APPENDIX 3.AA HI-TRAC 125 - ROTATION TRUNNION WELD ANALYSIS

3.AA.1 Introduction

HI-TRAC has a lower pocket trunnion attached to the HI-TRAC outer shell and to the water jacket outer shell. In this appendix, the weld stresses and the stress distribution in the adjacent metal structure (inner and outer shell, water jacket, etc.) are analyzed. Drawing 1880, Sheet 10 shows the configuration.

3.AA.2 Methodology

Strength of Materials formulae are used to evaluate resistance of the weld group. The weld evaluation is performed with HI-TRAC in either vertical or horizontal orientation. The applied loading is a force applied in any direction at the center of the pocket. This force is resisted by the assemblage of weld which is loaded by a force and by a bending moment due to the offset of the point of application to the centroid of the weld group. Figure 3.AA.1 shows the configuration.

The stress distribution in the surrounding metal structure is determined using the FEA code ANSYS (Version 5.4). The finite element model is shown in Figure 3.AA.2. The model is one-quarter symmetric and extends longitudinally 15" above the rotation trunnion where the edges are restrained. The bottom flange, the inner and outer shells, and the radial channels are modeled using SHELL63 elements; the pocket trunnion and lead shield are modeled with 4-node solid elements. Linear elastic material behavior is assumed; material properties are obtained from Section 3.3.

The acceptance criteria is from ASME Code Subsection NF, Level A for base material SA516-Gr70 (per Table NF-3324.5(a)-1). Since this is a component that can be construed as being active during a low-speed "lift" operation, a dynamic amplifier of 1.15 is included. The trunnion static load associated with a "Normal" lift assumes that the upper trunnions and the rotation trunnions all carry load in proportion to the centroid location per Table 3.2.3. The trunnion load associated with an "Off-Normal" or "Upset Service Level B" lift is 50% of the total load (i.e., the lifting cables are assumed completely slack but no additional dynamic amplifier is included).

3.AA.3 Input Data for Weld Group Analysis

Holtec drawing 1880, shows the weld group connecting the trunnion to the HI-TRAC outer shell structure. Dimensions are per Holtec drawing no. 1880, sheet 10. The rotation trunnion is welded to the HI-TRAC outer shell around the four sides and is also welded to the water jacket around three sides. All welds are full penetration.

The input weight is the heaviest fully loaded HI-TRAC dry weight.

$Weight := 243000 \cdot lbf$ Table 3.2.2

From the drawing, the length, the width, the minimum depth of the block to the outer shell weld, and the depth of the pocket are, respectively:

$L := 13 \cdot in$ $W := 12.375 \cdot in$

$D := 11.8469 \cdot in$ $d := 3.9375 \cdot in$

The distance between trunnions (per drawing 1880) is $L_{total} := 178.25 \cdot in$

The minimum centroidal distance from the bottom trunnion of a loaded HI-TRAC 125 is calculated as (Table 3.2.3 and drawing 1880)

$L_{cent} := 91.66 \cdot in - 8.25 \cdot in$

Therefore the bottom rotation trunnions will normally be subjected to the load fraction "f" where

$$f := 1 - \frac{L_{cent}}{L_{total}} \qquad f = 0.532$$

Per ASME Section III, Subsection NF for Class 3 construction, the allowable stress for full penetration welds is equal to the base metal allowable strength (NF-3256.2) for Level A service loading. An increase of 33% is permitted for Level B service loading per Table 3552(b)-1.

The weld thickness to the outer shell is

$t_w := .625 \cdot in$

The weld thickness to the water jacket shell is the same as the base metal shell.

$t_{wj} := 0.5 \cdot in$

3.AA.4 Allowable strength

The allowable stress is $S_a := 17500 \cdot psi$ Table 3.1.10

3.AA.5 Calculation of Load and Moment on Weld Group for Level A Condition

The force and moment on the single weld group is computed assuming a dynamic amplifier equal to 15% of the load on the trunnion. This is standard practice for crane low-speed lifting operations.

Dynamic Amplifier $\quad DAF := 1.15$

$Force := \frac{f \cdot Weight}{2} \cdot DAF \qquad Force = 74342.323\,lbf$

$Moment\_arm := D - .5 \cdot d \qquad Moment\_arm = 9.878\,in$

Note that we have conservatively calculated the bending moment on the weld group by using the largest moment arm (i.e., the distance to the outer shell weld).

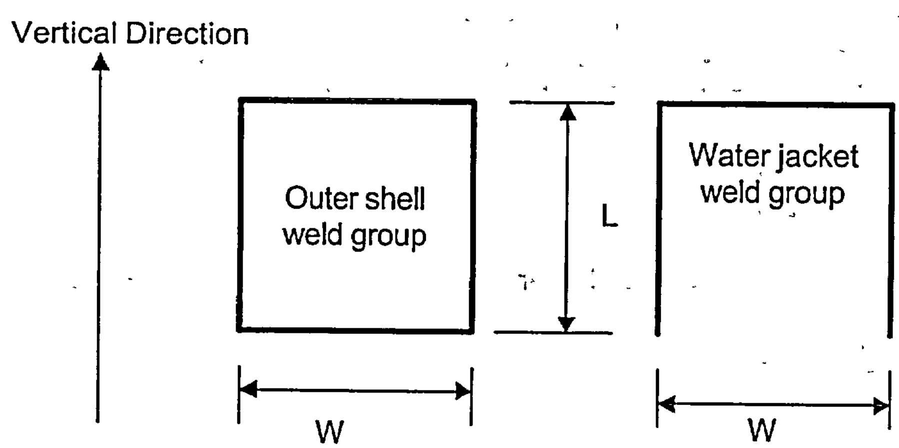
$Bending\_Moment := Force \cdot Moment\_arm$

$Bending\_Moment = 7.344 \times 10^{5}\,in \cdot lbf$

3.AA.6 Calculation of Metal Area and Inertia of Weld Groups

The area and inertia properties of the welds to the outer shell and to the water jacket are computed as follows:

The configuration of the two weld groups is shown below:



The area and inertia properties of the outer shell weld group are:

$$Area := 2 \cdot t_w \cdot (L + W) \qquad Area = 31.719\, in^2$$

$$Inertia_{vert} := 2 \cdot t_w \cdot \frac{L^3}{12} + 2 \cdot (t_w \cdot W) \cdot (.5 \cdot L)^2 \qquad Inertia_{vert} = 882.409\, in^4$$

$$Inertia_{horiz} := 2 \cdot t_w \cdot \frac{W^3}{12} + 2 \cdot (t_w \cdot L) \cdot (.5 \cdot W)^2 \qquad Inertia_{horiz} = 819.542\, in^4$$

The inertia properties of the weld to the water jacket is conservatively computed by considering only two opposing weld lines (the legs of the "U" shaped weld group) in the computation of moment of inertia for a vertically oriented cask:

$$Area_{wj} := t_{wj} \cdot (2 \cdot L + W) \qquad Area_{wj} = 19.187\, in^2$$

$$Inertia_{wjvert} := 2 \cdot t_{wj} \cdot \frac{L^3}{12} \qquad Inertia_{wjvert} = 183.083\, in^4$$

$$Inertia_{wjhoriz} := 2 \cdot (t_{wj} \cdot L) \cdot (.5 \cdot W)^2 + \frac{t_{wj} \cdot W^3}{12} \qquad Inertia_{wjhoriz} = 576.67\, in^4$$

It is assumed that the distribution of force and moment to the water jacket weld group and to the outer shell weld group is based on the ratios of the inertia properties of the individual weld group. Therefore, for the trunnion load oriented along the cask axis (HI-TRAC vertical), then the distribution ratio is

$$r_v := \frac{Inertia_{wjvert}}{Inertia_{vert} + Inertia_{wjvert}} \qquad r_v = 0.172$$

For the trunnion load oriented perpendicular to the cask axis (HI-TRAC horizontal)

$$r_h := \frac{Inertia_{wjhoriz}}{Inertia_{horiz} + Inertia_{wjhoriz}} \qquad r_h = 0.413$$

3.AA.7 Weld Stress Calculations

3.AA.7.1 Cask Vertical - Maximum loading on the outer shell

Stress due to "Force"

$$\sigma_1 := \frac{Force \cdot (1 - r_v)}{Area} \qquad \sigma_1 = 1941\,psi$$

Stress due to "Bending_Moment"

$$\sigma_2 := \frac{Bending\_Moment \cdot L \cdot (1 - r_v)}{2 \cdot Inertia_{vert}} \qquad \sigma_2 = 4480\,psi$$

The maximum stress for calculation of weld safety factor for the outer shell is assumed to be the SRSS of the two stress components (in reality, these two stresses are at right angles to one another at any point in the weld group, and only the maximum of the two stresses need be considered if maximum normal stress theory is used.

$$\sigma_{max} := \sqrt{\sigma_1^{\,2} + \sigma_2^{\,2}} \qquad \sigma_{max} = 4.882 \times 10^3\,psi$$

The safety factor on the maximum weld stress in the outer shell of the HI-TRAC 125 is

$$SF_1 := \frac{S_a}{\sigma_{max}} \qquad SF_1 = 3.58$$

3.AA.7.2 Cask Horizontal - Maximum loading on the water jacket shell

Stress due to "Force"

$$\sigma_1 := \frac{Force \cdot (r_h)}{Area_{wj}} \qquad \sigma_1 = 1600\,psi$$

Stress due to "Bending_Moment"

$$\sigma_2 := \frac{Bending\_Moment \cdot W \cdot (r_h)}{2 \cdot Inertia_{wjhoriz}} \qquad \sigma_2 = 3254\,psi$$

The maximum stress for calculation of weld safety factor for the water jacket is assumed to be the SRSS of the two stress components (in reality, these two stresses are at right angles to one another at any point in the weld group, and only the maximum of the two stresses need be considered if maximum normal stress theory is used.

$$\sigma_{max} := \sqrt{\sigma_1^2 + \sigma_2^2} \qquad \sigma_{max} = 3.627 \times 10^3 \text{psi}$$

The safety factor on the maximum weld stress in the water jacket of the HI-TRAC 125 is

$$SF_2 := \frac{S_a}{\sigma_{max}} \qquad SF_2 = 4.83$$

3.AA.8 Weld Stress Analysis for Level B Condition of Loading

Under this load condition, all of the weight is assumed supported on the two rotation trunnions. Therefore, the Level A load on each trunnion increases in magnitude by

$$Increase := \frac{\frac{0.5}{DAF}}{\left(\frac{f}{2}\right)} \qquad Increase = 1.634$$

Since the allowable stress increases by 33% for this condition, the safety factors are

$$SF_{1B} := SF_1 \cdot \frac{1.33}{Increase} \qquad SF_{1B} = 2.917$$

$$SF_{2B} := SF_2 \cdot \frac{1.33}{Increase} \qquad SF_{2B} = 3.927$$

3.AA.9 Finite Element Analysis

The linear elastic finite element analysis is performed for a trunnion load, applied vertically, having the magnitude appropriate to the off-normal load condition:

$$Load := \frac{243000 \cdot lbf}{2} \qquad Load = 1.215 \times 10^5 \text{lbf}$$

The resulting stress distributions for the inner and outer shells and for the radial channels are shown in Figures 3.AA.3-3.AA.8. To determine the actual state of stress under Level A loading, the following amplifier need to be incorporated to account for centroid position and inertia load amplifier.

Level B stress intensity amplifier

$Amp_B := 1.0$ Note that no dynamic amplifier is assumed for this off-normal case where the lift trunnions support no load

Level A stress intensity amplifier (including the inertia load facor)

$Amp_A := f \cdot DAF$ $Amp_A = 0.612$

The following table is constructed from the figures. Peak stresses at the corner are not included in the stress evaluation. Away from the immediate vicinity of the trunnion, the state of stress is considered as "primary". At the trunnion discontinuity, the stress away from the corner is considered as "primary plus secondary". The initial superscript "a" means axial stress directed along the cask axis; the initial superscript "c" means circumferential stress directed around the cask periphery. No NF Code limits are set for "primary plus secondary stress states"; the values are posted for information only.

LEVEL B

Component - Outer shell

Primary Stress

$\sigma_{apo} := -830 \cdot psi \cdot Amp_B$ $\sigma_{apo} = -830\,psi$

$\sigma_{cpo} := -436 \cdot psi \cdot Amp_B$ $\sigma_{cpo} = -436\,psi$

Primary plus Secondary Stress

$\sigma_{aso} := 2484 \cdot psi \cdot Amp_B$ $\sigma_{aso} = 2.484 \times 10^3\,psi$

$\sigma_{cso} := -2973 \cdot psi \cdot Amp_B$ $\sigma_{cso} = -2.973 \times 10^3\,psi$

Component - Inner shell

Primary Stress

$\sigma_{api} := -956 \cdot psi \cdot Amp_B$ $\sigma_{api} = -956\,psi$

$\sigma_{cpi} := -1501 \cdot psi \cdot Amp_B$ $\sigma_{cpi} = -1.501 \times 10^3\,psi$

Primary plus Secondary Stress

$\sigma_{asi} := 1734 \cdot psi \cdot Amp_B$ $\sigma_{asi} = 1.734 \times 10^3\,psi$

$\sigma_{csi} := -1501 \cdot psi \cdot Amp_B$ $\sigma_{csi} = -1.501 \times 10^3\,psi$

Component - Radial Channel

Primary Stress

$\sigma_{apc} := 2305 \cdot psi \cdot Amp_B$ $\sigma_{apc} = 2.305 \times 10^3\,psi$

$\sigma_{cpc} := -631 \cdot psi \cdot Amp_B$ $\sigma_{cpc} = -631\,psi$

Primary plus Secondary Stress

$\sigma_{asc} := -13867 \cdot psi \cdot Amp_B$ $\sigma_{asc} = -1.387 \times 10^4\,psi$

$\sigma_{csc} := -2303 \cdot psi \cdot Amp_B$ $\sigma_{csc} = -2.303 \times 10^3\,psi$

To obtain values appropriate to the Level A condition, all results are multiplied by

$$\frac{Amp_A}{Amp_B} = 0.612$$

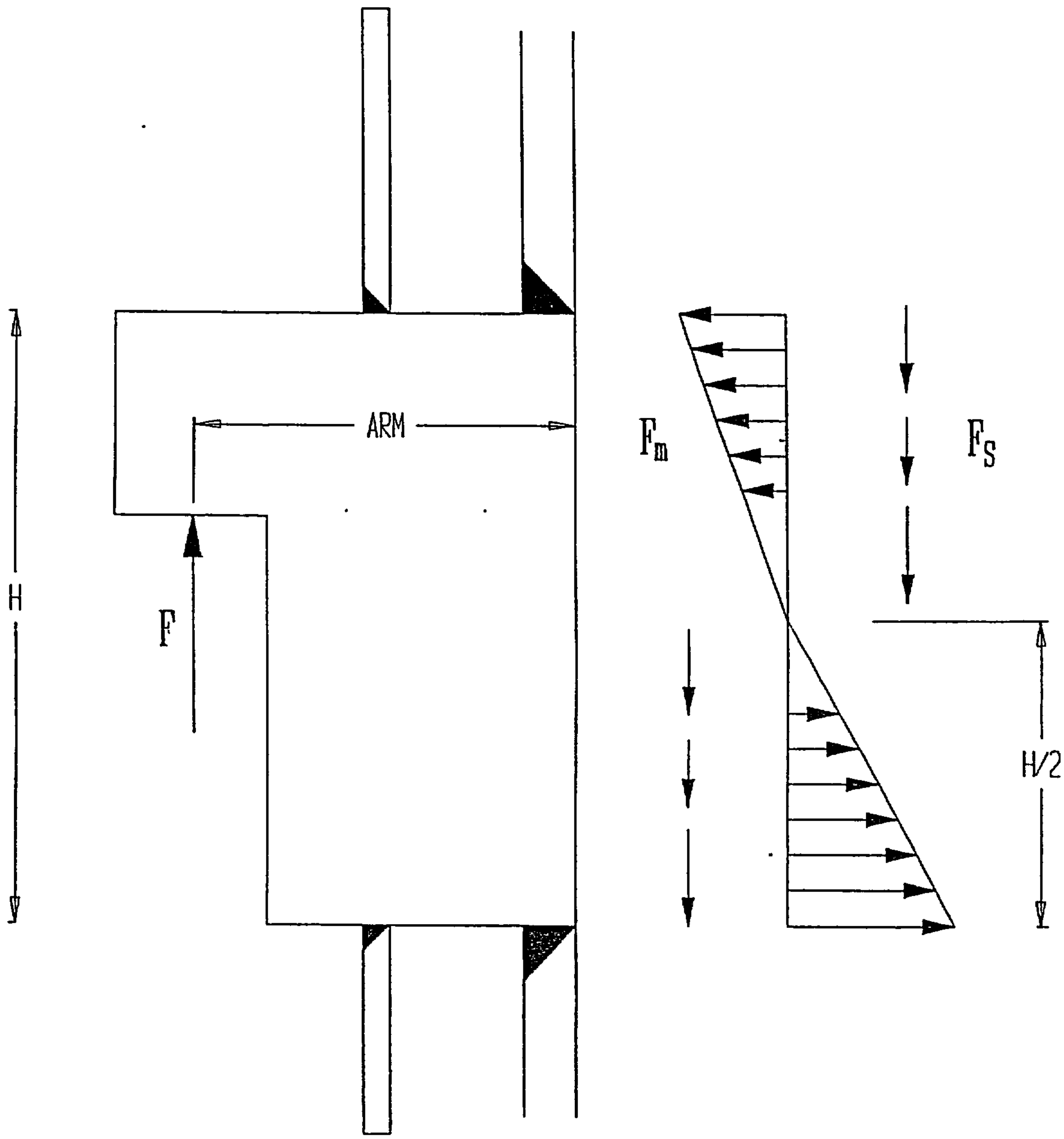


FIGURE 3.AA.1 FORCES AND MOMENTS ON
125 TON ROTATION TRUNNION WELD

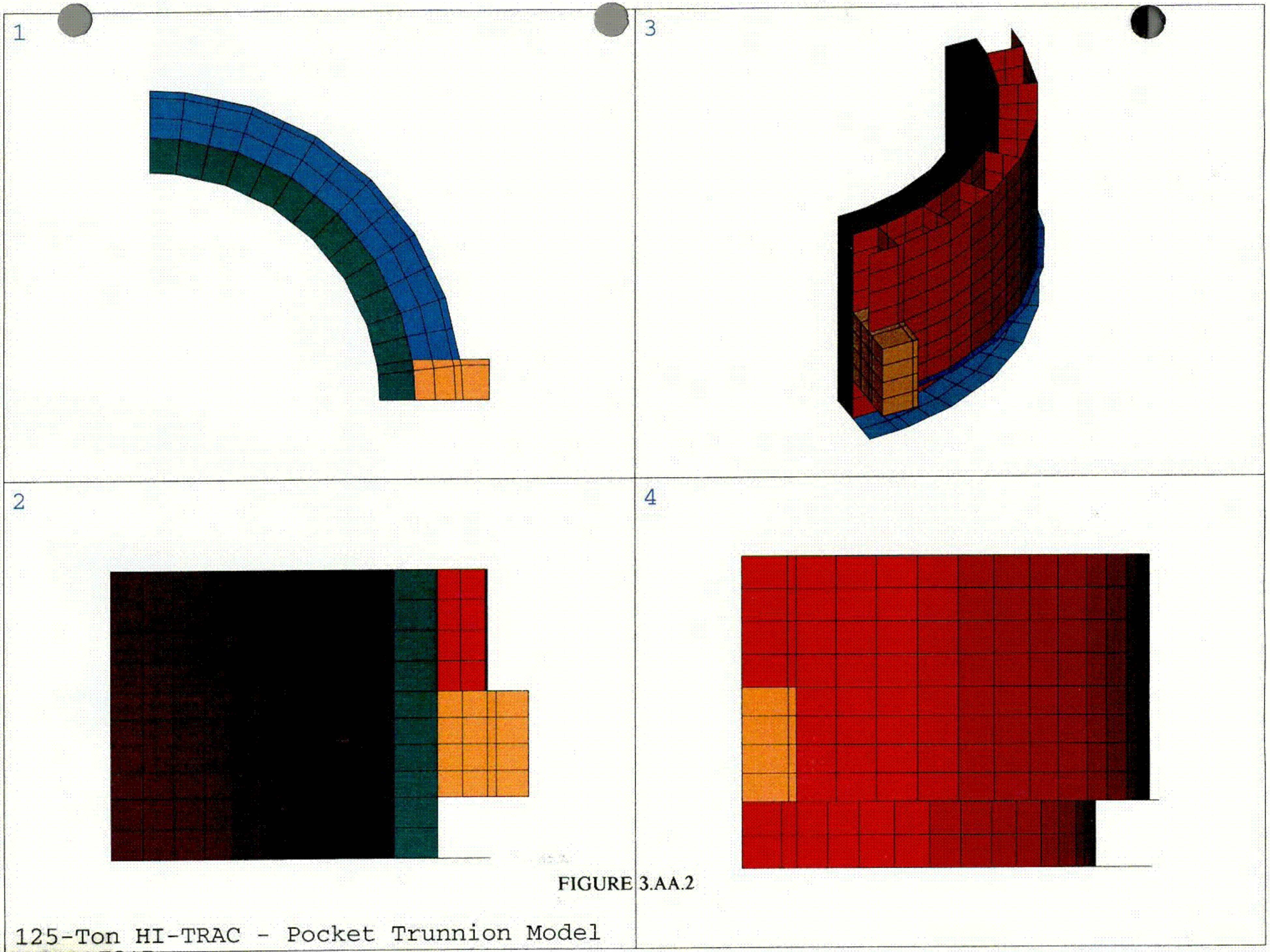


FIGURE 3.AA.2

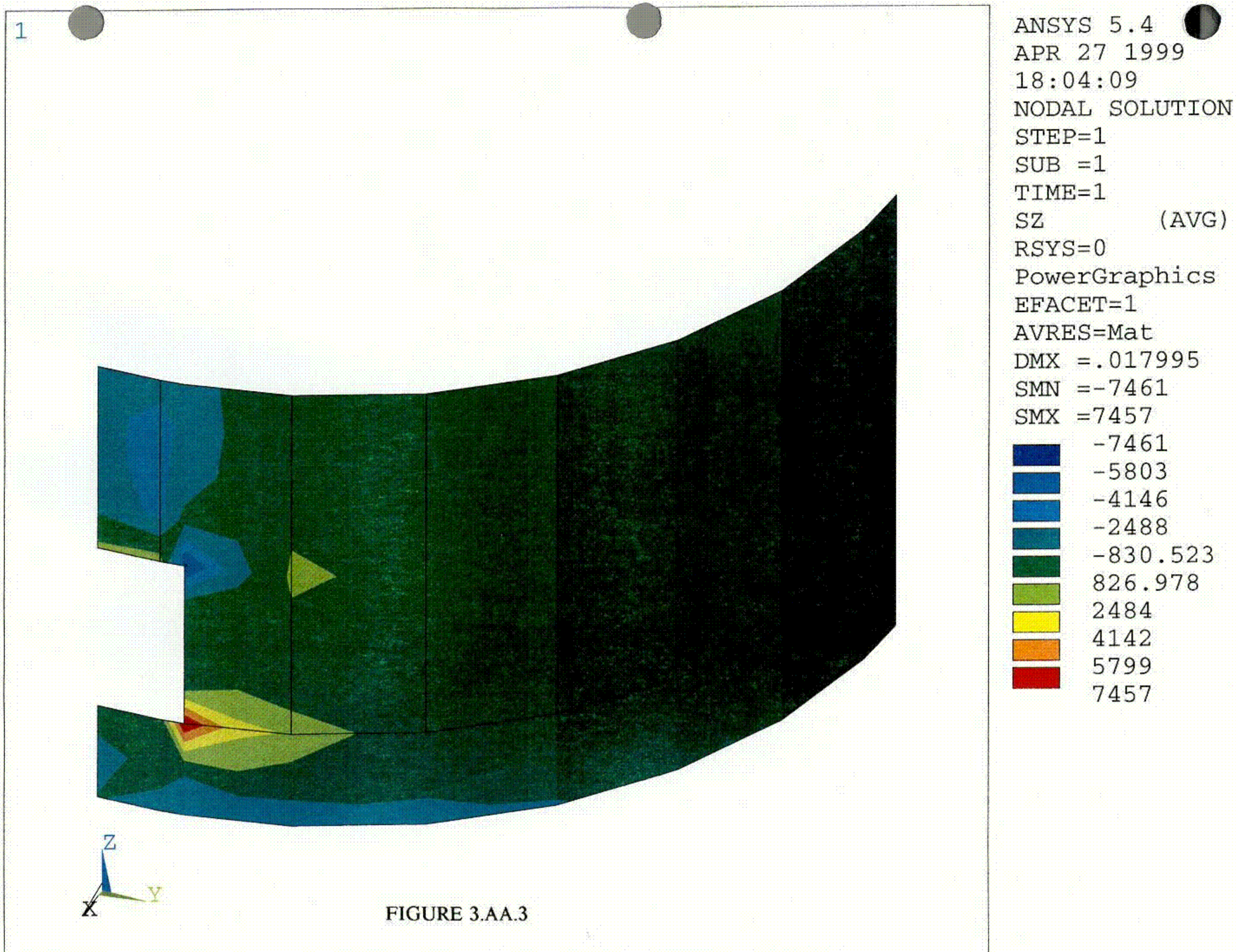
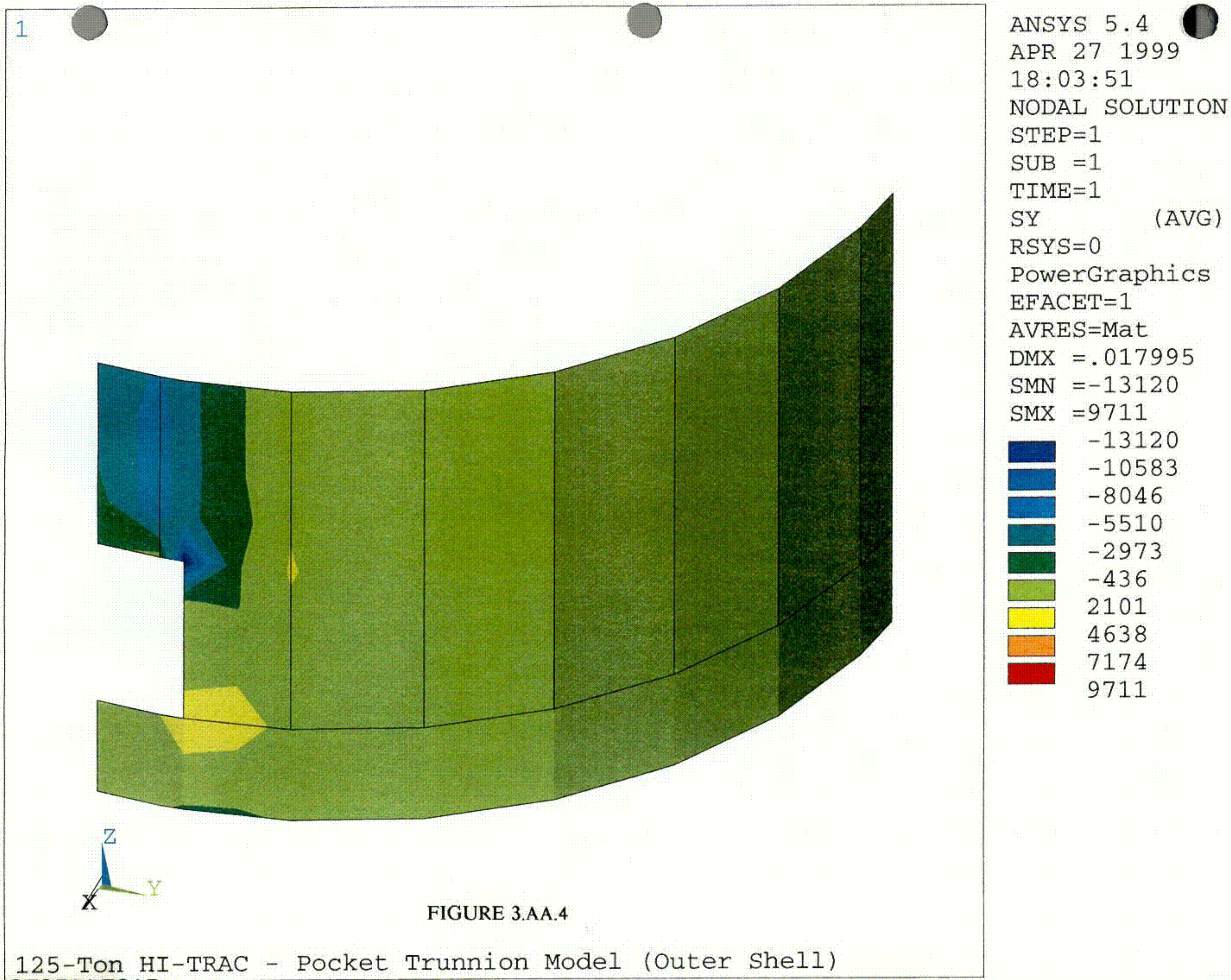


FIGURE 3.AA.3

125-Ton HI-TRAC - Pocket Trunnion Model (Outer Shell)


1
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APR 27 1999
18:03:51
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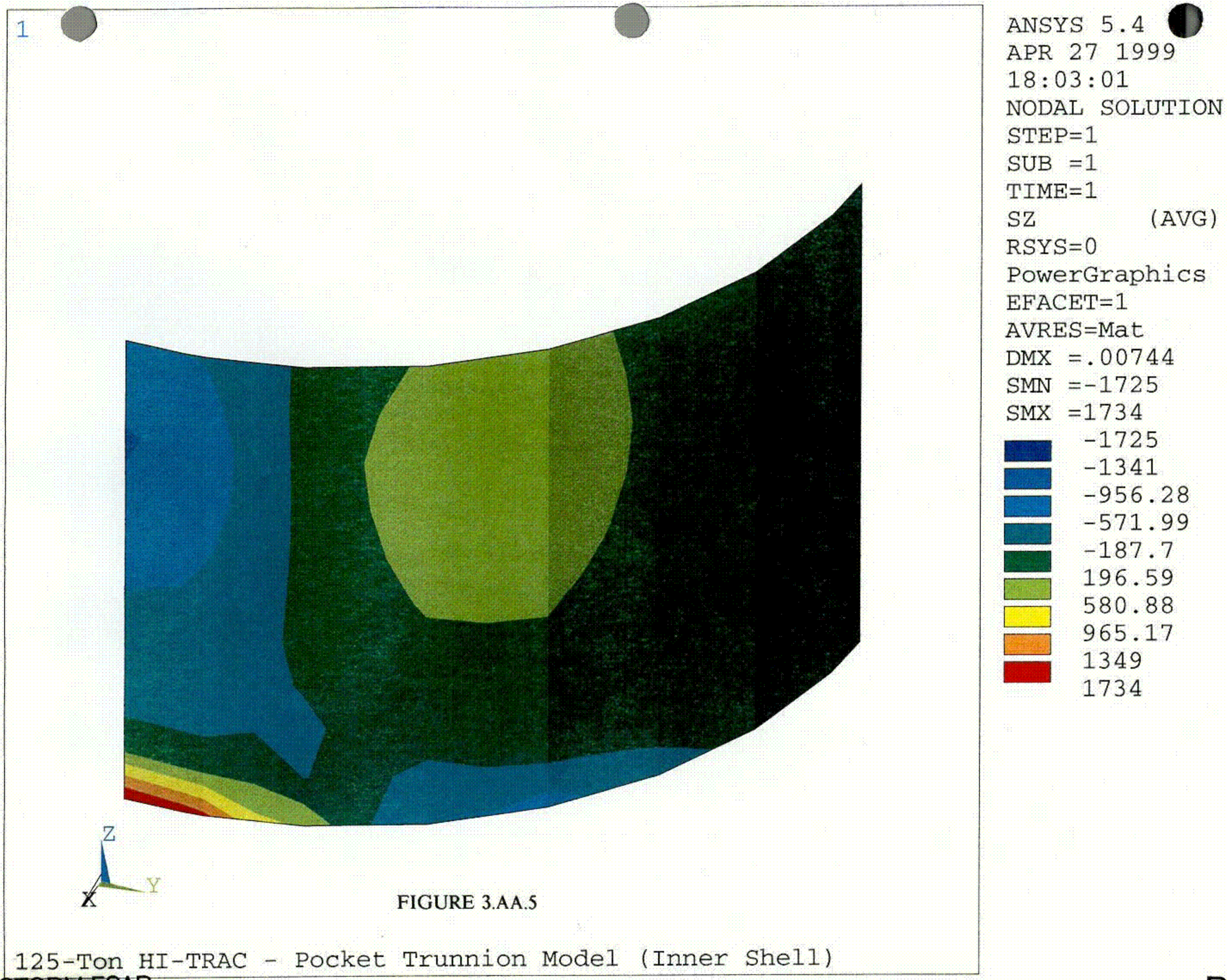
SUB =1
TIME=1
SY (AVG)
RSYS=0
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.017995
SMN =-13120
SMX =9711
-13120
-10583
-8046
-5510
-2973
-436
2101
4638
7174
9711
Z
Y
X
125-Ton HI-TRAC - Pocket Trunnion Model (Outer Shell)

FIGURE 3.AA.4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RSYS=0
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.00744
SMN =-1725
SMX =1734
-1725
-1341
-956.28
-571.99
-187.7
196.59
580.88
965.17
1349
1734
Z
Y
X
125-Ton HI-TRAC - Pocket Trunnion Model (Inner Shell)

FIGURE 3.AA.5

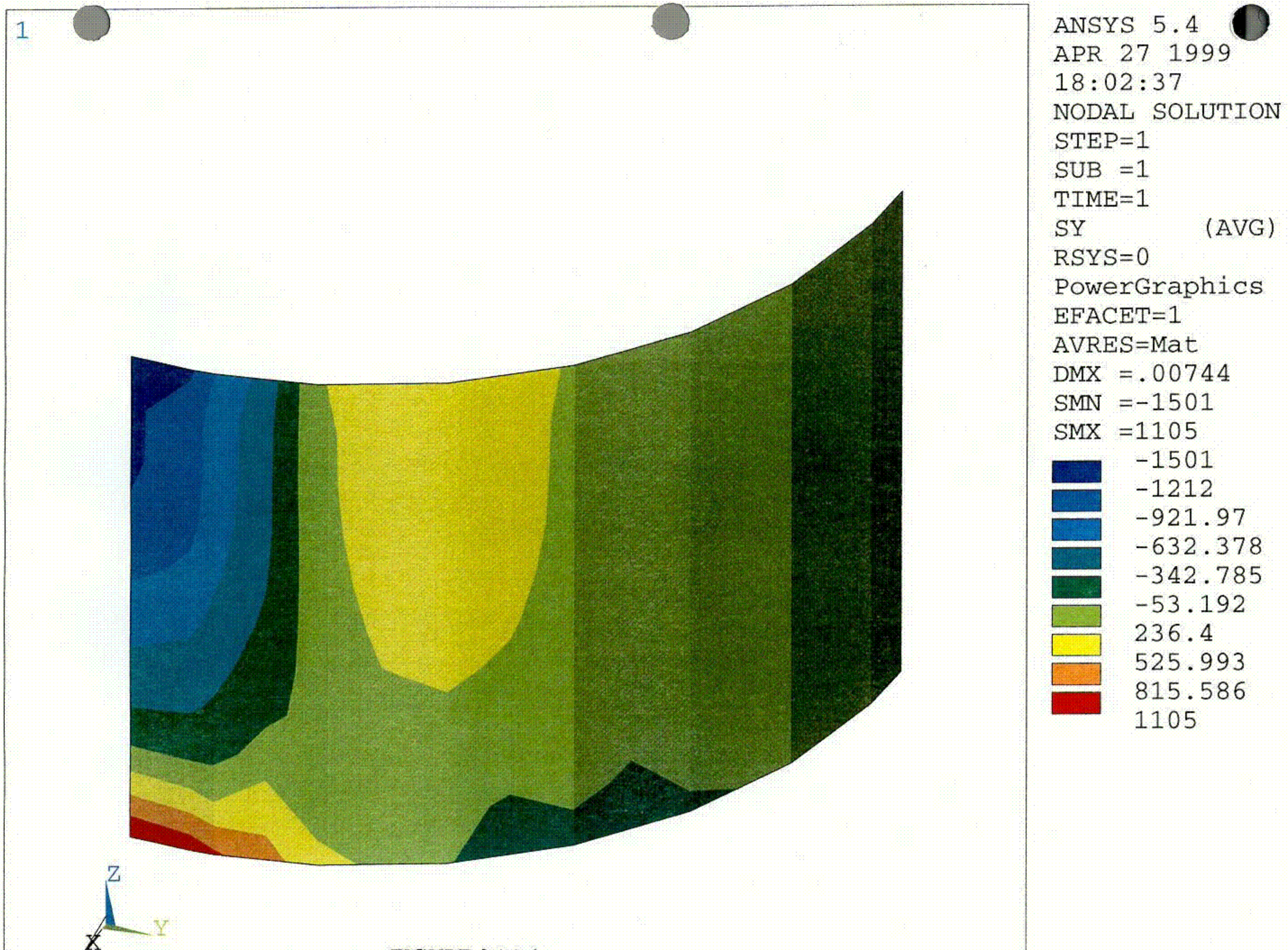


FIGURE 3.AA.6

125-Ton HI-TRAC - Pocket Trunnion Model (Inner Shell)

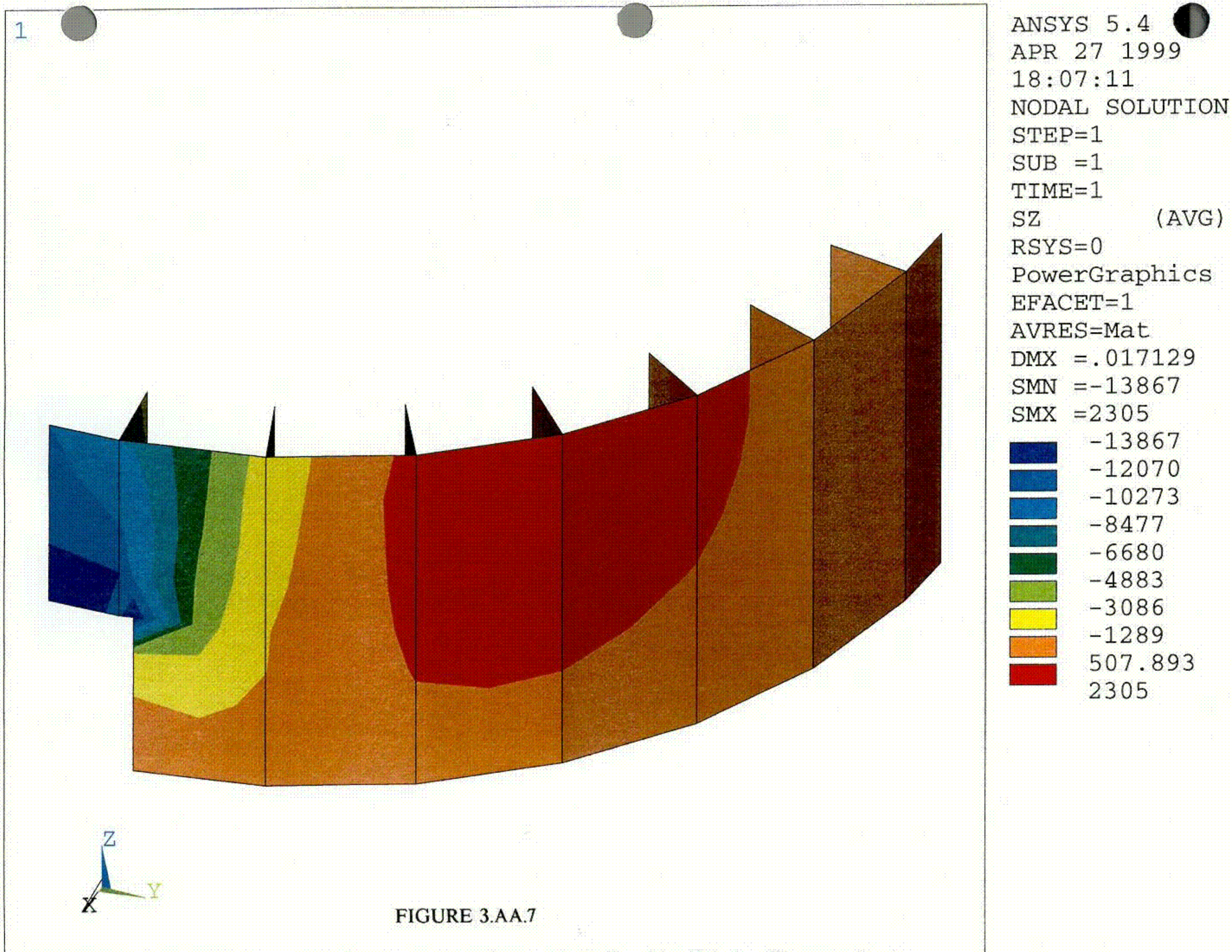
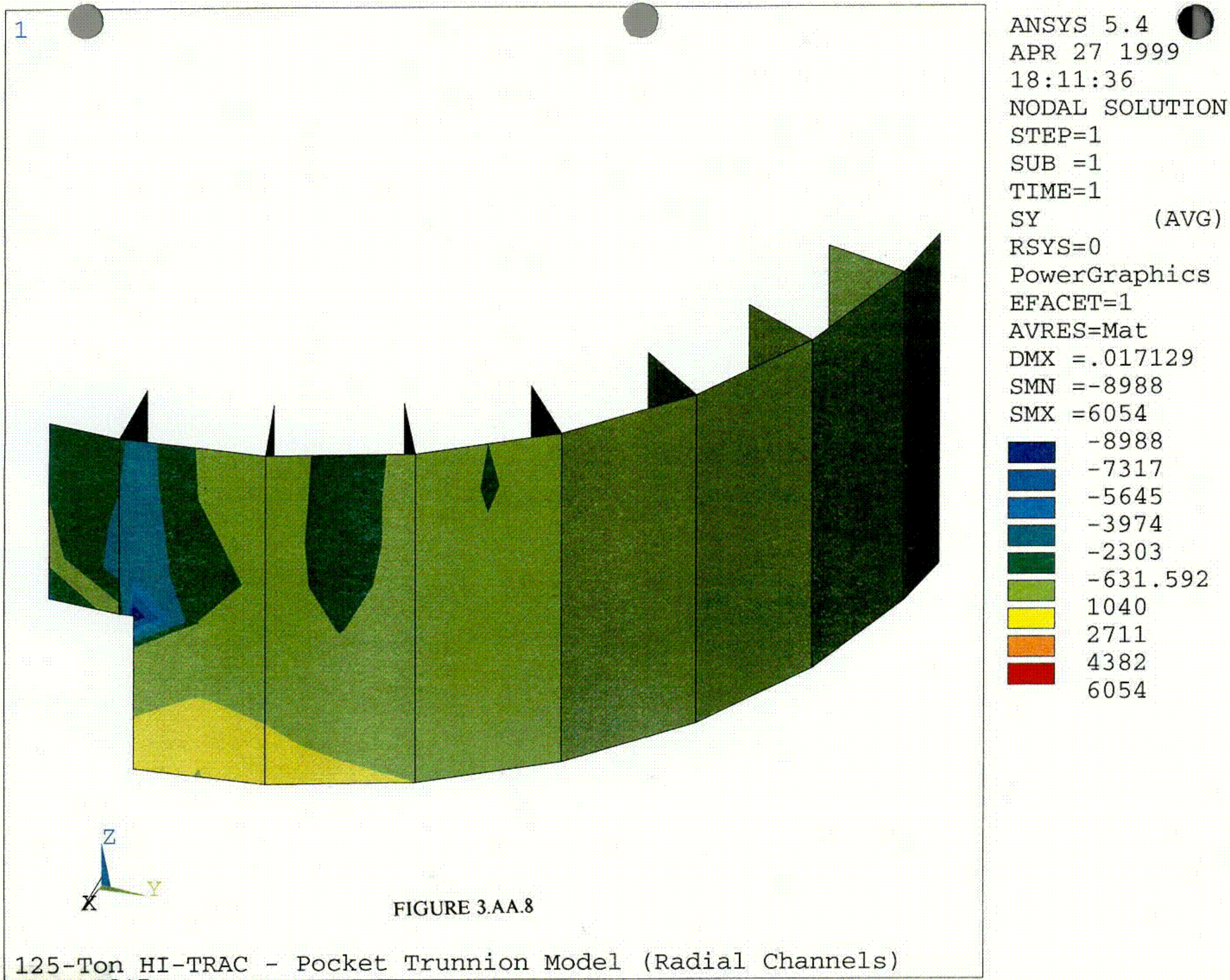


FIGURE 3.AA.7

125-Ton HI-TRAC - Pocket Trunnion Model (Radial Channels)



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TIME=1
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RSYS=0
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.017129
SMN =-8988
SMX =6054
-8988
-7317
-5645
-3974
-2303
-631.592
1040
2711
4382
6054
Z
Y
X
125-Ton HI-TRAC - Pocket Trunnion Model (Radial Channels)

FIGURE 3.AA.8

APPENDIX 3.AB HI-TRAC POOL LID STRESS AND CLOSURE ANALYSIS

3.AB.1 Introduction

The 125 ton HI-TRAC pool lid is made up of a top plate, a bottom plate, and gamma shielding material. The pool lid is bolted to the HI-TRAC bottom flange with 36 bolts designed to maintain a water seal during lifting from the spent fuel cask pit. The 100 ton HI-TRAC pool lid has the same construction with different thicknesses. This appendix demonstrates that the stress in the pool lid does not exceed Level A allowable strength under a pressure equivalent to the heaviest MPC, the contained water, and the self weight of the lid. To account for lifting dynamics, a 15% increase in the pressure is assumed. Analysis is also included to demonstrate that the yield strength is not exceeded under three times the lifted load. This calculation demonstrates compliance with Regulatory Guide 3.61.

3.AB.2 Methodology

Classical formulae for plate stress are used, together with an equilibrium analysis of the bolting, to compute the stress in the lid and the stress in the bolts. The 125 ton HI-TRAC pool lid is analyzed first in detail for the two steel plates which make up the pool lid; the calculations are repeated for the 100 ton HI-TRAC for the appropriate weight and dimensions.

3.AB.3 References

[3.AB.3.1] ASME Code Subsection NF, 1995.

[3.AB.3.2] J. Shigley and C. Mischke, Mechanical Engineering Design, 5th Edition, McGraw-Hill, 1989.

[3.AB.3.3] S.P. Timoshenko, Strength of Materials, Volume 2, Third Edition, McGraw-Hill, 1958, p.99.

3.AB.4 Assumptions

For the HI-TRAC pool lid, It is assumed that both plates supports the applied loading with the shield material acting only as a pressure transfer medium to load the bottom plate. The analysis is performed for the heaviest loaded MPC, with water in the HI-TRAC. The water weight is assumed distributed uniformly; the weight of the loaded MPC is conservatively imposed as a ring load at diameter 2/3*68.375" to represent the fact that the interface pressure is minimal at the center and maximized at the periphery of the interface between the MPC and the lid plate (i.e. assumed linearly distributed from center to periphery) .

For the 125 ton HI-TRAC and the 100 ton HI-TRAC, the lid diameter is taken as the bolt circle diameter.

Sections 3.AB.5-3.AB.10 show calculations for the 125 ton unit; Section 3.AB.11 and beyond contain the similar calculations for the 100 ton unit.

3.AB.5 Input Data for the 125 ton HI-TRAC

Diameter at lid flange $d_f := 90 \cdot in$

Diameter of MPC $d_{mpc} := 68.375 \cdot in$

Thickness of top plate $t_{plate} := 2 \cdot in$

Thickness of bottom plate $t_{bot} := 1 \cdot in$

Number of bolts $nb := 36$ Bolt Diameter $d_b := 1.0 \cdot in$

Stress area of 1" bolts $A_r := .6051 \cdot in^2$ [3.AB.3.2, Table 8.2]

The mechanical properties of SA193 Grade B7 for 2.5" to 4" diameters are conservatively used for the HI-TRAC lid bolts. The ultimate and yield strength of bolt material (SA-193 B7 @ 200 deg. F) are:

$S_u := 115000 \cdot psi$ Table 3.3.4

$S_{yb} := 95000 \cdot psi$

Allowable stress of SA516, Gr.70 @ 350 deg. F (membrane plus bending)

$S_a := 26300 \cdot psi$ Table 3.1.10

Dynamic Load Factor $DLF := .15$

Total Weight of pool lid $W_{lid} := 12500 \cdot lbf$ (Table 3.2.2)

Weight of Lead $W_{lead} := 4526 \cdot lbf$

Weight of Bottom plate $W_{bp} := 1800 \cdot lbf$

Bounding MPC weight $W_{MPC} := 90000 \cdot lbf$ Table 3.2.2

Weight of water in HI-TRAC $W_{water} := 17000 \cdot lbf$ (Table 3.2.4)

3.AB.6 Calculation of Lid Pressure Load

The bending moment from the ring load imposed by the loaded MPC is set equal to the expression for the maximum bending moment for a simply supported plate.

This defines an effective uniform pressure over the surface of the plate that can be used to simplify the calculation of maximum stress in the lid. This is an acceptable calculation since it is only maximum stress that is of interest, not stress distribution or deflection.

Calculation of effective pressure due to MPC loading. First, define the offset distance for computing the ring moment.

$$x := \frac{(d_f - .667 \cdot d_{mpc})}{2} \qquad x = 22.197\,in$$

$$M_{ring} := W_{MPC} \cdot \frac{x}{\pi \cdot d_{mpc}} \qquad M_{ring} = 9.3 \times 10^3\,in \cdot \frac{lbf}{in}$$

Let Poisson's ratio be $\nu := 0.3$

Then the effective uniform pressure that gives the same moment as that from the ring load is

$$p_1 := \frac{M_{ring}}{\frac{(3+\nu)}{8} \cdot \left(\frac{d_f}{2}\right)^2} \qquad p_1 = 11.134\,psi$$

Note that this effective pressure can only be used for bending stress calculation. It does not represent the pressure that would arise if the MPC load were assumed uniformly distributed

Pressure from water $$p_2 := \frac{W_{water}}{\pi \cdot \frac{d_f^2}{4}} \qquad p_2 = 2.672\,psi$$

Pressure from top lid plate self weight

$$p_3 := \frac{(W_{lid} - W_{lead} - W_{bp})}{\pi \cdot \frac{d_f^2}{4}} \qquad p_3 = 0.97\,psi$$

Lateral pressure for calculation of top lid stress, amplified by the DLF, is

$$q := (1 + DLF)\cdot(p_2 + p_3 + p_1) \qquad q = 16.993\,psi$$

The pressure on the bottom plate due to the lead and the bottom lid plate self weight is

$$dp := \frac{(W_{lead} + W_{bp})\cdot(1 + DLF)}{\pi \cdot \frac{d_f^2}{4}} \qquad dp = 1.144\,psi$$

It is assumed that both plates deflect by the same amount under the applied load.

The pressure load on the top plate is q-p while the pressure load on the bottom plate is p. p is the pressure transmitted by the shielding material. On the basis of equality of lateral displacement of the two plates under the pressures q-p and p+dp, respectively, the pressure p is:

$$p := \left[\frac{1}{\left(\frac{1}{t_{plate}^3} + \frac{1}{t_{bot}^3}\right)}\right]\cdot\left(\frac{q}{t_{plate}^3} - \frac{dp}{t_{bot}^3}\right) \qquad p = 0.872\,psi$$

3.AB.7 Calculation of Lid Stress

The maximum bending stress in the pool lid top plate is obtained from the equation

$$\sigma := .375\cdot(3.3)\cdot(q - p)\cdot\left(\frac{d_f}{2\cdot t_{plate}}\right)^2 \qquad [3.AB.3.3]$$

$$\sigma = 1.01 \times 10^4\,psi$$

The safety factor is $\frac{S_a}{\sigma} = 2.604$

The maximum bending stress in the pool lid bottom plate is obtained from the equation

$$\sigma := .375 \cdot (3.3) \cdot (p + dp) \cdot \left(\frac{d_f}{2 \cdot t_{bot}} \right)^2 \qquad [3.AB.3.3]$$

$\sigma = 5.05 \times 10^3\,psi$

The safety factor is $\frac{S_a}{\sigma} = 5.208$

3.A.B.8 Calculation of Bolt Stress and Shear Stress

$$Total\_Load := (W_{lid} + W_{water} + W_{MPC}) \cdot (1 + DLF)$$

$Total\_Load = 1.374 \times 10^5\,lbf$

$$Load\_per\_Bolt := \frac{Total\_Load}{nb}$$

$Load\_per\_Bolt = 3.817 \times 10^3\,lbf$

Bolt tensile stress to support applied pressure

$$\sigma_{bolt} := \frac{Load\_per\_Bolt}{A_r}$$

$\sigma_{bolt} = 6.309 \times 10^3\,psi$

From [3.AB.3.1), Section NF-3324.6, the allowable bolt stress is 50% of the ultimate strength of the bolting material. Therefore, the bolt safety factor is

$$SF_{bolt} := .5 \cdot \frac{S_u}{\sigma_{bolt}} \qquad SF_{bolt} = 9.114$$

The safety factor computed in accordance with Regulatory Guide 3.61 is

$$SF_{3.61} := \frac{S_{yb}}{3 \cdot \sigma_{bolt}} \qquad SF_{3.61} = 5.02$$

Note that the bolt size is not set by this calculation; the size is set by the requirement that the transfer lid and the HI-TRAC remain together during a side drop (see Appendix AD). In this lifting application, bolt tension is the only load.

The shear stress developed to support the total load is

$$\tau := \frac{Total\_Load}{\pi \cdot d_f \cdot t_{plate}} \qquad \tau = 243.02\,psi$$

This is well below the allowable stress in shear; we conclude that no further shear checks are necessary.

3.AB.9 Bolt Torque Requirements [3.AB.3.2]

$$T := .2 \cdot Load\_per\_Bolt \cdot d_b \qquad T = 63.623\,ft \cdot lbf$$

For bolts applied with Anti-Seize

$$T := .12 \cdot Load\_per\_Bolt \cdot d_b \qquad T = 38.174\,ft \cdot lbf$$

These are calculated minimum torques. In chapter 8, an increased initial torque is specified to provide a safety factor of 1.5 minimum on initial bolt torque.

3.AB.10 Lid Stresses Under 3 Times Lifted Load

The lid stress calculation is now repeated neglecting the lid weight, and increasing the lifted load by a factor of 3.0. It is desired to demonstrate that the safety factor against material yield is greater than 1.0. This requirement is imposed by USNRC Regulatory Guide 3.61.

$$DLF_1 := 3 \cdot 1.15$$

Lateral pressure for calculation of lid stress

$$q := (DLF_1)\cdot(p_1 + p_2 + p_3) \qquad q = 50.979\,psi$$

It is assumed that both plates deflect by the same amount under the applied load.

$$dp := \frac{(W_{lead} + W_{bp})\cdot DLF_1}{\pi\cdot\frac{d_f^2}{4}} \qquad dp = 3.431\,psi$$

The pressure load on the top plate is q-p while the pressure load on the bottom plate is p. p is the pressure transmitted by the shielding material. On the basis of equality of lateral displacement of the two plates under the pressures q-p and p+dp, respectively, the pressure p is calculated as:

$$p := \left[\frac{1}{\left(\frac{1}{t_{plate}^3} + \frac{1}{t_{bot}^3}\right)}\right]\cdot\left(\frac{q}{t_{plate}^3} - \frac{dp}{t_{bot}^3}\right) \qquad p = 2.615\,psi$$

Calculation of Lid Stress

The maximum bending stress in the pool lid top plate is obtained from the equation

$$\sigma := .375\cdot(3.3)\cdot(q - p)\cdot\left(\frac{d_f}{2\cdot t_{plate}}\right)^2 \qquad [3.AB.3.3]$$

$$\sigma = 3.03\times 10^4\,psi$$

Here the safety factor is calculated by a comparison with material yield strength

The safety factor is $\frac{33150\cdot psi}{\sigma} = 1.094$ Table 3.3.2

The maximum bending stress in the pool lid bottom plate is obtained from the equation

$$\sigma := .375 \cdot (3.3) \cdot (p + dp) \cdot \left(\frac{d_f}{2 \cdot t_{bot}} \right)^2$$ [3.AB.3.3]

$\sigma = 1.515 \times 10^4 \text{psi}$ $\frac{33150 \cdot \text{psi}}{\sigma} = 2.188$

Therefore, the lid plates maximum tensile stresses at the extreme fiber of the plate like members are below yield under 3 times the lifted load.

3.AB.11 Input Data for the 100 ton HI-TRAC

Diameter at lid bolts $d_f := 86.5 \cdot \text{in}$

Diameter of MPC $d_{mpc} := 68.375 \cdot \text{in}$

Thickness of top plate $t_{plate} := 2 \cdot \text{in}$

Thickness of bottom plate $t_{bot} := 0.5 \cdot \text{in}$

Number of bolts $nb := 36$ Bolt Diameter $d_b := 1.0 \cdot \text{in}$

Stress area of 1" bolts $A_r := .6051 \cdot \text{in}^2$ [3.AB.3.2, Table 8.2]

The mechanical properties of SA193 Grade B7 for 2.5" to 4" diameters are conservatively used for the HI-TRAC lid bolts. The ultimate and yield strength of bolt material (SA-193 B7 @ 200 deg. F) are:

$S_u := 115000 \cdot \text{psi}$ $S_{yb} := 95000 \cdot \text{psi}$ Table 3.3.4

Allowable stress of SA516, Gr.70 @ 350 deg. F (membrane plus bending)

$S_a := 26300 \cdot \text{psi}$ Table 3.1.10

Dynamic Load Factor $DLF := .15$

Total Weight of pool lid $W_{lid} := 8000 \cdot \text{lbf}$ (Table 3.2.2)

Weight of Lead $W_{lead} := 2715 \cdot \text{lbf}$

Weight of Bottom plate $W_{bp} := 831 \cdot \text{lbf}$

Bounding MPC weight	$W_{MPC} := 90000 \cdot lbf$	Table 3.2.2
Weight of water in HI-TRAC	$W_{water} := 17000 \cdot lbf$	(Table 3.2.4)

3.AB.12 Calculation of Lid Pressure Load

The bending moment from the ring load imposed by the loaded MPC is set equal to the expression for the maximum bending moment for a simply supported plate.

This defines an effective uniform pressure over the surface of the plate that can be used to simplify the calculation of maximum stress in the lid. This is an acceptable calculation since it is only maximum stress that is of interest, not stress distribution or deflection.

Calculation of effective pressure due to MPC loading. First, define the offset distance for computing the ring moment.

$$x := \frac{(d_f - .667 \cdot d_{mpc})}{2} \qquad x = 20.447\,in$$

$$M_{ring} := W_{MPC} \cdot \frac{x}{\pi \cdot d_{mpc}} \qquad M_{ring} = 8.567 \times 10^3\,in \cdot \frac{lbf}{in}$$

Let Poisson's ratio be $\nu := 0.3$

Then the effective uniform pressure that gives the same moment as that from the ring load is

$$p_1 := \frac{M_{ring}}{\frac{(3+\nu)}{8} \cdot \left(\frac{d_f}{2}\right)^2} \qquad p_1 = 11.103\,psi$$

Note that this effective pressure can only be used for bending stress calculation. It does not represent the pressure that would arise if the MPC load were assumed uniformly distributed

Pressure from water $$p_2 := \frac{W_{water}}{\pi \cdot \frac{d_f^2}{4}} \qquad p_2 = 2.893\,psi$$

Pressure from top lid plate self weight

$$p_3 := \frac{(W_{lid} - W_{lead} - W_{bp})}{\pi \cdot \frac{d_f^2}{4}} \qquad p_3 = 0.758\,psi$$

Lateral pressure for calculation of top lid stress, amplified by the DLF, is

$$q := (1 + DLF) \cdot (p_2 + p_3 + p_1) \qquad q = 16.966\,psi$$

The pressure on the bottom plate due to the lead and the bottom lid plate self weight is

$$dp := \frac{(W_{lead} + W_{bp}) \cdot (1 + DLF)}{\pi \cdot \frac{d_f^2}{4}} \qquad dp = 0.694\,psi$$

It is assumed that both plates deflect by the same amount under the applied load.

The pressure load on the top plate is q-p while the pressure load on the bottom plate is p. p is the pressure transmitted by the shielding material. On the basis of equality of lateral displacement of the two plates under the pressures q-p and p+dp, respectively, the pressure p is:

$$p := \left[\frac{1}{\left(\frac{1}{t_{plate}^3} + \frac{1}{t_{bot}^3} \right)} \right] \cdot \left(\frac{q}{t_{plate}^3} - \frac{dp}{t_{bot}^3} \right) \qquad p = -0.422\,psi$$

Note that a negative p is computed here because of the plate flexibility. Since it is not clear that the lead can support a tensile stress to maintain the assumption, we neglect p whenever its inclusion is non-conservative.

3.AB.13 Calculation of Lid Stress

The maximum bending stress in the pool lid top plate is obtained from the equation

$$\sigma := .375 \cdot (3.3) \cdot (q - p) \cdot \left(\frac{d_f}{2 \cdot t_{plate}} \right)^2 \qquad [3.AB.3.3]$$

$$\sigma = 1.006 \times 10^4 \text{psi}$$

The safety factor is $\frac{S_a}{\sigma} = 2.614$

The maximum bending stress in the pool lid bottom plate is obtained from the equation

$$\sigma := .375 \cdot (3.3) \cdot (dp) \cdot \left(\frac{d_f}{2 \cdot t_{bot}} \right)^2 \qquad [3.AB.3.3]$$

$$\sigma = 6.425 \times 10^3 \text{psi}$$

The safety factor is $\frac{S_a}{\sigma} = 4.093$

3.A.B.14 Calculation of Bolt Stress and Shear Stress

$$\text{Total\_Load} := \left(W_{lid} + W_{water} + W_{MPC} \right) \cdot (1 + DLF)$$

$$\text{Total\_Load} = 1.323 \times 10^5 \text{lbf}$$

$$\text{Load\_per\_Bolt} := \frac{\text{Total\_Load}}{nb}$$

$$\text{Load\_per\_Bolt} = 3.674 \times 10^3 \text{lbf}$$

Bolt tensile stress to support applied pressure

$$\sigma_{bolt} := \frac{Load\_per\_Bolt}{A_r}$$

$$\sigma_{bolt} = 6.071 \times 10^3 psi$$

From [3.AB.3.1), Section NF-3324.6, the allowable bolt stress is 50% of the ultimate strength of the bolting material. Therefore, the bolt safety factor is

$$SF_{bolt} := .5 \cdot \frac{S_u}{\sigma_{bolt}} \qquad SF_{bolt} = 9.471$$

The safety factor computed in accordance with Regulatory Guide 3.61 is

$$SF_{3.61} := \frac{S_{yb}}{3 \cdot \sigma_{bolt}} \qquad SF_{3\ 61} = 5.216$$

Note that the bolt size is not set by this calculation; the size is set by the requirement that the transfer lid and the HI-TRAC remain together during a side drop (see Appendix AD). In this lifting application, bolt tension is the only load.

The shear stress developed to support the total load is

$$\tau := \frac{Total\_Load}{\pi \cdot d_f \cdot t_{plate}} \qquad \tau = 243.332 psi$$

This is well below the allowable stress in shear; we conclude that no further shear checks are necessary.

3.AB.15 Bolt Torque Requirements [3.AB.3.2]

$$T := .2 \cdot Load\_per\_Bolt \cdot d_b \qquad T = 61.227 ft \cdot lbf$$

For bolts applied with Anti-Seize

$$T := .12 \cdot Load\_per\_Bolt \cdot d_b \qquad T = 36.736 ft \cdot lbf$$

These are calculated minimum torques. In chapter 8, an increased initial torque is specified to provide a safety factor of 1.5 minimum on initial bolt torque.

3.AB.16 Lid Stresses Under 3 Times Lifted Load

The lid stress calculation is now repeated neglecting the lid weight, and increasing the lifted load by a factor of 3.0. It is desired to demonstrate that the safety factor against material yield is greater than 1.0. This requirement is imposed by USNRC Regulatory Guide 3.61.

$$DLF_1 := 3 \cdot 1.15$$

Lateral pressure for calculation of lid stress

$$q := (DLF_1) \cdot (p_1 + p_2 + p_3) \qquad q = 50.899\,\text{psi}$$

It is assumed that both plates deflect by the same amount under the applied load.

$$dp := \frac{(W_{lead} + W_{bp}) \cdot DLF_1}{\pi \cdot \frac{d_f^2}{4}} \qquad dp = 2.082\,\text{psi}$$

The pressure load on the top plate is q-p while the pressure load on the bottom plate is p. p is the pressure transmitted by the shielding material. On the basis of equality of lateral displacement of the two plates under the pressures q-p and p+dp, respectively, the pressure p is calculated as:

$$p := \left[\frac{1}{\left(\frac{1}{t_{plate}^3} + \frac{1}{t_{bot}^3}\right)}\right] \cdot \left(\frac{q}{t_{plate}^3} - \frac{dp}{t_{bot}^3}\right) \qquad p = -1.267\,\text{psi}$$

As before, we include p only where the use of the negative sign is conservative.

Calculation of Lid Stress

The maximum bending stress in the pool lid top plate is obtained from the equation

$$\sigma := .375 \cdot (3.3) \cdot (q - p) \cdot \left(\frac{d_f}{2 \cdot t_{plate}} \right)^2 \qquad [3.AB.3.3]$$

$$\sigma = 3.019 \times 10^4 \text{psi}$$

Here the safety factor is calculated by a comparison with material yield strength

The safety factor is $\frac{33150 \cdot \text{psi}}{\sigma} = 1.098$ Table 3.3.2

The maximum bending stress in the pool lid bottom plate is obtained from the equation

$$\sigma := .375 \cdot (3.3) \cdot (dp) \cdot \left(\frac{d_f}{2 \cdot t_{bot}} \right)^2 \qquad [3.AB.3.3]$$

$$\sigma = 1.928 \times 10^4 \text{psi}$$

The safety factor is $\frac{33150 \cdot \text{psi}}{\sigma} = 1.72$ Table 3.3.2

Therefore, the lid plates maximum tensile stresses at the extreme fiber of the plate like members are below yield under 3 times the lifted load.

3.AB.18 Conclusions

Calculations have been performed for the pool lids for the 125 ton HI-TRAC and for the 100 ton HI-TRAC.

The pool lid and the bolts have acceptable safety factors even when 3 times the lifted load is applied.

The specified bolting is adequate to support the load. The actual bolt preload may vary according to gasket seating requirements, but adequate margins are listed in Chapter 8.

The lid plates maximum tensile stresses, at the extreme fiber of the plate like members, are below yield under 3 times the lifted load.

APPENDIX 3.AC - LIFTING CALCULATIONS

3.AC.1 Scope of Appendix

In this Appendix, the attachment locations that are used for lifting various lids are analyzed for strength and engagement length.The mating lifting device is not a part of this submittal but representative catalog items are chosen for analysis to demonstrate that commercially available lifting devices suffice to meet the required safety margins.

3.AC.2 Configuration

The required data for analysis is 1) the number of bolts NB; 2) the bolt diameter db; 3) the lifted weight; and 4), the details of the individual bolts.

3.AC.3 Acceptance Criteria

The lifting bolts are considered as part of a special lifting device; therefore, NUREG-0612 applies. The acceptance criteria is that the bolts and the adjacent lid threads must have stresses less than 1/3 x material yield strength and 1/5 x material ultimate strength. These reduced requirements are acceptable since the outer diameters of the lifted parts are larger than the inside diameter of the cavity under the lifted parts; therefore, the lifted parts cannot impact stored fuel directly as long as sufficient controls are maintained on carry heights to preclude inordinant lid rotations in the event of a handling accident.

3.AC.4 Composition of Appendix

This appendix is created using the Mathcad (version 2000) software package. Mathcad uses the symbol ':=' as an assignment operator, and the equals symbol '=' retrieves values for constants or variables.

3.AC.5 References

[3.AC.1] E. Oberg and F.D. Jones, *Machinery's Handbook*, Fifteenth Edition, Industrial Press, 1957, pp987-990.

[3.AC.2] FED-STD-H28/2A, *Federal Standard Screw-Thread Standards for Federal Services*, United States Government Printing Office, April, 1984.

3.AC.6 Input Data for Lifting of Overpack Top Lid (HI-STORM 100S bounds)

Lifted Weight (Table 3.2.1): $W_{lift} := (25500 \cdot 1.15) \cdot lbf$ includes 15% inertia load factor

The following input parameters are taken from Holtec Dwgs. for 100S lid.

Bolt diameter $db := 1.5 \cdot in$ (Dwg. 3072)

$N := 6 \cdot \frac{1}{in}$ is the number of threads per inch (UNC)

$L_{eng} := 1.5 \cdot in$ is the length of engagement (lower of two 2" top plates, Dwg. 1561).

Number of Bolts $NB := 4$

Lifting of the HI-STORM 100 lid is limited to a straight (90 deg) lift. For conservatism the minimum lift angle (from the horizontal) is assumed to be 65 degees:

$$ang := 65 \cdot deg$$

$A_d := \pi \cdot \frac{db^2}{4}$ $A_d = 1.767\,in^2$ is the area of the unthreaded portion of the bolt

$A_{stress} := 1.405 \cdot in^2$ is the stress area of the bolt

$d_{pitch} := 1.3917 \cdot in$ is the pitch diameter of the bolt

$dm_{ext} := 1.2955 \cdot in$ is the minor diameter of the bolt

$dm_{int} := 1.3196 \cdot in$ is the minor diameter of the hole

The design temperature of the top lid, located atop the overpack, is 350 deg. F. The lid lifting bolts, will not see this temperature under normal circumstances. For conservatism, the material properties and allowable stresses for the lid used in the qualification are taken at 350 deg F.

The yield and ultimate strengths of the overpack top lid are reduced by factors of 3 and 5, respectively. The eyebolt working load limit(not part of the HI-STORM 100 System) will have a safety factor of 5.

$S_{ulid} := \frac{70000}{5} \cdot psi$ (Table 3.3.2) $S_{ylid} := \frac{33150}{3} \cdot psi$ (Table 3.3.2)

The yield stress criteria governs the analysis.

3.AC.7 Calculations

3.AC.7.1 Length of Engagement/Strength Calculations

In this section, it is shown that the length of thread engagement is adequate The method and terminology of Reference 3.AC.2 is followed.

$p := \frac{1}{N}$ is the thread pitch

$H := 4 \cdot 0.21651 \cdot p$ $H = 0.144\,in$

$Depth_{ext} := \frac{17}{24} \cdot H$ $Depth_{ext} = 0.102\,in$

$Depth_{int} := \frac{5}{8} \cdot H$ $Depth_{int} = 0.09\,in$

$dmaj_{ext} := dm_{ext} + 2 \cdot Depth_{ext}$ $dmaj_{ext} = 1.5\,in$

Using page 103 of reference 3.AC.2,

$$Bolt\_thrd\_shr\_A := \pi \cdot N \cdot L_{eng} \cdot dm_{int} \cdot \left[\frac{1}{2 \cdot N} + .57735 \cdot \left(d_{pitch} - dm_{int} \right) \right]$$

$$Bolt\_thrd\_shr\_A = 4.662\,in^2$$

$$Ext\_thrd\_shr\_A := \pi \cdot N \cdot L_{eng} \cdot dmaj_{ext} \cdot \left[\frac{1}{2 \cdot N} + 0.57735 \cdot \left(dmaj_{ext} - d_{pitch} \right) \right]$$

$$Ext\_thrd\_shr\_A = 6.186\,in^2$$

The normal stress capacities of the bolt, and load capacity of the top lid material, based on yield strength, are (the shear area is taken as the stress area here since the lifting bolt that also fits into this hole is not part of the HI-STORM 100 System. The representative lid lifting bolt specification for the analysis is assumed as equivalent to Crosby S-279, Part Number 9900271):

$Load\_Capacity_{bolt} := 21400 \cdot lbf$ $Load\_Capacity_{bolt} = 2.14 \times 10^4\,lbf$

$$Load\_Capacity_{lid} := (0.577 \cdot S_{ylid}) \cdot Ext\_thrd\_shr\_A$$

$$Load\_Capacity_{lid} = 3.944 \times 10^{4} \, lbf$$

Therefore, the lifting capacity of the configuration is based on bolt shear due to lid thread capacity or the actual catalog rated capacity of the bolt adjusted for the angled lift.

$$Max\_Lift\_Load := NB \cdot Load\_Capacity_{lid} \qquad Max\_Lift\_Load = 1.578 \times 10^{5} \, lbf$$

$$SF := \frac{Max\_Lift\_Load}{W_{lift}} \qquad SF = 5.38 \qquad > 1$$

Even though a vertical lift is required, the safety factor is consistently and conservatively computed based on the assumed lift angle:

or

$$SF := \frac{NB \cdot Load\_Capacity_{bolt} \cdot 0.611}{W_{lift}} \qquad SF = 1.784 \qquad > 1$$

Note that the minimum safety factor based on bolt rated capacity does not include the built-in catalog rated safety factor of 5. The factor of 0.611 is based on an interpolation of the reduction factor stated in the Crosby Catalog (p. 72) for off angle lifts as computed below:

For a 45 degree off-angle, the reduction factor is 0.70; therefore for the assumed 25 degree off-angle,

$$\frac{(90 \cdot deg - ang)}{45 \cdot deg} \cdot 0.70 = 0.389 \qquad 1 - 0.389 = 0.611$$

3.AC.8 Input Data for Lifting of HI-TRAC Pool Lid

Lifted Weight: (the HI-TRAC 125 pool lid bounds all other lids - this is the only load)

$Weight := 12500 \cdot lbf$ Table 3.2.2. This load bounds all other lids that may be lifted.

$ang := 45 \cdot deg$ Minimum Lift Angle from Horizontal (to bound all lifts other than the HI-STORM 100 top lid)

$inertia\_load\_factor := .15$

$W_{lift} := Weight \cdot (1.0 + inertia\_load\_factor)$

$W_{lift} = 1.437 \times 10^4 lbf$ includes any anticipated inertia load factor

The assumed representative lifting bolts used for the analysis herein are High-Load Lifting Bolts per McMaster-Carr Catalog 104, p. 929, Part Number 3026T34.

$Working\_Load := 17000 \cdot lbf$ These lifting bolts are designed for off-vertical lifts

Bolt diameter $db := .875 \cdot in$

Number of Bolts $NB := 4$

$N := 9 \cdot \frac{1}{in}$ is the number of threads per inch

$L_{eng} := 1.375 \cdot in$ is the length of engagement (per M-C catalog)

The material properties are those of SA 516 Grade 70 @ 350 deg. F. From Table 3.3.2,

$S_{ulid} := \frac{70000 \cdot psi}{5}$ $S_{ylid} := \frac{33150 \cdot psi}{3}$

$A_d := \pi \cdot \frac{db^2}{4}$ $A_d = 0.601 in^2$ is the area of the unthreaded portion of the bolt

$A_{stress} := .462 \cdot in^2$ is the stress area of the bolt

$d_{pitch} := .8028 \cdot in$ is the pitch diameter of the bolt

$dm_{ext} := .7427 \cdot in$ is the major diameter of the bolt

$dm_{int} := .7547 \cdot in$ is the minor diameter of the threaded hole

Thread properties are from Machinery's Handbook, 23rd Edition, Table 3a, p.1484

3.AC.9 Calculations

Length of Engagement/Strength Calculations

In this section, it is shown that the length of thread engagement is adequate The method and terminology of reference 3.AC.2 is followed.

$p := \frac{1}{N}$ is the thread pitch

$H := 4 \cdot 0.21651 \cdot p$ $H = 0.096\,in$

$Depth_{ext} := \frac{17}{24} \cdot H$ $Depth_{ext} = 0.068\,in$

$Depth_{int} := \frac{5}{8} \cdot H$ $Depth_{int} = 0.06\,in$

$dmaj_{ext} := dm_{ext} + 2 \cdot Depth_{ext}$ $dmaj_{ext} = 0.879\,in$

Using page 103 of reference 3.AC.2,

$$Bolt\_thrd\_shr\_A := \pi \cdot N \cdot L_{eng} \cdot dm_{int} \cdot \left[\frac{1}{2 \cdot N} + .57735 \cdot \left(d_{pitch} - dm_{int} \right) \right]$$

$$Bolt\_thrd\_shr\_A = 2.445\,in^2$$

$$Ext\_thrd\_shr\_A := \pi \cdot N \cdot L_{eng} \cdot dmaj_{ext} \cdot \left[\frac{1}{2 \cdot N} + 0.57735 \cdot \left(dmaj_{ext} - d_{pitch} \right) \right]$$

$$Ext\_thrd\_shr\_A = 3.402\,in^2$$

The load capacity of the lid material based on yield strength is:

$$Load\_Capacity_{lid} := \left(0.577 \cdot S_{ylid}\right) \cdot Ext\_thrd\_shr\_A$$

$$Load\_Capacity_{lid} = 2.169 \times 10^4\,lbf$$

Therefore, the lifting capacity of the configuration, based on lid shear, is.

$$Max\_Lift\_Load_{lidshear} := NB \cdot Load\_Capacity_{lid}$$

$$Max\_Lift\_Load_{lidshear} = 8.677 \times 10^{4}\,lbf$$

The safety factor is defined as

$$SF := \frac{Max\_Lift\_Load_{lidshear}}{W_{lift}} \qquad SF = 6.036 \quad > 1$$

The safety factor, based on the working load limit specified in the McMaster-Carr Catalog, is

$$SF_b := \frac{Working\_Load}{0.25 \cdot W_{lift}} \qquad SF_b = 4.73$$

3.AC.10 Input Data for Lifting of HI-TRAC Top Lid

Lifted Weight: (the HI-TRAC 125 top lid bounds all other lids - this is the only load)

$Weight := 2750 \cdot lbf$ Table 3.2.2

$ang := 45 \cdot deg$ Minimum Lift Angle from Horizontal (to bound all lifts other than the HI-STORM 100 top lid)

$inertia\_load\_factor := .15$

$$W_{lift} := Weight \cdot (1.0 + inertia\_load\_factor)$$

$W_{lift} = 3.163 \times 10^{3}\,lbf$ includes any anticipated inertia load factor

The lifting bolts assumed as representative for the analysis herein are High-Load Lifting Bolts per McMaster-Carr Catalog 104, p. 929, Part Number 3026T32.

$Working\_Load := 9000 \cdot lbf$ These lifting bolts are designed for off-vertical lifts

Bolt diameter $db := .625 \cdot in$

Number of Bolts $NB := 4$

$N := 11 \cdot \frac{1}{in}$ is the number of threads per inch

$L_{eng} := 1.0 \cdot in$ is the length of engagement (per M-C catalog)

For , the material properties are those of SA 516 Grade 70 @ 350 deg. F. From Table 3.3.2,

$$S_{ulid} := \frac{70000 \cdot psi}{5} \qquad S_{ylid} := \frac{33150 \cdot psi}{3}$$

$A_d := \pi \cdot \frac{db^2}{4}$ $\quad A_d = 0.307\,in^2$ is the area of the unthreaded portion of the bolt

$A_{stress} := .226 \cdot in^2$ is the stress area of the bolt

$d_{pitch} := .566 \cdot in$ is the pitch diameter of the bolt

$dm_{ext} := .5168 \cdot in$ is the major diameter of the bolt

$dm_{int} := .5266 \cdot in$ is the minor diameter of the threaded hole

Thread properties are from Machinery's Handbook, 23rd Edition, Table 3a, p.1484

3.AC.11 Calculations

Length of Engagement/Strength Calculations

In this section, it is shown that the length of thread engagement is adequate The method and terminology of reference 3.AC.2 is followed.

$p := \frac{1}{N}$ is the thread pitch

$H := 4 \cdot 0.21651 \cdot p$ $\quad H = 0.079\,in$

$Depth_{ext} := \frac{17}{24} \cdot H$ $\quad Depth_{ext} = 0.056\,in$

$Depth_{int} := \frac{5}{8} \cdot H$ $\quad Depth_{int} = 0.049\,in$

$dmaj_{ext} := dm_{ext} + 2 \cdot Depth_{ext}$ $\quad dmaj_{ext} = 0.628\,in$

Using page 103 of reference 3.AC.2,

$$\text{Bolt\_thrd\_shr\_A} := \pi \cdot N \cdot L_{eng} \cdot dm_{int} \cdot \left[\frac{1}{2 \cdot N} + .57735 \cdot \left(d_{pitch} - dm_{int} \right) \right]$$

$$\text{Bolt\_thrd\_shr\_A} = 1.241\ \text{in}^2$$

$$\text{Ext\_thrd\_shr\_A} := \pi \cdot N \cdot L_{eng} \cdot dmaj_{ext} \cdot \left[\frac{1}{2 \cdot N} + 0.57735 \cdot \left(dmaj_{ext} - d_{pitch} \right) \right]$$

$$\text{Ext\_thrd\_shr\_A} = 1.768\ \text{in}^2$$

The load capacity of the lid material based on yield strength is:

$$\text{Load\_Capacity}_{lid} := \left(0.577 \cdot S_{ylid}\right) \cdot \text{Ext\_thrd\_shr\_A}$$

$$\text{Load\_Capacity}_{lid} = 1.128 \times 10^4\ \text{lbf}$$

Therefore, the lifting capacity of the configuration, based on lid shear, is.

$$\text{Max\_Lift\_Load}_{lidshear} := NB \cdot \text{Load\_Capacity}_{lid}$$

$$\text{Max\_Lift\_Load}_{lidshear} = 4.51 \times 10^4\ \text{lbf}$$

The safety factor is defined as

$$SF := \frac{\text{Max\_Lift\_Load}_{lidshear}}{W_{lift}} \qquad SF = 14.261 \quad > 1$$

The safety factor, based on the working load limit specified in the McMaster-Carr Catalog, is

$$SF_b := \frac{\text{Working\_Load}}{0.25 \cdot W_{lift}} \qquad SF_b = 11.383$$

3.AC.12 Conclusion

The preceding analysis demonstrates that the length of thread engagement at the lifting locations are conservatively set. When lifting of the component is not being performed, plugs of a non-galling material with properties equal to or better than the base material shall be in-place to provide a filler material.

3.AC.13 Length of Engagement for Circumferential Bolts in HI-TRAC Pool Lid

Input Data for Check of thread engagement

Total supported load: $W_{lift} := 119500 \cdot 1.15 \cdot lbf$ From Appendix 3.AB

with a 15% dynamic load factor

Bolt diameter $db := 1.0 \cdot in$ Holtec drawing no. 1880

Number of Bolts $NB := 36$ Holtec drawing no. 1880

$N := 8 \cdot \frac{1}{in}$ is the number of threads per inch Holtec drawing no. 1880

$L_{eng} := 0.5 \cdot in$ is the length of engagement Holtec drawing no. 1880

$A_d := \pi \cdot \frac{db^2}{4}$ $A_d = 0.785\,in^2$ is the area of the unthreaded portion of the bolt

$A_{stress} := 0.606 \cdot in^2$ is the stress area of the bolt

$d_{pitch} := 0.9188 \cdot in$ is the pitch diameter of the bolt

Per Table 3a of Machinery's Handbook, 23rd Edition, p. 1484

$dm_{ext} := 0.8512 \cdot in$ is the minor diameter of the bolt

$dm_{int} := 0.8647 \cdot in$ is the minor diameter of the hole

For conservatism, the material properties and allowable stresses for the pool lid bolts and the lid used in the qualification are taken at 350 deg F for the lid, and 300 deg. F for the bolts. The mechanical properties of SA193 Grade B7 for 2.5" to 4" diameters are conservatively used for the HI-TRAC lid bolts.

The yield and ultimate strengths of the lid, and the bolts are:

$Su_{lid} := \frac{70000}{5} \cdot psi$ $Su_{bolt} := \frac{103016 \cdot psi}{5}$

$Sy_{lid} := \frac{33150}{3} \cdot psi$ $Sy_{bolt} := \frac{85100 \cdot psi}{3}$

SA-193-B7 bolts Table 3.3.4

3.AC.13.1 Length of Engagement/Strength Calculations

In this section, it is shown that the length of thread engagement is adequate The method and terminology of reference 3.AC.2 is followed.

$p := \frac{1}{N}$ is the thread pitch $p = 0.125\,in$

$H := 4 \cdot 0.21651 \cdot p$ $H = 0.108\,in$

$Depth_{ext} := \frac{17}{24} \cdot H$ $Depth_{ext} = 0.077\,in$

$Depth_{int} := \frac{5}{8} \cdot H$ $Depth_{int} = 0.068\,in$

$dmaj_{ext} := dm_{ext} + 2 \cdot Depth_{ext}$ $dmaj_{ext} = 1.005\,in$

Using page 103 of reference 3.AC.2,

$$Bolt\_thrd\_shr\_A := \pi \cdot N \cdot L_{eng} \cdot dm_{int} \cdot \left[\frac{1}{2 \cdot N} + .57735 \cdot \left(d_{pitch} - dm_{int} \right) \right]$$

$$Bolt\_thrd\_shr\_A = 1.019\,in^2$$

$$Ext\_thrd\_shr\_A := \pi \cdot N \cdot L_{eng} \cdot dmaj_{ext} \cdot \left[\frac{1}{2 \cdot N} + 0.57735 \cdot \left(dmaj_{ext} - d_{pitch} \right) \right]$$

$$Ext\_thrd\_shr\_A = 1.414\,in^2$$

The load capacities of the bolt and the lid material based on yield strength are:

$Load\_Capacity_{bolt} := S_{ybolt} \cdot A_{stress}$ $Load\_Capacity_{bolt} = 1.719 \times 10^4\,lbf$

$$Load\_Capacity_{boltthrd} := \left(0.577 \cdot S_{ybolt} \right) \cdot Bolt\_thrd\_shr\_A$$

$$Load\_Capacity_{boltthrd} = 1.667 \times 10^4\,lbf$$

$$Load\_Capacity_{lid} := (0.577 \cdot S_{ylid}) \cdot Ext\_thrd\_shr\_A$$

$$Load\_Capacity_{lid} = 9.016 \times 10^{3}\,lbf$$

Therefore, the capacity of the configuration is based on base metal thread shear.

$$Max\_Lift\_Load := NB \cdot Load\_Capacity_{lid} \qquad Max\_Lift\_Load = 3.246 \times 10^{5}\,lbf$$

The safety factor is

$$SF := \frac{Max\_Lift\_Load}{W_{lift}} \qquad SF = 2.362 \qquad >1$$

The load capacities of the bolt and the lid material based on ultimate strength are:

$$Load\_Capacity_{bolt} := S_{ubolt} \cdot A_{stress}$$

$$Load\_Capacity_{bolt} = 1.249 \times 10^{4}\,lbf$$

$$Load\_Capacity_{boltthrd} := (0.577 \cdot S_{ubolt}) \cdot Bolt\_thrd\_shr\_A$$

$$Load\_Capacity_{boltthrd} = 12108\,lbf$$

$$Load\_Capacity_{lid} := (0.577 \cdot S_{ulid}) \cdot Ext\_thrd\_shr\_A$$

$$Load\_Capacity_{lid} = 1.142 \times 10^{4}\,lbf$$

Therefore, the load capacity is based on base metal shear.

$$Max\_Lift\_Load := NB \cdot Load\_Capacity_{lid} \qquad Max\_Lift\_Load = 4.112 \times 10^{5}\,lbf$$

and the safety factor is

$$SF := \frac{Max\_Lift\_Load}{W_{lift}} \qquad SF = 2.992 \qquad > 1$$

Therefore, it is shown that the HI-TRAC pool lid bolts have adequate engagement length into the lid to permit the transfer of the required load.

APPENDIX 3.AD 125 TON HI-TRAC TRANSFER LID STRESS ANALYSES

3.AD.1 Introduction

This appendix considers the structural analysis of the HI-TRAC transfer lid under the following limiting conditions:

Lifting of fully loaded MPC - Normal Condition
Horizontal Drop of HI-TRAC - Accident Condition

In the first case, it is shown that the sliding doors adequately support a loaded MPC plus the door weight, both being amplified by a dynamic load factor associated with a low speed lifting operation, and that the loads are transferred to the transfer cask body without overstress.

In the second case, analysis is performed to show that the transfer lid and the transfer cask body do not separate during a HI-TRAC horizontal drop which imposes a deceleration load on the connection. In this case, because of the geometry of the transfer lid housing, the force of separation is from the HI-TRAC since the housing impacts the ground before the HI-TRAC body; i.e., the connection needs to withstand an amplified load from the HI-TRAC loaded weight, amplified by the deceleration. Analysis is also performed to show that the bolts that act as "door stops" will keep the doors from opening due to deceleration from a side drop.

3.AD.2 References

[3.AD.2.1] Young, Warren C., Roark's Formulas for Stress and Strain, 6th Edition, McGraw-Hill,1989.

[3.AD.2.2] Holtec Drawing 1928 (two sheets)

[3.AD.2.3] J.Shigley and C. Mischke, Mechanical Engineering Design, McGraw Hill, 1989.

[3.AD.2.4] McMaster-Carr Supply Company, Catalog No. 101, 1995.

[3.AD.2.5] Machinery's Handbook, 23rd Edition, Industrial Press

3.AD.3 Composition

This appendix was created using the Mathcad (version 8.0) software package. Mathcad uses the symbol ':=' as an assignment operator, and the equals symbol '=' retrieves values for constants or variables.

3.AD.4 General Assumptions

1. Formulas taken from Reference [3.AD.2.1] are based on assumptions that are delineated in that reference.

2. During lifting operation, the MPC is supported on a narrow rectangular section of the door. The width of the section in each of two doors is set at the span of the three wheels. Beam theory is used to calculate stresses.

3. The loading from the MPC on the door is simulated by a uniform pressure acting on the total surface area of the postulated beam section of the door.

3.AD.5 Methodology and Assumptions

Strength of Materials analysis are performed to establish structural integrity. Stresses in the transfer lid door are computed based on simplified beam analysis, where the width of the top plate beam is taken as the span of the door support wheels (see drawing 1928).

For all lifting analyses, the acceptance criteria is the more severe of ASME Section III, Subsection NF (allowable stresses per tables in Chapter 3),or USNRC Regulatory Guide 3.61 (33.3% of yield strength at temperature).

3.AD.6 Input Data (per BM-1928 and drawing 1928; weights are from Table 3.2.2, with detailed door component weights from the calculation package HI-981928)

Unsupported door top plate length	$L := 72.75 \cdot in$
Half Door top plate width	$w := 25 \cdot in$
Door top plate thickness	$t_{tp} := 2.25 \cdot in$
Thickness of middle plate	$t_{mp} := .5 \cdot in$
Thickness of bottom plate	$t_{bp} := 0.75 \cdot in$
HI-TRAC bounding dry weight	$W := 243000 \cdot lbf$
MPC bounding weight	$W_{mpc} := 90000 \cdot lbf$
Transfer Lid Bounding Weight (with door)	$W_{tl} := 24500 \cdot lbf$
Weight of door top plate (2 items)	$W_{tp} := 3762 \cdot lbf$
Door Lead shield weight (2 items)	$W_{lead} := 3839 \cdot lbf$

Weight of door bottom plate (2 items) $W_{bp} := 994 \cdot lbf$ |

Weight of Holtite A (2 items) $W_{ha} := 691 \cdot lbf$ |

Weight of door middle plate (2 items) $W_{mp} := 663 \cdot lbf$ |

Total door weight (2 components) excluding wheels and trucks

$W_{td} := W_{tp} + W_{lead} + W_{bp} + W_{ha} + W_{mp}$ $\qquad W_{td} = 9.949 \times 10^3 lbf$

Weight of wheels, trucks and miscellaneous pieces $W_{misc} := 2088 \cdot lbf$ |

Total Load transferred by 1 set of 3 wheels including wheels, trucks, and miscellaneous items

$$W_{door} := \frac{.5 \cdot (W_{td} + W_{misc})}{2} \qquad W_{door} = 3.009 \times 10^3 lbf$$

Dynamic Load Factor for low speed lift $DLF := 0.15$

Young's Modulus SA-516-Gr70 @ 350 deg. F $E := 28 \cdot 10^6 \cdot psi$

Allowable membrane stress for Level A condition @ 350 deg. F(Table 3.3.2) (Use allowable of SA-516-Gr 70 to be conservative) $S_a := 17500 \cdot psi$

Yield strength of SA-350-LF3 @ 350 deg. F to be conservative (Table 3.3.3) $S_y := 32700 \cdot psi$

Maximum Deceleration g level per design basis $G_{max} := 45$

3.AD.7 Analysis of Door plates Under Lift of MPC - Level A Event

The transfer lid door has a top and bottom plate connected by side plates that act as stiffeners in the loaded section. The top plate is 2.25" thick and the total span between wheel centers is 73". The bottom plate is 0.75" thick and spans 73". The side plates that connect the plates are 1" thick.

The lid door acts as a composite beam between wheel sets. To ensure conservatism, the effective width of the composite beam is taken as the distance between the outermost stiffeners. Beam theory is valid up to 1/8 of the span [Ref. 3.AD.2.1]. Beyond this value, a beam begins to act as a stronger two-way plate. Therefore, a one-way beam approximation for the dimensions of this lid underestimates the capacity of the lid. The load acting on the beam is taken as the bounding weight from a fully loaded MPC plus the bounding weight of the transfer lid door assembly. The load is applied as a uniform pressure and the beam is assumed simply supported.

The geometric parameters of the system are (drawing 1928, sheet 2):

$b := w$

$h := 8 \cdot in$	overall beam height	
$htp := t_{tp}$	thickness of top plate	$htp = 2.25\,in$
$hg := 5.75 \cdot in$	height of side plate	
$hbp := t_{bp}$	thickness of bottom plate	$hbp = 0.75\,in$
$tg := 1 \cdot in$	thickness of each side plate	

The centroid (measured from the top surface) and area moment of inertia of the composite beam are:

$$yc := \frac{3 \cdot hg \cdot tg \cdot \left(htp + \frac{hg}{2}\right) + htp \cdot b \cdot \frac{htp}{2} + hbp \cdot (b - 3 \cdot tg) \cdot \left(h - \frac{hbp}{2}\right)}{htp \cdot b + 3 \cdot hg \cdot tg + hbp \cdot (b - 3 \cdot tg)}$$

$yc = 3.083\,in$

$$Inertia := \frac{b \cdot htp^3}{12} + htp \cdot b \cdot \left(yc - \frac{htp}{2}\right)^2 + \frac{tg \cdot hg^3}{4} + 3 \cdot hg \cdot tg \cdot \left(yc - htp - \frac{hg}{2}\right)^2 \ldots$$
$$+ \frac{(b - 3 \cdot tg) \cdot hbp^3}{12} + hbp \cdot (b - 3 \cdot tg) \cdot \left(yc - htp - hg - \frac{hbp}{2}\right)^2$$

$Inertia = 821.688\,in^4$

The maximum stress is due to the moment:

$$Moment := \frac{(W_{mpc} + W_{td})}{2} \cdot \frac{L}{8} \qquad Moment = 4.545 \times 10^{5}\,lbf \cdot in$$

The bending stress is

$$\sigma := \frac{Moment \cdot (h - yc) \cdot (1 + DLF)}{Inertia} \qquad \sigma = 3.127 \times 10^{3}\,psi$$

The stress must be less than the 33.3% of the yield strength of the material. This acceptance criteria comes from Reg. Guide 3.61. The safety factor is,

$$Sy := S_y$$

$$SF_{3.61} := \frac{Sy}{3 \cdot \sigma} \qquad SF_{3.61} = 3.486$$

The safety factor as defined by ASME Section III, Subsection NF for Class 3 components is

$$SF_{nf} := \frac{1.5 \cdot S_a}{\sigma} \qquad SF_{nf} = 8.394$$

Now consider the plate section between stiffeners and check to see if plate stress is acceptable. The span of the plate between stiffeners is

$$span := 12.5 \cdot in$$

Calculate the pressure on each half of lid door due to MPC.

$$p := \frac{.5 \cdot W_{mpc} \cdot (1 + DLF)}{L \cdot w} \qquad p = 28.454\,psi$$

Calculate the pressure due to self weight

$$p_d := .5 \cdot (W_{tp}) \cdot \frac{1 + DLF}{L \cdot w} \qquad p_d = 1.189\,psi$$

Bending moment due to pressure

$$Moment := \frac{(p + p_d) \cdot L \cdot span^2}{8} \qquad Moment = 4.212 \times 10^{4}\,lbf \cdot in$$

Maximum bending stress

$$\sigma_{bending} := \frac{6 \cdot Moment}{L \cdot t_{tp}^2}$$

$\sigma_{bending} = 686.179\,psi$

(Small!!!)

Now perform a Weld Check

$$Load := (p + p_d) \cdot L \cdot w$$

$Load = 5.391 \times 10^4\,lbf$

The shear stress at the weld connection is (conservatively neglect stiffener welds)

$$\tau := \frac{Load}{2 \cdot w \cdot t_{tp}}$$

$\tau = 479.227\,psi$ Low!

It is concluded that the significant stresses arise only by the action of the member as a composite beam composed of plates and stiffeners. Local bending stresses in the plate are small and can be neglected

3.AD.8 Wheel Loads on Housing

$W_{door} = 3.009 \times 10^3\,lbf$ From weight calculation - 50% of 1 half-door

Load per wheel

$$Load_{wheel} := \frac{(W_{door} + .25 \cdot W_{mpc}) \cdot (1 + DLF)}{3}$$

$Load_{wheel} = 9.779 \times 10^3\,lbf$

Note that working capacities of wheels are 10000 lb per McMaster Carr Catalog [3.AD.2.4].

The wheel rides on an angle track (item 7 in dwg. 1928). The thickness of the angle is

$$t_a := 0.125 \cdot in$$

The wheel span (three wheels) is (see sheet 2, side view of Dwg. 1928)

$$s := 18.5 \cdot in$$

Therefore the direct stress in the leg of the angle is

$$\sigma_a := \frac{1}{2\cdot\cos(45\cdot deg)\cdot s\cdot t_a}\cdot 3\cdot Load_{wheel}$$

$$\sigma_a = 8.97\times 10^3 psi$$

Overstress in this track does not impede ready retrievability of the fuel. Nevertheless, for conservatism, the safety factor in accordance with Regulatory Guide 3.61 is evaluated for the material specified for the angle.

$$SF_{angle} := \frac{36000\cdot psi}{3\cdot\sigma_a} \qquad SF_{angle} = 1.338$$

3.AD.9 Housing Stress Analysis

The most limiting section that sets the minimum safety factor for the door housing under a lifting condition is the box structure adjacent to the track that serves as the direct load path to the bolts. In this section, a conservative estimate of the stress levels in this region is obtained and the safety factor established. The door load is transferred to the bottom plate by the wheels running on an angle track. The load is then transferred to two vertical stiffeners that form the side of the box. The top plate, forming the top of the box, serves as the structure that moves the load to the bolts.

The lid bottom plate of the housing (item 2 of Dwg. 1928) that directly supports the wheel loading can be conservatively considered as a wide plate supporting the load from one of the sliding doors. The applied load is transferred to the two vertical plates (items 3 and 4 of Dwg. 1928). Figure 3.AD.2 shows the configuration for analysis. The following dimensions are obtained from the drawing:

Length of analyzed section	$L_H := 25\cdot in$	
Thickness of item 2	$t_{bottom} := 2\cdot in$	From BM-1928
Thickness of item 3	$t_1 := 1.5\cdot in$	
Thickness of item 4	$t_2 := 1\cdot in$	
Width of item 21	$t_{21} := 3.5\cdot in$	

With respect to Figure 3.AD.2, referring to the drawing, the length x is defined as a+b

$x := (.5 \cdot 93) \cdot in - 36.375 \cdot in$ $\qquad x = 10.125 in$

dimension "b" $\qquad b := x - t_1 - t_{21} - .5 \cdot t_1$ $\qquad b = 4.375 in$

dimension "a" $\qquad a := x - b$ $\qquad a = 5.75 in$

Compute the moment of inertia of item 2 at the root assuming a wide beam

$$I := L_H \cdot \frac{t_{bottom}^3}{12} \qquad I = 16.667 in^4$$

The maximum bending moment in the bottom plate is given as,

$$Moment := 3 \cdot Load_{wheel} \cdot b \qquad Moment = 1.283 \times 10^5 lbf \cdot in$$

The maximum bending stress is

$$\sigma_{bending} := \frac{Moment \cdot t_{bottom}}{2 \cdot I} \qquad \sigma_{bending} = 7.701 \times 10^3 psi$$

The safety factor, based on primary bending stress (ASME Code evaluation), is

$$1.5 \cdot \frac{S_a}{\sigma_{bending}} = 3.409$$

It is concluded that this region is not limiting.

The safety factor based on Reg. Guide 3.61 (compare to 33% of yield strength) is

$$\frac{S_y}{3 \cdot \sigma_{bending}} = 1.415$$

The reactions at the two support points for the section are

$$F_1 := 3 \cdot Load_{wheel} \cdot \left(1 + \frac{b}{a}\right) \qquad F_1 = 5.166 \times 10^4 lbf$$

$$F_2 := 3 \cdot Load_{wheel} \cdot \frac{b}{a} \qquad F_2 = 2.232 \times 10^4 lbf$$

Therefore, consistent with the support assumptions, the direct stress in the two stiffeners is

$$\sigma_1 := \frac{F_1}{L_H \cdot t_1} \qquad \sigma_1 = 1.377 \times 10^3 \text{psi}$$

$$\sigma_2 := \frac{F_2}{L_H \cdot t_2} \qquad \sigma_2 = 892.822 \text{psi}$$

Safety factors, using the more conservative Reg. Guide 3.61 criteria, are

$$SF_1 := \frac{S_y}{3 \cdot \sigma_1} \qquad SF_1 = 7.913$$

$$SF_2 := \frac{S_y}{3 \cdot \sigma_2} \qquad SF_2 = 12.208$$

3.AD.10 Bolt Stress

Figure 3.AD.3 shows the bolt array assumed to resist the lifted load when the doors are closed and when the fully loaded MPC is being supported by the doors.

The bolt tensile stress area is, for the 1" diameter bolts

$$A_b := 0.605 \cdot \text{in}^2 \qquad d_{bolt} := 1 \cdot \text{in}$$

The bolt circle radius is

$$R_b := 45 \cdot \text{in}$$

The bolt angular spacing is $\theta := 10 \cdot \text{deg}$

The centroid of the nine bolts point P* in Figure 3.AD.3, assumed to carry 100% of the wheel load, is computed as follows:

$$A_{total} := 9 \cdot A_b \qquad A_{total} = 5.445 \text{in}^2$$

Compute the following sum:

$$Sum := 2\cdot A_b\cdot R_b\cdot(1-\cos(4\cdot\theta)) + 2\cdot A_b\cdot R_b\cdot(1-\cos(3\cdot\theta)) \ldots$$
$$+ 2\cdot A_b\cdot R_b\cdot(1-\cos(2\cdot\theta)) + 2\cdot A_b\cdot R_b\cdot(1-\cos(\theta))$$

$$Sum = 24.145\,in^3$$

Then the centroid of the bolts is $X_{bar} := \frac{Sum}{A_{total}}$ $X_{bar} = 4.434\,in$

Compute the bolt moment of inertia about the centroid by first locating each bolt relative to the centroid. First compute some distances "z":

$z_1 := R_b\cdot(1-\cos(4\cdot\theta)) - X_{bar}$ $z_1 = 6.094\,in$

$z_2 := R_b\cdot(1-\cos(3\cdot\theta)) - X_{bar}$ $z_2 = 1.595\,in$

$z_3 := R_b\cdot(1-\cos(2\cdot\theta)) - X_{bar}$ $z_3 = -1.72\,in$

$z_4 := R_b\cdot(1-\cos(\theta)) - X_{bar}$ $z_4 = -3.751\,in$

Then the bolt group moment of inertia about the centroid is,

$$I_{bolts} := 2\cdot A_b\cdot z_1^2 + 2\cdot A_b\cdot z_2^2 + 2\cdot A_b\cdot z_3^2 + 2\cdot A_b\cdot z_4^2 + A_b\cdot X_{bar}^2$$

$$I_{bolts} = 80.507\,in^4$$

The bolts must support the total wheel load acting on one rail, plus the additional load necessary to resist the moment induced about the bolt group centroid.

The moment arm is the distance from the bolt centroid to the angle guide rail

$moment\_arm := R_b - X_{bar} - 36.375\cdot in$ $moment\_arm = 4.191\,in$

Therefore, the bolt array must resist the following moment

$Moment_{bolts} := 6\cdot Load_{wheel}\cdot moment\_arm$ $Moment_{bolts} = 2.459\times10^5\,in\cdot lbf$

The bolt stress due to the direct load is:

$$stress_{direct} := 6 \cdot \frac{Load_{wheel}}{A_{total}} \qquad stress_{direct} = 1.078 \times 10^{4}\,psi$$

Compute $\quad y_1 := R_b \cdot (1 - \cos(4 \cdot \theta)) - X_{bar} \qquad y_1 = 6.094\,in \qquad > Xbar$

Therefore, the highest bolt stress due to the bending moment is,

$$stress_{moment} := \frac{Moment_{bolts} \cdot y_1}{I_{bolts}} \qquad stress_{moment} = 1.861 \times 10^{4}\,psi$$

Therefore, the total bolt stress to support lifting, on the heaviest loaded bolt, is

$$\sigma_{bolt} := stress_{direct} + stress_{moment} \qquad \sigma_{bolt} = 2.939 \times 10^{4}\,psi$$

The above calculation has considered only the stress induced by the MPC and the door; that is, the stress induced in the bolts by the load transmitted through the wheels. The entire set of bolts acts to support the door housing and this induces an additional component of stress in the bolts. This is computed below:

The total bounding weight of the transfer lid is

$$W_{tl} = 2.45 \times 10^{4}\,lbf$$

The total door load already accounted for in the bolt analysis is

$$W_{td} := 4 \cdot W_{door} \qquad W_{td} = 1.204 \times 10^{4}\,lbf$$

Therefore the additional average stress component in the 36 bolts is

$$\sigma_{avg} := \frac{(W_{tl} - W_{td})}{36 \cdot A_b} \qquad \sigma_{avg} = 572.221\,psi$$

Therefore the absolute maximum bolt stress is

$$\sigma_{bolt\_max} := \sigma_{bolt} + \sigma_{avg} \qquad \sigma_{bolt\_max} = 2.996 \times 10^{4}\,psi$$

The allowable bolt load is obtained from the ASME Code, Subsection NF, NF-3324.6 as 50% of the ultimate strength of the bolts. The bolts are assumed to be at a temperature below 200 degrees F because of their location. The mechanical properties of SA193 Grade B7 for 2.5" to 4" diameters are conservatively used for the HI-TRAC lid bolts.

$S_{ubolt} := 115000 \cdot psi$ @200 deg. F Table 3.3.4

$S_{ybolt} := 95000 \cdot psi$

Therefore, the bolt safety factor is

$$SF_{bolts} := \frac{.5 \cdot S_{ubolt}}{\sigma_{bolt\_max}} \qquad SF_{bolts} = 1.919$$

The transfer lid bolt preload required is

$$T := .12 \cdot \sigma_{bolt\_max} \cdot A_b \cdot d_{bolt} \qquad [3.AD.3] \qquad T = 181.246 ft \cdot lbf$$

Note that this exceeds the value calculated for the pool lid.

The safety factor using the Reg. Guide 3.61 criteria is

$$SF_{3.61} := \frac{S_{ybolt}}{3 \cdot \sigma_{bolt\_max}} \qquad SF_{3.61} = 1.057$$

Calculation of Thread Capacity

The following calculations are taken from Machinery's Handbook, 23rd Edition, pp. 1278-1279 plus associated screw thread Table 4, p 1514.

Input Geometry Data - 1" UNC, 8 threads/inch, 2A class

$L_e := 1.0 \cdot in$ Thread engagement length

$N := \frac{8}{in}$ Threads per inch

$D_m := 1 \cdot in$ Basic Major Diameter of threads

$D := .9755 \cdot in$ Minimum Major Diameter of External Threads

$E_{min} := .91 \cdot in$ Minimum Pitch Diameter of External Threads

$E_{max} := .9276in$ Maximum Pitch Diameter of Internal Threads

$K_n := .89 \cdot in$ Maximum Minor Diameter of Internal Threads

Input Yield Strength-Internal Threads (lid or forging); External Threads (bolts)

Values are obtained from ASME Code,Section II

$S_{ylid} := 38000 \cdot psi$ $Su_{lid} := 70000 \cdot psi$ $Su_{bolt} := S_{ubolt}$

Calculation of Tensile stress area (high-strength bolt, ultimate strength exceeding 100,000 psi)

$$A_{th} := \pi \cdot \left(.5 \cdot E_{min} - \frac{0.16238}{N} \right)^2 \qquad A_{tl} := .7854 \cdot \left(D_m - \frac{.9743}{N} \right)^2$$

$A_{th} = 0.594\,in^2$ $A_{tl} = 0.606\,in^2$

$A_t := if(Su_{bolt} > 100000 \cdot psi, A_{th}, A_{tl})$ $A_t = 0.594\,in^2$

Calculation of Shear Stress Area per the Handbook

$$A_{ext} := \pi \cdot N \cdot L_e \cdot K_n \cdot \left[\frac{0.5}{N} + 0.57735 \cdot (E_{min} - K_n) \right]$$ $A_{ext} = 1.656\,in^2$

$$A_{int} := \pi \cdot N \cdot L_e \cdot D \cdot \left[\frac{0.5}{N} + 0.57735 \cdot (D - E_{max}) \right]$$ $A_{int} = 2.21\,in^2$

Required Length of Engagement per Machinery's Handbook

$$L_{req} := 2 \cdot \frac{A_t}{\frac{A_{ext}}{L_e}}$$ $L_{req} = 0.717\,in$

Capacity Calculation Using Actual Engagement Length

For the specified condition, the allowable tensile stress in the bolt is per ASME NF

$$\sigma_{bolt} := Su_{bolt} \cdot 0.5 \qquad \sigma_{bolt} = 5.75 \times 10^{4} psi$$

The allowable shear stress in the bolt is:

$$\tau_{bolt} := \frac{.62 \cdot Su_{bolt}}{3} \qquad \tau_{bolt} = 2.377 \times 10^{4} psi$$

The allowable shear stress in the lid (or flange) is

$$\tau_{lid} := 0.4 \cdot Sy_{lid} \qquad \tau_{lid} = 1.52 \times 10^{4} psi$$

$$F_{shear\_lid} := \tau_{lid} \cdot A_{int} \qquad F_{shear\_lid} = 3.36 \times 10^{4} lbf$$

For the bolt, the allowable strength is the yield strength

$$F_{tensile\_bolt} := \sigma_{bolt} \cdot A_{t} \qquad F_{tensile\_bolt} = 3.414 \times 10^{4} lbf$$

$$F_{shear\_bolt} := \tau_{bolt} \cdot A_{ext} \qquad F_{shear\_bolt} = 3.936 \times 10^{4} lbf$$

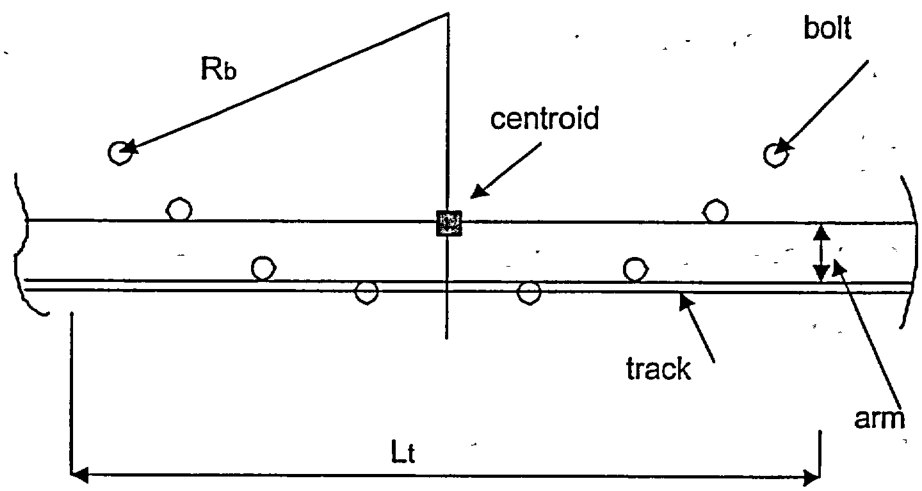
Therefore, thread shear in lid governs the design. The safety factors computed above should by multiplied by the ratio

$$\frac{F_{shear\_lid}}{F_{tensile\_bolt}} = 0.984$$

3.AD.11 Estimate of Primary Bending Stress in Lid Top Plate

The lid top plate maximum primary stresses develop due to the structural requirement of transferring the wheel loads to the bolt array. Based on the assumptions above as to the number of bolts participating in the support of the load, a total direct load and a bending moment is reacted by the bolt array. The active bolts have been assumed to be only those bolts in an 80 degree arc (see Figure 3.AD.3). To estimate the minimum safety factor inherent in the top plate, it is assumed that the same bending moment must also be reacted by the the lid top plate. The sketch below aids in the analysis:

The analysis is conservative as it neglects any support from either plate or bolts outside of the section identified.



The view shown is similar to the view in Figure 3.AD.3 with identification of terms for use in the following analysis;

$arm := moment\_arm$ $\qquad$ $arm = 4.191\,in$

$Moment := Moment_{bolts}$ $\qquad$ $Moment = 2.459 \times 10^{5}\,in \cdot lbf$

$L_t := R_b \cdot 2 \cdot \sin(45 \cdot deg)$ $\qquad$ $L_t = 63.64\,in$

The thickness of the lid top plate is

$t_p := 1.5 \cdot in$ $\qquad$ item 1 in BM-1928

The safety factor is established by considering the bending moment in the section of top plate a distance "arm" away from the track.

$I_p := \dfrac{L_t \cdot t_p^{\,3}}{12}$ $\qquad$ $I_p = 17.899\,in^4$

The primary bending stress is

$$\sigma_{tp} := \frac{Moment \cdot t_p}{2 \cdot I_p} \qquad \sigma_{tp} = 1.03 \times 10^4 psi$$

The limiting safety factor is obtained by consideration of the Regulatory Guide 3.61 criteria. Therefore,

$$SF_{tp} := \frac{S_y}{3 \cdot \sigma_{tp}} \qquad SF_{tp} = 1.058$$

Similarly, the average shear stress developed across the section is

$$\tau_{tp} := 6 \cdot \frac{Load_{wheel}}{t_p \cdot L_t} \qquad \tau_{tp} = 614.619 psi$$

The safety factor against primary shear overstress is large.

$$SF_{shear} := .6 \cdot \frac{S_y}{3 \cdot \tau_{tp}} \qquad SF_{shear} = 10.641$$

In the above safety factor calculation, the yield strength in shear is assumed as 60% of the yield strength in tension for the Reg. Guide 3.61 evaluation.

The validity of the approximate strength of materials calculation has been independently verified by a finite element analysis (see calculation package HI-981928).

3.AD.12 Separation of Transfer Lid from HI-TRAC

In the event of a side drop while HI-TRAC is in a horizontal position, the transfer lid housing will impact the ground, and the HI-TRAC body, including the MPC, will attempt to separate from the lid. Appendix 3.AN provides a detailed dynamic analysis of the handling accident and provides the interface load that must be transferred by the bolts.

From Appendix 3.AN, Section 3.AN.2.7, we find the following results for the 125-ton HI-TRAC:

$$Interface\_Force := 1272000 \cdot lbf$$

We now demonstrate that this load can be transferred by a combination of bolt shear and interface friction.

3.AD.12.1 Shear Capacity of 36 SA 193 B7 bolts

Number of bolts $nb := 36$

$S_{ubolt} = 1.15\times 10^{5}\text{psi}$ $A_b := A_t$

$Bolt\_Capacity := nb\cdot .6\cdot S_{ubolt}\cdot A_b$ $Bolt\_Capacity = 1.475\times 10^{6}\text{lbf}$

Note that here we are performing a failure analysis

3.AD.12.2 Shear Capacity due to Friction - 125 Ton HI-TRAC

Table 8.1.5 lists the actual preload torque as $T_{act} := 270\cdot ft\cdot lbf$

The calculated bolt torque requirement is $T = 181.246\,ft\cdot lbf$

Therefore the actual clamping force per bolt is:

$$T_{clamp} := \frac{T_{act}}{T}\cdot\sigma_{bolt\_max}\cdot A_b \qquad T_{clamp} = 2.649\times 10^{4}\text{lbf}$$

Following ASME, Section III, Subsection NF, NF-3324.6(4) for a blast cleaned joint, the frictional resistance for the assemblage of bolts is:

$$P_s := nb\cdot T_{clamp}\cdot 0.31 \qquad P_s = 2.957\times 10^{5}\text{lbf}$$

Note that since we are evaluating a side drop, the actual value of the clamping force may be used since there is no other tensile load acting on the bolts.

Therefore, the total shear capacity, based on ultimate strength in shear, is

$$Shear\_Capacity := Bolt\_Capacity + P_s$$

$$Shear\_Capacity = 1.77\times 10^{6}\text{lbf}$$

The safety factor for lid separation is defined as

$$SF := \frac{Shear\_Capacity}{Interface\_Force} \qquad SF = 1.392$$

It is concluded that there will be no separation of the HI-TRAC 125 from the transfer lid.

3.AD.13 Analysis of Door Lock Bolts (Item 22 of Dwg. 1928, Sheet 1)

Under the design basis side drop handling accident, the transfer lid doors (both) are restrained only by the two door lock bolts. Since the doors must remain closed to maintain shielding, these bolts need to have sufficient shear capacity to resist the door deceleration loading. The following calculation demonstrates that the door lock bolts have the desired shear capacity. The following input data is required to obtain a result:

$G_{max} = 45$

$D_{bolt} := 3.0 \cdot in$ Door lock bolt diameter per 125 ton transfer cask bill of materials.

$S_{abolt} := .42 \cdot S_{ubolt}$ Level D event per Appendix F of ASME Code

$Total\_Load := 4 \cdot W_{door}$ $Total\_Load = 1.204 \times 10^4\, lbf$

Recall that W_{door} has been defined in 3.AD.8 as 50% of the weight of one(of two) doors. The door bolt area is

$D_{bolt} = 3\, in$ $n := 4$ Threads/inch

The stress area is computed from the following formula (Machinery's Handbook, Industrial Press, NYC, 23rd Edition, p. 1279,)

$$A_{bolt} := \pi \cdot \left(\frac{D_{bolt}}{2} - \frac{0.16238}{n} \cdot in \right)^2 \qquad A_{bolt} = 6.691\, in^2$$

There are two bolts which support load and there are two shear faces per bolt (see section B-B on Dwg. 1928). The shear stress in the bolt section is

$$\tau_{bolt} := Total\_Load \cdot \frac{G_{max}}{2 \cdot 2 \cdot A_{bolt}} \qquad \tau_{bolt} = 2.024 \times 10^4 psi$$

Therefore, the safety factor on bolt shear stress is

$$SF_{bolt\_shear} := \frac{S_{abolt}}{\tau_{bolt}} \qquad SF_{bolt\_shear} = 2.387$$

and no loss of shielding will occur since the doors will be retained in place.

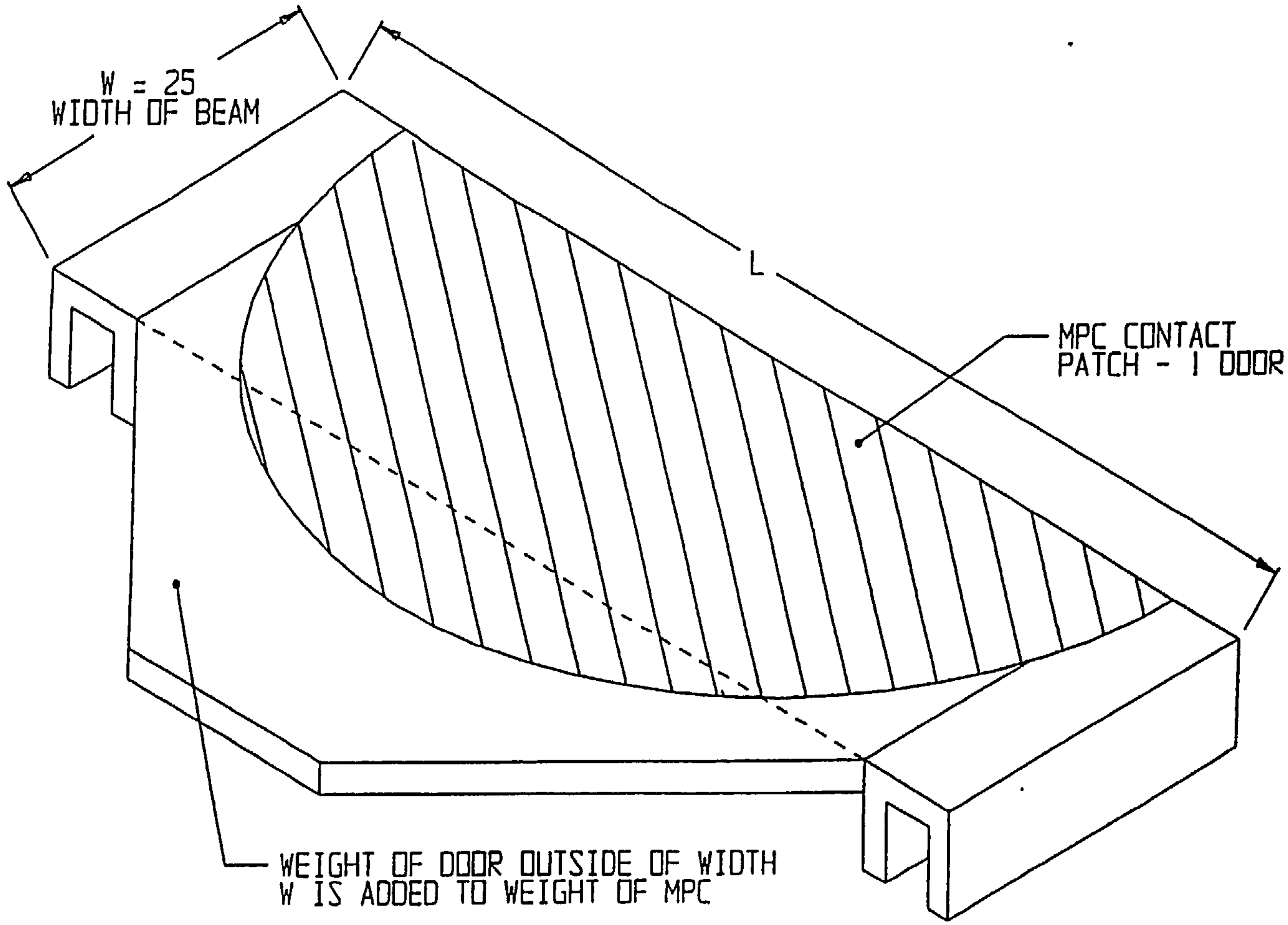


FIGURE 3.AD.1; DOOR PLATE SIMPLY SUPPORTED BEAM MODEL

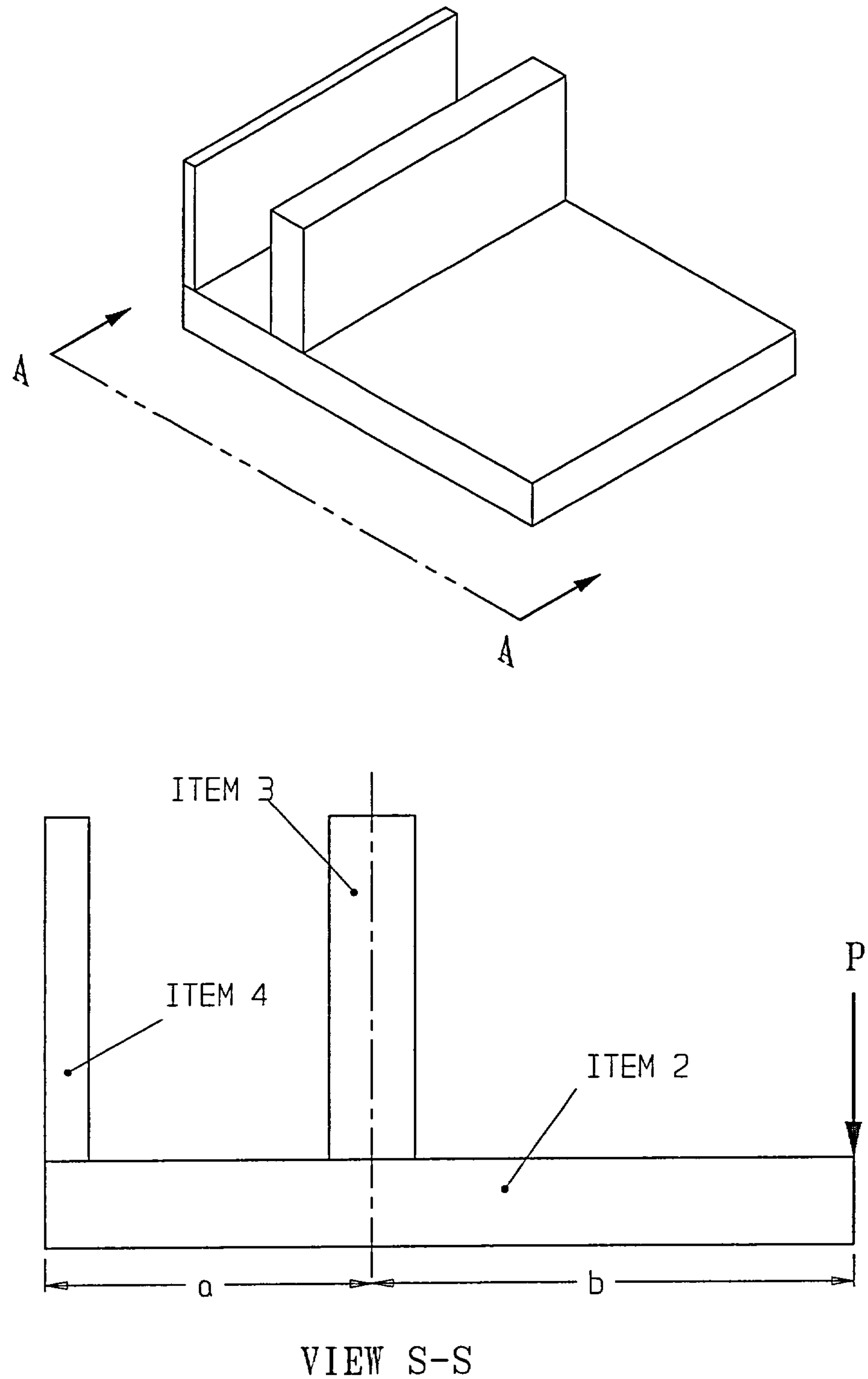


FIGURE 3.AD.2; SECTION OF BOTTOM PLATE FOR STRESS ANALYSIS

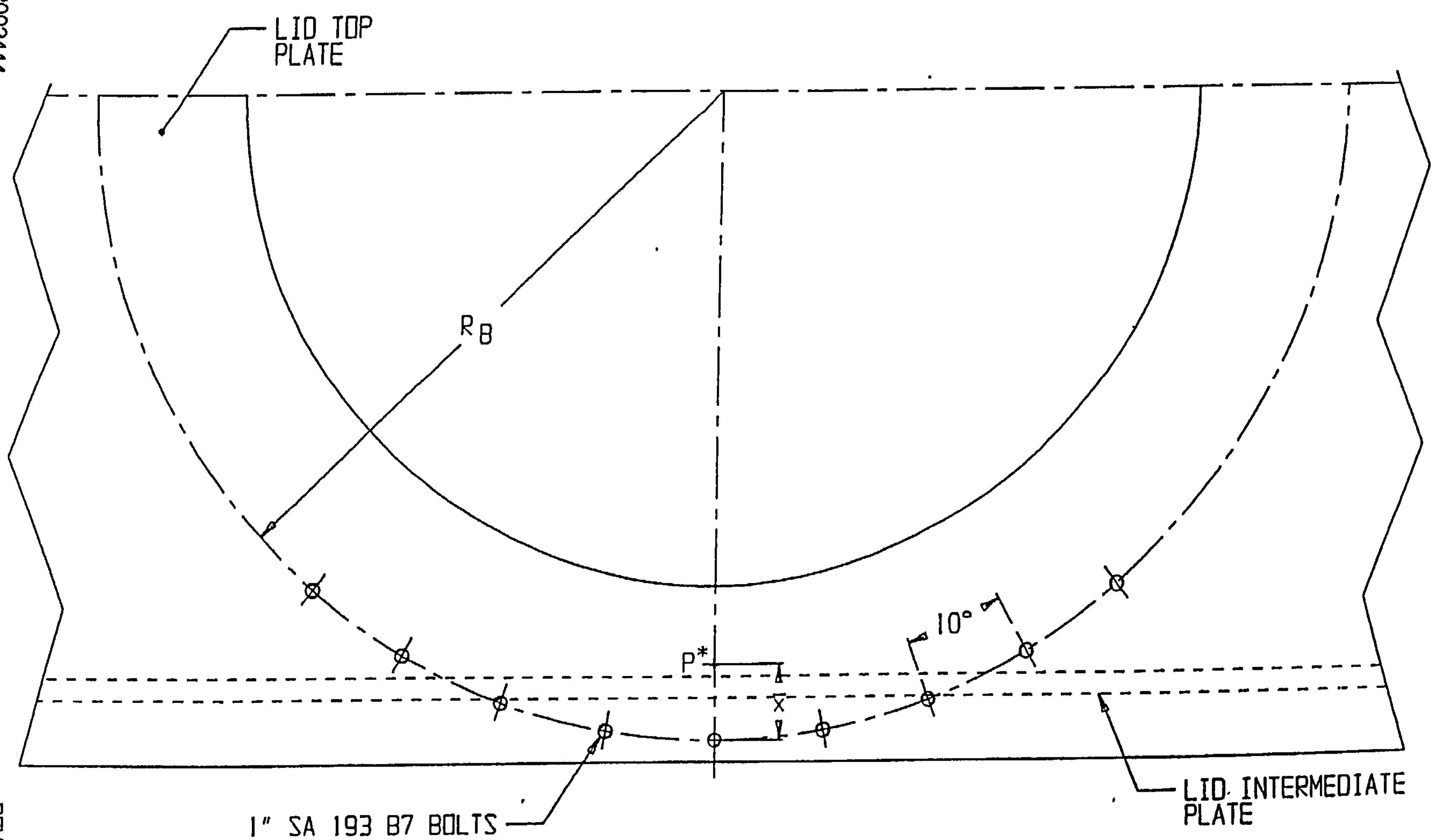


FIGURE 3.AD.3; HOUSING BOLT ARRAY TO SUPPORT LIFT OPERATION

APPENDIX 3.AE: GLOBAL ANALYSIS OF HI-TRAC LIFT

3.AE.1 Introduction

The global analysis of the 125 ton HI-TRAC lift is performed in this Appendix to show that the general primary stresses in the top flange, the inner shell, and the outer shell in the vicinity of the trunnion attachment do not exceed 17,500 psi and 1.5 x 17,500 psi = 26,250 psi for membrane and membrane plus bending stress, respectively, in accordance with requirements of the ASME Code, Section III, Subsection NF, for Level A conditions. In addition, we show in this appendix that the primary membrane stress, conservatively averaged over the width of the interface between the base of the trunnion block and the outer shell, does not exceed one-third of the material yield stress at temperature; this is in keeping with the requirements of USNRC Regulatory Guide 3.61. The trunnion and the trunnion block are modeled only to the extent necessary to insure that the proper moment arm is present. The analysis of the threaded lifting trunnions and the trunnion weldments at the top end of the 125 ton HI-TRAC are documented in Appendix 3.E.

A separate analysis is also performed in this appendix to evaluate the stress state in the lower part of the HI-TRAC flange when the bounding lid is in place. Specifically, it is shown that the bottom flange of HI-TRAC and the inner and outer shells meet the allowable stress limits of ASME Section III, Subsection NF, for Class 3 plate and shell structures. It is also demonstrated that the allowable stress limits imposed by Regulatory Guide 3.61 for a lifting operation are met. The imposed loading on the flange is the limiting bolt loading obtained from the analyses of the HI-TRAC pool lid (Appendix 3.AB) and the HI-TRAC transfer lid analyses (Appendices 3.AD and 3.AJ).

3.AE.2 Assumptions for Analysis of Upper Portion of HI-TRAC-125

The analysis in this appendix is based on the following conservative assumptions:

1. The analysis does not take any structural credit for the lead shielding annulus between the inner shell and the outer shell that is in close proximity to the trunnion.

2. The analysis does not take any structural credit for the steel water jacket at the outer surface of the outer shell.

3. The cask component temperature during lifting operation is taken as 200 degrees F. This is based on an evaluation of actual MPC temperatures at the top of the cask.

4. The weight of the loaded HI-TRAC is the bounding weight amplified by 15% to account for dynamic effects.

5. The load on the upper trunnion is positioned at the midpoint of a 2.5" wide contact interface located at the outermost position of the trunnion barrel. This is conservative, since during a heavy lift, the load will shift toward the inner edge of the initial contact interface area.

3.AE.3 Finite Element Model

A 3-D, 1/4-symmetry model of the HI-TRAC structure near the lifting trunnion is constructed using the ANSYS [1] 3-D isoparametric element SOLID45 as shown in Figures 3.AE.1 and 3.AE.2. The finite element plots are coded according to the particular properties of each HI-TRAC component modeled, i.e., shades of blue for the shells and the top flange, red for the threaded trunnion, and purple for the trunnion block. The Young's moduli for the three structural components are assigned values commensurate with the assumed operating temperature. The base of the finite element model is restrained from vertical movement while a concentrated vertical force equal to 1/4 of the assumed loaded weight is applied to a node point located on the trunnion axis at the appropriate position near the end of the trunnion elements. Note that the trunnion stress analysis is performed in Appendix 3.E, consistent with NRC accepted methodology, so a detailed local stress analysis is not required here.

3.AE.4 Stress Evaluation From Finite Element Analysis

The applied load is 0.25 x 250,000 lb. x 1.15 = 71,875 lb. The load is positioned at a node point on the trunnion centerline that is a radial distance of 45.0 inch – 1.25 inch = 43.75 inch from the longitudinal (vertical) centerline of the HI-TRAC. The subtraction of 1.25 inch reflects the geometry of the lift yoke arm that attaches to the trunnion during a lifting operation. The full width and longitudinal dimension of the trunnion block in the model are 10 inches. A static stress analysis is performed and the stress distributions evaluated. For this loading scenario, the largest stress (a normal stress parallel to the cask longitudinal axis) occurs at the interface between the base of the trunnion block and the interface with the edge of the outer shell (1 inch thick). The interface contact stress results show the local character of this stress with a significant variation occurring both through the thickness of the outer shell and along the circumferential length of the interface (10 inch). In the following, the evaluation of the safety factors existing in the structure is consistent with ASME, Section III, Subsection NF for a Class 3 plate and shell structure. As this analysis involves a non-axisymetric geometry and loading, a comparison of the primary stress state with NF allowable values can only be performed once a characteristic width of section is defined (In a pressure vessel, such a definition is not required as the stresses are independent of peripheral position). To this end, we note that ASME Code Section III, NB-3213.10 provides guidance on the extent of the region over which local stresses are categorized. Specifically, a characteristic length L in the circumferential direction no smaller than

$$L = 2\sqrt{Rt}$$

need be considered in the calculation of primary stresses for comparison with NF allowable stress levels. In the above equation, R is the radius of curvature of the mid-surface of the outer shell and t is the outer shell thickness. For the HI-TRAC 125,

R = (.5 x 81.25"-0.5") = 40.125" and t=1.0".

Therefore the characteristic circumferential length, over which the stress state is averaged, prior to comparing with Code allowable stress values, is:

L = 12.67"

Noting that this characteristic circumferential length exceeds the actual interface circumferential length, we conservatively evaluate the stress state by averaging over the entire 10" interface width along the base of the trunnion block and the outer shell. By virtue of the rapid decay in the stress magnitude as we move away from the centerline of the trunnion, the use of a lower characteristic length leads to a conservatively larger stress value.

We seek safety factors on primary membrane stress and surface membrane stress plus bending stress associated with the above section as defined by the ASME Code. The interface nodes are identified and the normal stresses in the global "Z" direction identified and averaged to obtain the longitudinal primary membrane stress for the outer shell section. Since moment equilibrium is primarily provided by the force associated with this stress component (and an opposing force on the inner shell), the stress variation through the thickness of the individual shells at the interface is most properly characterized as a secondary in the Code nomenclature. Nevertheless, in the evaluation of safety factors associated with satisfaction of ASME Code NF stress levels for a single shell acting as a pressure vessel, we conservatively include this local through thickness variation in the safety factor calculation. To this end, the subset of nodes associated with the outer surface of the outer shell at the interface is separately identified and the normal stresses in the global "Z" direction identified and averaged to obtain the membrane plus bending stress for the section. The following results are obtained:

Membrane stress (averaged over the characteristic circumferential width) = 6,185.9 psi
Surface stress (averaged over the characteristic circumferential width) = 8,191.9 psi

An evaluation of safety factors at this location provides the lower bound to safety factors at all other sections of the region modeled.

Comparison with Level A allowables (Table 3.1.10) is provided below:

SF(primary membrane) = 17,500 psi /6,185.9 psi = 2.83

SF(primary membrane plus primary bending) = 26,250 psi /8,191.9 psi = 3.2

Consistent with the definition of safety factors in other sections of this FSAR, the safety factor is defined as the allowable value divided by the calculated value.

Consistent with the intent of Regulatory Guide 3.61, we compare the primary membrane stress in the outer shell with 1/3 of the material yield strength of the shell material. Yield strength data for SA-516 at 200 degrees F is used in the calculation.

SF(Reg. Guide 3.61) = 34,600 psi/(3 x 6185.9) = 1.86

We conclude that the construction satisfies the intent of Regulatory Guide 3.61, Section 3.4.3.

3.AE.5 Analysis of HI-TRAC 125 Bottom Flange

Appendix 3.AD contains an analysis of the transfer lid for HI-TRAC 125 to demonstrate structural integrity during the postulated lifting operation. The bounding lifted load at that location is the bounding weight of the loaded MPC together with the bounding weight of the transfer lid. The results from Appendix 3.AD bound the results for the smaller HI-TRAC in Appendix 3.AJ. The transfer lid establishes the bolt preload for the transfer cask application. Appendix 3.AB examines the pool lid under the same conditions. Since there is water in the cask during this operation, the lifted weight, including the water, exceeds the lifted weight when the transfer lid is in place. Therefore, a bounding flange analysis is undertaken that uses the results from the pool lid evaluation as the input to the calculation.

From Appendix 3.AB, for the 125 Ton HI-TRAC, the total bolt load, including a 15% amplification factor, is

T = 137,400 lb. (Appendix. 3.AB, subsection 3.AB.8)

Conservatively assuming that all of the bolt preload is removed when the lift commences, the flange is modeled as an annular plate subjected to a total peripheral load applied at the outer diameter (bolt circle diameter = 90") and clamped by the HI-TRAC inner and outer shells at a diameter equal to 72". Figure 3.AE.3 shows a free body of a section of the flange with the total load from the bolts "T" and the reaction loads "T1" and "T2" in the inner and outer shells, respectively. This annular plate solution giving maximum bending stress, due to the load "T", is available in the classical plate literature [2]. It is conservatively assumed that the outer periphery of the flange is free to rotate. In the actual loaded configuration, there is some restraint to flange rotation provided by the flange of the fastened lid. Specifically, Case 8 in Figure 38 of [2] (Table 3) gives the maximum bending stress in the form

Stress = k x T/h^2

where k is a constant depending on the flange inner and outer diameters, T is the total bolt load, and "h" is the flange thickness. For the diameter ratio 90"/72" = 1.25, the constant k = 0.227. The bottom flange thickness "h" is

h = 2" (Bill-of-Materials -1880)

Therefore, under the amplified lifted load, the maximum bending stress in the flange is

Stress = 0.227 x 137,400 lb./4 sq.inch = 7797.5 psi

The allowable stress permitted by the governing ASME "NF" subsection is 1.5 x 17,500 psi = 26,250 psi. Therefore, the safety factor, considering this flange as an NF plate component, is

Safety Factor = 26,250psi/7798psi = 3.37

Alternately, applying the Regulatory Guide 3.61 criteria (a comparison with 33.3% of tensile yield strength at temperature) to establish the safety factor gives

Safety Factor = 33,150 psi/(3 x 7797.5psi) = 1.42

This result for the HI-TRAC 125 bounds the similar result that would be obtained for the HI-TRAC 100 since the lifted load is lower and the bolt circle diameter is smaller.

The peripheral loading from the bolts is resisted by direct loads in the inner and outer shells to maintain equilibrium. These loads develop primary membrane stress in the respective shells. The connecting welds are partial penetration groove welds so consideration need only be given to the stress through the welds. Conservatively neglecting any reduction in the moment due to circumferential bending stresses induced in the narrow plate, the loads and stresses in the shells can be determined using the free body sketch shown in Figure 3.AE.3.

Let r_1, r_2, and r_o be the loaded radius of the inner shell, outer shell, and bolt circle, respectively.

From BM-1880 and the associated drawings, the values are:

$r_1 = .5 \times (68.75" + 0.625") = 34.6875"$
$r_2 = .5 \times (81.25" - 0.625") = 40.3125"$
$r_o = 45"$

Then if T1 and T2 are the total tensile forces in the inner and outer shells, respectively, force and moment equilibrium equations applied to a unit peripheral section of the annular ring, yield

$$T1 + T2 = T$$

$$T(r_o - r_1) - T2(r_2 - r_1) = 0$$

The solution for T1 and T2 are

$$T1 = -(r_o - r_2)T/(r_2 - r_1) \qquad T2 = (r_o - r_1)T/(r_2 - r_1)$$

or

T1 = -114,500 lb.

T2 = +251,900 lb.

Therefore, the average shear stresses in the shell partial penetration welds are

Inner shell Stress = T1/(3.14159 x 69.375" x 0.625") = -841 psi

Outer shell Stress = T2/(3.14159 x 80.625" x 0.625") = +1591 psi

Large factors of safety exist in the shells under this lifting condition. Under the more limiting Regulatory Guide 3.61 limit, the safety factor is:

Safety Factor = 33,150psi/(3 x 1,591 psi) = 6.94

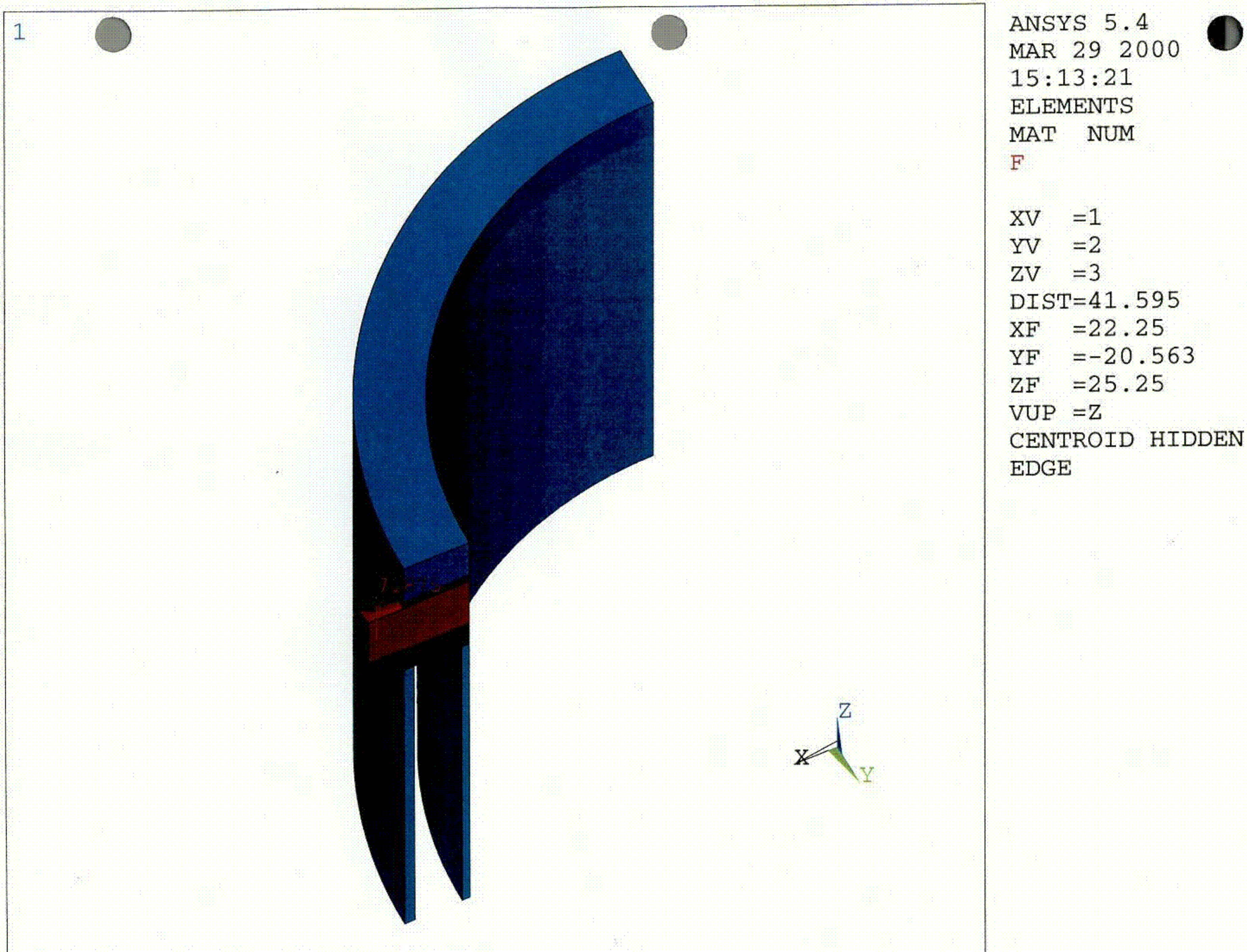
3.AE.6 Conclusion

The analysis in this appendix shows that the design of the 125 ton HI-TRAC is adequate for lifting and meets the requirements imposed by ASME Section III, Subsection NF for Class 3 plate and shell structures. Further, the intent of Regulatory Guide 3.61 to limit stresses under a lift to 33.3% of tensile yield strength is also satisfied. The safety factors calculated in this appendix provide a lower bound to the safety factors existing at all sections of the HI-TRAC subject to primary stresses during the lifting operation.

3.AE.7 Reference

1. ANSYS, General Purpose Finite Element Code, Revision 5.3, ANSYS Inc.

2. Timoshenko and Woinowsky-Kreiger, Theory of Plates and Shells, 2nd Edition, McGraw-Hill, 1959, Chapter 3, Table 3.



Global Analysis of HI-TRAC Lift

FIGURE 3.AE.1

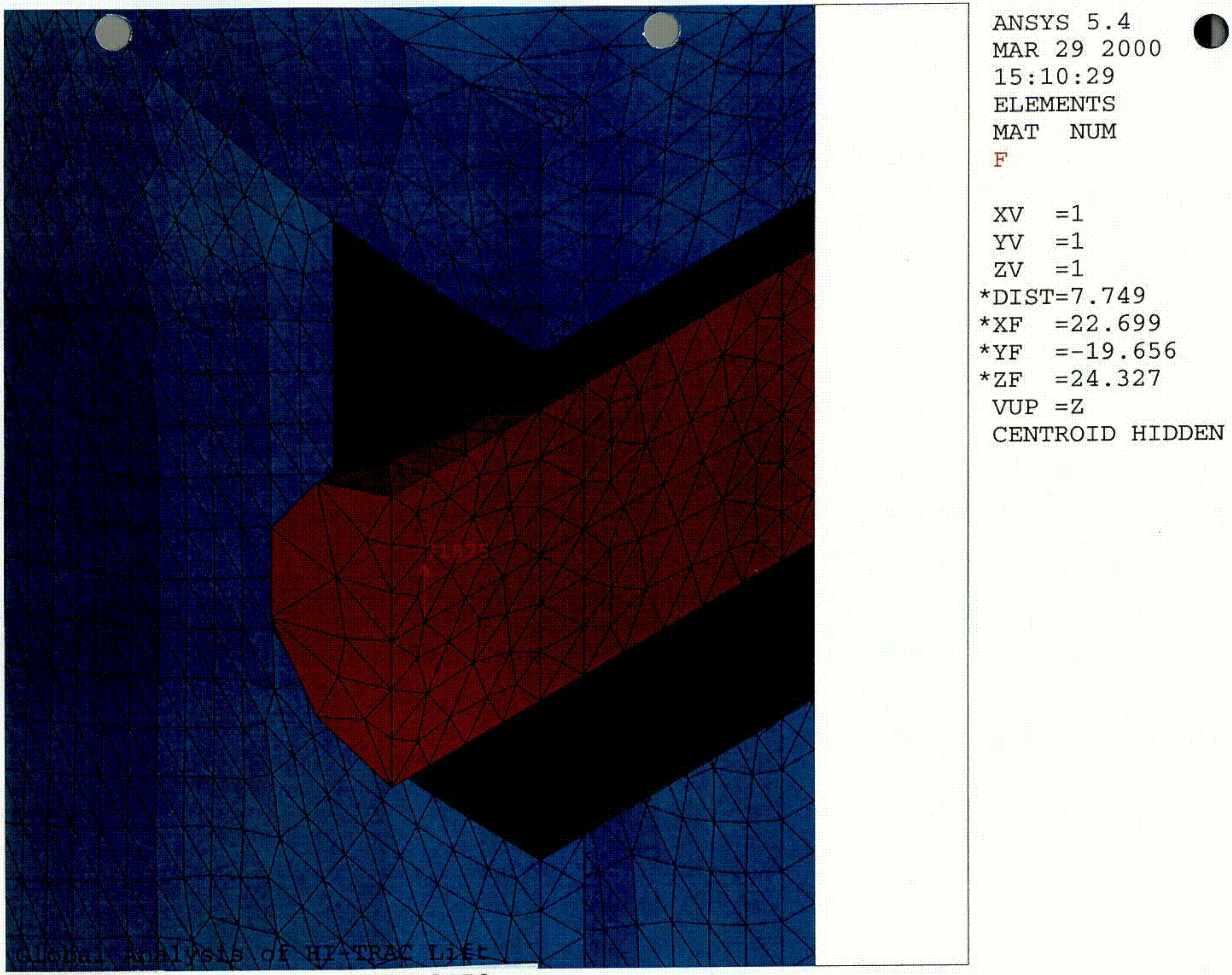

ANSYS 5.4
MAR 29 2000
15:10:29
ELEMENTS
MAT NUM
F
XV =1
YV =1
ZV =1
*DIST=7.749
*XF =22.699
*YF =-19.656
*ZF =24.327
VUP =Z
CENTROID HIDDEN
Global Analysis of HI-TRAC Lift

FIGURE 3.AE.2

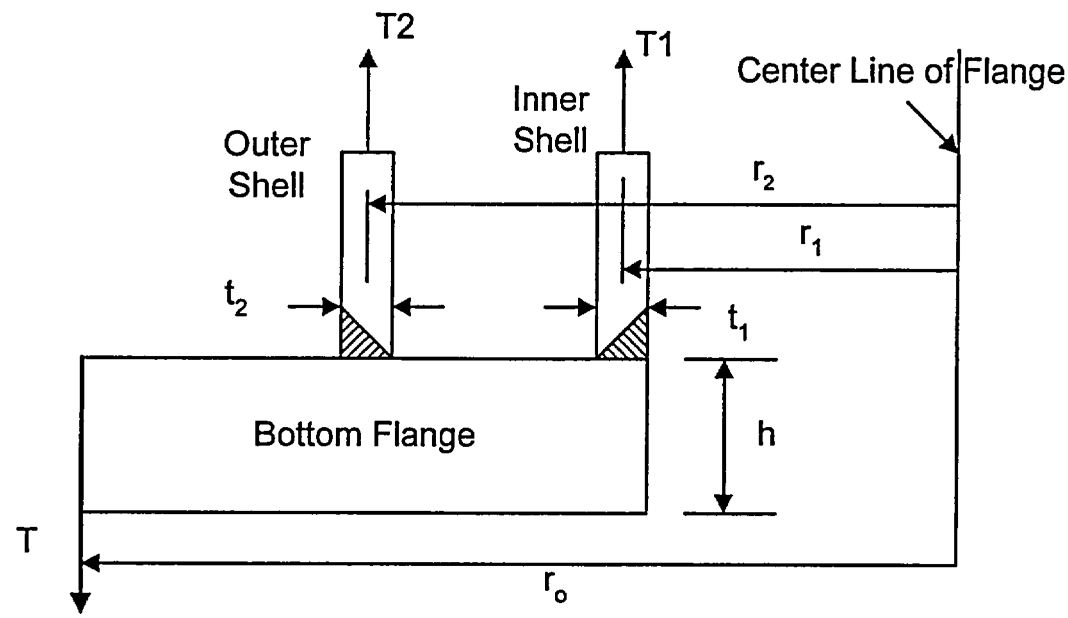


FIGURE 3.AE.3; FREE-BODY OF HI-TRAC 125 BOTTOM FLANGE SHOWING LOAD FROM LID BOLTS "T" AND EQUILIBRIUM LOADS "T1" AND "T2" IN THE INNER AND OUTER SHELLS

APPENDIX 3.AF: MPC TRANSFER FROM HI-TRAC TO HI-STORM 100 UNDER COLD CONDITIONS OF STORAGE

3.AF.1 Scope

In this calculation, estimates of operating gaps, both radially and axially, are computed for the fuel basket-to-MPC shell, and for the MPC shell-to-overpack. This calculation is in support of the results presented in Section 3.4.5. A hot MPC is lowered from a HI-TRAC transfer cask into a storage overpack assumed to be at steady state temperatures appropriate to cold conditions of storage.

3.AF.2 Methodology

Bounding temperatures are used to construct temperature distributions that will permit calculation of differential thermal expansions both radially and axially for the basket-to-MPC gaps, and for the MPC-to-overpack gaps. Reference temperatures are set at 70°F for all components. A comprehensive nomenclature listing is provided in Section 3.AF.6.

3.AF.3 References

[3.AF.1] Boley and Weiner, Theory of Thermal Stresses, John Wiley, 1960, Sec. 9.10, pp. 288-291.

[3.AF.2] Burgreen, Elements of Thermal Stress Analysis, Arcturus Publishers, Cherry Hill NJ, 1988.

3.AF.4 Calculations

3.AF.4.1 Input Data

Based on thermal calculations in Chapter 4 and results from Appendix 3.I, the following temperatures are appropriate at the hottest location of the HI-TRAC (see Figure 3.I.1 and Table 4.5.2).

The temperature change at the overpack inner shell, $\Delta T_{1h} := 0 - 70$

The temperature change at the overpack outer shell, $\Delta T_{2h} := 0 - 70$

The temperature change at the mean radius of the MPC shell, $\Delta T_{3h} := 455 - 70$ |

The temperature change at the outside of the MPC basket, $\Delta T_{4h} := (600 - 70) \cdot 1.1$ |

The temperature change at the center of the basket (helium gas), $\Delta T_{5h} := 852 - 70$ |

Note that the outer basket temperature is conservatively amplified by 10% to insure a bounding parabolic distribution. This conservatism serves to maximize the growth of the basket.
The geometry of the components are as follows (referring to Figure 3.U.1) |

The outer radius of the overpack, $b := 66.25 \cdot in$

The inner radius of the overpack, $a := 34.75 \cdot in$

The mean radius of the MPC shell, $R_{mpc} := \frac{68.375 \cdot in - 0.5 \cdot in}{2}$ $\quad R_{mpc} = 33.938\,in$

The initial MPC-to-storage overpack radial clearance, $RC_{mo} := .5 \cdot (69.5 - 68.5) \cdot in$

$RC_{mo} = 0.5\,in$

This initial radial clearance value, used to perform a radial growth check, is conservatively based on the channel radius (see Dwg. 1495, Sh. 5) and the maximum diameter of the MPC. For axial growth calculations for the MPC-to-overpack lid clearance, the axial length of the overpack is defined as the distance from the top of the pedestal platform to the bottom of the lid bottom plate, and the axial length of the MPC is defined as the overall MPC height.

The axial length of the overpack, $L_{ovp} := 191.5 \cdot in$

The axial length of the MPC, $L_{mpc} := 190.5 \cdot in$

The initial MPC-to-overpack nominal axial clearance, $AC_{mo} := L_{ovp} - L_{mpc}$

$AC_{mo} = 1\,in$

For growth calculations for the fuel basket-to-MPC shell clearances, the axial length of the basket is defined as the total length of the basket and the outer radius of the basket is defined as the mean radius of the MPC shell minus one-half of the shell thickness minus the initial basket-to-shell radial clearance.

The axial length of the basket, $L_{bas} := 176.5\,in$

The initial basket-to-MPC lid nominal axial clearance, $AC_{bm} := 1.8125 \cdot in$

The initial basket-to-MPC shell nominal radial clearance, $RC_{bm} := 0.1875 \cdot in$

The outer radius of the basket, $R_b := R_{mpc} - \frac{0.5}{2} \cdot in - RC_{bm}$ $\quad R_b = 33.5\,in$

The coefficients of thermal expansion used in the subsequent calculations are based on the mean temperatures of the MPC shell and the basket (conservatively estimated high).

The coefficient of thermal expansion for the MPC shell, $\alpha_{mpc} := 9.338 \cdot 10^{-6}$

The coefficient of thermal expansion for the basket, $\alpha_{bas} := 9.90 \cdot 10^{-6}$ 600 deg. F

3.AF.4.2 Thermal Growth of the Overpack

Results for thermal expansion deformation and stress in the overpack are obtained here. The system is replaced by a equivalent uniform hollow cylinder with approximated average properties.

Based on the given inside and outside surface temperatures, the temperature solution in the cylinder is given in the form:

$$C_a + C_b \cdot \ln\left(\frac{r}{a}\right)$$

where

$$C_a := \Delta T_{1h} \qquad C_a = -70$$

$$C_b := \frac{\Delta T_{2h} - \Delta T_{1h}}{\ln\left(\frac{b}{a}\right)} \qquad C_b = 0$$

Next, form the integral relationship:

$$Int := \int_a^b \left[C_a + C_b \cdot \left(\ln\left(\frac{r}{a}\right)\right)\right] \cdot r\,dr$$

The Mathcad program, which was used to create this appendix, is capable of evaluating the integral "Int" either numerically or symbolically. To demonstrate that the results are equivalent, the integral is evaluated both ways in order to qualify the accuracy of any additional integrations that are needed.

The result obtained through numerical integration, $Int = -1.114 \times 10^5\, in^2$

To perform a symbolic evaluation of the solution the integral "Ints" is defined. This integral is then evaluated using the Maple symbolic math engine built into the Mathcad program as:

$$Int_s := \int_a^b \left[C_a + C_b \cdot \left(\ln\left(\frac{r}{a}\right)\right)\right] \cdot r\,dr$$

$$Int_s := \frac{1}{2} \cdot C_b \cdot \ln\left(\frac{b}{a}\right) \cdot b^2 + \frac{1}{2} \cdot C_a \cdot b^2 - \frac{1}{4} \cdot C_b \cdot b^2 + \frac{1}{4} \cdot C_b \cdot a^2 - \frac{1}{2} \cdot C_a \cdot a^2$$

$$Int_s = -1.114 \times 10^5\, in^2$$

We note that the values of Int and Ints are identical. The average temperature change in the overpack cylinder (T_{bar}) is therefore determined as:

$$T_{bar} := \frac{2}{\left(b^2 - a^2\right)} \cdot Int \qquad T_{bar} = -70$$

In this case, the result of the calculation is obvious and simply affords an independent check!!

We estimate the average coefficient of thermal expansion for the overpack by weighting the volume of the various layers. A total of four layers are identified for this calculation. They are:

1) the inner shell
2) the shield shell
3) the radial shield
4) the outer shell

Note that the shield shell was removed from the HI-STORM 100 design as of 6/01. The replacement of the shield shell with concrete, however, has a negligible effect on the resultant coefficient of thermal expansion because (a) the difference in thermal expansion coefficients between concrete and carbon steel is small and (b) the shield shell accounts for a small percentage of the total overpack radial thickness.

Thermal properties are based on estimated temperatures in the component and coefficient of thermal expansion values taken from the tables in Chapter 3. The following averaging calculation involves the thicknesses (t) of the various components, and the estimated coefficients of thermal expansion at the components' mean radial positions. The results of the weighted average process yields an effective coefficient of linear thermal expansion for use in computing radial growth of a solid cylinder (the overpack).

The thicknesses of each component are defined as:

$$t_1 := 1.25 \cdot in$$

$$t_2 := 0.75 \cdot in$$

$$t_3 := 26.75 \cdot in$$

$$t_4 := 0.75 \cdot in$$

and the corresponding mean radii can therefore be defined as:

$$r_1 := a + .5\, t_1 + 2.0 \cdot in$$ (add the channel depth)

$$r_2 := r_1 + .5 \cdot t_1 + .5 \cdot t_2$$

$$r_3 := r_2 + .5 \cdot t_2 + .5 \cdot t_3$$

$$r_4 := r_3 + .5 \cdot t_3 + .5 \cdot t_4$$

To check the accuracy of these calculations, the outer radius of the overpack is calculated from r_4 and t_4, and the result is compared with the previously defined value (b).

$$b_1 := r_4 + 0.5 \cdot t_4$$

$$b_1 = 66.25\,in$$

$$b = 66.25\,in$$

We note that the calculated value b_1 is identical to the previously defined value b. The coefficients of thermal expansion for each component, estimated based on the temperature gradient, are defined as:

$$\alpha_1 := 5.53 \cdot 10^{-6}$$

$$\alpha_2 := 5.53 \cdot 10^{-6}$$

$$\alpha_3 := 5.5 \cdot 10^{-6}$$

$$\alpha_4 := 5.53 \cdot 10^{-6}$$

Thus, the average coefficient of thermal expansion of the overpack is determined as:

$$\alpha_{avg} := \frac{r_1 \cdot t_1 \cdot \alpha_1 + r_2 \cdot t_2 \cdot \alpha_2 + r_3 \cdot t_3 \cdot \alpha_3 + r_4 \cdot t_4 \cdot \alpha_4}{\frac{a + b}{2} \cdot (t_1 + t_2 + t_3 + t_4)}$$

$$\alpha_{avg} = 5.611 \times 10^{-6}$$

Reference 3.AF.1 gives an expression for the radial deformation due to thermal growth. At the inner radius of the overpack (r = a), the radial growth is determined as:

$$\Delta R_{ah} := \alpha_{avg} \cdot a \cdot T_{bar}$$

$$\Delta R_{ah} = -0.014\,in$$

Similarly, an overestimate of the axial growth of the overpack can be determined by applying the average temperature (T_{bar}) over the entire length of the overpack as:

$$\Delta L_{ovph} := L_{ovp} \cdot \alpha_{avg} \cdot T_{bar}$$

$$\Delta L_{ovph} = -0.075 \text{ in}$$

As expected, the drop in temperature causes a decrease in the inner radius and the axial length of the storage overpack.

3.AF.4.3 Thermal Growth of the MPC Shell

The radial and axial growth of the MPC shell (ΔR_{mpch} and ΔL_{mpch}, respectively) are determined as:

$$\Delta R_{mpch} := \alpha_{mpc} \cdot R_{mpc} \cdot \Delta T_{3h} \qquad \Delta R_{mpch} = 0.122 \text{ in}$$

$$\Delta L_{mpch} := \alpha_{mpc} \cdot L_{mpc} \cdot \Delta T_{3h} \qquad \Delta L_{mpch} = 0.685 \text{ in}$$

3.AF.4.4 Clearances Between the MPC Shell and Overpack

The final radial and axial MPC shell-to-overpack clearances (RG_{moh} and AG_{moh}, respectively) are determined as:

$$RG_{moh} := RC_{mo} + \Delta R_{ah} - \Delta R_{mpch} \qquad RG_{moh} = 0.364 \text{ in}$$

$$AG_{moh} := AC_{mo} + \Delta L_{ovph} - \Delta L_{mpch} \qquad AG_{moh} = 0.24 \text{ in}$$

Note that this axial clearance (AG_{moh}) is based on the temperature distribution at the hottest axial location of the system.

3.AF.5 Summary of Results

The previous results are summarized here.

MPC Shell-to-Overpack

Radial clearance $RG_{moh} = 0.364 \text{ in}$ Axial clearance $AG_{moh} = 0.24 \text{ in}$

3.AF.6 Nomenclature

a is the inner radius of the overpack
AC_{bm} is the initial fuel basket-to-MPC axial clearance.
AC_{mo} is the initial MPC-to-overpack axial clearance.
AG_{bmh} is the final fuel basket-to-MPC shell axial gap for the hot components.
AG_{moh} is the final MPC shell-to-overpack axial gap for the hot components.
b is the outer radius of the overpack.
L_{bas} is the axial length of the fuel basket.
L_{mpc} is the axial length of the MPC.
L_{ovp} is the axial length of the overpack.
r_1 (r_2,r_3,r_4) is mean radius of the overpack inner shell (shield shell, concrete, outer shell).
R_b is the outer radius of the fuel basket.
R_{mpc} is the mean radius of the MPC shell.
RC_{bm} is the initial fuel basket-to-MPC radial clearance.
RC_{mo} is the initial MPC shell-to-overpack radial clearance.
RG_{bmh} is the final fuel basket-to-MPC shell radial gap for the hot components.
RG_{moh} is the final MPC shell-to-overpack radial gap for the hot components.
t_1 (t_2,t_3,t_4) is the thickness of the overpack inner shell (shield shell, concrete, outer shell).
T_{bar} is the average temperature of the overpack cylinder.
α_1 (α_2,α_3,α_4) is the coefficient of thermal expansion of the overpack inner shell (shield shell, concrete, outer shell).
α_{avg} is the average coefficient of thermal expansion of the overpack.
α_{bas} is the coefficient of thermal expansion of the overpack.
α_{mpc} is the coefficient of thermal expansion of the MPC.
ΔL_{bh} is the axial growth of the fuel basket for the hot components.

ΔL_{mpch} the the axial growth of the MPC for the hot components.
ΔL_{ovph} is the axial growth of the overpack for the hot components.
ΔR_{ah} is the radial growth of the overpack inner radius for the hot components.
ΔR_{bh} is the radial growth of the fuel basket for the hot components.
ΔR_{mpch} is the radial growth of the MPC shell for the hot components.
ΔT_{1h} is the temperature change at the overpack inner shell for hot components.
ΔT_{2h} is the temperature change at the overpack outer shell for hot components.
ΔT_{3h} is the temperature change at the MPC shell mean radius for hot components.
ΔT_{4h} is the temperature change at the MPC basket periphery for hot components.
ΔT_{5h} is the temperature change at the MPC basket centerline for hot components.
ΔT_{bas} is the fuel basket centerline-to-periphery temperature gradient.
σ_{ca} is the circumferential stress at the overpack inner surface.
σ_{cb} is the circumferential stress at the overpack outer surface.
σ_r is the maximum radial stress of the overpack.
σ_{z1} is the axial stress at the fuel basket centerline.
σ_{zo} is the axial stress at the fuel basket periphery.

APPENDIX 3.AG - STRESS ANALYSIS OF THE HI-TRAC WATER JACKET

3.AG.1 Introduction: This calculation determines the stress level in the HI-TRAC water jacket and loaded water jacket welds under the combined effects of internal pressure caused by heating, by hydrostatic effects, and by dynamic effects during lifting and transport.

3.AG.2 Methodology: Formulas from theory of elastic plates are used to calculate maximum stress in the outer enclosure panels and in the top and bottom.

3.AG.3 References:

[3.1] Roark's Formulas for Stress and Strain, 6th Edition, McGraw-Hill, 1989.

[3.2] Mathcad 8.0, Mathsoft, 1998.

[3.3] Strength of Materials Part II, S.P. Timoshenko, McGraw-Hill, 3rd Edition, 1956.

3.AG.4 125 Ton HI-TRAC

3.AG.4.1 Input Data: All dimensions taken from Holtec drawing 1880 for HI-TRAC

Thickness of enclosure shell panels	$t_v := .5 \cdot in$
Thickness of bottom flange	$t_p := 1.0 \cdot in$
Bottom flange outer diameter	$OD := 94.625 \cdot in$
Bottom flange inner diameter	$ID := 81.25 \cdot in$

The allowable strength is (SA-516,Gr.70, Table 3.1.10 of the HI-STORM FSAR), For membrane stress, the allowable strength value is:

$S_a := 17500 \cdot psi$ For a bending stress evaluation, this value is increased 50%.

The ultimate strength of the base material (used to evaluate the welds) is

$S_u := 70000 \cdot psi$ (Table 3.3.2)

3.AG.4.2 Calculations

3.AG.4.2.1 Design pressure

For the purpose of this calculation, the design pressure must be first calculated by adding the saturation pressure at the peak jacket temperature (Table 2.2.1 of the HI-STORM FSAR) with the hydrodynamic pressure of the water.

The saturation pressure is:

$$p_{sat} := 60 \cdot psi$$

If the water density and the height of the water jacket are:

$\gamma_{water} := 62.4 \cdot \frac{lbf}{ft^3}$ $h_{jacket} := 168.75 \cdot in$ (Holtec BM-1880)

The hydrostatic pressure at the bottom of the water jacket is:

$p_{hs} := \gamma_{water} \cdot h_{jacket}$ $p_{hs} = 6.09\,psi$

The dynamic load factor for vertical transport of HI-TRAC is 0.15, so the pressure needs to be amplified by 1.15. The hydrodynamic pressure at the base of the water jacket is then calculated as:

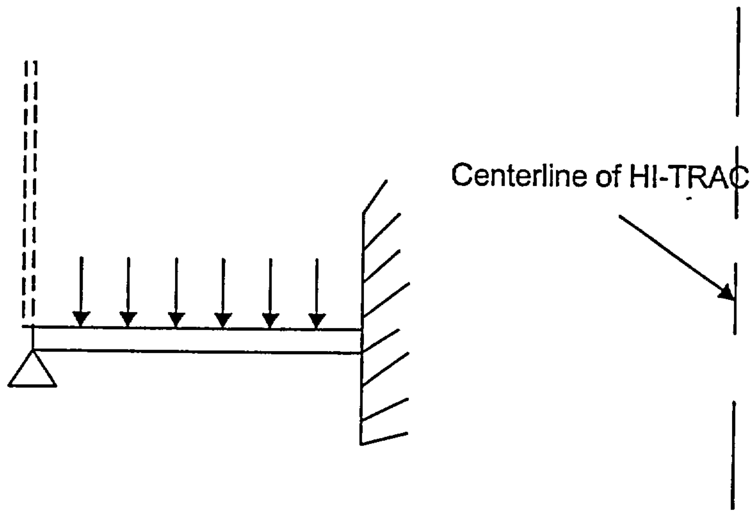
$p_{hd} := 1.15 \cdot p_{hs}$ $p_{hd} = 7.01\,psi$

The design pressure is:

$q := p_{sat} + p_{hd}$ $q = 67.01\,psi$

3.AG.4.2.2 Bottom annular flange:

The flange is considered as an annular plate clamped at the inside diameter, and conservatively assumed pinned at the outside diameter. That is, there is no welded connection, capable of transmitting a moment, assumed to exist at the connection of the bottom flange with the enclosure shell panels of the outer enclosure shell panels to the annulus. Intermediate vertical support from the radial ribs is conservatively neglected. The results are obtained from Section 23, case 4 of Figure 72 in [3.3].



$radius\_ratio := \frac{OD}{ID}$ $\qquad$ $radius\_ratio = 1.165$

From Table 5 of the reference $\qquad$ $k := 0.122$ $\qquad$ $k_1 := 0.00343$

The maximum bending stress in the annular flange is obtained from [3.3] as:

$$\sigma_{plate} := k \cdot q \cdot \left(\frac{OD}{2 \cdot t_p}\right)^2 \qquad \sigma_{plate} = 1.83 \times 10^4 psi \qquad < 1.5S_a$$

The safety factor in the annular flange is. $\qquad$ $\frac{1.5 \cdot S_a}{\sigma_{plate}} = 1.434$

This is a conservative result as it neglects the effect of partial clamping action at the connection with the outer enclosure panels. The result for the bottom flange bounds the result for the top flange since the applied pressure is less.

3.AG.4.2.3 Outer Enclosure Flat Panels

Revised Analysis of HI-TRAC Water Jacket Enclosure Shell

Scope: Number of ribs has been reduced and enclosure panels are now curved. Therefore, analysis model becomes a clamped shallow arch under internal pressure.

Cases 5-14 Loading Terms

Partial Uniformly Distributed Radial Loading

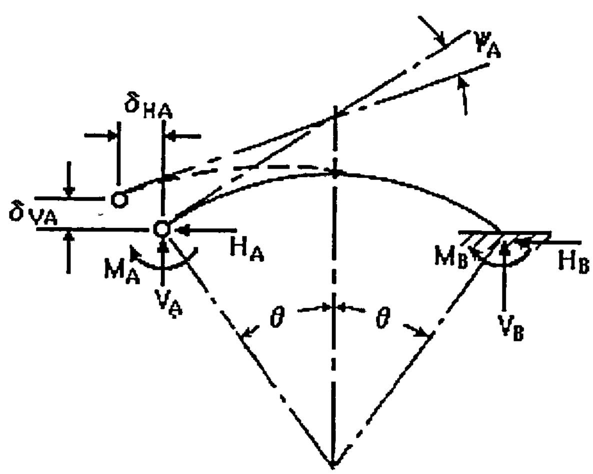
This file corresponds to Cases 5j, 6j, 7j, 8j, 9j, 10j, 11j, 12j, 13j and 14j in *Roark's Formulas for Stress and Strain.*

This file contains the general formulas for the reaction moment, horizontal end reaction, vertical end reaction, horizontal deflection, vertical deflection and angular rotation for a circular arch with a partial uniformly distributed radial loading. Because the constants and loading terms necessary to calculate these formulas remain the same under certain conditions, the following 10 restraint conditions with the above load have been included in this file:

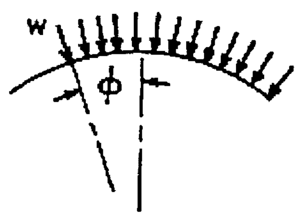
Case 5: Left end fixed, right end fixed

The following sketches should be referred to for definitions of dimensions and loadings:

Circular arch



Partial uniformly distributed radial loading



Notation file Provides a description of Table 18 and the notation used.

Enter properties of cross section Before progressing further, it is necessary to flip to Table 16 to calculate the following values for your cross section:

- principal centroidal moment of inertia (I_c)
- area of the cross section (A)
- shape constant (F)
- distance from centroidal axis to neutral axis (h)

Table 16 Once this is done, enter the computed values below.

Principal centroidal moment of inertia:	$I_c := 0.01 \cdot in^4$
Area of cross section:	$A := 0.5 \cdot in^2$
Shape constant:	$F := \frac{6}{5}$
Distance between axes:	$h := .0004301 \cdot in$

Enter dimensions, properties and loading of arch

Radius of curvature: $R := 46.5 \cdot in$

Half-span of the beam: $\theta := 15 \cdot deg$

Height of cross section: $d := 0.5 \cdot in$

Modulus of elasticity: $E := 27 \cdot 10^6 \cdot \frac{lbf}{in^2}$

Poisson's ratio: $\nu := 0.3$

Load: $w := -75 \cdot \frac{lbf}{in}$

Angle from vertical to load: $\phi := 15 \cdot deg$

Constants

These constants are used in the formulas to calculate the reaction moment, horizontal and vertical end reactions, horizontal and vertical deflections and angular rotation at A:

$$G := \frac{E}{2 \cdot (1 + \nu)} \qquad thin := if\left(\frac{R}{d} \geq 8, 1, 0\right) ck := if\left(\frac{R}{d} < 8, 1, 0\right)$$

$$\alpha := thin \cdot \left(\frac{I_c}{A \cdot R^2}\right) + thick \cdot \left(\frac{h}{R}\right)$$

$$\beta := thin \cdot \left(\frac{F \cdot E \cdot I_c}{G \cdot A \cdot R^2}\right) + thick \cdot \left[\frac{2 \cdot F \cdot (1 + \nu) \cdot h}{R}\right]$$

$$k_1 := 1 - \alpha + \beta \qquad k_2 := 1 - \alpha$$

$$s := \sin(\theta) \qquad n := \sin(\phi)$$

$$c := \cos(\theta) \qquad e := \cos(\phi)$$

$$B_{HH} := 2\cdot\theta\cdot c^2 + k_1\cdot(\theta - s\cdot c) - k_2\cdot 2\cdot s\cdot c$$

$$B_{HV} := -2\cdot\theta\cdot s\cdot c + k_2\cdot 2\cdot s^2 \qquad B_{VH} := B_{HV}$$

$$B_{HM} := -2\cdot\theta\cdot c + k_2\cdot 2\cdot s \qquad B_{MH} := B_{HM}$$

$$B_{VV} := 2\cdot\theta\cdot s^2 + k_1\cdot(\theta + s\cdot c) - k_2\cdot 2\cdot s\cdot c$$

$$B_{VM} := 2\cdot\theta\cdot s \qquad B_{MV} := B_{VM}$$

$$B_{MM} := 2\cdot\theta$$

Loading terms

$$LF_H := w\cdot R\cdot\left[\frac{k_1}{2}\cdot\left(s\cdot c\cdot e + c^2\cdot n - \theta\cdot e - \phi\cdot e\right) \ldots \atop + k_2\cdot(s + n - \theta\cdot c - \phi\cdot c)\right]$$

$$LF_V := w\cdot R\cdot\left[\frac{k_1}{2}\cdot\left(\theta\cdot n + \phi\cdot n + s\cdot c\cdot n + s^2\cdot e\right) \ldots \atop + k_2\cdot\left(\theta\cdot s + \phi\cdot s - 2\cdot s\cdot c\cdot n + 2\cdot c^2\cdot e - c - e\right)\right]$$

$$LF_M := w\cdot R\cdot\left[k_2\cdot(\theta + \phi - s\cdot e - c\cdot n)\right]$$

Formulas for horizontal and vertical deflections, reaction moment, horizontal and vertical end reactions and angular rotation at the left edge

Case 5j **Left end fixed, right end fixed**

Horizontal deflection: $\delta_{HA} := 0 \cdot in$

Vertical deflection: $\delta_{VA} := 0 \cdot in$

Angular rotation: $\psi_A := 0 \cdot deg$

Because the above equal zero, the following three equations are solved simultaneously using Mathcad's solve block for H_A, V_A and M_A:

Enter guess values:

$H_A := 1000 \cdot lbf$ $V_A := 1000 \cdot lbf$ $M_A := 100 \cdot lbf \cdot ft$

Define the three linear expressions:

Given

$$B_{HH} \cdot H_A + B_{HV} \cdot V_A + \frac{B_{HM} \cdot M_A}{R} = LF_H$$

$$B_{VH} \cdot H_A + B_{VV} \cdot V_A + \frac{B_{VM} \cdot M_A}{R} = LF_V$$

$$B_{MH} \cdot H_A + B_{MV} \cdot V_A + \frac{B_{MM} \cdot M_A}{R} = LF_M$$

Solve for H_A, V_A and M_A:

$$\begin{pmatrix} H_A \\ V_A \\ M_A \end{pmatrix} := \text{Find}(H_A, V_A, M_A)$$

Horizontal end reaction: $H_A = 3.087 \times 10^3 \, lbf$

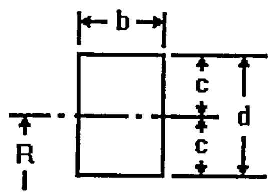
Vertical end reaction: $V_A = -902.631 \, lbf$

Reaction moment: $M_A = 24.915 \, lbf \cdot ft$

Table 16 Formulas for curved beams subjected to bending in the plane of the curve

Case 1 Solid Rectangular Section

Solid rectangular section



Notation file Provides a description of Table 16 and the notation used.

Enter dimensions

Radius of curvature measured to centroid of section: $R := 46.5 \cdot in$

Height of rectangular section: $d := 0.5 \cdot in$

Width of rectangular section: $b := 1 \cdot in$

Conditions

If R/d ≥ 8, then the beam should generally be considered thin.
If R/d < 8, then the beam can generally be considered thick.

$$\frac{R}{d} = 93 \qquad thin := if\left(\frac{R}{d} \geq 8, 1, 0\right) \quad thick := if\left(\frac{R}{d} < 8, 1, 0\right)$$

Constants

Half-height: $c := \frac{d}{2}$ $c = 0.25\,in$

$\int_{area} \frac{dA}{r}$ is equal to $b \cdot \ln\left(\frac{\frac{R}{c}+1}{\frac{R}{c}-1}\right) = 0.011\,in$

Shape constant for rectangle:
See article 7.10 on page 201 in Roark. $F := \frac{6}{5}$

For all of the cross sections shown in this table this centroidal axis perpendicular to the plane of bending is a principal axis of the cross section.

Moment of inertia of section about centroidal axis perpendicular to the plane of bending:

$I_c := \frac{b \cdot d^3}{12}$ $I_c = 0.01\,in^4$

Area:

$A := b \cdot d$ $A = 0.5\,in^2$

Distance from centroidal axis to neutral axis measured toward center of curvature:

$$h := c \cdot \left(\frac{R}{c} - \frac{2}{\ln\left(\frac{\frac{R}{c}+1}{\frac{R}{c}-1}\right)}\right) \cdot thick + \frac{I_c}{R \cdot A} \cdot thin \qquad h = 4.48 \times 10^{-4}\,in$$

k_i is the ratio of actual stress in extreme fiber on the concave side (σ_i) to unit stress (σ) in corresponding fiber as computed by ordinary flexure formula for a straight beam (σ_i/σ).

k_o is the ratio of actual stress in extreme fiber on the convex side (σ_o) to unit stress (σ) in corresponding fiber as computed by ordinary flexure formula for a straight beam (σ_o/σ).

$$k_i := \left(\frac{1}{\frac{3\cdot h}{c}}\right)\cdot\left(\frac{1-\frac{h}{c}}{\frac{R}{c}-1}\right) \qquad k_i = 1.004$$

$$k_o := \left(\frac{1}{\frac{3\cdot h}{c}}\right)\cdot\left(\frac{1+\frac{h}{c}}{\frac{R}{c}+1}\right) \qquad k_o = 0.996$$

At 75 psi

$$S := -w\cdot\frac{R}{b\cdot d} \qquad S = 6.975\times 10^{3}\,\text{psi}$$

$$\sigma := 6\cdot\frac{M_A}{b\cdot d^2}$$

$$\sigma = 7.175\times 10^{3}\,\text{psi}$$

Maximum Combined Stress on the Panel:

Membrane Stress (S) + Bending Stress (σ)

$$\sigma_{max} := S + k_i\cdot\sigma \qquad \sigma_{max} = 1.418\times 10^{4}\,\text{psi}$$

$$SF := \frac{1.5\cdot S_a}{\sigma_{max}} \qquad SF = 1.852$$

3.AG.4.2.4 Weld Stress and Panel Direct Stress

Enclosure outer panel weld

This is a full penetration weld. Therefore, the stress in the weld is equal to the stress in the base metal.

Top and Bottom Flange weld to outer shell

This is a double fillet weld. The assumed load is the maximum bending moment developed in the bottom flange.

$\sigma_{plate} = 1.83 \times 10^4 \text{psi}$

$$M_{flange} := \frac{\sigma_{plate} \cdot t_p^2}{6} \qquad M_{flange} = 3.05 \times 10^3 \text{in} \cdot \frac{\text{lbf}}{\text{in}}$$

The fillet weld size at this location is

$$t_{fweld} := 0.25 \cdot \text{in}$$

The moment capacity of the weld is the effective force through the throat of the weld on each surface of the flange multiplied by the distance between the centroids of each of the welds (flange thickness + 2/3 weld leg size).

Therefore, the force at the weld throat is

$$F_{throat} := \frac{M_{flange} \cdot 1 \cdot \text{in}}{\left(t_p + .667 \cdot t_{fweld}\right)} \qquad F_{throat} = 2.614 \times 10^3 \text{lbf}$$

Therefore the shear stress in the throat of each weld in

$$\tau_{fweld} := \frac{F_{throat}}{0.7071 \cdot t_{fweld} \cdot 1 \cdot \text{in}} \qquad \tau_{fweld} = 1.479 \times 10^4 \text{psi}$$

Therefore, the safety factor for this weld is

$$SF_{fweld} := \frac{.3 \cdot S_u}{\tau_{fweld}} \qquad SF_{fweld} = 1.42$$

The primary membrane stress developed in the radially oriented portion of of the outer enclosure panels is computed as follows for the 1" strip:

$$Load := V_A \cdot \sin(\theta) + H_A \cdot \cos(\theta)\ d = 2.748 \times 10^3\ lbf$$

The panel radially oriented direct stress is

$$\sigma_{direct} := \frac{Load}{1.25in \cdot 1in} \qquad \sigma_{direct} = 2.198 \times 10^3\ psi$$

The safety factor is

$$SF_{radial} := \frac{S_a}{\sigma_{direct}} \qquad SF_{radial} = 7.961$$

3.AG.4.2.5 Conclusion

The design is acceptable for a maximum water jacket pressure of

$q = 67.008\ psi$

Based on the result that the minimum safety factor is 1.42, a 10% overpressure will be acceptable for a hydrotest for leaks.

$$p_{test} := 1.1 \cdot q \qquad p_{test} = 73.709\ psi$$

3.AG.5 100 Ton HI-TRAC

3.AG.5.1 Input Data: All dimensions taken from Holtec drawing 2145 for HI-TRAC

Thickness of outer enclosure panel	$t_v := .375 \cdot in$
Thickness of bottom flange	$t_p := 1.0 \cdot in$
Bottom flange outer diameter	$OD := 91.0 \cdot in$
Bottom flange inner diameter	$ID := 78.0 \cdot in$

The allowable membrane strength is (SA-516,Gr.70, Table 3.1.10 of the HI-STORM FSAR),

$$S_a := 17500 \cdot psi$$

3.AG.5.2 Calculations

3.AG.5.2.1 Design pressure

For the purpose of this calculation, the design pressure must be first calculated by adding the saturation pressure at the peak jacket temperature (Table 2.2.1 of the HI-STORM FSAR) with the hydrodynamic pressure of the water.

The saturation pressure is: $p_{sat} := 60 \cdot psi$

If the water density and the height of the water jacket are:

$\gamma_{water} := 62.4 \cdot \frac{lbf}{ft^3}$ $h_{jacket} := 168.75 \cdot in$ (Holtec drawing no. 2145)

The hydrostatic pressure at the bottom of the water jacket is:

$p_{hs} := \gamma_{water} \cdot h_{jacket}$ $p_{hs} = 6.09\,psi$

Incorporating the dynamic amplification assumed during the lifting operation,

$p_{hd} := 1.15 \cdot p_{hs}$ $\qquad$ $p_{hd} = 7.01\,\text{psi}$

The design pressure is, therefore:

$q := p_{sat} + p_{hd}$ $\qquad$ $q = 67.01\,\text{psi}$

3.AG.5.2.2 Bottom annular flange:

The flange is considered to be clamped at the inside diameter, and pinned at the outside diameter. That is, there is no welded connection of the side plates to the annulus. The results are obtained from Section 23, case 4 of Figure 72 in [3.3].

$\text{radius\_ratio} := \dfrac{OD}{ID}$ $\qquad$ $\text{radius\_ratio} = 1.167$

From Table 5 of the reference $\qquad$ $k := 0.122$ $\qquad$ $k_1 := 0.00343$

The maximum stress in the annular flange is

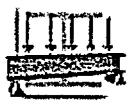
$\sigma_{plate} := k \cdot q \cdot \left(\dfrac{OD}{2 \cdot t_p}\right)^2$ $\qquad$ $\sigma_{plate} = 1.692 \times 10^4\,\text{psi}$ $\qquad$ $< 1.5S_a$

The calculated safety factor is. $\qquad$ $\dfrac{1.5 \cdot S_a}{\sigma_{plate}} = 1.551$

3.AG.5.2.3 Outer Enclosure Panels

Revised Analysis of HI-TRAC Water Jacket Enclosure Shell

Scope: Number of ribs has been reduced and enclosure panels are now curved. Therefore, analysis model becomes a clamped shallow arch under internal pressure.



Cases 5-14 Loading Terms

Partial Uniformly Distributed Radial Loading

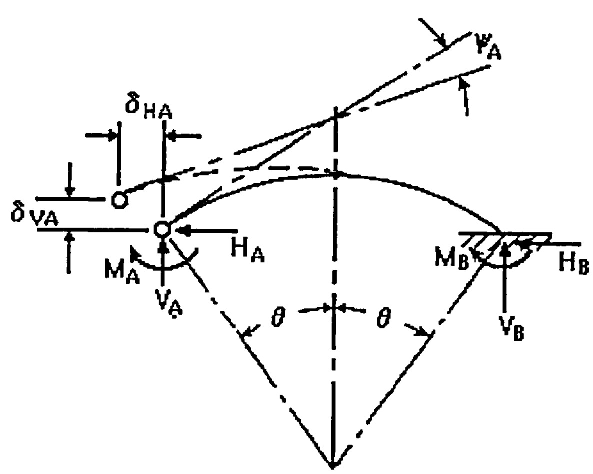
This file corresponds to Cases 5j, 6j, 7j, 8j, 9j, 10j, 11j, 12j, 13j and 14j in *Roark's Formulas for Stress and Strain.*

This file contains the general formulas for the reaction moment, horizontal end reaction, vertical end reaction, horizontal deflection, vertical deflection and angular rotation for a circular arch with a partial uniformly distributed radial loading. Because the constants and loading terms necessary to calculate these formulas remain the same under certain conditions, the following 10 restraint conditions with the above load have been included in this file:

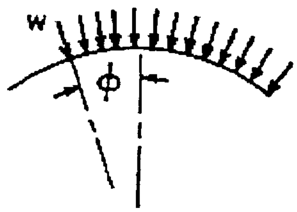
Case 5: Left end fixed, right end fixed

The following sketches should be referred to for definitions of dimensions and loadings:

Circular arch



Partial uniformly distributed radial loading



Notation file Provides a description of Table 18 and the notation used.

Enter properties of cross section **Before progressing further, it is necessary to flip to Table 16 to calculate the following values for your cross section:**

- **principal centroidal moment of inertia (I_c)**
- **area of the cross section (A)**
- **shape constant (F)**
- **distance from centroidal axis to neutral axis (h)**

Table 16 **Once this is done, enter the computed values below.**

Principal centroidal moment of inertia: $I_c := 4.395 \times 10^{-3} \cdot in^4$

Area of cross section: $A := .375 \cdot in^2$

Shape constant: $F := \frac{6}{5}$

Distance between axes: $h := .000434 \cdot in$

Enter dimensions, properties and loading of arch

Radius of curvature: $R := 44.375 \cdot in$

Half-span of the beam: $\theta := 18 \cdot deg$

Height of cross section: $d := 0.375 \cdot in$

Modulus of elasticity: $E := 27 \cdot 10^6 \cdot \frac{lbf}{in^2}$

Poisson's ratio: $\nu := 0.3$

Load: $w := -71 \cdot \frac{lbf}{in}$

Angle from vertical to load: $\phi := 18 \cdot deg$

Constants

These constants are used in the formulas to calculate the reaction moment, horizontal and vertical end reactions, horizontal and vertical deflections and angular rotation at A:

$$G := \frac{E}{2 \cdot (1 + \nu)} \qquad thin := if\left(\frac{R}{d} \geq 8, 1, 0\right) \quad thick := if\left(\frac{R}{d} < 8, 1, 0\right)$$

$$\alpha := thin \cdot \left(\frac{I_c}{A \cdot R^2}\right) + thick \cdot \left(\frac{h}{R}\right)$$

$$\beta := thin \cdot \left(\frac{F \cdot E \cdot I_c}{G \cdot A \cdot R^2}\right) + thick \cdot \left[\frac{2 \cdot F \cdot (1 + \nu) \cdot h}{R}\right]$$

$$k_1 := 1 - \alpha + \beta \qquad k_2 := 1 - \alpha$$

$$s := \sin(\theta) \qquad n := \sin(\phi)$$

$$c := \cos(\theta) \qquad e := \cos(\phi)$$

$$B_{HH} := 2\cdot\theta\cdot c^2 + k_1\cdot(\theta - s\cdot c) - k_2\cdot 2\cdot s\cdot c$$

$$B_{HV} := -2\cdot\theta\cdot s\cdot c + k_2\cdot 2\cdot s^2 \qquad B_{VH} := B_{HV}$$

$$B_{HM} := -2\cdot\theta\cdot c + k_2\cdot 2\cdot s \qquad B_{MH} := B_{HM}$$

$$B_{VV} := 2\cdot\theta\cdot s^2 + k_1\cdot(\theta + s\cdot c) - k_2\cdot 2\cdot s\cdot c$$

$$B_{VM} := 2\cdot\theta\cdot s \qquad B_{MV} := B_{VM}$$

$$B_{MM} := 2\cdot\theta$$

Loading terms

$$LF_H := w\cdot R\cdot\left[\frac{k_1}{2}\cdot\left(s\cdot c\cdot e + c^2\cdot n - \theta\cdot e - \phi\cdot e\right) \ldots \\ + k_2\cdot(s + n - \theta\cdot c - \phi\cdot c)\right]$$

$$LF_V := w\cdot R\cdot\left[\frac{k_1}{2}\cdot\left(\theta\cdot n + \phi\cdot n + s\cdot c\cdot n + s^2\cdot e\right) \ldots \\ + k_2\cdot\left(\theta\cdot s + \phi\cdot s - 2\cdot s\cdot c\cdot n + 2\cdot c^2\cdot e - c - e\right)\right]$$

$$LF_M := w\cdot R\cdot\left[k_2\cdot(\theta + \phi - s\cdot e - c\cdot n)\right]$$

Formulas for horizontal and vertical deflections, reaction moment, horizontal and vertical end reactions and angular rotation at the left edge

Case 5j **Left end fixed, right end fixed**

Horizontal deflection: $\delta_{HA} := 0 \cdot in$

Vertical deflection: $\delta_{VA} := 0 \cdot in$

Angular rotation: $\psi_A := 0 \cdot deg$

Because the above equal zero, the following three equations are solved simultaneously using Mathcad's solve block for H_A, V_A and M_A:

Enter guess values:

$H_A := 1000 \cdot lbf$ $V_A := 1000 \cdot lbf$ $M_A := 100 \cdot lbf \cdot ft$

Define the three linear expressions:

Given

$$B_{HH} \cdot H_A + B_{HV} \cdot V_A + \frac{B_{HM} \cdot M_A}{R} = LF_H$$

$$B_{VH} \cdot H_A + B_{VV} \cdot V_A + \frac{B_{VM} \cdot M_A}{R} = LF_V$$

$$B_{MH} \cdot H_A + B_{MV} \cdot V_A + \frac{B_{MM} \cdot M_A}{R} = LF_M$$

Solve for H_A, V_A and M_A:

$$\begin{pmatrix} H_A \\ V_A \\ M_A \end{pmatrix} := \text{Find}(H_A, V_A, M_A)$$

Horizontal end reaction: $H_A = 2.913 \times 10^3 \, lbf$

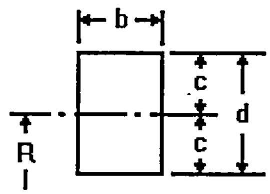
Vertical end reaction: $V_A = -973.597 \, lbf$

Reaction moment: $M_A = 10.177 \, lbf \cdot ft$

Table 16 Formulas for curved beams subjected to bending in the plane of the curve

Case 1 Solid Rectangular Section

Solid rectangular section



Notation file Provides a description of Table 16 and the notation used.

Enter dimensions

Radius of curvature measured to centroid of section: $R := 44.375 \cdot in$

Height of rectangular section: $d := 0.375 \cdot in$

Width of rectangular section: $b := 1 \cdot in$

Conditions

If R/d ≥ 8, then the beam should generally be considered thin.
If R/d < 8, then the beam can generally be considered thick.

$$\frac{R}{d} = 118.333 \quad thin := if\left(\frac{R}{d} \geq 8, 1, 0\right) \quad thick := if\left(\frac{R}{d} < 8, 1, 0\right)$$

Constants

Half-height: $c := \frac{d}{2}$ $\quad c = 0.188\,\text{in}$

$\int_{area} \frac{dA}{r}$ is equal to $b \cdot \ln\left(\frac{\frac{R}{c}+1}{\frac{R}{c}-1}\right) = 8.451 \times 10^{-3}\,\text{in}$

Shape constant for rectangle:
See article 7.10 on page 201 in Roark. $\quad F := \frac{6}{5}$

For all of the cross sections shown in this table this centroidal axis perpendicular to the plane of bending is a principal axis of the cross section.

Moment of inertia of section about centroidal axis perpendicular to the plane of bending:

$$I_c := \frac{b \cdot d^3}{12} \qquad I_c = 4.395 \times 10^{-3}\,\text{in}^4$$

Area:

$$A := b \cdot d \qquad A = 0.375\,\text{in}^2$$

Distance from centroidal axis to neutral axis measured toward center of curvature:

$$h := c \cdot \left(\frac{R}{c} - \frac{2}{\ln\left(\frac{\frac{R}{c}+1}{\frac{R}{c}-1}\right)}\right) \cdot thick + \frac{I_c}{R \cdot A} \cdot thin \qquad h = 2.641 \times 10^{-4}\,\text{in}$$

k_i is the ratio of actual stress in extreme fiber on the concave side (σ_i) to unit stress (σ) in corresponding fiber as computed by ordinary flexure formula for a straight beam (σ_i/σ).

k_o is the ratio of actual stress in extreme fiber on the convex side (σ_o) to unit stress (σ) in corresponding fiber as computed by ordinary flexure formula for a straight beam (σ_o/σ).

$$k_i := \left(\frac{1}{\frac{3 \cdot h}{c}}\right) \cdot \left(\frac{1 - \frac{h}{c}}{\frac{R}{c} - 1}\right) \qquad k_i = 1.003$$

$$k_o := \left(\frac{1}{\frac{3 \cdot h}{c}}\right) \cdot \left(\frac{1 + \frac{h}{c}}{\frac{R}{c} + 1}\right) \qquad k_o = 0.997$$

At 71 psi

$$S := -w \cdot \frac{R}{b \cdot d} \qquad S = 8.402 \times 10^3 \text{psi}$$

$$\sigma := 6 \cdot \frac{M_A}{b \cdot d^2}$$

$$\sigma = 5.21 \times 10^3 \text{psi}$$

<u>Maximum Combined Stress on the Panel:</u>

Membrane Stress (S) + Bending Stress (σ)

$$\sigma_{max} := S + k_i \cdot \sigma \qquad \sigma_{max} = 1.363 \times 10^4 \text{psi}$$

$$SF := \frac{1.5 \cdot S_a}{\sigma_{max}} \qquad SF = 1.926$$

The welds between the bottom flange and the outer shell are the same as in the HI-TRAC 125. Therefore, since the moment is larger there, the previous result is bounding.

The primary membrane stress developed in the radially oriented portion of of the outer enclosure panels is computed as follows for the 1" strip:

$$Load := V_A \cdot \sin(\theta) + H_A \cdot \cos(\theta)\ d = 2.469 \times 10^3\,lbf$$

The panel radially oriented direct stress is

$$\sigma_{direct} := \frac{Load}{1.25in \cdot 1 \cdot in} \qquad \sigma_{direct} = 1.975 \times 10^3\,psi$$

The safety factor is

$$SF_{radial} := \frac{S_a}{\sigma_{direct}} \qquad SF_{radial} = 8.86$$

3.AG.5.2.5 Conclusion

The design is acceptable for a maximum water jacket pressure of

$$q = 67.008\,psi$$

Based on the result that the minimum safety factor is 1.168, a 10% overpressure will be acceptable for a hydrotest for leaks.

$$p_{test} := 1.1 \cdot q \qquad p_{test} = 73.709\,psi$$

Hydrostatic test pressure for the HI-TRAC water jackets is set at 74 psi.

APPENDIX 3.AH HI-TRAC TOP LID SEPARATION ANALYSES

3.AH.1 Introduction

This appendix considers the separation analysis of the 125 ton HI-TRAC top lid under the following condition:

Horizontal Drop of HI-TRAC - Accident Condition

In this case, analysis is limited to showing that the top lid and the transfer cask body do not separate during a HI-TRAC horizontal drop which imposes the design basis G load on the top lid. Results from analysis of the 125 ton unit analysis will bound the 100 ton HI-TRAC top lid results. We also show that under a drop, the top lid and the top lid stud array are sufficiently robust to insure that the MPC is not ejected from the HI-TRAC during the secondary impact.

3.AH.2 References

[3.AH.2.1] J.Shigley and C. Mischke, Mechanical Engineering Design, McGraw Hill, 1989.

[3.AH.2.2] Roark's Handbook for Stress and Strain, 6th Edition, Electronic Version

3.AH.3 Composition

This appendix was created using the Mathcad (version 8.0) software package. Mathcad uses the symbol ':=' as an assignment operator, and the equals symbol '=' retrieves values for constants or variables.

3.AH.4 Input Data for Top Lid

Number of studs $nb := 24$ (Holtec drawing no. 1880)

Top Lid Weight $W := 2750 \cdot lbf$ (Table 3.2.2 for 125 ton HI TRAC)

Design Basis Deceleration $G := 45$

3.AH.5 Separation of Top Lid from HI-TRAC

In the event of a side drop while HI-TRAC is in a horizontal position, the top lid will attempt to separate from the body of the cask. Here, the ultimate shear load capacity of the top lid is computed and compared with the expected G load.

3.AH.5.1 Shear and Tensile Capacity of SA 193 B7 studs and stud Holes

Because of the location of the studs in the top flange (near the outer surface), 300 degrees F is assumed as an appropriate temperature to assess material properties for the studs and for the flange material surrounding the stud holes

$S_{ubolt} := 112020 \cdot psi$ @300 deg. F Table 3.3.4

$S_{ybolt} := 94100 \cdot psi$

Calculation of Thread Capacity

The following calculations are taken from Machinery's Handbook, 23rd Edition, pp. 1278-1279 plus associated screw thread Table 4, p 1514.

Input Geometry Data - 1" UNC, 8 threads/inch, 2A class

$L_e := 1.5 \cdot in$ Thread engagement length

$N := \frac{8}{in}$ Threads per inch

$D_m := 1 \cdot in$ Basic Major Diameter of threads

$D := .9755 \cdot in$ Minimum Major Diameter of External Threads

$E_{min} := .91 \cdot in$ Minimum Pitch Diameter of External Threads

$E_{max} := .9276 in$ Maximum Pitch Diameter of Internal Threads

$K_n := .89 \cdot in$ Maximum Minor Diameter of Internal Threads

Input Strength-Internal Threads (lid or forging); External Threads (bolts)

$$Su_{bolt} := S_{ubolt}$$

The ultimate strength of the top flange material, SA350 LF3 @ 300 degrees F, is

$$Su_{lid} := 66700 \cdot psi$$

Calculation of Tensile stress area (high-strength bolt, ultimate strength exceeding 100,000 psi)

$$A_{th} := \pi \cdot \left(.5 \cdot E_{min} - \frac{0.16238}{N} \right)^2 \qquad A_{tl} := .7854 \cdot \left(D_m - \frac{.9743}{N} \right)^2$$

$$A_{th} = 0.594\, in^2 \qquad A_{tl} = 0.606\, in^2$$

$$A_t := if(Su_{bolt} > 100000 \cdot psi, A_{th}, A_{tl}) \qquad A_t = 0.594\, in^2$$

Calculation of Shear Stress Area per the Handbook

$$A_{ext} := \pi \cdot N \cdot L_e \cdot K_n \cdot \left[\frac{0.5}{N} + 0.57735 \cdot (E_{min} - K_n) \right] \qquad A_{ext} = 2.484\, in^2$$

$$A_{int} := \pi \cdot N \cdot L_e \cdot D \cdot \left[\frac{0.5}{N} + 0.57735 \cdot (D - E_{max}) \right] \qquad A_{int} = 3.315\, in^2$$

Required Length of Engagement per Machinery's Handbook

$$L_{req} := 2 \cdot \frac{A_t}{\frac{A_{ext}}{L_e}} \qquad L_{req} = 0.717\, in$$

Capacity Calculation Using Actual Engagement Length

For the specified (limit) condition, the allowable tensile stress in the bolt is per ASME III, Appendix F

$\sigma_{bolt} := Su_{bolt} \cdot 0.7$ $\qquad \sigma_{bolt} = 7.841 \times 10^4 psi$

The allowable shear stress in the bolt is (use 60% of ultimate since we are performing failure analysis:

$\tau_{bolt} := .6 \cdot Su_{bolt}$ $\qquad \tau_{bolt} = 6.721 \times 10^4 psi$

The allowable shear stress in the lid (or flange) is taken as (here we are examining for safety against failure; hence we use ultimate shear strength of lid material.

$\tau_{lid} := 0.6 \cdot Su_{lid}$ $\qquad \tau_{lid} = 4.002 \times 10^4 psi$

$F_{shear\_lid} := \tau_{lid} \cdot A_{int}$ $\qquad F_{shear\_lid} = 1.327 \times 10^5 lbf$

For the bolt, the allowable strength is the yield strength

$F_{tensile\_bolt} := \sigma_{bolt} \cdot A_t$ $\qquad F_{tensile\_bolt} = 4.655 \times 10^4 lbf$

$F_{shear\_bolt} := \tau_{bolt} \cdot A_{ext}$ $\qquad F_{shear\_bolt} = 1.67 \times 10^5 lbf$

Therefore, bolt tension governs the design.

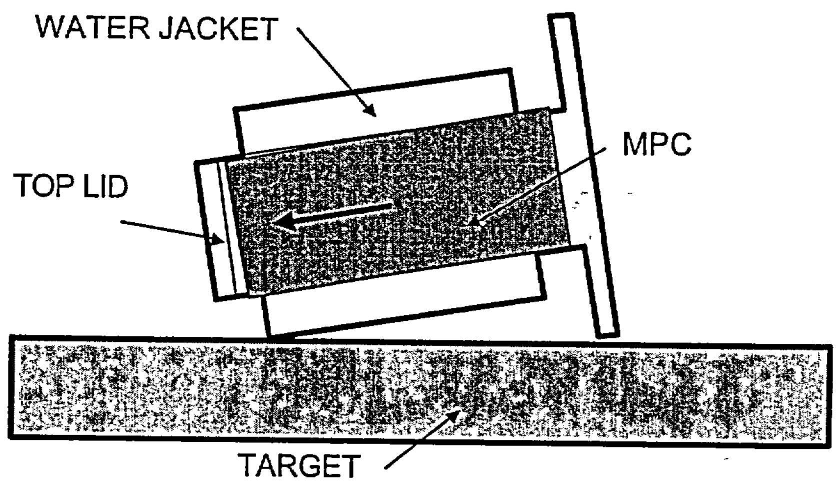
$Bolt\_Capacity\_in\_Tension := F_{tensile\_bolt}$

3.AH.6 CONTAINMENT OF THE MPC - Stud and Lid Evaluation

Appendix 3.AN contains results of the side drop of the HI-TRAC transfer cask from an initial orientation with the trunnions horizontal. This drop accident has been postulated as a bounding drop accident during handling of the HI-TRAC in a horizontal orientation. The results of the analysis have shown that the maximum interface longitudinal load that develops between the MPC and the HI-TRAC top lid is

$$Load := 132000 \cdot lbf$$

The interface load develops because their is a difference in the centrifugal accelerations values for transfer cask and the MPC that results in the MPC moving towards and impacting with the top lid. The sketch below describes the scenario:



The MPC/Top Lid interface force tends to stretch the studs and bend the lid. In the following section, we investigate:

1. The ability of the studs to resist the tensile interface load and the stud shear force due to the impact with the target.

2. The ability of the top lid (an annular plate) to resist the ring loading at the interface developed by the impact.

The safety factor on stud tensile load is

$$SF_{bolt\_tension} := \frac{nb \cdot Bolt\_Capacity\_in\_Tension}{Load} \qquad SF_{bolt\_tension} = 8.464$$

The total shear load that must be resisted by the bolts is

$$Load_{shear} := W \cdot G \qquad Load_{shear} = 123750\,lbf$$

$$SF_{shear} := \frac{nb \cdot \left(\tau_{bolt} \cdot A_t\right)}{Load_{shear}} \qquad SF_{shear} = 7.738$$

The interaction equation for combined shear and tension is

$$I := \left(\frac{1}{SF_{bolt\_tension}}\right)^2 + \left(\frac{1}{SF_{shear}}\right)^2 \qquad I = 0.031$$

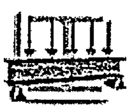
< 1.0 OK

It is clear that sufficient margin exists in the bolts to prevent lid separation even without consideration of any interface friction due to preload.

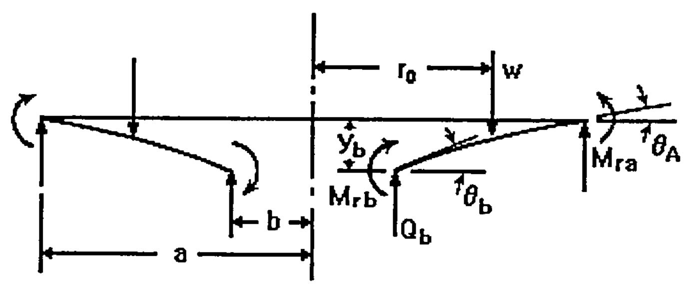
Lid Stress Evaluation

To evaluate the bending capacity of the lid, we assume a simply supported annular plate subject to the impact load from the MPC. The load is applied as a ring load at the location of the outer diameter of the MPC. This is appropriate since the top lid of the HI-TRAC is considerably more flexible in bending than the MPC lid. The appropriate plate solution is given in [3.AH.2.2, Table 24] and the calculations detailed in the Calculation Package for HI-STORM 100 (HI-981928). The summary of calculations is given below:

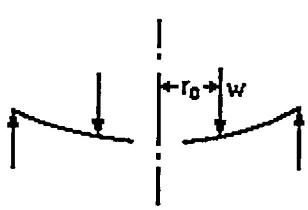
Table 24 Formulas for shear, moment and deflection of flat circular plates of constant thickness



Cases 1a - 1d Annular Plate With Uniform Annular Line Load w at Radius r_0; Outer Edge Simply Supported



Outer edge simply supported, inner edge free



For this analysis, only Case 1a is of interest

$a \equiv 39.375 \cdot in \qquad w := \frac{Load}{2 \cdot \pi \cdot a} \qquad w = 533.548 \frac{lbf}{in}$

**Enter dimensions,
properties and loading**

Plate dimensions:

thickness: $t \equiv 1.0 \cdot in$

outer radius: $a \equiv 39.375 \cdot in$

inner radius: $b \equiv 13.5 \cdot in$

Applied unit load: $w \equiv 533.548 \cdot \frac{lbf}{in}$

Modulus of elasticity: $E \equiv 28.5 \cdot 10^6 \cdot \frac{lbf}{in^2}$

Poisson's ratio: $\nu \equiv 0.3$

Radial location of applied load: $r_o \equiv 33.8125 \cdot in$

The following results are obtained from the detailed calculation following Roark's handbook:

Maximum lateral deflection at edge of opening = 0.908"

Maximum radial bending stress = 14,930 psi
(at point of application of impact load)

Maximum tangential bending stress (at edge of opening) = 35,560 psi.

The maximum stress intensity, away from the impact circle is equal to the maximum of either the radial or tangential stresses since both stresses always have the same sign. Therefore, based on Level D stress allowable for Subsection NF, (Table 3.1.12), The safety factor on the lid bending stress is

$$SF_{lid\_bending} := \frac{58700psi}{35560psi} \qquad SF_{lid\_bending} = 1.651$$

The allowable shear load around the periphery of the lid is computed as:

$$Q_{all} := (.42 \cdot 70000 \cdot psi) \cdot 1 \cdot in \qquad Q_{all} = 2.94 \times 10^{4} \frac{lbf}{in}$$

The safety factor on a failure due to peripheral shear is

$$SF_{shear} := \frac{Q_{all}}{w} \qquad SF_{shear} = 55.103$$

3.AH.7 CONCLUSIONS

The lid will not separate from the top flange of HI-TRAC due to the design basis deceleration.

The lid bolts are adequate to maintain the MPC inside of the HI-TRAC.

The top lid meets Level D allowable when subject to the impact load from a side drop that induces maximum slapdown angle.