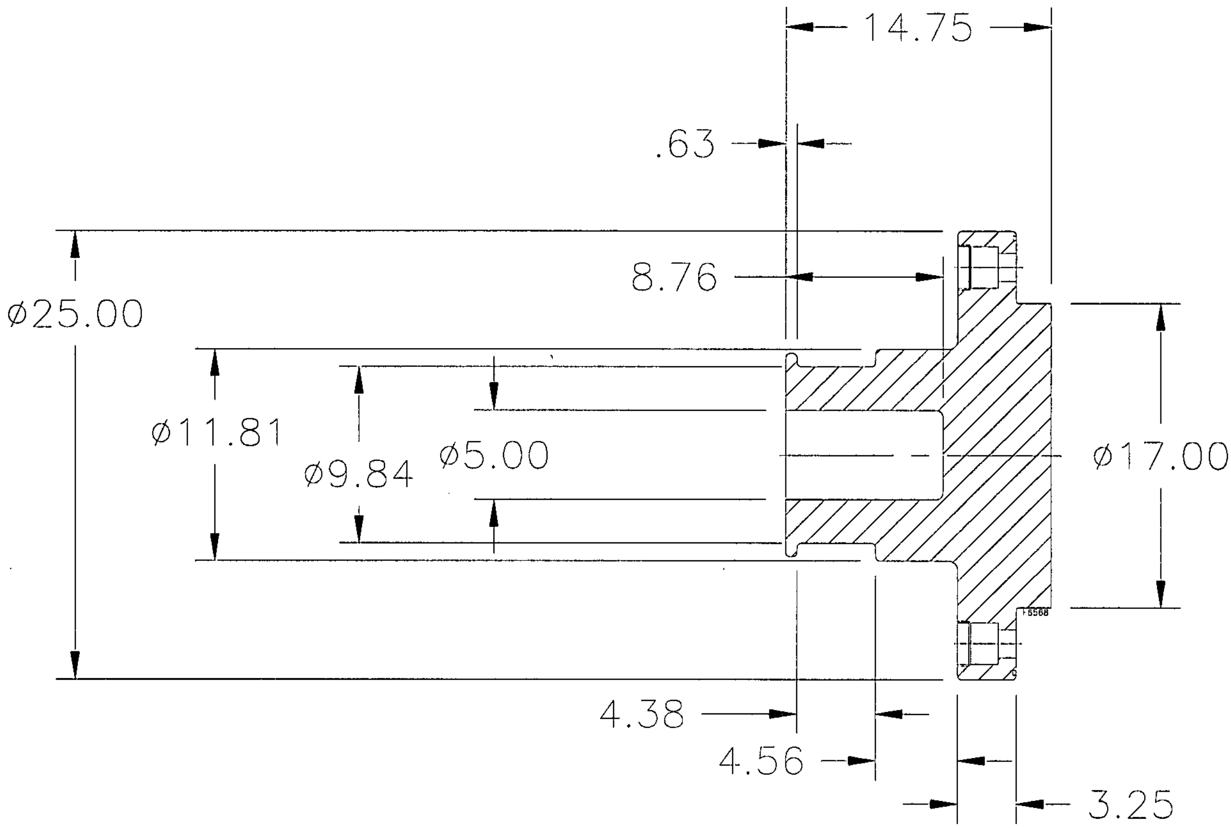

Figure A.2.13.5-1
Double Shoulder Trunnion Section

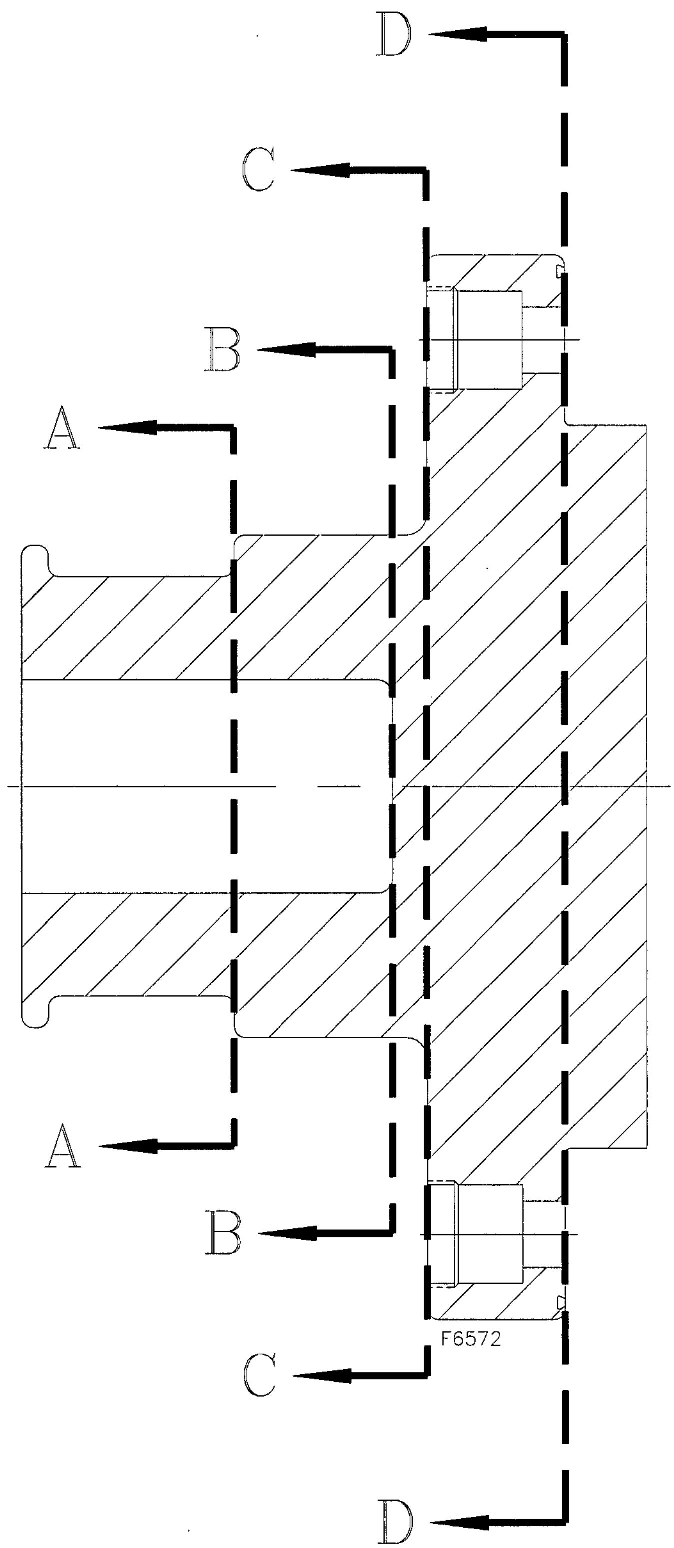

Figure A.2.13.5-2
Double Shoulder Trunnion Stress Sections

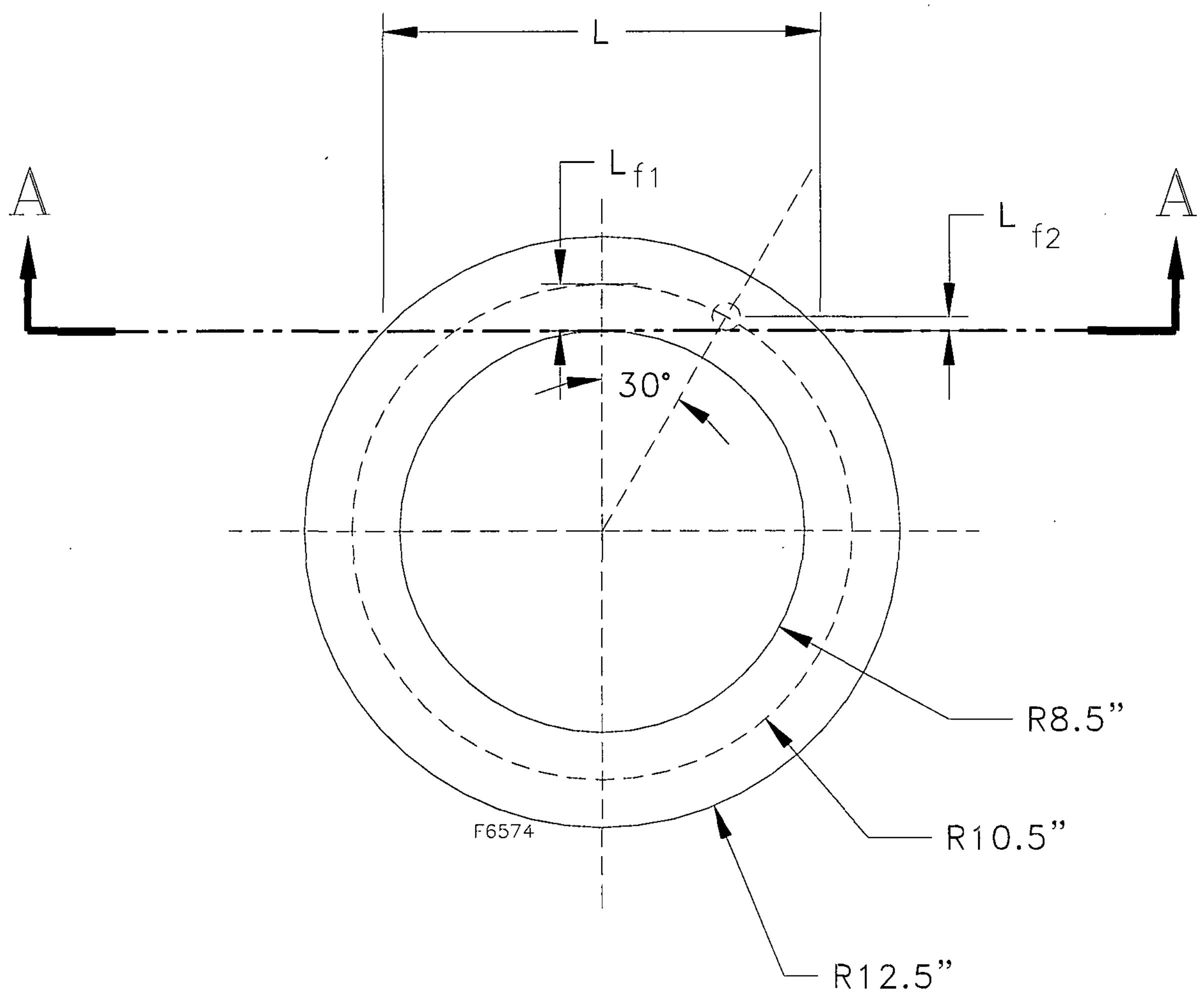

Figure A.2.13.5-3
Trunnion Flange

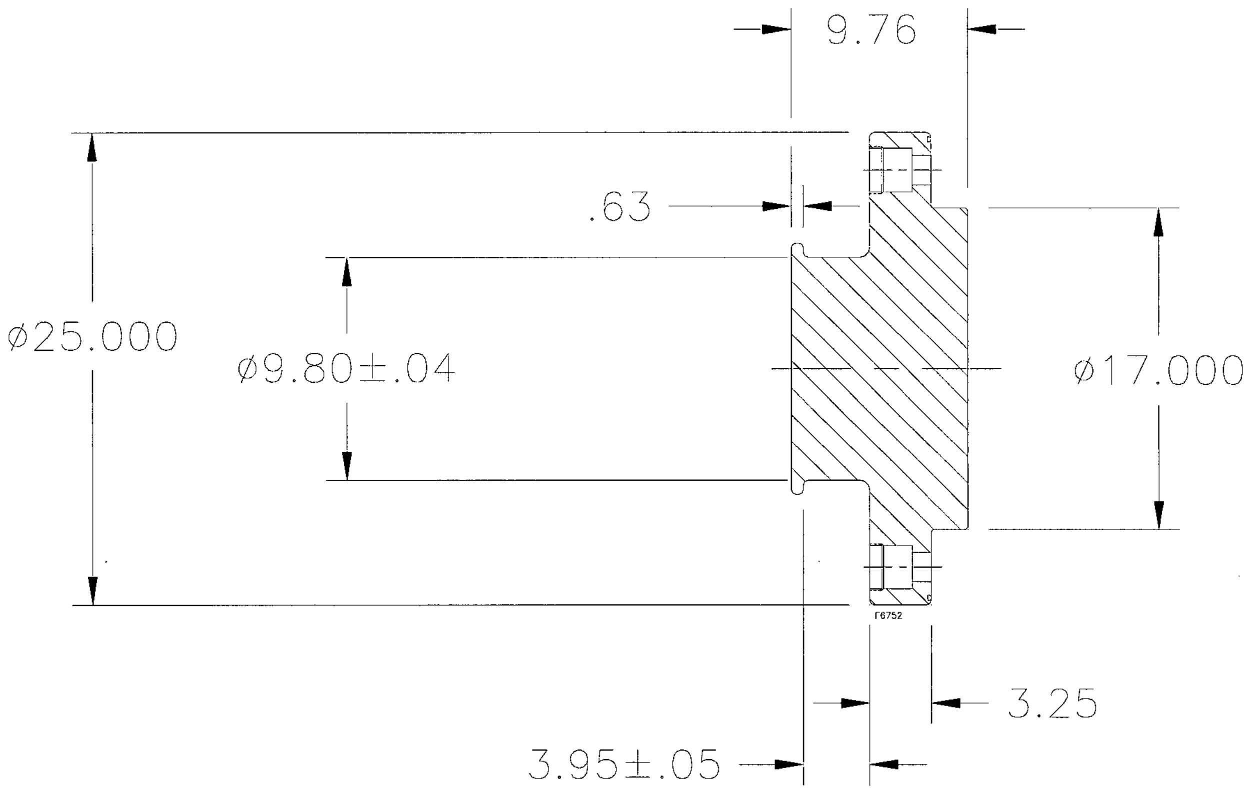

Figure A.2.13.5-4
Single Shoulder Trunnion Section

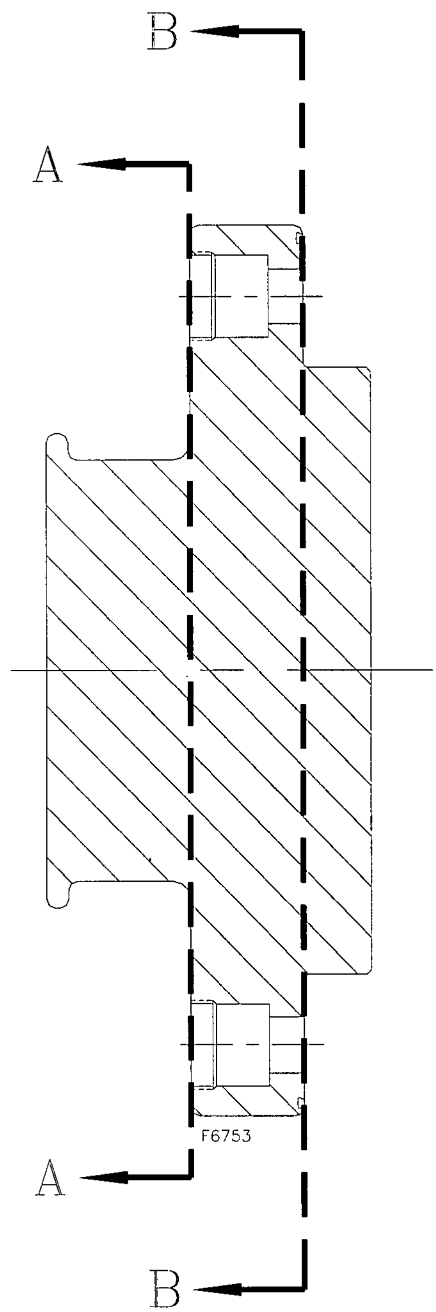

Figure A.2.13.5-5
Single Shoulder Trunnion Stress Sections

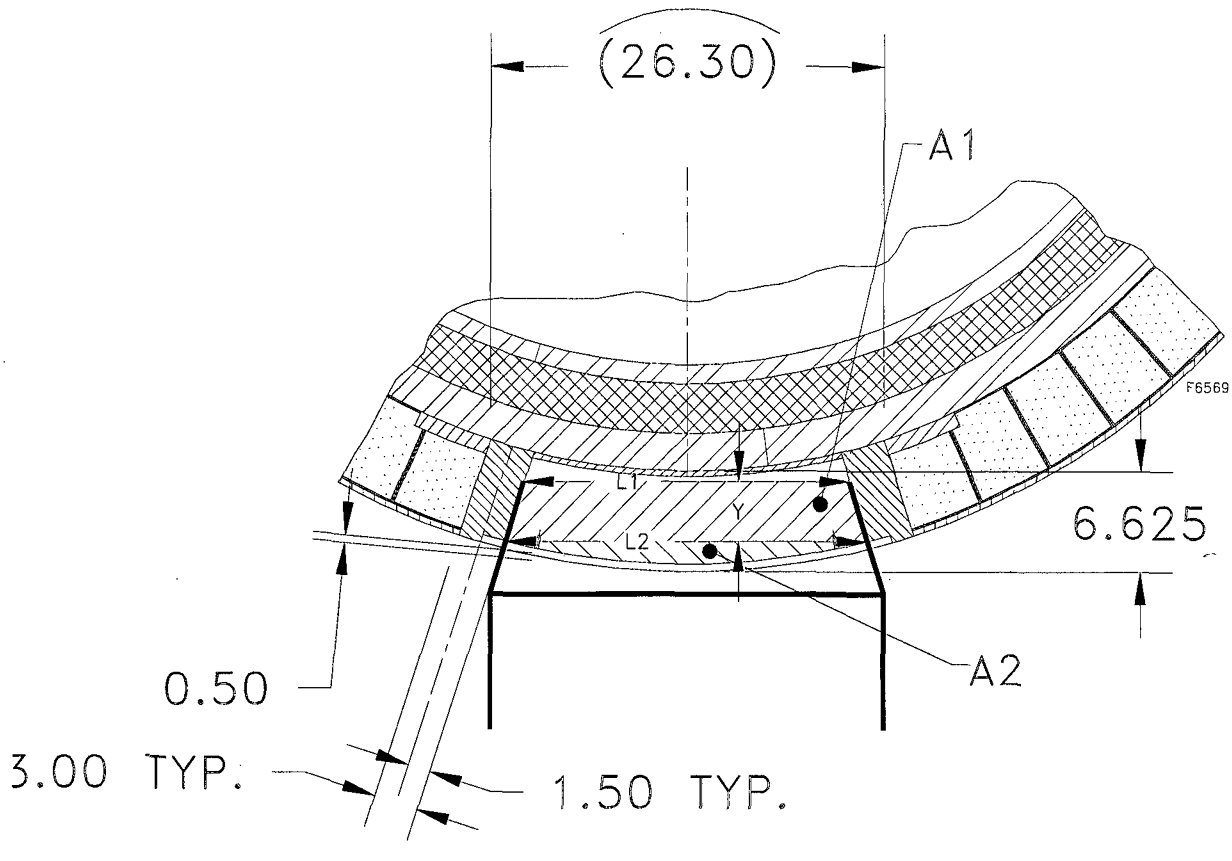

Figure A.2.13.5-6
Shear Key Bearing Area

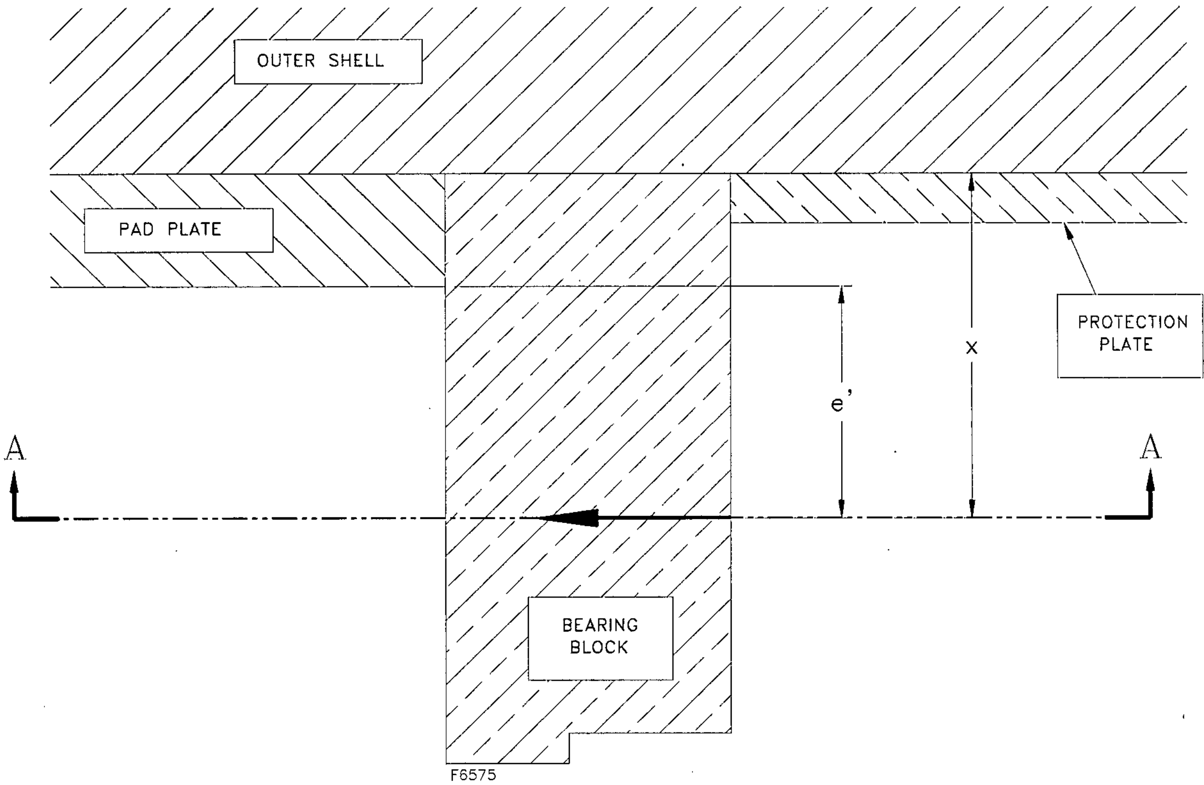

Figure A.2.13.5-7
Bearing Block Bending Length

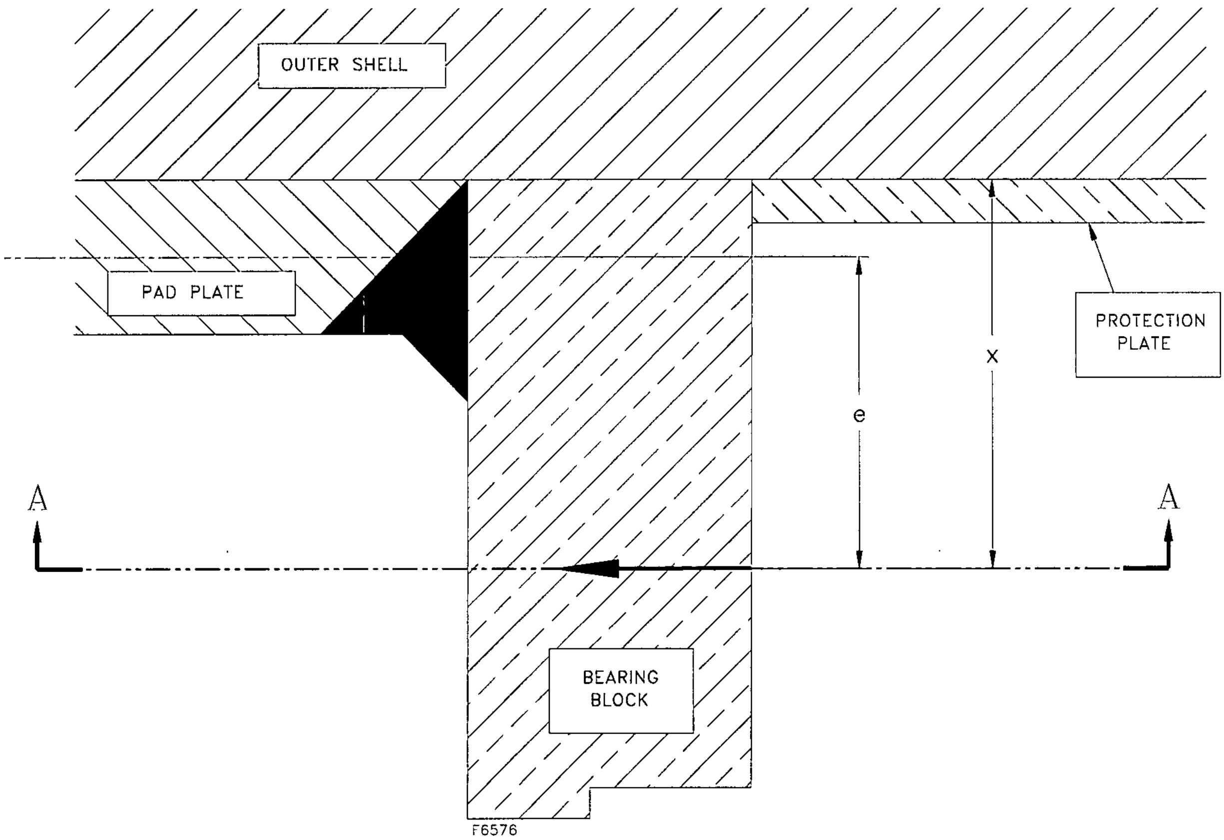

Figure A.2.13.5-8
Bearing Block and Pad Plate Weld Bending Length

Appendix A.2.13.6
MP197HB Cask Containment Boundary Fatigue Evaluation

TABLE OF CONTENTS

LIST OF TABLES

Appendix A.2.13.6
MP197HB Cask Containment Boundary Fatigue Evaluation

NOTE: References in this Appendix are shown as [1], [2], etc. and refer to the reference list in Section A.2.13.6.5.

A.2.13.6.1 Purpose

The purpose of the fatigue analysis is to show that the containment vessel stresses are within acceptable NCT fatigue limits. This is done by determining the fatigue usage factor for each NCT event for the lid and for the rest of the containment boundary at locations on the containment vessel with the highest stresses. The cumulative fatigue damage or usage factor for all of the events is conservatively determined by adding the fatigue usage factors for the individual events, both for the lid and for the rest of the containment boundary, assuming these maximum stress intensities occur at the same location.

The sum of the individual usage factors is checked to make certain that for a given number of round-trip shipments of the NUHOMS® - MP197HB transport cask, the total fatigue damage factor, for the lid and for the rest of the containment boundary, is less than one.

The number of round-trip shipments considered for the lid is 600. The number of round-trip shipments considered for the rest of the containment boundary is 1,000.

A.2.13.6.2 Assumptions

The fatigue analysis is based on the procedure described in Regulatory Guide 7.6 [4] and ASME Section III Appendices [3]. When determining the stress cycles, consideration is given to the superposition of individual loads which can occur together and produce a total stress intensity range greater than the stress intensity range of individual loads. Also, the maximum stress intensities for all individual loads are conservatively combined simultaneously. The sequence of events assumed for the fatigue evaluation is given below.

1. Operating bolt preload
2. Leak test
3. Pressure fluctuations
4. Temperature fluctuations
5. Vibration
6. Shock
7. 1 foot drop.

The maximum stresses in the NUHOMS® - MP197HB transport cask containment boundary for each individual load case are taken from Appendix A.2.13.1.

The NUHOMS® - MP197HB transport cask is only loaded for one of the two legs of a round trip shipment.

The bolt torque is applied twice every round trip.

The pressure cycle occurs twice every round trip.

In the lid case, since the maximum stress intensity occurs in the –40°F cold environment load case, it is assumed that the temperature cycle occurs twice per round-trip shipment.

For the rest of the containment boundary, it is assumed that the temperature cycle only occurs once per round trip shipment, when the cask is loaded.

It is conservatively assumed that the cask is dropped once per round trip shipment.

Each round trip shipment is assumed to average 3,000 miles each way.

A.2.13.6.3 Calculations

A.2.13.6.3.1 Bolt Preload

The number of preload cycles is two times the number of round-trip shipments.

The bolt preload specified to ensure a leak tight seal produces significant stresses in the lid. Therefore, this loading is conservatively included in the fatigue evaluation. The maximum stress intensity due to bolt preload is 12.5 ksi in the lid and 9.8 ksi in the rest of the transport cask containment boundary (see Table A.2.13.1-1).

A.2.13.6.3.2 Leak test

The proof test is 1.5 × 30 psig (The MNOP is 12.7 psig; 30 psig is conservatively used for design) = 45 psig.

The maximum stress intensity due to a normal condition pressure load of 30 psig is 2.7 ksi in the lid and 1.5 ksi (actual calculated stress is 1.4 ksi; 1.5 ksi is used) in the rest of the containment boundary (Table A.2.13.1-2).

Therefore, the maximum stress intensity due to the test pressure load in the lid is approximately 1.5 × 2.7 ksi = 4.1 ksi and 1.5 × 1.5 ksi = 2.3 ksi in the rest of the containment boundary

A.2.13.6.3.3 Pressure Fluctuations

It is assumed that the pressure cycle occurs twice per round-trip shipment.

The maximum stress intensity due to a pressure load of 30 psig is 2.7 ksi in the lid and 1.5 ksi in the rest of the containment boundary (Table A.2.13.1-2).

A.2.13.6.3.4 Temperature Fluctuations

The maximum stress intensity in the lid due to normal condition thermal loads occurs in the -40°F cold environment load case, and is 16.9 ksi (Table A.2.13.1-6). The maximum stress intensity in the rest of the containment boundary due to normal condition thermal loads occurs in the 100°F hot environment load case, and is 23.9 ksi (Table A.2.13.1-4).

A.2.13.6.3.5 Vibration

Since vibration accelerations are higher on a truck than on a rail car, the truck vibration loads are considered bounding. According to [1], the peak vibration load at the bed of a truck is 0.3 g longitudinal, 0.3 g transverse, and 0.6 g vertical. The maximum stress intensity generated by truck vibration is computed by extrapolating from the maximum stress intensity obtained in the railcar vibration load case.

According to [2], the peak vibration load on a railcar is 0.19 g longitudinal, 0.19 g transverse, and 0.42 g vertical. Therefore the truck vibration load is conservatively roughly 150% of the railcar vibration load.

The maximum stress intensity in the whole transport cask containment boundary due to railcar vibration calculated in Table A.2.13.1-12 is 1.0 ksi (outer shell is not containment boundary). Therefore, the maximum stress intensity in the transport cask containment boundary (including the lid) due to truck vibration would be roughly 1.5 ksi.

For the number of round trip shipments considered in this analysis, the number of truck vibration cycles would be very large.

A.2.13.6.3.6 Shock

The NUHOMS®-MP197HB transport cask may be shipped either by truck or by railcar. ANSI N14.23 [1] specifies a peak shock loading of 2.3 g longitudinal, 1.6 g lateral, 3.5 g vertical up, and 2.0 g vertical down for truck transport, while NUREG 766510 [2] specifies a peak shock loading of 4.7 g in all directions for rail car transport. Consequently, only the inertial loading caused by a railcar shock is considered, since it is bounding.

Each round trip shipments averages 3,000 miles each way. NUREG 766510 [2] reports that there are roughly 9 shock cycles per 100 miles of rail car transport. The maximum stress intensities are found when the contents of the cask load the containment boundary, which happens during only half of a round-trip shipment. Therefore the number of cycles is:

For the lid:

$$3{,}000 \text{ miles} \times 600 \text{ shipments} \times 0.09 \text{ shocks / mile} = 162{,}000 \text{ cycles.}$$

For the rest of the containment boundary:

$$3{,}000 \text{ miles} \times 1{,}000 \text{ shipments} \times 0.09 \text{ shocks / mile} = 270{,}000 \text{ cycles.}$$

The maximum stress intensity range due to railcar shock is equal to 13.0 + 1.7 = 14.7 ksi in the lid and 8.8 + 4.0 =12.8 ksi in the rest of the containment boundary (Tables A.2.13.1-9 and A.2.13.1-11).

A.2.13.6.3.7 1-foot Normal Condition Drop

The maximum stress intensity due to normal condition impact loads occurs in the 1 foot side drop load case, and is equal to 15.9 ksi in the lid and 24.0 ksi in the rest of the containment boundary (Table A.2.13.1-16).

A.2.13.6.3.8 Transport Cask Fatigue Evaluation-Usage Factor Calculation

The damage factors listed in Tables A.2.13.6-1 and A.2.13.6-2 are computed based on the stresses and cyclic histories described above, and the fatigue curves shown in Figure I-9.1 of [3] for UTS < 80 ksi.

Since the model used for stress analysis of the transport cask includes detailed meshing of corners and bolt holes, the fatigue strength reduction factor (KF) which accounts for stress concentrations, is already accounted for in the stresses reported above. However, for conservatism, a stress concentration factor of 2 is used to obtain peak stresses.

The NUHOMS®-MP197HB transport cask containment boundary is constructed from SA-203 Grade E and SA-350-LF3. The modulus of elasticity of SA-203 Grade E and SA-350-LF3 is 26.2×10^6 psi at 400°F.

Consequently, KE = 30.0 × 106 / 26.2 × 106 = 1.1450 [3].

Here n is the number of cycles, N is taken from Figure I-9.1 of [3], and S_a is defined in the following way:

If one cycle goes from 0 to +S.I., then Sa = (1/2) × S.I. × KF × KE.

If one cycle goes from -S.I. to +S.I., then Sa = S.I. × KF × KE.

Where KE is the correction factor for modulus of elasticity.

A.2.13.6.4 Conclusions

The total damage factor is less than one in both cases. Therefore, the NUHOMS® - MP197HB transport cask lid will not fail due to fatigue after 600 round trip shipments, and the rest of the NUHOMS® - MP197HB transport cask containment boundary will not fail due to fatigue after 1,000 round trip shipments.

A.2.13.6.5 References

1. American Standard Design Basis for Resistance to Shock and Vibration of Radioactive Material Packages Greater than One Ton in Truck Transport, ANSI, N14.23, May 1980.
2. Shock and Vibration Environments for Large Shipping Containers on Rail Cars and Trucks, NUREG 766510.
3. ASME Boiler and Pressure Vessel Code, Section III, Division 1, Appendix, 2004 with 2006 addendum.
4. USNRC Regulatory Guide 7.6, "Design Criteria for the Structural Analysis of Shipping Cask Containment Vessel," Rev. 1, March 1978.

Table A.2.13.6-1 Damage Factor Calculation (Lid)

Event	**Stress Intensity (ksi)**	**S.I. × K_F (ksi)**	**S_a (ksi)**	**Cycles**		**Damage Factor n / N**
				n	**N**	
Bolt preload	12.5	25.0	14.3	1,200	380,000	0.003
Leak test	4.1	8.2	4.7	1,200	∞	0.000
Pressure fluctuations	2.7	5.4	6.2	1,200	∞	0.000
Temperature fluctuations	16.9	33.8	19.4	1,200	113,000	0.011
Vibration load	1.5	3.0	3.4	-[1]	∞	0.000
Shock load	14.7	29.4	16.8	162,000	186,000	0.871
1 foot drop impact load	15.9	31.8	18.2	600	140,000	0.004
					Σ	**0.889**

[1] The number of truck vibration cycles is very large and difficult to estimate. However, since N for this load case is ∞, n / N = 0, for a finite number of shipments.

Table A.2.13.6-2 Damage Factor Calculation (Rest of Containment Boundary)

Event	Stress Intensity (ksi)	S.I. × K_F (ksi)	S_a (ksi)	Cycles		Damage Factor n / N
				n	N	
Bolt preload	9.8	19.6	11.2	2,000	2,640,000	0.001
Leak test	2.3	4.6	2.6	2,000	∞	0.000
Pressure fluctuations	1.5	3.0	3.4	2,000	∞	0.000
Temperature fluctuations	23.9	47.8	27.4	1,000	29,300	0.034
Vibration load	1.5	3.0	3.4	-[1]	∞	0.000
Shock load	12.8	25.6	14.7	270,000	344,000	0.785
1 foot drop impact load	24.0	48.0	27.5	1,000	29,000	0.034
					Σ	**0.854**

[1] The number of truck vibration cycles is very large and difficult to estimate. However, since N for this load case is ∞, n / N = 0, for a finite number of shipments.

Appendix A.2.13.7
MP197HB DSC (Shell Assembly) Structural Evaluation

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

Appendix A.2.13.7
MP197HB DSC (Shell Assembly) Structural Evaluation

A.2.13.7.1 Introduction

The DSC shell assemblies consist of a cylindrical shell, bottom and top cover plates (inner and outer, as applicable) and bottom and top shield plugs. Each DSC shell assembly functions to support a basket assembly and confine associated fuel assemblies that are contained within the DSC shell assembly.

Multiple DSC shell assembly designs are evaluated. Each design is categorized into one of four groups based on similarity of geometry, plate thicknesses and compartment payload.

The four groups and the corresponding canisters are as follows:

Group	Grouped Canisters
1	69BTH, 37PTH, 32PTH, 32PTH Type 1, 32PTH1
2	61BT, 61BTH Type 1
3	61BTH Type 2, 32PT, 24PTH
4	24PT4, 24PTH-S-LC

For each group, the bounding payload weight (basket plus fuel assembly) and bounding design configuration are used for the analyses. DSC design features and dimensions are provided in Appendices A.1.4.1 through A.1.4.10. Appendices A.1.4.1 through A.1.4.9 provide detailed descriptions of each DSC. Appendix A.1.4.10 contains reference drawings for all of the above listed DSCs. The following paragraphs highlight the DSC similarities and the bases for the four canister groups.

Group 1

The top and bottom end assembly dimensions for the Group 1 DSCs are given in the following table.

Group 1 DSC Top and Bottom End Assembly Dimensions (in.)

Component	32PTH	32PTH Type 1	32PTH1-S	32PTH1-M	32PTH1-L	69BTH	37PTH
Outer Top Cover	2.00	2.00	2.00	2.00	2.00	2.00	2.00
Inner Top Cover		2.00	2.00	2.00	2.00	2.00	2.00
Top Shield Plug	10.00	8.00	8.00	8.00	6.00	5.75	5.75
Inner Bottom Cover	1.75	2.25	2.25	2.25	2.25	1.75	1.75
Bottom Shield Plug	5.25	4.50	4.50	4.50	2.25	3.50	3.50
Outer Bottom Cover	1.75	2.00	2.00	2.00	2.00	2.00	2.00

For side drop analyses, thinner cover plates result in more limiting (i.e., bounding) stress results Hence, thinner cover plate configuration is used for side drop analyses.

For end drop analyses, the thicker cover plates would result in more limiting stress results due to the additional weight of the thicker plates. Hence thicker cover plate configuration is used for end drop and buckling analyses.

Group 2

The top and bottom end dimensions for the Group 2 DSCs are given in the following table.

Group 2 DSC Top and Bottom End Dimensions (in.)

Component	61BT/61BTH Type 1	61BT/61BTH Type 1 (Optional)
Outer Top Cover	1.25	1.25
Inner Top Cover	0.75	0.75
Top Shield Plug	7.00	7.00
Inner Bottom Cover	0.75	1.75
Bottom Shield Plug	5.00	4.00
Outer Bottom Cover	1.75	1.75

Note that the 61BT/61BTH Type 1 DSC has a thinner inner bottom cover plate than the optional design. Side drop analyses for thinner cover plates result in more limiting stress results than those for the DSC designs with thicker end plates. Therefore, the side drop 3D finite element models used top and bottom end plate thicknesses associated with the thinner inner bottom cover plate design.

The same plate thicknesses are used for the 2D finite element end drop and bucking load evaluation models because the total thickness (and therefore, weight) of each set of end plates is the same for the two designs and give essentially the same limiting results for shell stresses and for shell buckling loads.

Group 3

The top and bottom end dimensions for the Group 3 DSCs are given in the following table.

Group 3 DSC Top and Bottom End Dimensions (in.)

Component	61BTH Type 2	32PT-S100 32PT-L100	32PT-S125 32PT-L125	24PTH
Outer Top Cover	1.50	1.50	1.50	1.50
Inner Top Cover	1.25	1.25	1.25	1.25
Top Shield Plug	6.25	6.25	7.50	6.25
Inner Bottom Cover	1.75	1.75	1.75	1.75
Bottom Shield Plug	4.00	4.00	5.25	4.00
Outer Bottom Cover	1.75	1.75	1.75	1.75

Thinner, conservative cover plate thicknesses are used for the side drop 3D finite element models as follows:

Outer Top Cover: 1.46 in.
Inner Top Cover: 1.19 in.

Inner Bottom Cover: 1.69 in.
Outer Bottom Cover: 1.70 in.

The same cover plate thicknesses and thicker shield plug thicknesses are used for the end drop and buckling load 2D finite element models.

Group 4

The top and bottom end dimensions for the Group 4 DSCs are given in the following table.

Group 4 DSC Top and Bottom End Assembly Dimensions (in.)

Component	**24PT4**	**24PTH-S-LC**
Outer Top Cover	1.25	1.50
Inner Top Forging	2.00	1.74
Lead Plug Top Cover Plate	1.5	0.3
Inner Bottom Cover	2.00	1.86
Outer Bottom Cover Plate	2.00	0.3

The bounding geometry details used in the analyses are as follows:

Top End 3-D Model:

Outer Top Cover Plate = 1.25 in
Inner Top Forging = 1.74 in
Lead Plug Top Cover Plate = 0.3 in

Bottom End 3-D Model:

Bottom Forging = 1.85 in
Outer Bottom Cover Plate = 0.3 in

2-D Axisymmetric Model:

Outer Top Cover Plate = 1.25 in
Inner Top Forging = 1.74 in
Lead Plug Top Cover Plate = 0.30 in
Bottom Forging = 1.85 in
Outer Bottom Cover Plate = 0.30 in

A.2.13.7.2 Dynamic Load Factors and Maximum Drop Accelerations

A. Dynamic Load Factors

Two load cases are considered for development of dynamic load factors and maximum drop accelerations; one due to longitudinal loading (end drop) and one due to transverse loading (side drop). During an end drop, the fundamental natural periods of the DSC components are taken to be that of simply supported cylindrical shells without axial constraint, under longitudinal vibration. During a side drop, the fundamental natural period of the canister shell is taken to be that of a cylinder in an ovalling mode and a simply supported cylindrical shell without axial constraint.

Since the canister is not modeled in detail in the transfer cask dynamic analysis, it is necessary to transfer the loads from the cask dynamic analysis model to the detailed models of the canister. The canisters are evaluated using quasi-static analyses with a dynamic load factor (DLF) computed from the transient dynamic analysis.

The development of the dynamic load factor applicable to each canister type is described in Appendix A.2.13.9. The results are summarized in the following table.

Canister Dynamic Load Factor Results Summary for Each Canister

MP197-HB Transport Package Drop Orientation	**61BT/61BTH/ 61BTHF**	**69BTH**	**24PTHF/ 24PTH**	**24PT4**	**32PT**	**32PTH/ 32PTH1**	**37PTH**
	DLF	DLF	DLF	DLF	DLF	DLF	DLF
Normal Condition Canister End Drop	1.32	1.31	1.32	1.18	1.33	1.31	1.30
Accident Condition Canister End Drop	1.17	1.16	1.16	1.17	1.17	1.16	1.15
Normal Condition Canister Side Drop	1.27	1.29	1.30	1.26	1.30	1.26	1.31
Accident Condition Canister Side Drop	1.01	1.00	1.00	1.01	1.00	1.01	1.02

The bounding dynamic load factor corresponding to each group of canister types is used to bound the analysis of the canister in each group. The bounding DLFs used in each group are summarized in the table below.

Bounding Dynamic Load Factor for Each DSC Group

MP197-HB Transport Package Drop Orientation	**DLF Group 1 DSCs**	**DLF Group 2 DSCs**	**DLF Group 3 DSCs**	**DLF Group4 DSCs**
Normal Condition Canister End Drop	1.31	1.32	1.33	1.32
Accident Condition Canister End Drop	1.16	1.17	1.17	1.17
Normal Condition Canister Side Drop	1.31	1.27	1.30	1.30
Accident Condition Canister Side Drop	1.02	1.01	1.01	1.01

B. Development of Maximum Accelerations

This section calculates the required accelerations for each of the DSC groups and for use in the canister structural evaluations.

The baseline g loads are calculated from LS-DYNA analyses of the transport cask and are described in Appendix A.2.13.12, Section A.2.13.12.10. The results are as follows:

Normal Condition Canister End Drop: 18g
Accident Condition Canister End Drop: 55g

Normal Condition Canister Side Drop: 19g
Accident Condition Canister Side Drop: 55g

Multiplying the above accelerations by the bounding DLFs in each DSC group results in the required accelerations to be used for the DSC analyses. Actual acceleration values used in the analyses are greater than the calculated values. The calculated required acceleration values and the values used in the analyses are shown in the following table. Using 75g for DSC canister HAC end drop and side drop analyses bound all other drop orientations.

Bounding g Loads for Each DSC Group

MP197-HB Transport Package Acceleration	Group 1	Group 2	Group 3	Group 4
Calculated Required Normal Condition Canister End Drop Acceleration	18x1.31= 23.58	18x1.32= 23.76	18x1.33= 23.94	18x1.32= 23.76
Baseline g load for Canister NCT End Drop Acceleration	30	30	30	30
Calculated Required Accident Condition Canister End Drop Acceleration	55x1.16= 63.8	55x1.17= 64.35	55x1.17= 64.35	55x1.17= 64.35
Baseline g load for Canister HAC End Drop Acceleration	75	75	75	75
Calculated Required Normal Condition Canister Side Drop Acceleration	19x1.31= 24.89	19x1.27= 24.13	19x1.30= 24.70	19x1.30= 24.70
Baseline g load for Canister NCT Side Drop Acceleration	30	25	25	25
Calculated Required Accident Condition Canister Side Drop Acceleration	55x1.02= 56.1	55x1.01= 55.55	55x1.01= 55.55	55x1.01= 55.55
Baseline g load for Canister HAC Side Drop Acceleration	75	75	75	75

A.2.13.7.3 Canister Structural Analysis

Finite element analyses are performed in order to quantify stresses in the DSCs generated by transport loads. The applied loads considered are normal and accident condition top end, bottom end, and side drops, combined with internal and external pressures and temperature distributions (thermal expansion stresses). Several finite element models are used to evaluate stresses for the normal and accident loads: 180 degree 3D models are used for side drop analyses; 2D axisymmetric models are used for end drop and thermal expansion analyses. Elastic material properties are used for normal condition stress analyses. Elastic-plastic material properties are used for normal condition limit load analyses and accident condition stress analyses.

A. Material Properties

Steel Material Properties for Group 1 through 4 DSCs

Material properties and allowable stresses for normal (NCT) and accident (HAC) drop analyses are based on 500 °F which bound -40 °F, -20 °F, and 100 °F ambient conditions. Thermal expansion analyses are based on temperature-dependent material properties shown in Tables A.2.13.7-1 and A.2.13.7-2 [3].

For the accident condition side drop cases where elastic-plastic analyses are performed, the tangent modulus is taken as 5% of the elastic modulus.

Lead Material Properties for Group 4 DSCs

The Group 4 DSCs have lead shield plugs instead of steel shield plugs. Material properties for the normal and accident drop analyses are based on 500 °F which bounds the maximum DSC temperatures. For accident condition load cases, dynamic stress-strain properties are used for lead. Material properties of lead are shown in Table A.2.13.7-3 [6], *[7]*.

B. Design Criteria

The steel component stresses are compared with the allowable stresses set forth by ASME B&PV Code Subsection NB [1]. The allowable stress values at 500 °F for the steel components are summarized in Table A2.13.7-4. Closure weld stress allowables are based on ISG-15 [5], which requires a design stress reduction factor to account for weld imperfection or flaws and recommends a stress reduction factor of 0.8 based on multi-level PT examination. The corresponding values at 500 °F are summarized in Table A.2.13.7-5. If the allowable stress limits are exceeded, a simplified fatigue analysis per NB- 3228.5 [1] is performed.

C. Loading Conditions

The load cases considered are normal and hypothetical accident condition drops, pressure loads, and temperature distributions (thermal expansion stresses). The normal condition drop loads are combined with internal and external pressure and the 100° F and -20° F ambient environment thermal loads. The accident condition drop loads are combined with internal and external pressure. The following tables summarize both normal and accident condition DSC individual load cases.

DSC Normal Condition (NCT) Load Cases

Loading	Analysis Type	Service Level	Load	Analysis Method
Hot Environment Thermal Load	Elastic Analysis	A	Hot Ambient	Finite Element Analysis (2D, axisymmetric model)
Cold Environment Thermal Load	Elastic Analysis	A	Cold Ambient	Finite Element Analysis (2D, axisymmetric model)
Internal Pressure	Elastic Analysis	A	Internal Pressure	Finite Element Analysis (included in drop analyses)
External Pressure	Elastic Analysis	A	External Pressure	Finite Element Analysis (included in drop analyses)
1 Foot Side Drop	Elastic Analysis	A	Lateral g-Load	Finite Element Analysis (3D, 180 deg. model)
1 Foot Top End Drop	Elastic Analysis	A	Axial g-Load	Finite Element Analysis (2D, axisymmetric model)
1 Foot Bottom End Drop	Elastic Analysis	A	Axial g-Load	Finite Element Analysis (2D, axisymmetric model)

DSC Accident Condition (HAC) Load Cases

Loading	Analysis Type	Service Level	Load	Analysis Method
30 Foot Side Drop	Elastic-Plastic Analysis	D	Lateral g-Load	Finite Element Analysis (3D, 180 deg. model)
30 Foot Top End Drop	Elastic Analysis	D	Axial g-Load	Finite Element Analysis (2D, axisymmetric model)
30 Foot Bottom End Drop	Elastic Analysis	D	Axial g-Load	Finite Element Analysis (2D, axisymmetric model)

The individual loads are combined as shown in the following tables.

DSC Normal Condition (NCT) Load Combinations

Load Case	Individual Loads						
	25g(1) Side Drop	30g Top End Drop	30g Bottom End Drop	15 psig(2) Internal Pressure	15 psig External Pressure	Hot Ambient Environment	Cold Ambient Environment
1	X			X		X	
2	X				X		X
3		X		X		X	
4		X			X		X
5			X	X		X	
6			X		X		X

*Notes:

1. The Group 1 DSC analyses conservatively used 30g for the normal side drop.
2. The internal pressures used in the analyses are as follow:
   Group 1 DSCs: 30 psig
   Group 2 DSCs: 15 psig
   Group 3 DSCs: 15 psig
   Group 4 DSCs: 20 psig

DSC Accident Condition (HAC) Load Combinations

Load Case	Individual Loads						
	75g Side Drop	75g Top End Drop	75g Bottom End Drop	15 psig(1) Internal Pressure	15 psig External Pressure	Hot Ambient Environment	Cold Ambient Environment
7	X			X		X	
8	X				X		X
9		X		X		X	
10		X			X		X
11			X	X		X	
12			X		X		X

Note:

1. The internal pressures used in the analyses are as follow:
   Group 1 DSCs: 30 psig
   Group 2 DSCs: 15 psig
   Group 3 DSCs: 15 psig
   Group 4 DSCs: 20 psig

D. Finite Element Analysis

Finite Element Model

Finite element models are constructed to evaluate stresses for the normal and accident loads using ANSYS computer program [4]. A separate set of models is used for side drop analyses of the top end and bottom end of each DSC group. For side drop load combinations, 180 degree 3D models are used. For end drop load combinations and thermal expansion stresses, 2D axisymmetric models are used. The models for end drop loading are extended to include the full length of the DSC.

The 3D finite element models are developed using SOLID45 solid elements and 3D point-to-point CONTA178 and CONTAC52 contact elements. Contact between cover plates and shield plugs are modeled using CONTA178 elements. The initial gaps between these components are considered to be closed. In addition to contact elements at the interface of the cylindrical shell and shield plates, or between plates, the models include CONTA178 contact elements at the interface of the cylindrical shell to the cask. The nodes of these elements are located at the OD of the cylindrical shell and ID of the cask. The initial locations of these nodes define the nominal centered gap between cask and canister. The gaps at cask rails, and between the canister and the cask, are input by contact element real constants. The contact element nodes located at the ID of the cask are held fixed in all directions, simulating a rigid cask on which the canister drops.

For end drop load cases and thermal expansion stresses, 2D axisymmetric finite element models are developed using ANSYS PLANE42 elements. Contact between cover plates and shield plugs are modeled using CONTAC12 elements.

The welds between the shell and inner cover plates are modeled by coupling the contacting nodes in all directions. Welds between the shell and the outer top cover plate and between shell and outer bottom cover are modeled with PLAN42 elements.

Symmetry boundary conditions are defined for all nodes at symmetry planes.

Geometry plots of the finite element analytical models are given in Figures A.2.13.7-1 through A.2.13.7-8.

Load Cases

Accelerations used in the analyses are provided in Section A.2.13.7.3. In general, accelerations of 25g, 30g and 75g are defined for the normal side drop, normal end drop, and accident drops (side and end), respectively, in the appropriate direction for each of the drop conditions. The exception to this is for the Group 1 DSCs, where a normal side drop acceleration of 30g was conservatively used (see Section A.2.13.7.3). Load cases used in the analyses are shown in the following tables.

Load Cases for Side Drop Normal Condition of Transport (NCT)

Load Case Number**	Loading Condition	Service Level	Case Description
1NCT (1)	Top End DSC Model Lateral Load + Internal Pressure (Top Norails IP)	A	DSC in Cask, horizontal, supported on side, Impact away from transport cask rails.
2NCT (2)	Top End DSC Model Lateral Load + External Pressure (Top Norails EP)	A	DSC in Cask, horizontal, supported on side, Impact away from transport cask rails.
3NCT (1)	Top End DSC Model Lateral Load + Internal Pressure (Top Rails IP)	A	DSC in Cask, horizontal, supported on side, Impact onto the two transport cask rails.
4NCT (2)	Top End DSC Model Lateral Load + External Pressure (Top Rails EP)	A	DSC in Cask, horizontal, supported on side, Impact onto the two transport cask rails.
5NCT (1)	Bottom End DSC Model Lateral Load + Internal Pressure (Bottom Norails IP)	A	DSC in Cask, horizontal, supported on side, Impact away from transport cask rails.
6NCT (2)	Bottom End DSC Model Lateral Load + External Pressure (Bottom Norails EP)	A	DSC in Cask, horizontal, supported on side, Impact away from transport cask rails.
7NCT (1)	Bottom End DSC Model Lateral Load + Internal Pressure (Bottom Rails IP)	A	DSC in Cask, horizontal, supported on side, Impact onto the two transport cask rails.
8NCT (2)	Bottom End DSC Model Lateral Load + External Pressure (Bottom Rails EP)	A	DSC in Cask, horizontal, supported on side, Impact onto the two transport cask rails.

Note:
** Number in ( ) represents the DSC normal condition load combination number shown in previous table.

Load Cases for Side Drop Hypothetical Accident Condition (HAC)

Load Case Number**	Loading Condition	Service Level	Case Description
1HAC (7)	Top End DSC Model Lateral Load + Internal Pressure (Top Norails IP)	D	DSC in Cask, horizontal, supported on side, Impact away from transport cask rails.
2HAC (8)	Top End DSC Model Lateral Load + External Pressure (Top Norails EP)	D	DSC in Cask, horizontal, supported on side, Impact away from transport cask rails.
3HAC (7)	Top End DSC Model Lateral Load + Internal Pressure (Top Rails IP)	D	DSC in Cask, horizontal, supported on side, Impact onto the two transport cask rails.
4HAC (8)	Top End DSC Model Lateral Load + External Pressure (Top Rails EP)	D	DSC in Cask, horizontal, supported on side, Impact onto the two transport cask rails.
5HAC (7)	Bottom End DSC Model Lateral Load + Internal Pressure (Bottom Norails IP)	D	DSC in Cask, horizontal, supported on side, Impact away from transport cask rails.
6HAC (8)	Bottom End DSC Model Lateral Load + External Pressure (Bottom Norails EP)	D	DSC in Cask, horizontal, supported on side, Impact away from transport cask rails.
7HAC (7)	Bottom End DSC Model Lateral Load + Internal Pressure (Bottom Rails IP)	D	DSC in Cask, horizontal, supported on side, Impact onto the two transport cask rails.
8HAC (8)	Bottom End DSC Model Lateral Load + External Pressure (Bottom Rails EP)	D	DSC in Cask, horizontal, supported on side, Impact onto the two transport cask rails.

Note:
** Number in ( ) represents the DSC accident condition load combination number shown in previous table.

Load Cases for End Drop Normal Condition of Transport (NCT)

Load Case Number**	Loading Condition	Service Level	Case Description
1NCT(ED) (3)	Lid End Drop, Axial Load + Internal Pressure (Top End Drop IP)	A	Cask vertical, supported at top, axial acceleration + internal pressure.
2NCT(ED) (4)	Lid End Drop, Axial Load + External Pressure (Top End Drop EP)	A	Cask vertical, supported at top, axial acceleration + external pressure.
3NCT(ED) (5)	Bottom End Drop, Axial Load + Internal Pressure (Bottom End Drop IP)	A	Cask vertical, supported at bottom, axial acceleration + internal pressure.
4NCT(ED) (6)	Bottom End Drop, Axial Load + External Pressure (Bottom End Drop EP)	A	Cask vertical, supported at bottom, axial acceleration + external pressure.

Note:
** Number in ( ) represents the DSC normal condition load combination number shown in previous table.

Load Cases for End Drop Accident Condition of Transport (HAC)

Load Case Number**	Loading Condition	Service Level	Case Description
1HAC(ED) (9)	Lid End Drop, Axial Load + Internal Pressure (Top End Drop IP)	D	Cask vertical, supported at top, axial acceleration + internal pressure.
2HAC(ED) (10)	Lid End Drop, Axial Load + External Pressure (Top End Drop EP)	D	Cask vertical, supported at top, axial acceleration + external pressure
3HAC(ED) (11)	Bottom End Drop, Axial Load + Internal Pressure (Bottom End Drop IP)	D	Cask vertical, supported at bottom, axial acceleration + internal pressure.
4HAC(ED) (12)	Bottom End Drop, Axial Load + External Pressure (Bottom End Drop EP)	D	Cask vertical, supported at bottom, axial acceleration + external pressure.

Note:
** Number in ( ) represents the DSC accident condition load combination number shown in previous table.

For the side drop and end drop analyses, the weight of the internals (basket + fuel assemblies) is accounted for by applying a pressure to the inner surface of the canister. Bounding weight is used for the analysis of each DSC group. The internal weights used for the analyses are listed in the following table.

	Group 1 DSCs	Group 2 DSCs	Group 3 DSCs	Group 4 DSCs
Bounding Calculated Internal Weight for the DSCs in the Group	85,000 lb.	67,000 lb.	77,000 lb.	68,000 lb.
Internal Weight used for the Canister Analysis in Each Group	90,000 lb.	80,000 lb.	80,000 lb.	90,000 lb.

E. Side Drop Load Analysis

For side drop load cases away from transfer cask rails (1NCT, 2NCT, 5NCT, 6NCT, 1HAC, 2HAC, 5HAC & 6HAC), inertia loads for canister internals is accounted for by applying a cosine varying pressure on the inside surface of the canister shell. Assuming that the canister internals react upon a 90° arc of the inside surface, then the inertial load of the internals, $P_{(\theta)}$, which varies with angle, θ, (θ = 0 is at the impact point), is governed by the following expression:

$$P_{(\theta)} = P_{max}\cos(2\theta) \qquad (-45^{\circ} < \theta < 45^{\circ})$$

Where P_{max} is the maximum pressure at the impact point (θ = 0). Assuming the axial length of the applied load is L, the inside radius of the canister shell is R, and the load distribution, $P_{(\theta)}$ above, then the total inertial load in the drop direction generated by the internals, F, is the following:

$$F = \int_{\frac{-\pi}{4}}^{\frac{\pi}{4}} P_{\max}\cos(2\theta)\cos(\theta)LRd\theta$$

$$\Rightarrow \qquad F = \frac{P_{\max}LR}{2}\int_{\frac{-\pi}{4}}^{\frac{\pi}{4}}(\cos((2+1)\theta)+\cos((2-1)\theta))d\theta$$

By integrating the equation above we get the following.

$$F = \left[\frac{P_{\max}LR}{2}\right]\left[\frac{\sin(3\theta)}{3}+\sin(\theta)\right]_{\frac{-\pi}{4}}^{\frac{\pi}{4}}$$

Therefore,

$$F = \left[\frac{P_{\max}LR}{2}\right]\left[\frac{\sin\left(\frac{3\pi}{4}\right)}{3}+\sin\left(\frac{\pi}{4}\right)-\frac{\sin\left(\frac{-3\pi}{4}\right)}{3}-\sin\left(\frac{-\pi}{4}\right)\right]$$

$$F = P_{\max}LR\left[\frac{\sin\left(\frac{3\pi}{4}\right)}{3}+\sin\left(\frac{\pi}{4}\right)\right]$$

An example calculation for the Group 1 DSCs is provided below:

The canister shell inner diameter, R = 34.375. The axial length of the applied load (basket length), L = 162 in.

Total weight of canister internals of 90,000 lb. (basket + fuel assemblies) is used for the calculation.

$F = 90{,}000 \times 30g = 2{,}700{,}000$ [for NCT]
$F = 90{,}000 \times 75g = 6{,}750{,}000$ [for HAC]

Therefore, P_{max} for Normal Condition of Transport (NCT) is:

$$P_{max} = \frac{2700000}{(162.00)(34.375)}\left[\frac{\sin\left(\frac{3\pi}{4}\right)}{3} + \sin\left(\frac{\pi}{4}\right)\right]^{-1} = 514.3 \text{ psi}$$

Therefore, P_{max} for Hypothetical Accident Condition (HAC) is:

$$P_{max} = \frac{6750000}{(162.00)(34.375)}\left[\frac{\sin\left(\frac{3\pi}{4}\right)}{3} + \sin\left(\frac{\pi}{4}\right)\right]^{-1} = 1285.7 \text{ psi}$$

Therefore, the equivalent pressure applied on the canister inside shell surface for load cases away from transfer cask rails is: $P_{(\theta)} = 514.3 \cos(2\theta)$ and $1285.7 \cos(2\theta)$, respectively, for the Group 1 DSC NCT and HAC load cases where θ = angle from the bottom ($\theta = 0$ for a half symmetric model) of the horizontal canister shell to the center of the finite element model, up to 45°.

For side drop load cases onto the two transfer cask rails, inertia loads for the basket assembly are accounted for by applying an equivalent pressure onto the first (or innermost) rail only. The base value of pressure, at 1g load, is obtained after doing a few iterations with only pressure load on the first rail (simulating the basket assembly weight) and then checking the reaction loads in appropriate direction. The value obtained for the Group 1 DSCs is 93.6 psi which is then multiplied by the appropriate g loads.

F. End Drop Load Analysis

The weight of the canister internals (basket and fuel assemblies) during end drop is accounted for by applying equivalent pressures on the supporting surfaces of the canister components. For example, the weight of the canister internals used in the end drop analyses for the Group 2 DSCs is conservatively taken to be 80,000 lb. The corresponding pressure loads equivalent to the inertial load of the internals at 30g and 75g for the NCT and HAC end drops are:

$P = 23.2075 \times 30g = 696.22$ psi [For NCT at 30g]

$P = 23.2075 \times 75g = 1740.56$ psi [For HAC at 75g]

For end drop buckling analyses, where g-loads exceed 75 g, the g values and corresponding canister internals loads are appropriately increased according to the formula shown above for accelerations beyond the 75 g load.

G. Internal and External Pressure Analyses

Internal and external pressures are applied to the appropriate surfaces of the cylindrical shell and cover plates using ANSYS pressure loading on the solid element surfaces.

H. Temperature (Thermal Expansion) Analysis

Temperature distributions, from Chapter A.3, are conservatively mapped onto the DSC structural models and stresses due to temperature distributions are calculated..

A.2.13.7.4 Stress Analysis Results

The maximum stress intensities in the canister are extracted from the ANSYS results, for the applicable load combinations. These stresses are compared to the normal and accident condition code allowable stress intensities.

The following notation is applicable to the tables of this Section:

Top:	Top End Model
Bot:	Bottom End Model
IP:	Loading with internal pressure
EP:	Loading with external pressure
No-P:	Loading without pressure
NoRails:	Side drop away from cask rails
Rails:	Side drop on cask rails
NCT:	Loading with normal side drop acceleration
HAC:	Loading with accident side drop acceleration
TSA:	Bounding thermal stress analysis
S.No.:	Section or component ID
ED:	End Drop

A.2.13.7.4.1 Group 1 DSC Stress Analysis Results

A. NCT Side Drop Results – Group 1 DSCs

A.1 Top End Model Stress Evaluation

The following tables summarize the linearized bounding stress for the main DSC components for the NCT side drop load combinations stress results for the Group 1 DSCs Top End Model.

NCT Loads Maximum Stress Intensities for Group 1 DSCs - Top End Model
(Away From Impact Zone)

Service Level A: Top End Model			
S.No.	**Component**	**P_m (17.5 ksi)**	**$P_m + P_b$ (26.3 ksi)**
1	Cylindrical Shell	14.4	24.5
2	Outer Top Cover Plate	4.8	9.1
3	Inner Top Cover Plate	5.7	10.2
4	Shield Plug	4.4	4.9
5	Support Ring	8.8	12.8
Note: The corresponding allowable stress limits are shown in brackets			

NCT Loads Maximum Stress Intensities for Group 1 DSCs - Top End Model
(At the Impact Zone)

Service Level A: Top End Model(1)			
S.No.	**Component**	**P_L (26.3 ksi)**	**$P_L + P_b + Q$ (52.5 ksi)**
1	Cylindrical Shell	31.1(2)	35.0
2	Outer Top Cover Plate	10.9	25.6
3	Inner Top Cover Plate	12.7	26.9
4	Shield Plug	14.2	23.8
5	Support Ring	7.0	10.2
Note: The corresponding allowable stress limits are shown in the brackets			

Notes:

1. The maximum primary membrane and primary membrane plus bending stresses listed above for the shell and the cover plates are located at the impact zone at the junction of the shell and the cover plates. In accordance with the provisions of Table NB-3217-1 (and Figure NB-3222-1), the primary membrane stresses are classified as local membrane (P_L) stresses. The membrane plus bending stresses are classified as local primary membrane plus secondary bending stresses ($P_L + P_b + Q$). Q is secondary stress due to mechanical load only. Secondary stress due to thermal will be combined separately to obtain the maximum stress intensity.
2. Limit analysis for a bounding case is performed and is presented in Section A.2.13.7.4.5.

Bounding Thermal Stress Intensity for Group 1 DSCs - Top End Model

Load Case	Bounding Thermal Stress Intensity (ksi)
	Top End Model
TSA-Group 1 DSCs	5.0 ksi

Stresses Results Away From the Impact Zone

All the maximum membrane stress intensities and maximum membrane plus bending stress intensity are within the code allowable membrane and membrane plus bending stress intensity limits.

The maximum membrane plus bending stress intensity is also combined with the bounding thermal stress intensity as follows:

$$P_m + P_b + Q = 24.5 + 5.0 = 29.5 \text{ ksi} \leq 3\ S_m = 52.5 \text{ ksi}$$

Therefore, the combined stress intensity meets the code allowable stresses.

Stresses Results at the Impact Zone

All of the local maximum membrane stress intensities and the local membrane plus secondary bending stress intensities are within the allowable stress intensity limits except the maximum local membrane stress intensity for the shell component, which occurs at the cylindrical shell and outer top cover plate junction. The limit analysis provisions of NB-3228.1 are used in accordance with NB-3222. Limit analysis for a bounding case is performed and is presented in Section A.2.13.7.4.5.

The maximum local membrane plus secondary bending stress intensity is also combined with the bounding thermal stress intensity as follows:

$$(P_m + P_b + Q_{mechanical\ \&\ pressure}) + Q_{thermal} = 35.0 + 5.0 = 40.0 \text{ ksi} \leq 3\ S_m = 52.5 \text{ ksi}$$

Therefore, the combined stress intensity meets the code allowable stresses.

A.2 Bottom End Model Stress Evaluation

The following table summarizes the linearized bounding stress for the main DSC components for the NCT side drop load combinations stress results for the Group 1 DSCs Bottom End Model. For the bottom end model, the stresses are lower than the stresses from the Top End Model, therefore only the maximum stress results at the impact zone are reported.

NCT Loads Maximum Linearized Stress Intensities for Group 1 DSCs - Bottom End Model
(At the Impact Zone)

Service Level A: Bottom End Model[1]			
S.No.	Component	P_L (26.3 ksi)	$P_L + P_b + Q$ (52.5 ksi)
1	Cylindrical Shell	26.1	34.9
2	Outer Bottom Cover Plate	12.0	15.8
3	Inner Bottom Cover Plate	14.2	17.5
4	Shield Plug	9.6	15.5
Note: The corresponding allowable stress limits are shown in brackets			

Note: 1. The maximum primary membrane and primary membrane plus bending stresses listed above for the shell and the cover plates are located near the impact zone at the junction of the shell and the cover plates. In accordance with the provisions of Table NB-3217-1 (and Figure NB-3222-1), the primary membrane stresses are classified as local membrane (P_L) stresses. The membrane plus bending stresses are classified as local primary membrane plus secondary bending stresses ($P_L + P_b + Q$).

Bounding Thermal Stress Intensity for Group 1 DSCs - Bottom End Model

Load Case	Bounding Thermal Stress Intensity (ksi)
	Bottom End Model
TSA-Group 1 DSCs	20.5 ksi

Stresses Results at the Impact Zone

All of the local maximum membrane stress intensities are within the allowable local membrane stress intensity limit.

The maximum local membrane plus secondary bending stress intensity is also combined with the bounding thermal stress intensity as follows:

$$(P_m + P_b + Q) + Q_{thermal} = 34.9 + 20.5 = 55.4 \text{ ksi} \geq 3\ S_m = 52.5 \text{ ksi}$$

Since the above stress intensity exceeds the $3S_m$ limit, the guidelines provided in NB-3228.5 Simplified Elastic-Plastic Analysis [1] are invoked to qualify the bottom end shell stress due to NCT side drop load cases.

For purposes of applying the provisions of the NB-3228.5, the thermal stresses in the shell are linearized in order to obtain the maximum primary membrane stress intensity. The maximum primary thermal membrane stress intensity is 7.4 ksi.

Application of the provisions of NB-3228.5 Simplified Elastic-Plastic Analysis [1]

The $3S_m$ limit on the range of the primary plus secondary stress intensity may be exceeded provide the requirements of (a) through (f) below are met.

(a) The range of primary plus secondary membrane plus bending stress intensity, excluding thermal bending stresses, shall be $\leq 3S_m$.

Check: Structural (Membrane + Bending) + Thermal (Membrane) = 34.9 + 7.4

$$= 42.3 \text{ ksi} \leq 3 \text{ Sm} = 52.5 \text{ ksi}$$

(b) The value of S_a used for entering the design fatigue curve is multiplied by the factor K_e.

Where:

$K_e = 1.0$ For $S_n < 3\ S_m$

or

$K_e = 1.0 + [(1\text{-}n)/(n(m\text{-}1))][(S_n/(3S_m)) - 1]$ For $3\ S_m < S_n < 3\ (m)\ S_m$

S_n = range of primary plus secondary stress intensity (both S_n and S_a will used in step "c" below)

Check: $S_n = (P_m + P_b + Q) + Q_{thermal} = 34.9 + 20.5 = 55.4$ ksi < 3 (m) $S_m = 89.3$ ksi

(where m = 1.7; refer to [1] Table NB-3228.5(b)-1, page 65)

Since S_n (55.4 ksi) is less than 3 (m) S_m (89.3 ksi) but greater than $3S_m$ (52.5 ksi)

Therefore,

$K_e = 1.0 + [(1\text{-}n)/(n(m\text{-}1))][(S_n/(3S_m)) - 1]$ is used

(where n = 0.3; refer to [1] Table NB-3228.5(b)-1, page 65)

$K_e = 1.0 + [(1\text{-}0.3)/((0.3)(1.7\text{-}1))][(55.4/52.5) - 1] = 1.18$

(c) The rest of the fatigue evaluation stays the same as required in NB-3222.4, except that the procedure of NB-3227.6 need not be used.

Check: A simplified and conservative fatigue evaluation is performed in consideration of the maximum full range of pressure and thermal primary plus secondary stress intensity for 10 cycles. The actual number of cycles for this full range of loading is much less (conservatively assumed 10 cycles over the life of the DSC). Higher cycle loadings are due to variations in ambient temperatures and vibrations during handling and develop much lower stresses (below the material endurance limit), such that the contribution to the cumulative usage factor for these loadings is negligible.

Using conservative values of $K_f = 4$; $K_e = 1.18$; and $S_n = 55.4$ ksi:

$S_a = K_f K_e (S_n / 2) E_{fc} / E$

$E_{fc} = 30{,}000$ ksi (Figure I-9.1 of [2]); $E_a = 25{,}800$ ksi (at 500 °F)

So, $S_a = 4(1.18)(55.4/2)(30000/25800) = 152.0$ ksi

From the design fatigue curve in Figure I-9.1 of [2] and NB-3224.5, the allowable number of cycles, N, for this value of S_a is approximately 300 cycles and the cumulative usage factor is conservatively calculated as follows:

$$U = 10 / 300 = 0.03 \ll 1.0$$

(d) The component meets the thermal ratcheting requirement of NB-3222.5.

Check: Overall thermal ratcheting can only occur with cyclic loading. The thermally induced stresses in this analysis are due to a single occurrence; therefore no thermal ratcheting is possible

(e) The temperature does not exceed those listed in Table NB-3228.5(b)-1 for the various classes of materials.

Check: The maximum temperature of the canister is much less than 800 °F listed in Table NB-3228.5(b).

(f) The material shall have a specified minimum yield strength to specified minimum tensile strength ratio of less than 0.80.

Check: The maximum $S_y/S_u = 30/75 = 0.40$ at 100 °F which is less than 0.80.

Based on the results of the above analysis, the design meets the requirements of the NB-3228.5 Simplified Elastic-Plastic Analysis.

A.3 Weld Stresses Due to NCT Side Drop Loads – Group 1 DSCs

The canister design incorporates two closure welds (top end model). One weld joins the cylindrical shell and the outer top cover plate, the other one joins the cylindrical shell and the inner top cover plate.

For the cylindrical shell and outer top cover plate weld, the maximum stress along with the corresponding limits are shown in the following table. Weld stresses are well below the corresponding stress limits.

Weld Stress - Cylindrical Shell & Outer Top Cover Plate (NCT) – Group 1 DSCs

Service Level A: Normal Condition of Transport (30g)			
		Weld Stress [ksi]	
Load Case	Load Case Detail	At Impact Zone (42.0 ksi)	Away from Impact Zone (21.0 ksi)
1NCT	Top Norails IP	24.8	12.0
2NCT	Top Norails EP	21.8	3.6
3NCT	Top Rails IP	22.2	10.5
4NCT	Top Rails EP	8.9	0.7
Note: The corresponding stress limits are shown in the brackets.			

The weld between the cylindrical shell and inner top cover plate has been modeled as a line weld by coupling the nodes. To determine the maximum weld stress for this weld, the following procedure is used:

Step 1: All nodes belonging to the weld coupling and inner top cover plate are selected. Nodal loads for all the nodes are listed in cylindrical coordinate system.

Step 2: All compressive radial stresses are ignored as the stresses of interest for the weld are tension and shear.

Step 3: The maximum resultant force is calculated using $F_{resultant} = \sqrt{(f_x^2 + f_y^2 + f_z^2)}$.

Step 4: The maximum weld stress is calculated next using

$$f_w = F_{resultant} / (L_{tributary}) (T_{weld})$$

where $L_{tributary}$ = minimum tributary length associated with the node
= π (34.375)/ (number of nodes -1)
= 1.80
T_{weld} = Weld Throat = 3/16.

The weld stress (f_w) is compared to the corresponding limits.

The table below summarizes the weld stresses for the weld between the cylindrical shell and the inner top cover plate. Weld stresses are well below the corresponding stress limits.

Weld Stress – Cylindrical Shell & Inner Top Cover Plate (NCT) – Group 1 DSCs

Service Level A: Normal Condition of Transport (30g)			
		Weld Max. Stress [ksi]	
Load Case	**Load Case Detail**	**At Impact Zone (42.0 ksi)**	**Away from Impact Zone (21.0 ksi)**
1NCT	Top Norails IP	15.95	12.98
2NCT	Top Norails EP	10.6	12.1
3NCT	Top Rails IP	6.3	1.5
4NCT	Top Rails EP	4.8	1.5
Note: The corresponding stress limits are shown in the brackets			

B. HAC Side Drop Results – Group 1 DSCs

The following tables summarize the HAC side drop load combinations stress results for the Group 1 DSCs. The stresses reported in this table are the maximum stress intensity (P_m+P_b) irrespective of the impact location. All stresses are well below the corresponding stress limits.

Stress Intensities for Side Drop HAC – Group 1 DSCs

Service Level D: Hypothetical Accident Condition (75g)				
		Max. Stress Intensity	**Stress Limits [ksi]**	
Load Case	**Load Case Detail**	**[ksi]**	P_m	P_m+P_b
1HAC	Top Norails IP	37.6	44.4	57.1
2HAC	Top Norails EP	38.7	44.4	57.1
3HAC	Top Rails IP	38.6	44.4	57.1
4HAC	Top Rails EP	39.9	44.4	57.1
5HAC	Bottom Norails IP	23.6	44.4	57.1
6HAC	Bottom Norails EP	25.5	44.4	57.1
7HAC	Bottom Rails IP	24.8	44.4	57.1
8HAC	Bottom Rails EP	22.5	44.4	57.1

B.1 Weld Stress Due to HAC Side Drop Loads – Group 1 DSCs

The following table summarizes the HAC side drop maximum weld stress results for the Group 1 DSCs. The maximum weld stresses are occurred at shell and outer top cover plate; therefore only weld stresses at these locations are reported. Weld stresses are below the corresponding stress limits.

Weld Stress - Side Drop HAC – Group 1 DSCs

Service Level D : Hypothetical Accident Condition (75g)			
Load Case	**Load Case Detail**	**Weld Max. Stress Intensity [ksi]**	**Weld Stress Limits [ksi]**
1HAC	Top Norails IP	31.7	35.5
2HAC	Top Norails EP	32.6	35.5
3HAC	Top Rails IP	33.7	35.5
4HAC	Top Rails EP	34.3	35.5

C. NCT End Drop Results – Group 1 DSCs

The following table summarizes the NCT end drop stress results for the Group 1 DSCs. The stresses reported in this table are the maximum stress intensity irrespective of the impact location. All stresses are within the allowable stress values.

Stress Intensities for End Drop NCT – Group 1 DSCs

Service Level A: End Drop Normal Condition of Transport (30g)					
			Stress Limits [ksi]		
Load Case	**Load Case Detail**	**Max. Stress Intensity [ksi]**	**P_m (S_m)**	**$P_m + P_b$ (1.5 S_m)**	**$P_m + P_b + Q$ (3.0 S_m)**
1NCT(ED)	Top End Drop IP	6.7	17.5	26.3	52.5
2NCT(ED)	Top End Drop EP	12.5	17.5	26.3	52.5
3NCT(ED)	Bottom End Drop IP	8.0	17.5	26.3	52.5
4NCT(ED)	Bottom End Drop EP	11.0	17.5	26.3	52.5

The maximum stress intensity is 12.5 ksi for Group 1 DSCs. Combining this value with the bounding thermal stress intensity of 20.5 ksi, the bounding $P_m + P_b + Q = 12.5 + 20.5 = 33.0$ ksi $\leq$ 52.5 ksi.

D. HAC End Drop Results – Group 1 DSCs

Stress intensities for end drop HAC load cases for the Group 1 DSCs are summarized in the following table. The stresses reported in this table are the maximum stress intensity irrespective to the impact locations. All stresses are within the allowable stress values.

Stress Intensities for End Drop HAC – Group 1 DSCs

Service Level D : End Drop Hypothetical Accident Condition (75g)				
Load Case	**Load Case Detail**	**Max. Stress Intensity [ksi]**	**Stress Limits [ksi]**	
			P_m	**P_m+P_b**
1HAC(ED)	Top End Drop IP	18.4	44.4	57.1
2HAC(ED)	Top End Drop EP	27.6	44.4	57.1
3HAC(ED)	Bottom End Drop IP	22.5	44.4	57.1
4HAC(ED)	Bottom End Drop EP	25.5	44.4	57.1

E. HAC End Drop Buckling Results – Group 1 DSCs

For the end drop hypothetical accident condition (HAC) load cases, the input load is increased beyond 75g until convergence is no longer achieved for the canister assembly model. The following table summarizes the allowable bucking loads for various load cases. The lowest allowable buckling load is 172g and the accident g load for group 1 DSCs is 56g, this gives a safety factor of 3.1.

HAC End Drop Allowable Buckling Loads - Group 1 DSCs

Load Case	Load Case Detail	Buckling Load (g)
1HAC(ED)	Top End Drop IP	216
2HAC(ED)	Top End Drop EP	216
3HAC(ED)	Bottom End Drop IP	173
4HAC(ED)	Bottom End Drop EP	172

A.2.13.7.4.2 Group 2 DSC Stress Analysis Results

A. NCT Side Drop Results – Group 2 DSCs

A.1 Top End Model Stress Evaluation

The following tables summarize the linearized bounding stress intensities for the main DSC components for the NCT side drop load combination for the Group 2 DSCs Top End Model

NCT Loads Maximum Stress Intensities for Group 2 DSCs - Top End Model
(Away From Impact Zone)

Service Level A: Top End Model			
S.No.	**Component**	**P_m (17.5 ksi)**	**$P_m + P_b$ (26.3 ksi)**
1	Cylindrical Shell	12.7	24.3
2	Outer Top Cover Plate	6.4	12.4
3	Inner Top Cover Plate	7.1	12.5
4	Shield Plug	3.6	3.9
5	Support Ring	7.3	11.6
Note: The corresponding stress limits are shown in the brackets.			

NCT Loads Maximum Stress Intensities for Group 2 DSCs - Top End Model
(At Impact Zone)

Service Level A: Top End Model(1)			
S.No.	**Component**	**P_L (26.3 ksi)**	**$P_L + P_b + Q$ (52.5 ksi)**
1	Cylindrical Shell	26.7(2)	34.5
2	Outer Top Cover Plate	13.7	29.4
3	Inner Top Cover Plate	14.9	31.7
4	Shield Plug	10.9	17.8
5	Support Ring	7.3	11.6
Note: The corresponding stress limits are shown in the brackets.			

Notes:

1. The maximum primary membrane and primary membrane plus bending stresses listed above for the shell and the cover plates are located at the impact zone at the junction of the shell and the cover plates. In accordance with the provisions of Table NB-3217-1 (and Figure NB-3222-1), the primary membrane stresses are classified as local membrane (P_L) stresses. The membrane plus bending stresses are classified as local primary membrane plus secondary bending stresses ($P_L + P_b + Q$).
2. Limit analysis for a bounding case is performed and is presented in Section A.2.13.7.4.5.

Bounding Thermal Stress Intensity for Group 2 DSCs - Top End Model

Load Case	Bounding Thermal Stress Intensity (ksi)
	Top End Model
TSA-Group 2 DSCs	5.0

Stresses Results Away From the Impact Zone

All the maximum membrane stress intensities and maximum membrane plus bending stress intensities are within the code allowable membrane and membrane plus bending stress intensity limits.

The maximum membrane plus bending stress intensity is also combined with the bounding thermal stress intensity as follows:

$$P_m + P_b + Q = 24.3 + 5.0 = 29.3 \text{ ksi} \leq 3\ S_m = 52.5 \text{ ksi}$$

Therefore, the combined stress intensity meets the code allowable stresses.

Stresses Results at the Impact Zone

All of the local maximum membrane stress intensities and local membrane plus secondary bending stresses are within the allowable stress intensity limits except the maximum stress intensity at the shell component, which occurs at the cylindrical shell and outer top cover plate junction. The limit analysis provisions of NB-3228.1 are used in accordance with NB-3222. Limit analysis for a bounding case is performed and is presented in Section A.2.13.7.4.5.

The maximum local membrane plus secondary bending stress intensity is combined with the bounding thermal stress intensity as follows:

$$(P_m + P_b + Q_{\text{mechanical \& pressure}}) + Q_{\text{thermal}} = 34.5 + 5.0 = 39.5 \text{ ksi} \leq 3\ S_m = 52.5 \text{ ksi}$$

Therefore, the combined stress intensity meets the code allowable stresses.

A.2 Bottom End Model Stress Evaluation

The following table summarizes the linearized bounding stress for the main DSC components for the NCT side drop load combinations stress results for the Group 2 DSCs Bottom End Model. For the bottom end model, the stresses are lower than the stresses from the Top End Model, therefore only the maximum stress results at the impact zone are reported.

NCT Loads Maximum Stress Intensities for Group 2 DSCs – Bottom End Model

Service Level A: Bottom End Model[1]			
S.No.	**Component**	**P_L (26.3 ksi)**	**$P_L + P_b + Q$ (52.5 ksi)**
1	Cylindrical Shell	17.3	26.3
2	Outer Bottom Cover Plate	12.5	15.7
3	Inner Bottom Cover Plate	15.4	18.8
4	Shield Plug	8.0	10.2
Note: The corresponding stress limits are shown in the brackets.			

Note: 1. The maximum primary membrane and primary membrane plus bending stresses listed above for the shell and the cover plates are located near the impact zone at the junction of the shell and the cover plates. In accordance with the provisions of Table NB-3217-1 (and Figure NB-3222-1), the primary membrane stresses are classified as local membrane (P_L) stresses. The membrane plus bending stresses are classified as local primary membrane plus secondary bending stresses ($P_L + P_b + Q$).

Bounding Thermal Stress Intensity for Group 2 DSCs Bottom End Model

Load Case	Bounding Thermal Stress Intensity (ksi) Bottom End Model
TSA-Group 2 DSCs	7.5

Stresses Results at the Impact Zone

All of the local maximum membrane stress intensities are within the allowable local membrane stress intensity limit.

The maximum local membrane plus secondary bending stress intensity is combined with the bounding thermal stress intensity as follows:

$$(P_m + P_b + Q_{mechanical\ \&\ pressure}) + Q_{thermal} = 26.3 + 7.5 = 33.8 \text{ ksi} \leq 3\ S_m = 52.5 \text{ ksi}$$

Therefore, the combined stress intensity meets the code allowable stresses.

A.3 Weld Stresses Due to NCT Side Drop Loads – Group 2 DSCs

The Group 2 DSC weld stresses are evaluated in the same way as those in Group 1 (in Section A.2.13.7.4.1).

For the cylindrical shell and outer top cover plate weld, the maximum stress intensities along with the corresponding limits are shown in the following table.

Weld Stress Intensity- Cylindrical Shell & Outer Top Cover Plate (NCT) – Group 2 DSCs

Service Level A: Normal Condition of Transport (25g)			
		Weld Max. Stress Intensity [ksi]	
Load Case	**Load Case Detail**	**At Impact Zone Stress Limit (42.0 ksi)**	**Away from Impact Zone Stress Limit (21.0 ksi)**
1NCT	Top Norails IP	32.2	16.3
2NCT	Top Norails EP	16.1	3.5
3NCT	Top Rails IP	34.2	16.4
4NCT	Top Rails EP	6.4	0.6
Note: The corresponding stress limits are shown in the brackets			

The maximum stresses for the weld between the cylindrical shell and inner top cover plate along with the corresponding limits, are shown in the following table.

Weld Stress – Cylindrical Shell & Inner Top Cover Plate (NCT) – Group 2 DSCs

Service Level A: Normal Condition of Transport (25g)			
		Weld Max. Stress [ksi]	
Load Case	Load Case Detail	At Impact Zone Stress Limit (42.0 ksi)	Away from Impact Zone Stress Limit (21.0 ksi)
1NCT	Top Norails IP	8.98	8.92
2NCT	Top Norails EP	6.92	6.60
3NCT	Top Rails IP	4.43	3.58
4NCT	Top Rails EP	4.97	1.20
Note: The corresponding stress limits are shown in the brackets			

As shown in the above tables, all weld stresses are below the corresponding stress limits.

B. HAC Side Drop Results - Group 2 DSCs

The following table summarizes the HAC side drop maximum stress intensities for the Group 2 DSCs. The stresses reported in this table are the maximum irrespective of the impact location.

Stress Intensities for Side Drop HAC – Group 2 DSCs

Load Case	Load Case Detail	Max. Stress Intensity [ksi]	Allow. P_m	Allow. $P_m + P_b$
1HAC	Top No Rails IP	36.7	44.4	57.1
2HAC	Top No Rails EP	38.5	44.4	57.1
3HAC	Top Rails IP	30.7	44.4	57.1
4HAC	Top Rails EP	31.0	44.4	57.1
5HAC	Bottom No Rails IP	26.4	44.4	57.1
6HAC	Bottom No Rails EP	25.3	44.4	57.1
7HAC	Bottom Rails IP	25.2	44.4	57.1
8HAC	Bottom Rails EP	24.9	44.4	57.1

B.1 Weld Stress Due to HAC Side Drop Loads – Group 2 DSCs

The following table summarizes the HAC side drop maximum weld stress results for the Group 2 DSCs. The maximum weld stresses are occurred at shell and outer top cover plate; therefore only weld stresses at these locations are reported. Weld stresses are below the corresponding stress limits.

Weld Stress - Side Drop HAC – Group 2 DSCs

Service Level D: Hypothetical Accident Side Drop Condition (75g)				
Load Case	**Load Case Detail**	**Outer Top Cover Weld Stress (ksi)**	**Inner Top Cover Weld Stress (ksi)**	**Weld Stress Limit (ksi)**
1HAC	Top Norails IP	26.35	18.43	35.5
2HAC	Top Norails EP	25.90	15.68	35.5
3HAC	Top Rails IP	26.03	15.78	35.5
4HAC	Top Rails EP	24.63	16.11	35.5

C. NCT End Drop Results – Group 2 DSCs

The following table summarizes the NCT end drop stress results for the Group 2 DSCs. The stresses reported in this table are the maximum irrespective of the impact location.

Stress Intensities for End Drop NCT – Group 2 DSCs

Service Level A : End Drop Normal Condition of Transport (30g)					
			Stress Limits [ksi]		
Load Case	**Load Case Detail**	**Max. Stress Intensity [ksi]**	**P_m (S_m)**	**$P_m + P_b$ (1.5 S_m)**	**$P_m + P_b + Q$ (3.0 S_m)**
1NCT (ED)	Top End Drop IP	5.7	17.5	26.3	52.5
2NCT (ED)	Top End Drop EP	13.7	17.5	26.3	52.5
3NCT (ED)	Bottom End Drop IP	**18.1**	17.5	26.3	52.5
4NCT (ED)	Bottom End Drop EP	7.8	17.5	26.3	52.5

All stresses except those for the 3NCT load case are within the allowable stress values. For load case 3NCT, the maximum stress intensity exceeds the P_m stress intensity. Therefore, stress linearization is performed for this load case and linearized stress intensities are tabulated in the following table. The maximum linearized $P_{m} + P_b$ stress intensity of 16.8 ksi is within the allowable stress value of 26.3 ksi.

Linearized Stress Intensities for End Drop 3NCT – Group 2 DSCs

Service Level A: Bottom End Drop IP [3NCT (ED)]			
S.No.	**Component**	**Membrane (17.5 ksi)**	**Membrane + Bending (26.3 ksi)**
1	Cylindrical Shell	7.6	16.8
2	Support Ring	2.4	3.1
3	Outer Top Cover Plate	2.6	7.6
4	Inner Top Cover Plate	1.9	5.2
5	Top Shield Plug	1.3	4.1
6	Inner Bottom Cover Plate	2.0	3.6
Note: The corresponding stress limits are shown in the brackets			

The bounding thermal stress case for the bottom end model gives a maximum stress intensity of 7.5 ksi. Combining the bounding thermal stress with the $P_m + P_b$ stress intensity of 16.8 ksi from the 3NCT load case, gives:

$P_m + P_b + Q = 16.8 + 7.5 = 24.3 \text{ ksi} \leq 3.0\ S_m = 52.5 \text{ ksi}.$

D. HAC End Drop Results – Group 2 DSCs

Stress intensities for end drop HAC load cases for the Group 2 DSCs are summarized in the following table. All stresses are within the allowable stress values.

Stress Intensities for End Drop HAC – Group 2 DSCs

Service Level D: End Drop Hypothetical Accident Condition (75g)				
Load Case	Load Case Detail	Max. Stress Intensity [ksi]	Stress Limits [ksi]	
			P_m	$P_m + P_b$
1HAC	Bottom End Drop IP	16.9	44.4	57.1
2HAC	Bottom End Drop EP	18.6	44.4	57.1
3HAC	Bottom End Drop No-P(1)	18.0	44.4	57.1
4HAC	Top End Drop IP	15.5	44.4	57.1
5HAC	Top End Drop EP	30.7	44.4	57.1
6HAC	Top End Drop No-P(1)	28.5	44.4	57.1

Note: 1. No-P represent load case without pressure in the analysis.

E. HAC End Drop Buckling Results – Group 2 DSCs

For the end drop hypothetical accident condition (HAC) load cases, the input load is increased beyond 75g until convergence is no longer achieved for the canister assembly model. The following table summarizes the bucking loads for various load cases. The lowest allowable buckling load is 219g and the accident g load for group 2 DSCs is 55g, this gives a safety factor of 4.0.

HAC End Drop Allowable Buckling Loads– Group 2 DSCs

Load Case	Load Case Detail	Buckling Load (g)
1HAC(ED)	Bottom End Drop IP	220
2HAC(ED)	Bottom End Drop EP	219
3HAC(ED)	Bottom End Drop No-P(1)	219
4HAC(ED)	Top End Drop IP	246
5HAC(ED)	Top End Drop EP	245
6HAC(ED)	Top End Drop No-P(1)	244

Note: 1. No-P represent load case without pressure in the analysis.

Load case of No-P is an additional load case performed without including either internal or external pressure in the end drop buckling analysis. This is to ensure that all the bounding load cases are included in the analysis.

A.2.13.7.4.3 Group 3 DSC Stress Analysis Results

A. NCT Side Drop Results – Group 3 DSCs

A.1 Top End Model Stress Evaluation

The following tables summarize the linearized bounding stresses for the main DSC components for the NCT side drop load combinations for the Group 3 DSCs Top End Model

NCT Loads Maximum Stress Intensities for Group 3 DSCs - Top End Model
(Away From Impact Zone)

Service Level A: Top End Model			
S.No.	**Component**	**P_m (17.5 ksi)**	**$P_m + P_b$ (26.3 ksi)**
1	Cylindrical Shell	12.0	21.4
2	Outer Top Cover Plate	5.4	10.6
3	Inner Top Cover Plate	6.1	11.2
4	Shield Plug	3.7	4.8
5	Support Ring	7.4	10.6
Note: The corresponding stress limits are shown in the brackets			

NCT Loads Maximum Stress Intensities for Group 3 DSCs - Top End Model
(At Impact Zone)

Service Level A: Top End Model(1)			
S.No.	**Component**	**P_L (26.3 ksi)**	**$P_L + P_b + Q$ (52.5 ksi)**
1	Cylindrical Shell	24.9	30.9
2	Outer Top Cover Plate	11.9	26.6
3	Inner Top Cover Plate	12.9	31.1
4	Shield Plug	12.1	20.0
5	Support Ring	6.0	9.5
Note: The corresponding stress limits are shown in the brackets			

Note: 1. The maximum primary membrane and primary membrane plus bending stresses listed above for the shell and the cover plates are located near the impact zone at the junction of the shell and the cover plates. In accordance with the provisions of Table NB-3217-1 (and Figure NB-3222-1), the primary membrane stresses are classified as local membrane (P_L) stresses. The membrane plus bending stresses are classified as local primary membrane plus secondary bending stresses (P_L+ P_b + Q).

Bounding Thermal Stress Intensity - Group 3 DSCs Top End Model

Load Case	Bounding Thermal Stress Intensity (ksi) Top End Model
TSA-Group 3 DSCs	13.0

Stresses Results Away From the Impact Zone

All the maximum membrane stress intensity and membrane plus bending stress intensity are within the code allowable membrane and membrane plus bending stress intensity limits.

The maximum membrane plus bending stress intensity is also combined with the bounding thermal stress intensity gives:

$$P_m + P_b + Q = 21.4 + 13.0 = 34.4 \text{ ksi} \leq 3\ S_m = 52.5 \text{ ksi}$$

Therefore, the combined stress intensity meets the code allowable.

Stresses Results at the Impact Zone

All the maximum local membrane stress intensity and local membrane plus secondary bending stress intensity are within the code allowable stress intensity limits.

The maximum local membrane plus secondary bending stress intensity is combined with the bounding thermal stress intensity as follows:

$$(P_m + P_b + Q_{\text{mechanical \& pressure}}) + Q_{\text{thermal}} = 31.1 + 13.0 = 44.1 \text{ ksi} \leq 3\ S_m = 52.5 \text{ ksi}$$

Therefore, the combined stress intensity meets the code allowable stresses.

A.2 Bottom End Model Stress Evaluations

The following table summarizes the linearized bounding stress for the main DSC components for the NCT side drop load combinations stress results for the Group 3 DSCs Bottom End Model. For the bottom end model, the stresses are lower than the stresses from the Top End Model, therefore only the maximum stress results at the impact zone are reported.

NCT Loads Stress Intensities for Group 3 DSCs – Bottom End Model
(At Impact Zone)

Service Level A: Bottom End Model(1)			
S.No.	**Component**	**P_L (26.3 ksi)**	**$P_L + P_b + Q$ (52.5 ksi)**
1	Cylindrical Shell	15.8	26.3
2	Outer Bottom Cover Plate	9.7	15.2
3	Inner Bottom Cover Plate	13.8	15.9
4	Shield Plug	8.0	10.2
Note: The corresponding stress limits are shown in the brackets			

Note: 1. The maximum primary membrane and primary membrane plus bending stresses listed above for the shell and the cover plates are located near the impact zone at the junction of the shell and the cover plates. In accordance with the provisions of Table NB-3217-1 (and Figure NB-3222-1), the primary membrane stresses are classified as local membrane (P_L) stresses. The membrane plus bending stresses are classified as local primary membrane plus secondary bending stresses ($P_L + P_b + Q$).

Bounding Thermal Stress Intensity - Group 3 DSCs Bottom End Model

Load Case	Bounding Thermal Stress Intensity (ksi)
	Bottom End Model
TSA-Group 3 DSCs	22.2

Stresses Results at the Impact Zone

All the maximum local membrane stress intensity and local membrane plus secondary bending stress intensity are within the code allowable for local membrane and local membrane plus secondary bending stress intensity limits.

The maximum local membrane plus secondary bending stress intensity is combined with the bounding thermal stress intensity as follows:

$$(P_m + P_b + Q_{mechanical\ \&\ pressure}) + Q_{thermal} = 26.3 + 22.2 = 48.5 \text{ ksi} \le 3\ S_m = 52.5 \text{ ksi}$$

Therefore, the combined stress intensity meets the code allowable.

A.3 Weld Stresses Due to NCT Side Drop Loads – Group 3 DSCs

The Group 3 DSC weld stresses are evaluated in the same way as those in Group 1 (in Section A.2.13.7.4.1).

For the cylindrical shell and outer top cover plate weld, the maximum stress intensities along with the corresponding limits are shown in the following table. All the weld stresses are within the allowable stresses.

Weld Stress Intensity- Cylindrical Shell & Outer Top Cover Plate (NCT) – Group 3 DSCs

Service Level A: Normal Condition of Transport (25g)			
		Weld Stress Intensity [ksi]	
Load Case	**Load Case Detail**	**At Impact Zone Stress Limit (42.0 ksi)**	**Away from Impact Zone Stress Limit (21.0 ksi)**
1NCT	Top Norails IP	28.2	14.8
2NCT	Top Norails EP	13.6	2.7
3NCT	Top Rails IP	28.0	15.3
4NCT	Top Rails EP	5.4	0.5
Note: The corresponding stress limits are shown in the brackets			

The weld between the cylindrical shell and inner top cover plate, the maximum stress intensities along with the corresponding limits are shown in the following table. All the weld stresses are within the allowable stresses.

Weld Stress – Cylindrical Shell & Inner Top Cover Plate (NCT) – Group 3 DSCs

Service Level A: Normal Condition of Transport (25g)			
		Weld Max. Stress [ksi]	
Load Case	**Load Case Detail**	**At Impact Zone Stress Limit (42.0 ksi)**	**Away from Impact Zone Stress Limit (21.0 ksi)**
1NCT	Top Norails IP	11.0	10.1
2NCT	Top Norails EP	8.2	8.0
3NCT	Top Rails IP	5.7	4.2
4NCT	Top Rails EP	4.7	1.3
Note: The corresponding stress limits are shown in the brackets.			

B. HAC Side Drop Results – Group 3 DSCs

The following table summarizes the HAC side drop stress intensities for the Group 3 DSCs. All the calculated stresses are within the allowable stress limits.

Stress Intensities for Side Drop HAC – Group 3 DSCs

Load Case	**Load Case Detail**	**Max. Stress Intensity [ksi]**	**Allow. P_m**	**Allow. $P_m + P_b$**
1HAC	Top No Rails IP	34.26	44.4	57.1
2HAC	Top No Rails EP	35.29	44.4	57.1
3HAC	Top Rails IP	31.37	44.4	57.1
4HAC	Top Rails EP	31.78	44.4	57.1
5HAC	Bottom No Rails IP	39.89	44.4	57.1
6HAC	Bottom No Rails EP	39.32	44.4	57.1
7HAC	Bottom Rails IP	24.95	44.4	57.1
8HAC	Bottom Rails EP	25.08	44.4	57.1

B.1 Weld Stress Due to HAC Side Drop Loads – Group 3 DSCs

The following table summarizes the HAC side drop weld stress results for the Group 3 DSCs. The maximum weld stresses are occurred at shell and outer top cover plate; therefore only weld stresses at these locations are reported. All the calculated stresses are within the allowable stress limits.

Weld Stress Intensity - Side Drop HAC – Group 3 DSCs

Service Level D: Hypothetical Accident Side Drop Condition (75g)				
Load Case	Load Case Detail	Outer Top Cover Weld Stress (ksi)	Inner Top Cover Weld Stress (ksi)	Weld Stress Limit (ksi)
1HAC	Top Norails IP	26.14	19.91	35.5
2HAC	Top Norails EP	26.05	17.85	35.5
3HAC	Top Rails IP	26.53	19.52	35.5
4HAC	Top Rails EP	25.64	20.49	35.5

C. NCT End Drop Results – Group 3 DSCs

The following table summarizes the NCT end drop stress results for the Group 3 DSCs. All stress intensities are below the allowable stress values.

Stress Intensities for End Drop NCT – Group 3 DSCs

Service Level A: End Drop Normal Condition of Transport (30g)					
			Stress Limits [ksi]		
Load Case	Load Case Detail	Max. Stress Intensity [ksi]	P_m (S_m)	$P_m + P_b$ (1.5 S_m)	$P_m + P_b + Q$ (3.0 S_m)
1NCT (ED)	Top End Drop IP	5.7	17.5	26.3	52.5
2NCT (ED)	Top End Drop EP	13.8	17.5	26.3	52.5
3NCT (ED)	Bottom End Drop IP	7.1	17.5	26.3	52.5
4NCT (ED)	Bottom End Drop EP	7.8	17.5	26.3	52.5

Combining the bounding thermal stress intensity of 13.0 ksi for the top end gives the following:

The top bounding end case: $P_m + P_b + Q = 13.8 + 13.0 = 26.8$ ksi $\leq 3.0\ S_m = 52.5$ ksi.

Combining the bounding thermal stress intensity of 22.2 ksi for the bottom end gives the following:

The bottom end bounding case: $P_m + P_b + Q = 7.8 + 22.2 = 30.0$ ksi $\leq 3.0\ S_m = 52.5$ ksi.

D. HAC End Drop Results – Group 3 DSCs

Stress intensities for end drop HAC load cases for the Group 3 DSCs are summarized in the following table. All stresses are within the allowable stress values.

Stress Intensities for End Drop HAC – Group 3 DSCs

Service Level D: End Drop Hypothetical Accident Condition (75g)				
Load Case	Load Case Detail	Max. Stress Intensity [ksi]	Stress Limits [ksi]	
			P_m	P_m+P_b
1HAC(ED)	Bottom End Drop IP	16.6	44.4	57.1
2HAC(ED)	Bottom End Drop EP	18.7	44.4	57.1
3HAC(ED)	Bottom End Drop No-P	18.0	44.4	57.1
4HAC(ED)	Top End Drop IP	15.6	44.4	57.1
5HAC(ED)	Top End Drop EP	31.2	44.4	57.1
6HAC(ED)	Top End Drop No-P	29.0	44.4	57.1

E. HAC End Drop Buckling Results – Group 3 DSCs

For the end drop hypothetical accident condition (HAC) load cases, the input load is increased beyond 75g until convergence is no longer achieved for the canister assembly model. The following table summarizes the bucking loads for various load cases. The lowest allowable buckling load is 203g and the accident g load for group 3 DSCs is 55g, this gives a safety factor of 3.7.

HAC End Drop Allowable Buckling Loads - Group 3 DSCs

Load Case	Load Case Detail	Buckling Load (g)
1HAC(ED)	Bottom End Drop IP	204
2HAC(ED)	Bottom End Drop EP	203
3HAC(ED)	Bottom End Drop No-P(1)	203
4HAC(ED)	Top End Drop IP	226
5HAC(ED)	Top End Drop EP	225
6HAC(ED)	Top End Drop No-P(1)	225

Note: 1. No-P represents load case without pressure in the analysis.

A.2.13.7.4.4 Group 4 DSC Stress Analysis Results

A. NCT Side Drop Results – Group 4 DSCs

A.1 Top End Model Stress Evaluation

The following tables summarize the linearized bounding stresses for the main DSC components for the NCT side drop load combinations for the Group 4 DSCs Top End Model.

NCT Loads Maximum Stress Intensities for Group 4 DSCs - Top End Model
(Away From Impact Zone)

Service Level A: Top End Model			
S.No.	**Component**	**P_m (17.5 ksi)**	**$P_m + P_b$ (26.3 ksi)**
1	Cylindrical Shell	25.2	25.9
2	Outer Top Cover Plate	12.3	16.0
3	Inner Top Cover Plate	16.0	16.9
4	Shield Plug	7.1	12.1
5	Support Ring	11.4	21.7

NCT Loads Maximum Stress Intensities for Group 4 DSCs - Top End Model
(At Impact Zone)

Service Level A: Top End Model(1)			
S.No.	**Component**	**P_L (26.3 ksi)**	**$P_L + P_b + Q$ (52.5 ksi)**
1	Cylindrical Shell	33.2(2)	40.7
2	Outer Top Cover Plate	20.1	27.3
3	Inner Top Cover Plate	19.2	21.4
4	Shield Plug	6.9	10.6
5	Support Ring	14.2	30.0

Note:

1. The maximum primary membrane and primary membrane plus bending stresses listed above for the shell and the cover plates are located near the impact zone at the junction of the shell and the cover plates. In accordance with the provisions of Table NB-3217-1 (and Figure NB-3222-1), the primary membrane stresses are classified as local membrane (P_L) stresses. The membrane plus bending stresses are classified as local primary membrane plus secondary bending stresses (P_L+ P_b +Q).
2. Limit analysis for a bounding case is performed and is presented in Section A.2.13.7.4.5.

Bounding Thermal Stress Intensity - Group 4 DSCs Top End Model

Load Case	Bounding Thermal Stress Intensity (ksi)
	Top End Model
TSA-Group 4 DSCs	35.7

Stresses Results Away From the Impact Zone

All the maximum membrane stress intensities and maximum membrane plus bending stress intensity are within the code allowable membrane and membrane plus bending stress intensity limits except the maximum membrane stress intensity for the shell component, which occurs at the cylindrical shell and outer top cover plate junction. The limit provisions of NB-3228.1 are used in accordance with NB-3222. Limit analysis for a bounding case is performed and is presented in Section A.2.13.7.4.5.

The maximum membrane plus bending stress intensity is also combined with the bounding thermal stress intensity as follows:

$$P_m + P_b + Q = 25.9 + 35.7 = 61.6 \text{ ksi} \geq 3\ S_m = 52.5 \text{ ksi}$$

Since the above stress intensity exceeds the $3S_m$ limit, the guidelines provided in NB-3228.5 Simplified Elastic-Plastic Analysis [1] are invoked to qualify the top end component stresses due to NCT side drop load cases.

Stresses Results at the Impact Zone

All of the local maximum membrane stress intensities are within the allowable local membrane stress intensity limit except the maximum stress intensity at the shell component, which occurs at the cylindrical shell and outer top cover plate junction. The limit provisions of NB-3228.1 are used in accordance with NB-3222. Limit analysis for a bounding case is performed and is presented in Section A.2.13.7.4.5.

All the maximum local membrane plus bending stress intensities are within the code allowables.

The maximum local membrane plus bending stress intensity is also combined with the bounding thermal stress intensity as follows:

$$(P_m + P_b + Q_{\text{mechanical \& pressure}}) + Q_{\text{thermal}} = 40.7 + 35.7 = 76.4 \text{ ksi} \geq 3\ S_m = 52.5 \text{ ksi}$$

Since the above stress intensity exceeds the $3S_m$ limit, the guidelines provided in NB-3228.5 Simplified Elastic-Plastic Analysis [1] are invoked to qualify the top end component stresses due to NCT side drop load cases.

For purposes of applying the simplified elastic-plastic analyses provisions of NB-3228.5, the thermal stresses in the shell are linearized in order to obtain the maximum primary membrane stress intensity due to thermal. The maximum thermal primary membrane stress intensity is 7.6 ksi.

Application of the provisions of NB-3228.5 Simplified Elastic-Plastic Analysis [1]

The $3S_m$ limit on the range of the primary plus secondary stress intensity may be exceeded provide the requirements of (a) through (f) below are met.

(a) The range of primary plus secondary membrane plus bending stress intensity, excluding thermal bending stresses, shall be $\leq 3S_m$.

The maximum membrane stress intensity for the TSA case at the top end of the cylindrical shell is 7.6 ksi. Therefore, the maximum thermal membrane stress intensity of 7.6 ksi at the top end of cylindrical shell is combined with the corresponding structural $P_m + P_b$.

Check: Mechanical stress (Membrane + Bending) + Thermal stress (Membrane) = 40.7 + 7.6 = 48.3 ksi.

Both structural $P_m + P_b$ and thermal membrane maximum stress intensities are at the cylindrical shell between the outer top cover plate and the cylindrical shell junction. The maximum temperature at this junction is 380° F. At this temperature, the value of Sm is taken to be that at 400° F, or 18.7 ksi.

3Sm = 3 x 18.7 ksi= 56.1 ksi

Mechanical stress (Membrane + Bending) + Thermal stress (Membrane) = 40.7 + 7.6 = 48.36 ksi. < 56.1 ksi

(b) The value of S_a used for entering the design fatigue curve is multiplied by the factor K_e.

Check: $S_n = P_m + P_b + Q = 40.7 + 35.7 = 76.4$ ksi $< 3(m)S_m = 95.37$ ksi

(Where m = 1.7; refer to [1] Table NB-3228.5(b)-1, page 65)

Since S_n (76.4 ksi) is less than 3 (m) S_m (95.37 ksi) but greater than $3S_m$ (56.1 ksi)

Therefore,

$$K_e = 1.0 + [(1-n)/(n(m-1))][(S_n/(3S_m)) - 1]$$
$$= 1.0 + [(1-0.3)/((0.3)(1.7-1))][(76.4/56.1) - 1] = 2.18$$

(Where n = 0.3; refer to [1] Table NB-3228.5(b)-1, page 65)

(c) The rest of the fatigue evaluation stays the same as required in NB-3222.4, except that the procedure of NB-3227.6 need not be used.

Check: A simplified and conservative fatigue evaluation is performed in consideration of the maximum full range of pressure and thermal primary plus secondary stress intensity for 10 cycles. The actual number of cycles for this full range of loading is much less (conservatively assumed 10 cycles over the life of the DSC). Higher cycle loadings are due to variations in ambient temperatures and vibrations during handling and develop much lower stresses (below the material endurance limit), such that the contribution to the cumulative usage factor for these loadings is negligible.

$S_a = K_f K_e (S_n / 2) E_{fc} / E_a$ (where $K_f = 4$; $K_e = 2.18$; $S_n = 76.4$ ksi

$E_{fc} = 30{,}000$ ksi (Figure I-9.1 of [2]); $E_a = 25{,}800$ ksi (at 500 °F)

So, $S_a = 4(2.18)(76.4/2)(30000/25800) = 387$ ksi

From the design fatigue curve in Figure I-9.1 of [2] and NB3224.5, the allowable number of cycles, N, for this value of S_a is approximately 40 cycles and the cumulative usage factor is conservatively calculated as follows:

$$U = 10 / 40 = 0.4 < 1.0$$

(d) The component meets the thermal ratcheting requirement of NB-3222.5.

Check: Overall thermal ratcheting can only occur with cyclic loading. The thermally induced stresses in this analysis are due to a single occurrence; therefore no thermal ratcheting is possible

(e) The temperature does not exceed those listed in Table NB-3228.5(b)-1 for the various classes of materials.

Check: The maximum temperature of the canister is much less than 800 °F listed in Table NB-3228.5(b)-1.

(f) The material shall have a specified minimum yield strength to specified minimum tensile strength ratio of less than 0.80.

Check: The maximum $S_y/S_u = 30/75 = 0.40$ at 100 °F which is less than 0.80.

Based on the results of the above analysis, the design meets the requirements of the "NB-3228.5 Simplified Elastic-Plastic Analysis".

A.2 Bottom End Model Stress Evaluation

The following table summarizes the linearized bounding stress for the main DSC components for the NCT side drop load combinations stress results for the Group 4 DSCs Bottom End Model. For the bottom end model, the stresses are lower than the stresses from the Top End Model, therefore only the maximum stress results at the impact zone are reported.

NCT Maximum Stress Intensities for Group 4 DSCs – Bottom End Model
(At Impact Zone)

Service Level A: Bottom End Model(1)			
S.No.	**Component**	**P_L (26.3 ksi)**	**$P_L + P_b + Q$ (52.5 ksi)**
1	Cylindrical Shell	10.5	26.4
2	Outer Bottom Cover Plate	10.0	14.1
3	Inner Bottom Cover Plate	9.2	14.7
4	Inner G-Ring Support	1.4	3.5
5	G-Ring Support	2.2	4.7

Note: 1. The maximum primary membrane and primary membrane plus bending stresses listed above for the shell and the cover plates are located at the impact zone at the junction of the shell and the cover plates. In accordance with the provisions of Table NB-3217-1 (and Figure NB-3222-1), the primary membrane stresses are classified as local membrane (P_L) stresses. The membrane plus bending stresses are classified as local primary membrane plus secondary bending stresses ($P_L + P_b + Q$).

Bounding Thermal Stress Intensity - Group 4 DSCs Bottom End Model

Load Case	Bounding Thermal Stress Intensity (ksi)
	Bottom End Model
TSA-Group 4 DSCs	25.2

Stresses Results at the Impact Zone

All of the local maximum membrane stress intensities are within the allowable local membrane stress intensity limit.

All the maximum local membrane plus secondary bending stress intensities are within the allowable local membrane plus bending stress intensity limit.

The maximum local membrane plus secondary bending stress intensity is also combined with the bounding thermal stress intensity as follows:

$$(P_m + P_b + Q) + Q_{thermal} = 26.4 + 25.2 = 51.6 \text{ ksi} \leq 3\ S_m = 52.5 \text{ ksi}$$

Therefore, the combined stress intensity meets the code allowable stresses.

A.3 Weld Stresses Due to NCT Side Drop Loads – Group 4 DSCs

The Group 4 DSC weld stresses are evaluated in the same way as those in Group 1 (in Section A.2.13.7.4.1).

For the cylindrical shell and outer top cover plate weld, the maximum stress intensities along with the corresponding limits are shown in the following table.

Weld Stress Intensity- Cylindrical Shell & Outer Top Cover Plate (NCT) – Group 4 DSCs

Service Level A: Normal Condition of Transport (25g)			
		Weld Max. Stress Intensity [ksi]	
Load Case	**Load Case Detail**	**At Impact Zone Stress Limit (42.0 ksi)**	**Away from Impact Zone Stress Limit (21.0 ksi)**
1NCT	Top Norails IP	21.7	8.0
2NCT	Top Norails EP	20.7	4.3
3NCT	Top Rails IP	15.6	7.5
4NCT	Top Rails EP	13.8	2.75
Note: The corresponding stress limits are shown in the brackets			

The weld stresses between the cylindrical shell and inner top cover plate are summarized in the following table.

Weld Stress – Cylindrical Shell & Inner Top Cover Plate (NCT) – Group 4 DSCs

Load Case	Load Case Detail	Max. Weld Stresses [ksi] At Impact Zone Stress Limit (42.0 ksi)	Max. Weld Stresses [ksi] Away from Impact Zone Stress Limit (21.0 ksi)
1NCT	Top Norails IP	9.22	3.76
2NCT	Top Norails EP	9.57	2.82
3NCT	Top Rails IP	7.44	3.68
4NCT	Top Rails EP	6.67	2.30
Note: The corresponding stress limits are shown in the brackets			

All the calculated weld stresses are within the stress limits.

B. HAC Side Drop Results – Group 4 DSCs

Following table summarizes the HAC side drop maximum stress intensity results for the Group 4 DSCs. The stresses reported in this table are the maximum irrespective of the impact location. All the calculated stresses are within the code allowables.

Stress Intensities for Side Drop HAC – Group 4 DSCs

Load Case	Load Case Detail	Max. Stress Intensity [ksi]	Allow. P_m	Allow. $P_m + P_b$
1HAC	Top No Rails IP	43.78	44.4	57.1
2HAC	Top No Rails EP	42.27	44.4	57.1
3HAC	Top Rails IP	42.83	44.4	57.1
4HAC	Top Rails EP	43.28	44.4	57.1
5HAC	Bottom No Rails IP	30.25	44.4	57.1
6HAC	Bottom No Rails EP	23.00	44.4	57.1
7HAC	Bottom Rails IP	32.04	44.4	57.1
8HAC	Bottom Rails EP	31.02	44.4	57.1

B.1 Weld Stress Due to HAC Side Drop Loads – Group 4 DSCs

The table below summarizes the HAC side drop maximum weld stress results for the Group 4 DSCs. The maximum weld stresses are occurred at shell and outer top cover plate; therefore only weld stresses at these locations are reported. All the calculated weld stresses are within the allowable weld stress limits.

Weld Stress - Side Drop HAC – Group 4 DSCs

Load Case	Load Case Detail	Outer Top Cover Weld (ksi)	Inner Top Cover Weld (ksi)	Weld Stress Limit (ksi)
1HAC	Top Norails IP	32.63	13.03	35.5
2HAC	Top Norails EP	31.13	12.75	35.5
3HAC	Top Rails IP	25.98	16.20	35.5
4HAC	Top Rails EP	26.42	15.45	35.5

C. NCT End Drop Results – Group 4 DSCs

The following table summarizes the NCT end drop maximum stress intensity results for the Group 4 DSCs.

Stress Intensities for End Drop NCT – Group 4 DSCs

Service Level A: End Drop Normal Condition of Transport (30g)					
			Stress Limits [ksi]		
Load Case	Load Case Detail	Max. Stress Intensity [ksi]	P_m (S_m)	$P_m + P_b$ (1.5 S_m)	$P_m + P_b + Q$ (3.0 S_m)
1NCT (ED)	Top End Drop IP	15.2	17.5	26.3	52.5
2NCT (ED)	Top End Drop EP	19.4	17.5	26.3	52.5
3NCT (ED)	Bottom End Drop IP	24.1	17.5	26.3	52.5
4NCT (ED)	Bottom End Drop EP	24.0	17.5	26.3	52.5

For load cases 2NCT, 3NCT and 4NCT, the maximum stress intensity exceeds the P_m stress intensity. Therefore, stress linearization is performed for these load cases and linearized stress intensities are tabulated in the following tables.

Linearized Stress Intensities for Top End Drop (Load Case 2NCT) – Group 4 DSCs

Service Level A: Top End Drop EP (2NCT)			
S.No.	Component	Membrane [ksi]	Membrane + Bending [ksi]
1	Cylindrical Shell	6.8	12.5
2	Inner Bottom Cover Plate	4.5	12.2
3	Outer Bottom Cover Plate	5.1	12.3
4	Grapple Ring	5.5	15.7

Linearized Stress Intensities for Bottom End Drop (Load Case 3NCT) – Group 4 DSCs

Service Level A: Bottom End Drop IP (3NCT)			
S.No.	Component	Membrane [ksi]	Membrane + Bending [ksi]
1	Cylindrical Shell	5.5	7.1
2	Outer Top Cover Plate	0.4	3.1
3	Inner Top Forging (Radial)	2.8	6.9
4	Inner Top Forging (Axial)	3.9	7.1
5	Lead Plug Top Cover Plate	5.0	10.8
6	Support Ring	2.8	4.2

Linearized Stress Intensities for Bottom End Drop (Load Case 4NCT) – Group 4 DSCs

Service Level A: Bottom End Drop EP (4NCT)			
S.No.	**Component**	**Membrane [ksi]**	**Membrane + Bending [ksi]**
1	Cylindrical Shell	5.0	11.1
2	Outer Top Cover Plate	0.8	5.5
3	Inner Top Forging (Radial)	5.0	12.0
4	Inner Top Forging (Axial)	6.5	12.4
5	Lead Plug Top Cover Plate	8.7	18.6
6	Support Ring	2.3	5.6

The linearized stresses meet the allowable stress intensity limit of 17.5 ksi for P_m and 26.3 ksi for $P_m + P_b$.

The maximum $P_m + P_b$ stress intensity for the above load case is 18.6 ksi (Load Case 4NCT). Combining the bounding thermal stress intensity of 35.70 ksi gives the following:

The bounding case, $P_m + P_b + Q = 18.6 + 35.7 = 54.3$ ksi > $3.0\ S_m = 52.5$ ksi.

The simplified elastic-plastic analysis performed in this section above (Top End Model, Stress Results at the Impact Zone) bounds this stress intensity.

D. HAC End Drop Results – Group 4 DSCs

Maximum stress intensities for end drop HAC load cases for the Group 4 DSCs have been summarized in the following table. All stresses are within the allowable stress values.

Stress Intensities for End Drop HAC – Group 4 DSCs

Service Level D: End Drop Hypothetical Accident Condition (75g)				
Load Case	**Load Case Detail**	**Max. Stress Intensity [ksi]**	**Stress Limits [ksi]**	
			P_m	**P_m+P_b**
1HAC(ED)	Bottom End Drop IP	36.9	44.4	57.1
2HAC(ED)	Bottom End Drop EP	36.4	44.4	57.1
3HAC(ED)	Bottom End Drop No-P	36.9	44.4	57.1
4HAC(ED)	Top End Drop IP	35.4	44.4	57.1
5HAC(ED)	Top End Drop EP	34.9	44.4	57.1
6HAC(ED)	Top End Drop No-P	35.4	44.4	57.1

E. HAC End Drop Buckling Results – Group 4 DSCs

For the end drop hypothetical accident condition (HAC) load cases, the input load is increased beyond 75g until convergence is no longer achieved for the canister assembly model. The following table summarizes the bucking loads for various load cases. The lowest allowable buckling load is 207g and the accident g load for group 4 DSCs is 55g, this gives a safety factor of 3.76.

HAC End Drop Allowable Buckling Loads– Group 4 DSCs

Load Case	Load Case Detail	Buckling Load (g)
1HAC(ED)	Bottom End Drop IP	208
2HAC(ED)	Bottom End Drop EP	207
3HAC(ED)	Bottom End Drop No-P	207
4HAC(ED)	Top End Drop IP	254
5HAC(ED)	Top End Drop EP	248
6HAC(ED)	Top End Drop No-P	246

A.2.13.7.5 Limit Analysis

A. Introduction

To address membrane stress intensity and membrane plus bending stress intensity exceedances for DSCs groups 1 to 4, in accordance with NB-3222 service level A, limit analyses are performed per NB-3228.1.

The limit analysis is described in NB-3228.1 and is repeated here for reference "The limits on general membrane stress intensity, local membrane plus primary bending stress intensity need not be satisfied at a specific location if it can be shown by limit analysis that the specified loadings do not exceed two-thirds of the lower bound collapse load (NB-3213.29). The yield strength to be used in these calculations is 1.5 S_m".

The lower bound limit load is calculated from an ideally elastic-plastic analysis. A bilinear stress-strain curve is used for all steel components where $S_y = 1.5\ S_m$ and $E_P/E = 0.0$ and all of applied the loads (g-load and pressures) are increased until the analysis solution is not converged.

B. Analysis

Load Cases Analyzed

As discussed in Section A.2.13.7.1, for the purposes of the analysis the canisters are divided into four groups. The canister groups and the corresponding sections documenting the NCT analyses are as follows:

Group	Canisters Groups	NCT Analysis Sections
1	69BTH, 37PTH, 32PTH, 32PTH Type 1, 32PTH1	A.2.13.7.4.1
2	61BT, 61BTH Type 1	A.2.13.7.4.2
3	61BTH Type 2, 32PT, 24PTH	A.2.13.7.4.3
4	24PT4, 24PTH-S-LC	A.2.13.7.4.4

These groups are evaluated for NCT loads as described in the sections listed in the above table. Based on the calculations, the most critical load cases are the load cases with internal pressure and external pressure for Group 4 on the (Top End Model). Thus these load cases are evaluated for Limit Analysis.

Since Group 4 DSC top end model includes a lead shield plug, therefore additional group (Group 2 with thinnest cover plate) was selected for the Limit Analysis to represent the DSCs with steel top shield plugs.

The baseline "g" loads used for NCT side drop load is 25g. Therefore, per NB-3228.1 the required lower bound collapse load should be at least 25g x 3/2 = 37.5g. The analyses are carrying out to 75g (3 times the baseline g load). The required maximum internal and external pressures for Group 2 DSC are 15 psig; these pressures are conservatively increased up to 45 psig in the analyses. Similarly for Group 4 DSCs the required internal and external pressures are 20 psig and 15 psig respectively. These values are also conservatively increased to 60 psig and 45 psig respectively in the analysis. The following table summarizes the load cases analyzed.

Load Case Number	Loading Condition
Group 2 DSCs -1	Top End Model, 75g + 45 psig internal pressure
Group 2 DSCs -2	Top End Model, 75g + 45 psig external pressure
Group 4 DSCs -1	Top End Model, 75g + 60 psig internal pressure
Group 4 DSCs -2	Top End Model, 75g + 45 psig external pressure

Material Properties Used in the Analysis

A bilinear elastic-perfectly plastic relationship and yield strength of $1.5S_m$ is used in the analysis for the steel components. For group 4 DSCs Top End Model, the canister shield plug is made of ASTM B29 Lead material. It is modeled using elastic perfectly plastic material properties with the specified yield equal to S_y. The following tables list the material properties for steel and lead. Material properties at 500°F are used for the analyses.

Bilinear Kinematic Material Properties used in Analysis – Group 2 DSCs

Material # 1 (Group 2 - 304 Stainless Steel)						
Temp. (°F)	100	200	300	400	500	600
Yield Stress($1.5S_m$) psi	30,000	30,000	30,000	28,050	26,250	24,600
Tangent Modulus (E_T)	0	0	0	0	0	0
Material # 2 (Group 2 – A36 Carbon Steel)						
Temp. (°F)	100	200	300	500	600	700
Yield Stress($1.5S_m$) psi	28,950	28,950	28,950	28,950	26,550	25,950
Tangent Modulus (E_T)	0	0	0	0	0	0

Bilinear Kinematic Material Properties used in Analysis – Group 4 DSCs

Material # 1 (Group 4 – Stainless Steel)						
Temp. (^{o}F)	100	200	300	400	500	600
Yield Stress(1.5Sm) psi	30,000	30,000	30,000	28,050	26,250	24,600
Tangent Modulus (E_T)	0	0	0	0	0	0
Material # 2 (Group 4 - Lead)						
Temp. (^{o}F)	100	175	250	325	440	620
Yield Stress - psi	490	428	391	320	320	320
Tangent Modulus (E_T)	0	0	0	0	0	0

C. Analysis Results

In all the four load cases analyzed, the solution was carry out up to 75g without loss of numerical convergence and, therefore, the collapse load was not reached. The deformation plots for these four load cases are shown in Figures A.2.13.7-9 through A.2.13.7-12, the structural deformation is small (0.34 inches) for the 75g load. Figures A.2.13.7-13 through A.2.13.7-16 show the maximum plastic strain for 25g NCT baseline g load is about 0.18% and occurs in the cylindrical shell. The resulting strains are small. The following table summarizes the results for limit load analysis. It indicates that the limit load is at least double the actual applied load.

Limit Load Result Summary for Group 2 and 4 DSCs

Load Case	Lower Bound Collapse Load[1]	Limit Load[2]	Actual g-Load	Factor of Safety
Group 2 DSCs -1	75	(2/3)*75 = 50	25	2.00
Group 2 DSCs -2	75	(2/3)*75 = 50	25	2.00
Group 4 DSCs -1	75	(2/3)*75 = 50	25	2.00
Group 4 DSCs -2	75	(2/3)*75 = 50	25	2.00

Notes:

1. This is a conservative estimate of the lower bound collapse load, since actual collapse was not achieved in the analysis.

2. Per NB-3228.1, the limit load is 2/3 of the lower bound collapse load.

A.2.13.7.6 References

1. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section III, Subsection NB, see Chapter A.2, Section A.2.1.2.1 for applicable editions.

2. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section III, Appendices, see Chapter A.2, Section A.2.1.2.1 for applicable editions.

3. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section II, Part D, see Chapter A.2, Section A.2.1.2.1 for applicable editions.

4. ANSYS Computer Code and User's Manual, Release 10A1.

5. ISG-15, Rev. 0, "Materials Evaluation."

6. *R.A. Robinson, et. al., " A survey of Strain-Rate Effects for some common Structural Materials used in Radioactive Material Packaging and Transportation Systems", Report BMI-1954, August 1976, Battelle Columbus Laboratories.*

7. *Tietz, T. E., "Determination of the Mechanical Properties of a High Purity Lead and a 0.058 % Copper-Lead Alloy," WADC Technical Report 57-695, ASTIA Document No. 151165, Stanford Research Institute, Menlo Park, CA, April 1958.*

8. *H.J. Rack, G.A. Knorovsky, "An Assessment of Stress-Strain Data Suitable for Finite-Element Elastic-Plastic Analysis of Shipping Containers," Sandia Laboratories, NUREG/CR-0481, SAND77-1872 R-7, 1978.*

Table A.2.13.7-1
Material Properties for Steels SA-240 304, SA-479 304, SA-182 F304, & SA-336 F304

Temp (^{o}F)	**E (10^3 ksi)**	**Sm (ksi)**	**Sy (ksi)**	**Su (ksi)**	**Coefficient of Thermal Expansion, α (10^{-6} in./in.°F)**	**Density, ρ (lb./in.^{3})**	**Poisson's ratio, ν**
-20		20.0	30.0	75.0			
70	28.3				8.5	0.29	0.3
100		20.0	30.0	75.0	8.6		
150			26.7		8.8		
200	27.6	20.0	25.0	71.0	8.9	0.29	0.3
250			23.6		9.1		
300	27.0	20.0	22.4	66.2	9.2	0.29	0.3
350					9.3		
400	26.5	18.7	20.7	64.0	9.5	0.29	0.3
450					9.6		
500	25.8	17.5	19.4	63.4	9.7	0.29	0.3
550					9.8		
600	25.3	16.4	18.4	63.4	9.8	0.29	0.3
650		16.2	18.0	63.4	9.9		
700	24.8	16.0	17.6	63.4	10.0		
750		15.6	17.2	63.3	10.0		
800	24.1	15.2	16.9	62.8	10.1		
850			16.5	62.0	10.1		
900	23.5		16.2	60.8	10.2		
950			15.9	59.3	10.3		
1000	22.8		15.5	57.4	10.3		

Table A.2.13.7-2
Material Properties for Steel A36

Temp (°F)	E (10^3 ksi)	Sm (ksi)	Sy (ksi)	Su (ksi)	Coefficient of Thermal Expansion, α (10^{-6} in./in.°F)	Density, ρ (lb./in.^{3})	Poisson's ratio, ν
-20		19.3	36.0	58.0			
70	29.5				6.4	0.29	0.3
100		19.3	36.0	58.0	6.5		
150			33.8		6.6		
200	28.8	19.3	33.0	58.0	6.7	0.29	0.3
250			32.4		6.8		
300	28.3	19.3	31.8	58.0	6.9	0.29	0.3
350					7.0		
400	27.7	19.3	30.8	58.0	7.1	0.29	0.3
450					7.2		
500	27.3	19.3	29.3	58.0	7.3	0.29	0.3
550					7.3		
600	26.7	17.7	27.6	58.0	7.4	0.29	0.3
650		17.4	26.7	58.0	7.5		
700	25.5	17.3	25.8	58.0	7.6		
750			24.9	57.3	7.7		
800	24.2		24.1	53.3	7.8		
850			23.4	48.5	7.9		
900	22.4		22.8	43.3	7.9		
950			22.1	38.0	8.0		
1000	20.4		21.4	33.4	8.1		

Table A.2.13.7-3
Material Properties for B-29 Lead

Static Mechanical Lead Properties

Temp. (°F)	Static Stress Properties (ksi) [7] Yield (S_Y) Tension	Static Stress Properties (ksi) [7] Yield (S_Y) Compression	Static Stress Properties (ksi) [7] Ultimate (S_U) Tension	E [8] (10^6 psi)	Coef. of Thermal Exp [8] (10^{-6} in/in/°F)
-99	-	-	-	2.50	15.28
70	-	-	-	2.34	16.07
100	0.584	0.490	1.570	2.30	16.21
175	0.509	0.428	1.162	2.20	16.58
250	0.498	0.391	0.844	2.09	16.95
325	0.311	0.320	0.642	1.96	17.54
440	-	-	-	1.74	18.50
620	-	-	-	1.36	20.39

Dynamic Stress-Strain Lead Properties

Strain (in/in)	Stress (ksi) [6] 100 °F	230 °F	300 °F	350 °F	500 °F
0.00048	1.14	1.06	1.00	0.97	0.86
0.03	2.2	2.0	1.7	1.5	1.1
0.1	3.3	2.8	2.38	2.1	1.26
0.3	4.9	3.2	2.72	2.4	1.44
0.5	5.6	3.6	3.06	2.7	1.62

Table A.2.13.7-4
Allowable Stress Values for Stainless Steel and Carbon Steel at 500 °F

Loading Condition	**Stress Category**	**Stress Criteria [1]**	**Material**	**Allowable Stress (ksi)**
Normal Conditions, Elastic Analysis	Membrane Stress, P_m	S_m	Stainless Steel	17.5
			A36	19.3
	Membrane + Bending Stress, $P_m + P_b$	1.5 S_m	Stainless Steel	26.3
			A36	29.0
	Primary + Secondary Stress, $P_m + P_b + Q$	3 S_m	Stainless Steel	52.5
			A36	57.9
Accident Conditions, Elastic-Analysis	Membrane Stress, P_m	min of (2.4 S_m, 0.7 S_u)	Stainless Steel	42.0
			A36	40.6
	Membrane + Bending Stress, $P_m + P_b$	min of (3.6 S_m, 1.0 S_u)	Stainless Steel	63.0
			A36	58.0
Accident Conditions, Elastic-Plastic Analysis	Membrane Stress, P_m	max of 0.7 S_u, $S_y + (S_u - S_y)/3$	Stainless Steel	44.4
			A36	40.6
	Membrane + Bending Stress, $P_m + P_b$	0.9 S_u	Stainless Steel	57.1
			A36	52.2

Table A.2.13.7-5
Allowable Weld Stresses

Service Level	Stress Region/Category	Stress Criteria	Allowable Stress Value at 500 °F [ksi]
A (Normal Condition of Transport)	Weld Stress away from Impact Zone	0.8 [1.5 S_m]	21.0
	Weld Stress in local area at Impact Zone	0.8 [3.0 S_m]	42.0
D (Hypothetical Accident Condition)	Primary Membrane Stress, P_m	0.8 [Max(0.7 S_u, $S_y + (S_u - S_y)/3$)]	35.5
	Primary Membrane (or Local Membrane) + Bending Stress, P_m *(or* P_l*)* + P_b	0.8 [0.9 S_u]	45.7

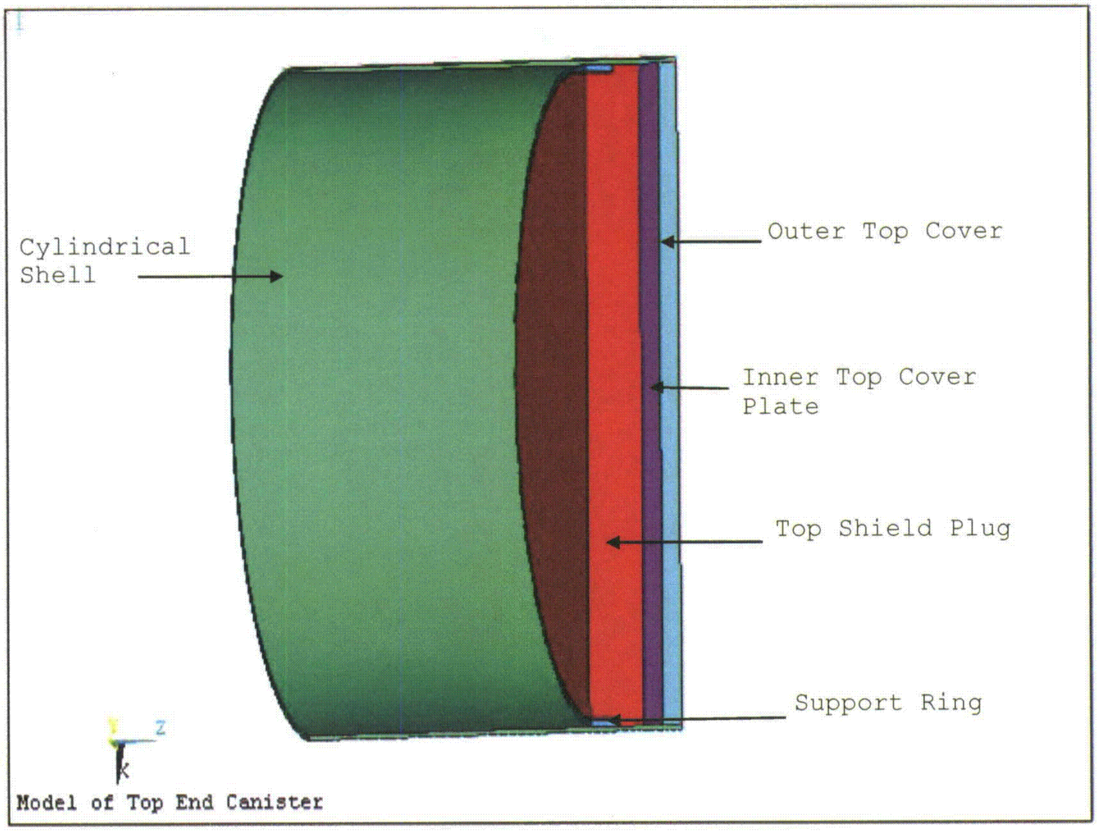

Figure A.2.13.7-1
DSC Groups 1, 2 and 3 Typical Top End 3D Finite Element Model

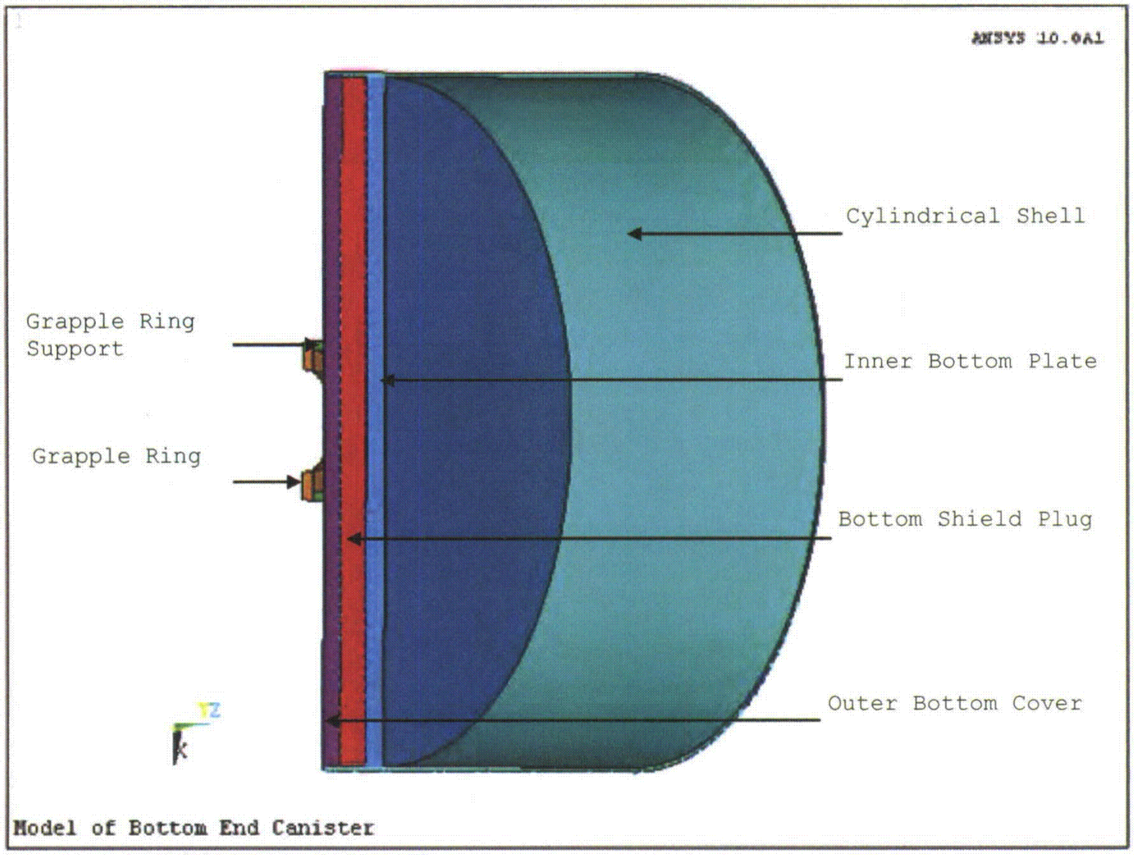

Figure A.2.13.7-2
DSC Groups 1, 2 and 3 Typical Bottom End 3D Finite Element Model

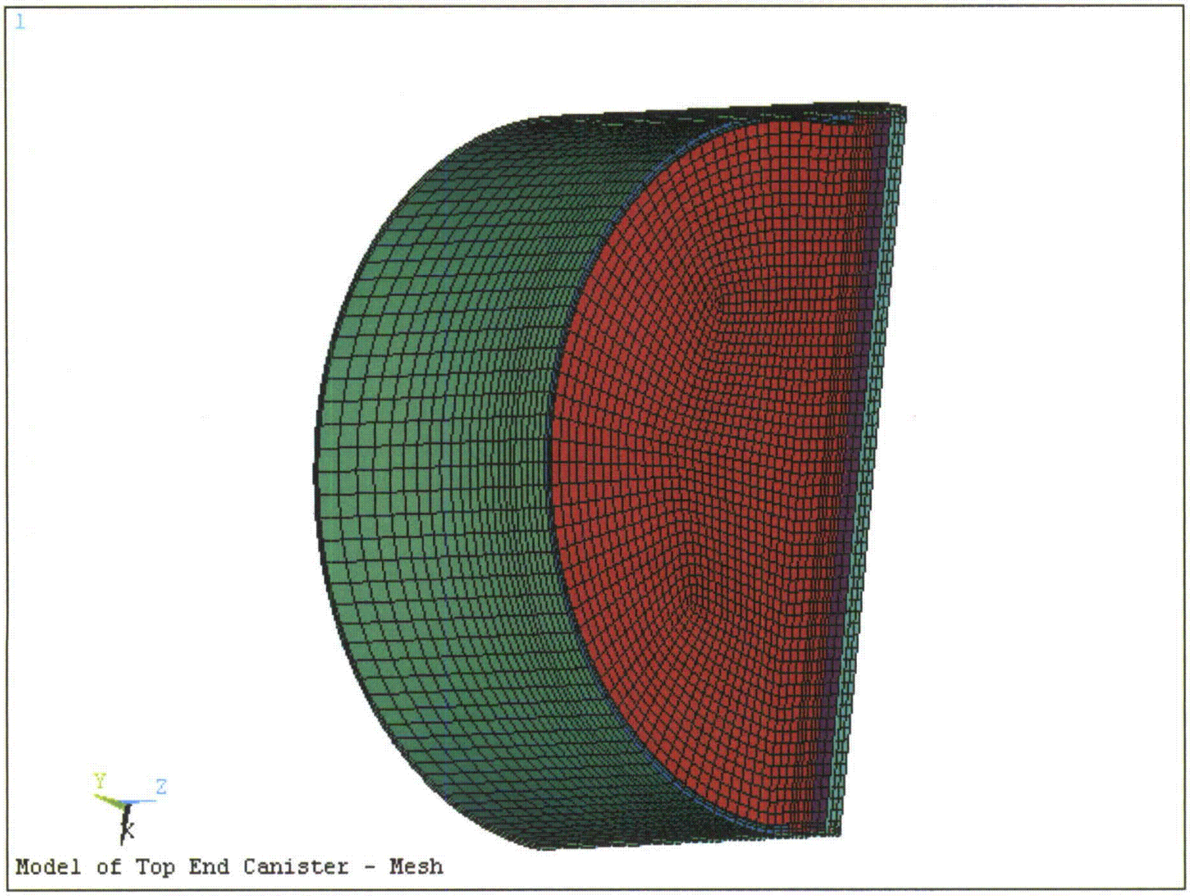

Figure A.2.13.7-3
DSC Groups 1, 2 and 3 Typical Top End 3D Finite Element Model Mesh

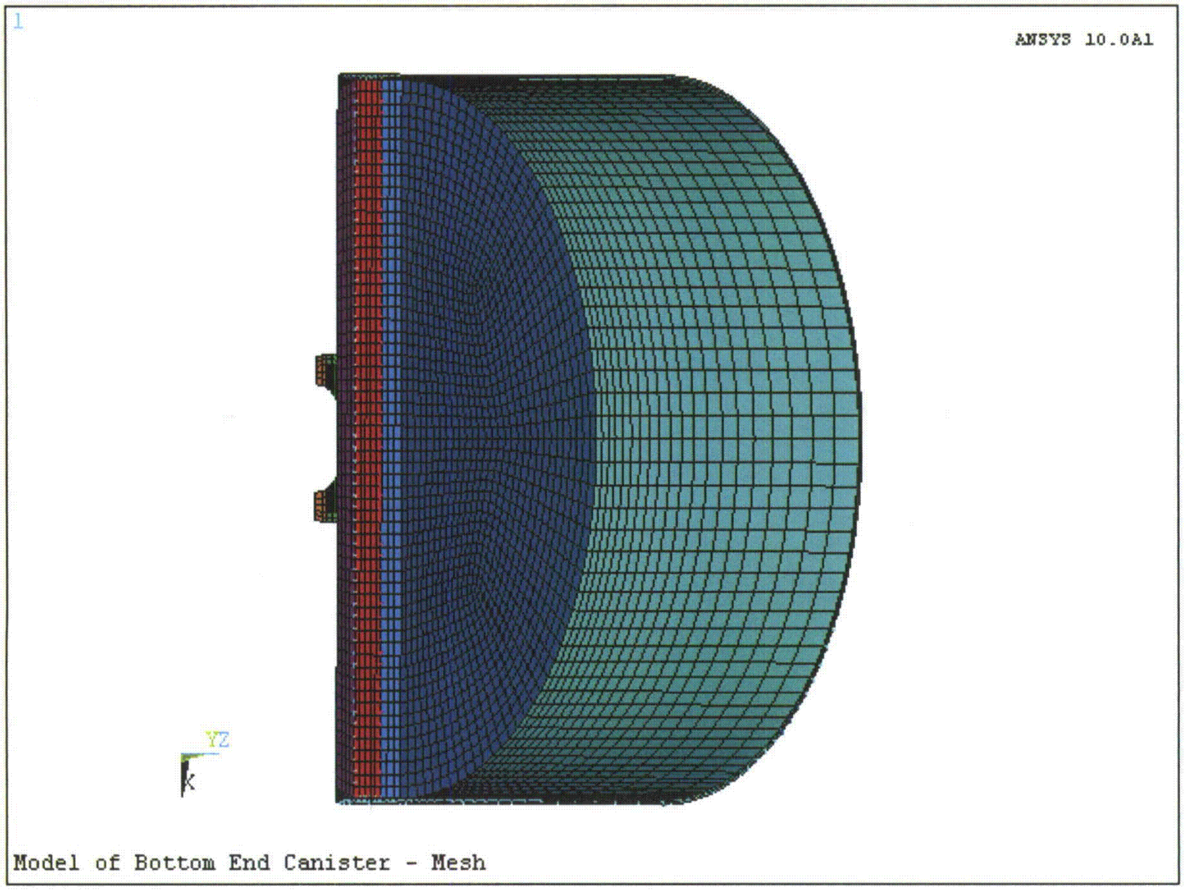

Figure A.2.13.7-4
DSC Groups 1, 2 and 3 Typical Bottom End 3D Finite Element Model Mesh

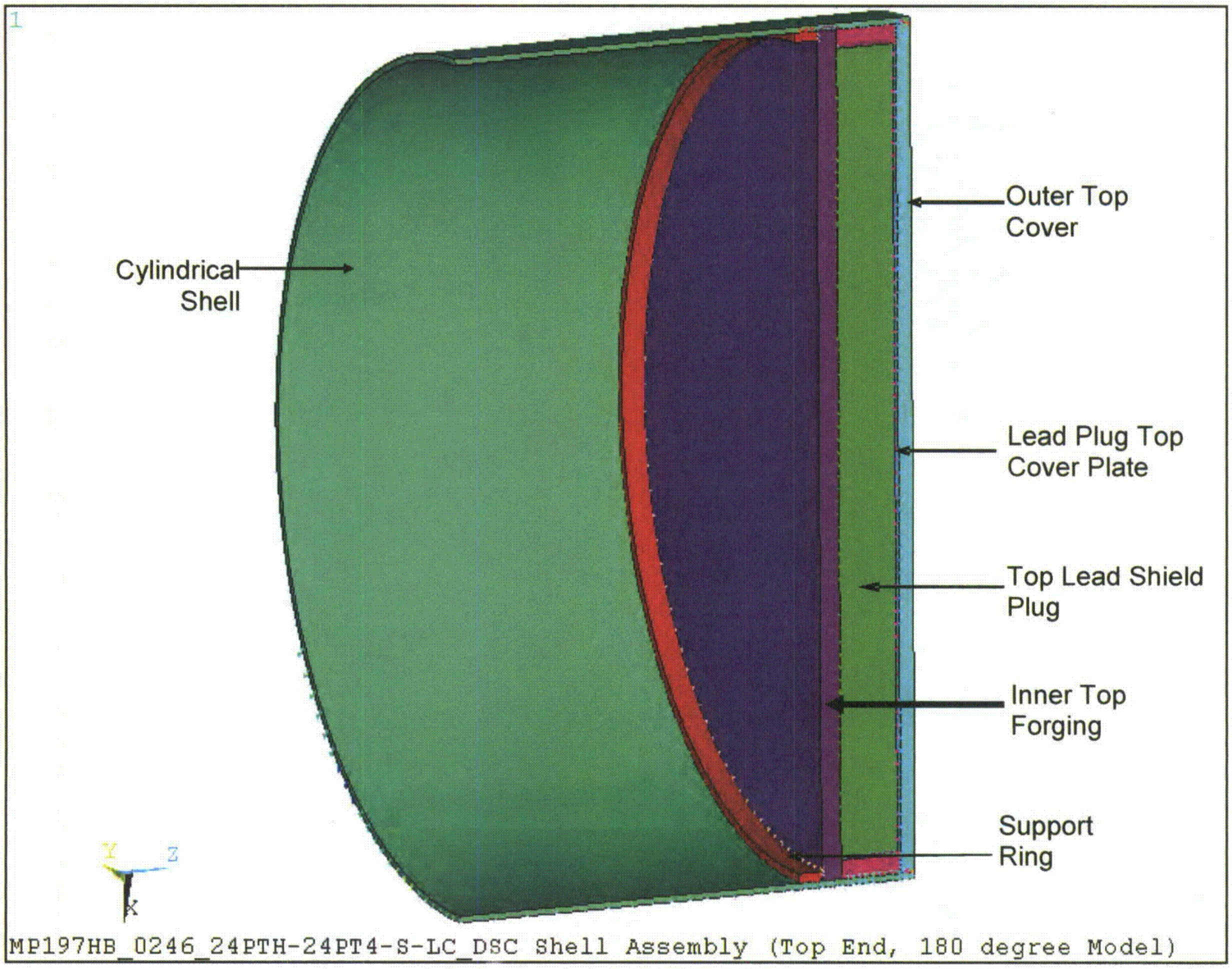

Figure A.2.13.7-5
DSC Group 4 Typical Top End 3D Finite Element Model

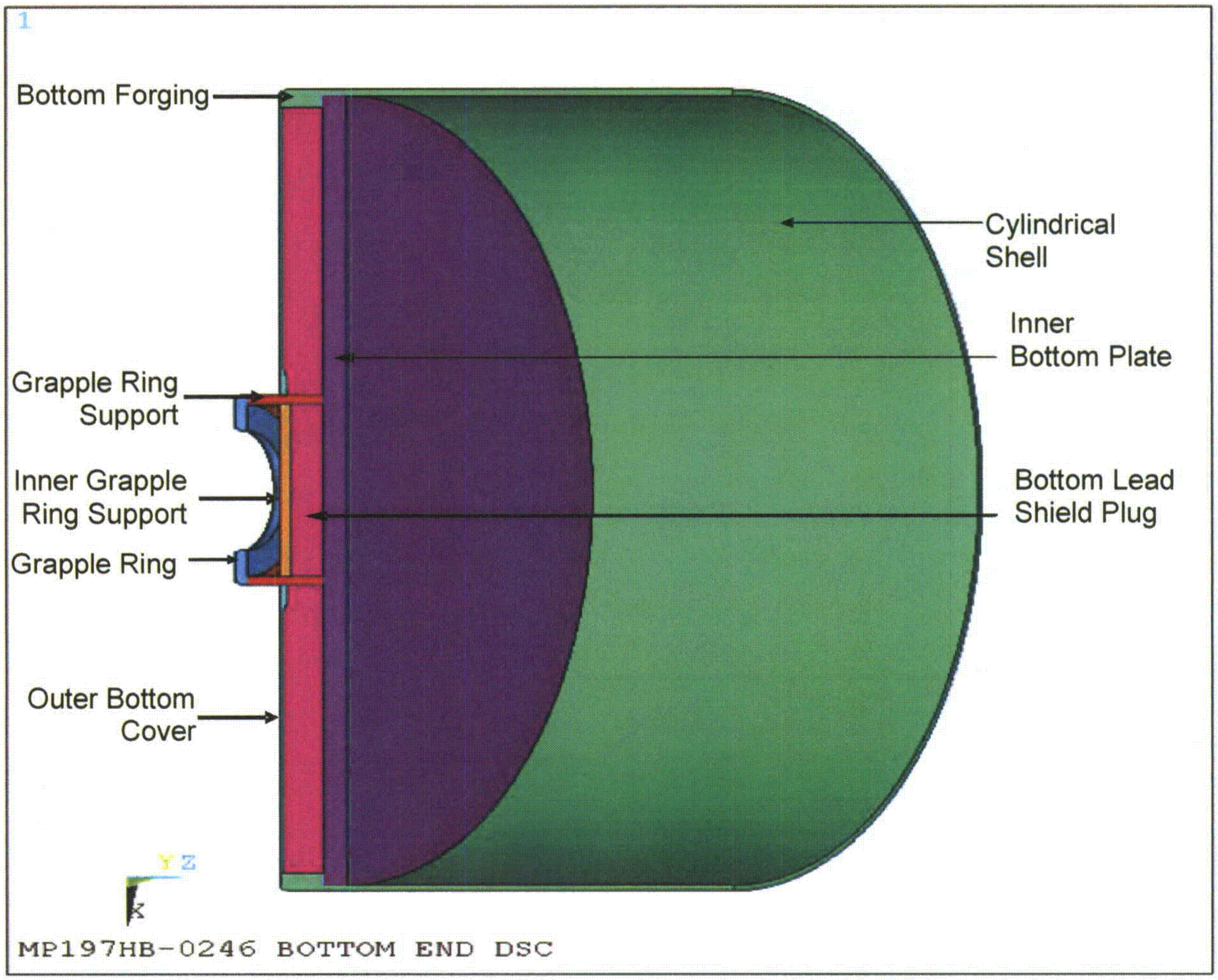

Figure A.2.13.7-6
DSC Group 4 Typical Bottom End 3D Finite Element Model

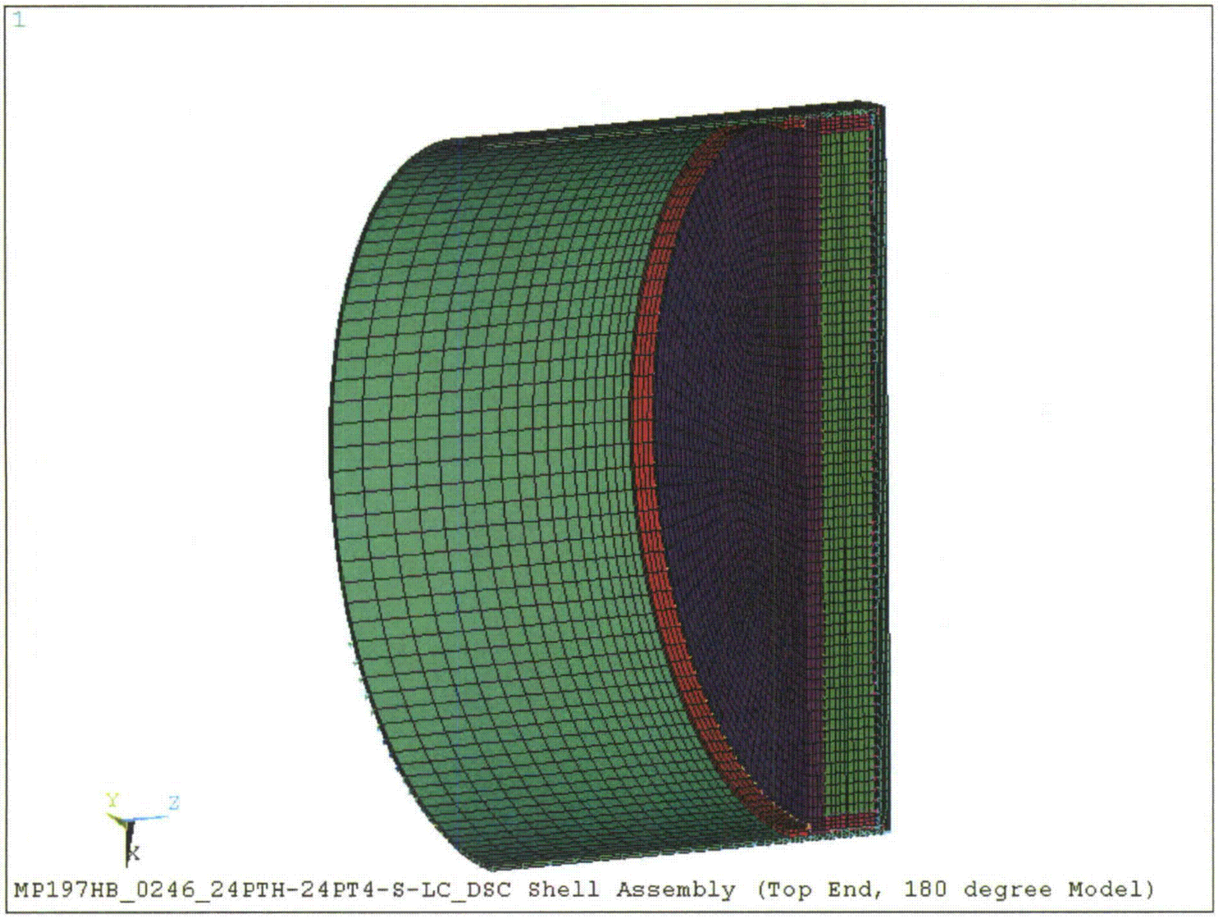

Figure A.2.13.7-7
DSC Group 4 Typical Top End 3D Finite Element Model Mesh

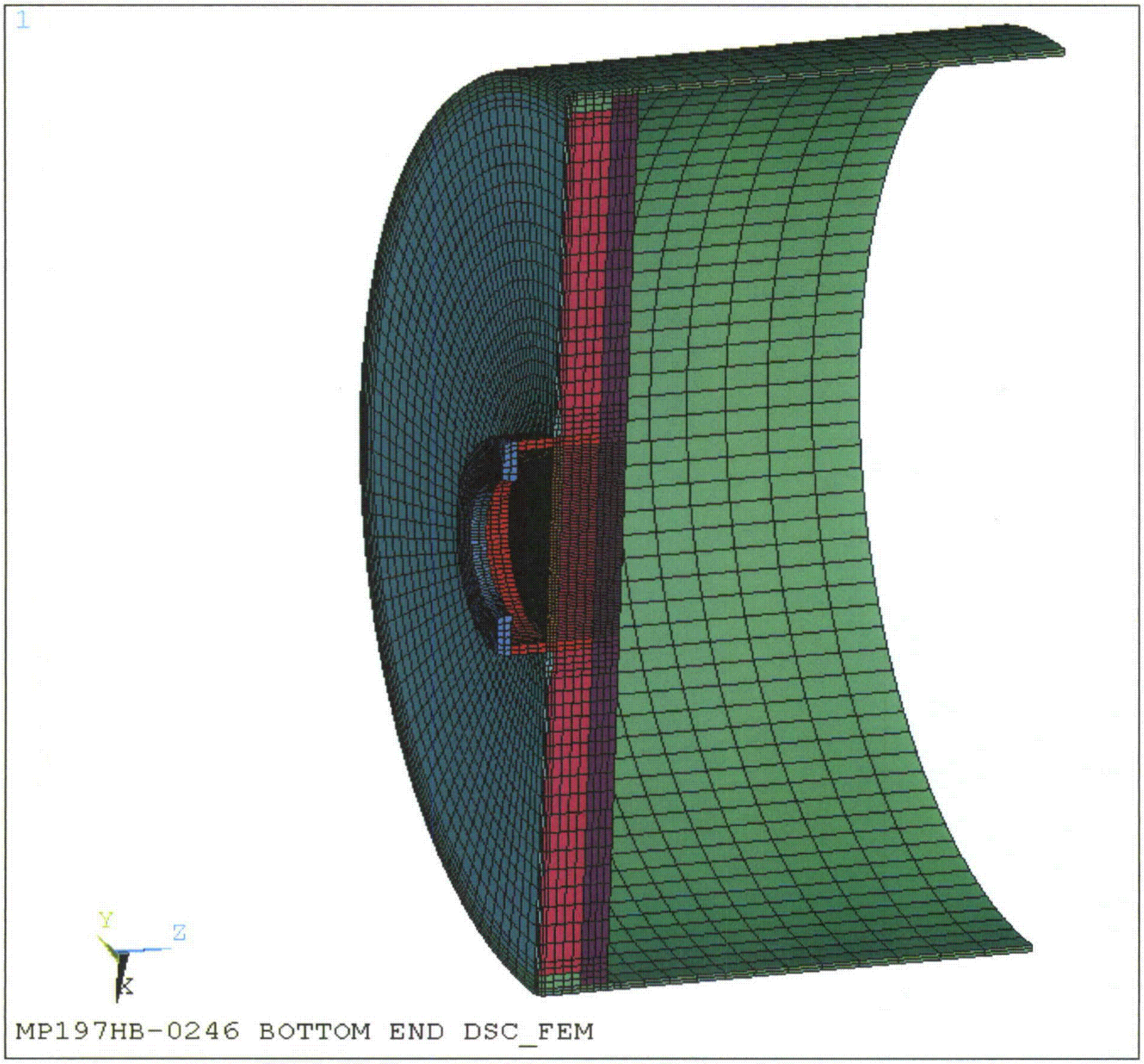

Figure A.2.13.7-8
DSC Group 4 Typical Bottom End 3D Finite Element Model Mesh

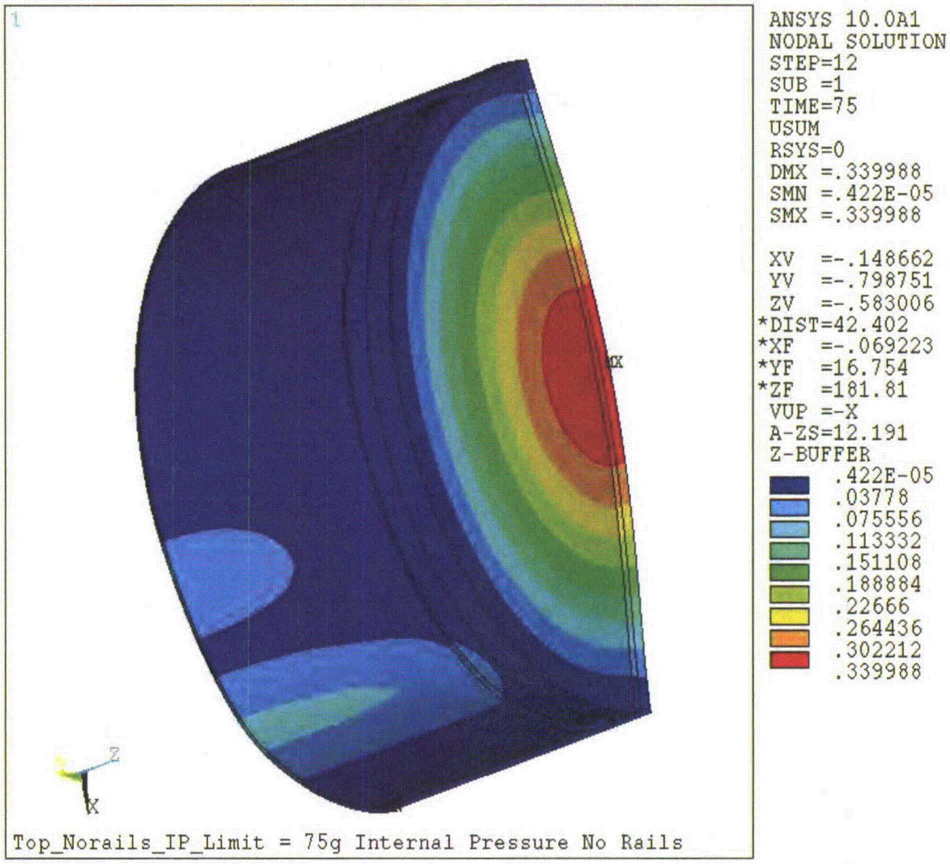

Figure A.2.13.7-9
Group 2 DSCs -1 Load Case Deflection Plot at 75g

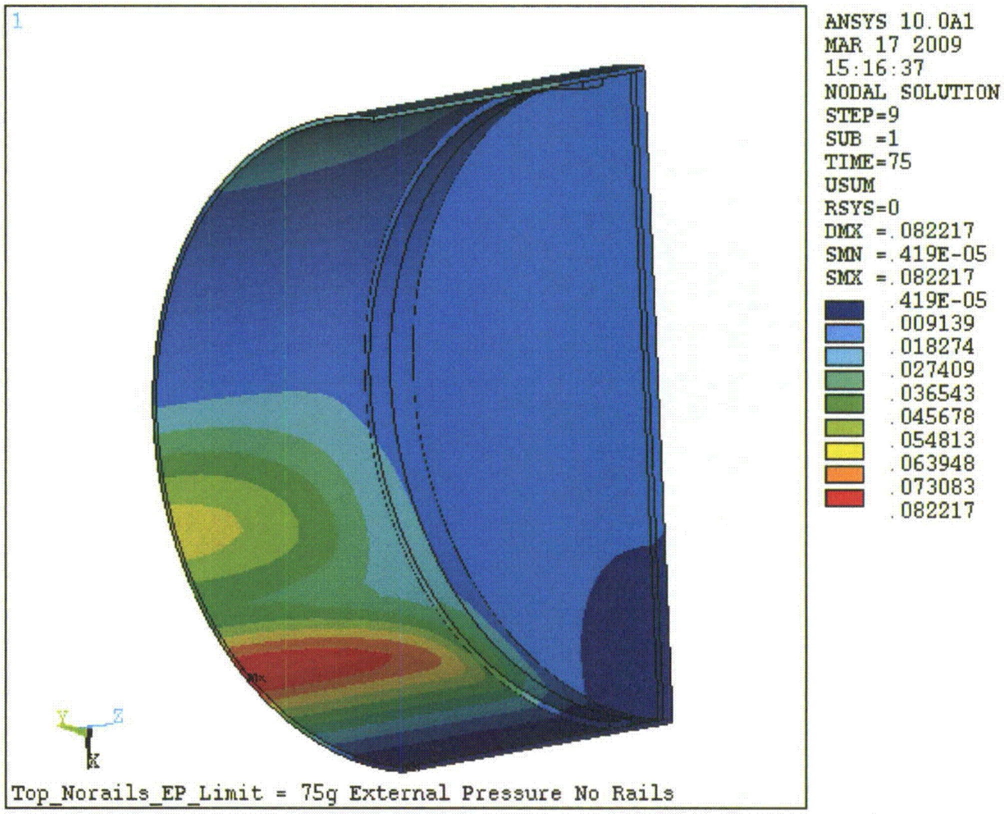

Figure A.2.13.7-10
Group 2 DSCs -2 Load Case Deflection Plot at 75g

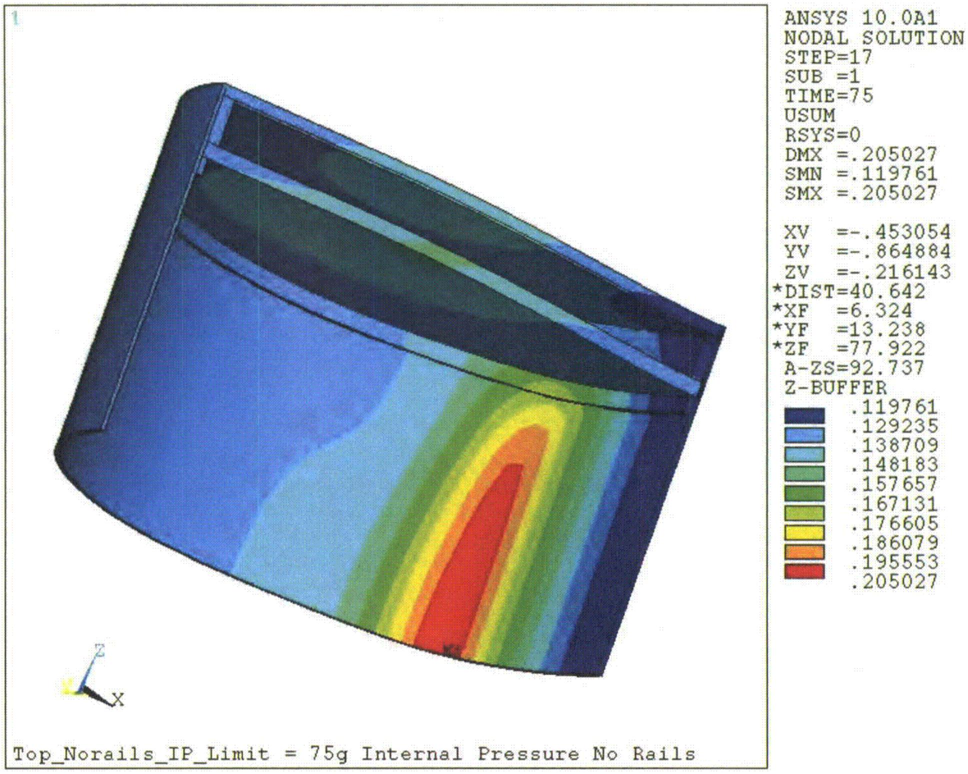

Figure A.2.13.7-11
Group 4 DSCs -1 Load Case Deflection Plot at 75g

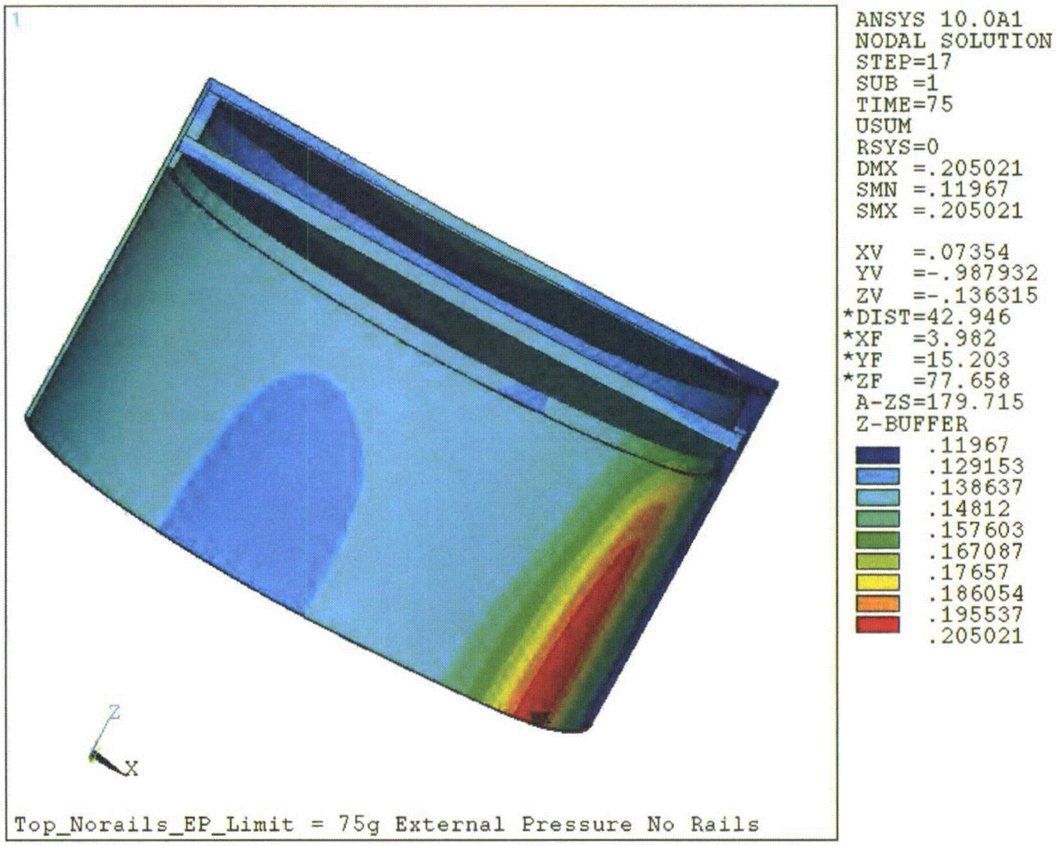

Figure A.2.13.7-12
Group 4 DSCs -2 Load Case Deflection Plot at 75g

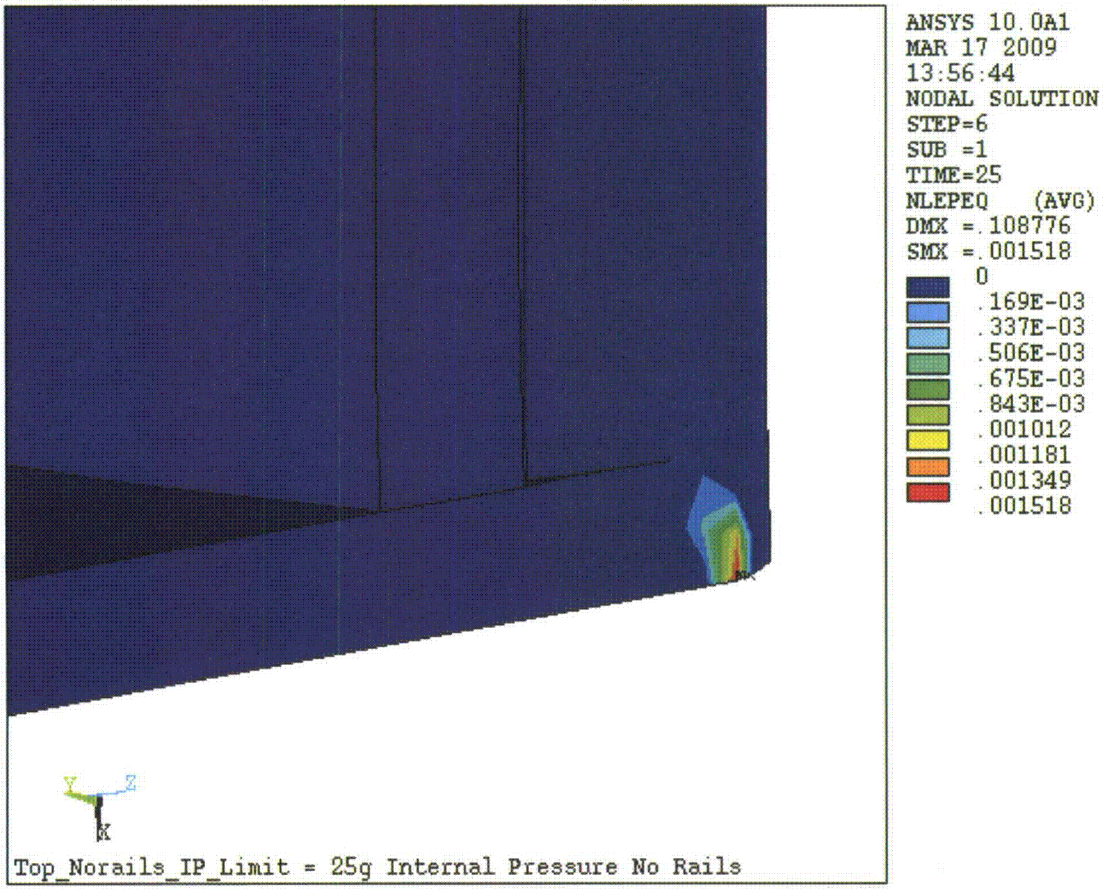

Figure A.2.13.7-13
Group 2 DSCs -1 Load Case Equivalent Plastic Strain at 25g

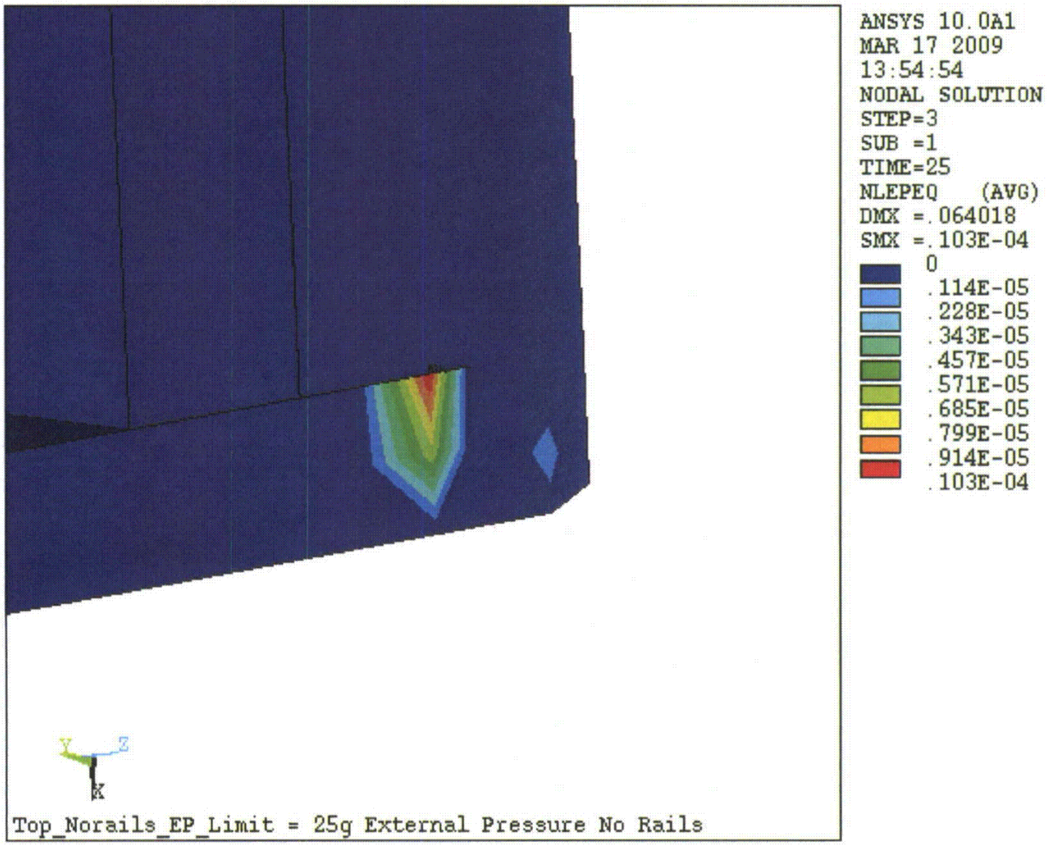

Figure A.2.13.7-14
Group 2 DSCs -2 Load Case Equivalent Plastic Strain at 25g

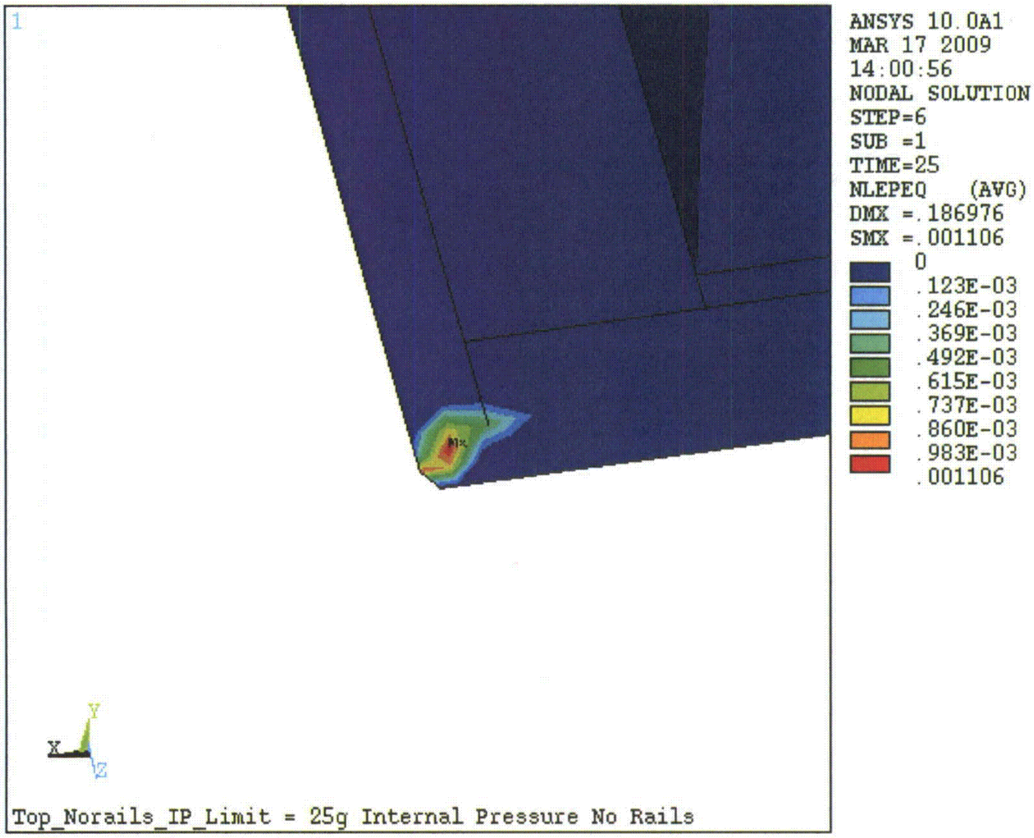

Figure A.2.13.7-15
Group 4 DSCs -1 Load Case Equivalent Plastic Strain at 25g

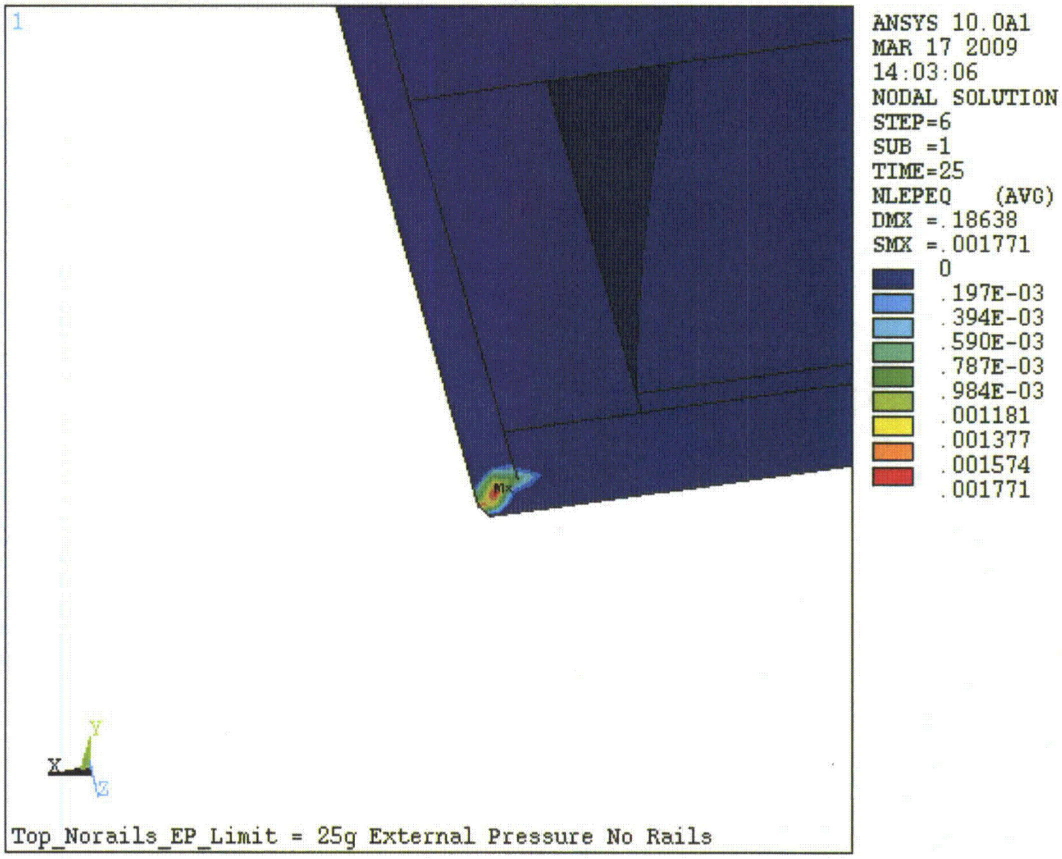

Figure A.2.13.7-16
Group 4 DSCs -2 Load Case Equivalent Plastic Strain at 25g