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Appendix A.1

Attachment 1

Revision of Lid Bolt Analysis
Supplants the Analyses in TN-32 TSAR, Revision 9A, Section 3A.3
and TN-32 FSAR, Revision 0, Section 3A.3

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3A.3 Lid Bolt Analyses

The lid bolt analysis presented below is performed using the weight of the TN-32A lid (including top neutron shield) since it is slightly heavier than the standard TN-32 lid assembly (with top neutron shield). The use of either aluminum or silver metallic o-ring seals is discussed in the analysis below. Both are acceptable for use in a TN-32 cask lid system. O-ring parameters from either an aluminum or silver o-ring are used interchangeably in the following analyses, depending on which of the parameters from a particular o-ring provides the more limiting result.

3A.3.1 Normal Conditions

3A.3.1.1 Bolt Preload

The lid is secured to the cask body by forty-eight 1.5-in. diameter bolts. The selected bolt preload is such that the metallic confinement seals are properly compressed and the lid is seated against the flange with sufficient force to resist the maximum cavity internal pressure and any dead weight loads acting to unseat the lid. The maximum corresponding tensile preload stress in the bolts at temperature is 51,930 psi (corresponding to a maximum applied torque during cask loading of 1230 ft-lb. with lubrication) which is less than the stress allowable for the bolt material for Normal (Level A) Conditions. The minimum tensile preload stress in the bolts at temperature is 37,150 psi (corresponding to a minimum applied torque of 880 ft-lb. with lubrication) The load per bolt is:

Maximum: $F_B = A_B \times 51{,}930$
$= 1.492 \times 51{,}930 = 77{,}480$ lb./bolt

Minimum: $F_B = A_B \times 37{,}150$
$= 1.492 \times 39{,}260 = 55{,}430$ lb./bolt

Since we have 48 bolts, the maximum total seating force of all 48 bolts is 48 F_B = 3,719,000 lb. and the minimum total seating force is 2,661,000 lb.

The force required to seat the metallic O-rings (also referred to as seals), from Reference 1, is a line load of 1399 pounds per inch of seal circumference for aluminum seals, or 2284 pounds per inch of seal circumference for silver seals. The diameter of the outer seal is 72.65 in. and the diameter of the inner seal 71.05 in. The seal seating force is then:

$F_{seating} = 1399\ \pi(72.65 + 71.05) = 631{,}570$ lb. (aluminum)

XI0021

$F_{seating}$ = 2284 π(72.65 + 71.05) = 1,031,000 lb. (silver)

The maximum cask cavity internal pressure is the Design Pressure of 100 psi. The force required to react the pressure load (conservatively assuming the pressure is applied over the outer seal diameter) is:

$$F_{Pressure} = 100\left(\frac{\pi}{4}\right)(72.65)^2 = 414{,}500 \text{ lbs.}$$

The TN-32 cask is always oriented vertical during loading, during transfer to the ISFSI and during storage on the pad. Dead weight of the lid and cask contents does not actually load the lid bolts. In fact the lid weight (and external pressure) helps to seat the lid. However, it is conservative to require that the bolt preload maintain lid seating in any cask orientation. The weights of the lid, fuel and basket are:

Lid Assembly Weight	=	14,480 lb.
Fuel Weight	=	49,060 lb.
Basket Weight	=	16,900 lb.
W_{Total}		80,440 lb.

The total of the seal seating force (conservatively assuming silver seals are used), pressure load and dead weight loads is:

$F_{seating}$	=	1,031,000 lb.
$F_{pressure}$	=	414,500
W_{total}	=	80,440
		1,525,940 lb.

This force is less than the preload stresses achieved for the torque value range of 880 to 1230 ft-lbs. and shows that ample lid seating force is provided under Normal Conditions. Furthermore, the range of preload stresses is well below the limiting value of $2S_m$ (63,800 psi) for the bolt material at 300°F.

3A.3.1.2 <u>Differential Thermal Expansion</u>

The 48 lid bolts preload the outer rim of the closure lid against the cask body flange. The 1.5 in. diameter bolts are installed through 1.56 in. diameter clearance holes in the 4.50 in. thick lid periphery. Preloading of the bolts against the lid is accomplished by tightening the bolts so that the shank

portions of the bolts within the clearance holes are stretched elastically. The bolt loads will therefore change from the initial installed values if any thermal expansion differences should occur between the lid (through thickness direction) and the bolts.

The bolt material is SA 320 Grade L43 (1 3/4 Ni 3/4 Cr 1/4 Mo). The lid and body flange are both SA 350 Grade LF3 (3 1/2 Ni). The Section III Code Appendices specify the same coefficient of thermal expansion for these materials. The bolts are in intimate contact with the lid and flange and will therefore operate at the same temperature as these components. Therefore there will be no thermal expansion differences between the lid and bolts, and the assembly preload will be maintained under all temperatures.

3A.3.1.3 Bolt Torsion

The torque required to preload the bolt is:

$T = K\ D_N\ F_B$ (Reference 2)

Where

T_{max} = 0.127(1.5)(77,480)=14,760 in-lb. = 1,230 ft-lb.
T_{min} = 0.127(1.5)(55,430)=10,560 in-lb. = 880 ft-lb.
K = Nut Factor = 0.127 for N-5000 lubricant (as determined experimentally on a TN-32 cask)
A_B = Bolt stress area = 1.492 in.^{2}
D_N = Bolt nominal diameter = 1.5 in.
F_{Bmax} = 51,930 psi preload stress x A_B = 77,480 lb.
F_{Bmin} = 37,150 psi preload stress x A_B = 55,130 lb.

The maximum residual torque in the bolt corresponds to the maximum applied torque and is:

T_R = 0.5625T = 0.5625 x 14,760 = 8,303 in-lb.

The minimum residual torque in the bolt corresponds to the minimum applied torque and is:

T_R = 0.5625T = 0.5625 x 10,560 = 5,940 in-lb.

The shear stress in the bolt due to the residual torque from the maximum preload given by Reference 3:

$$\tau_{torsion} = \frac{T_R\ r}{J}$$

XI0023

where r and J are based on the bolt effective radius for the above stress area.

r = 0.689 in. effective bolt radius

J = torsional moment of inertia of threaded bolt

$$J = \frac{\pi\ r^4}{2} = 0.354 \ \text{inch}^4$$

$\tau_{torsion}$ = 8,303(0.689)/0.354 = 16,160 psi (Torsional Shear Maximum)

$\tau_{torsion}$ = 5,940(0.689)/0.354 = 11,560 psi (Torsional Shear Minimum)

3A.3.1.4 <u>Bolt Bending</u>

It is assumed that bolt bending does not occur during seating of the lid against the cask body during assembly. The bolts are rotated as they are torqued so any slight relative movements between lid and body flange during preloading will not result in a net offset between the bolt head and tapped flange holes. In addition, since the lid, flange and bolt materials have the same coefficient of thermal expansion and will operate at essentially the same temperature, differential expansion between components will not produce bolt bending.

As internal pressure is applied to the cask cavity, the lid will bulge slightly and its edge will rotate. In addition, the body cylinder radius will increase slightly due to the internal pressure resulting in outward radial movement of the tapped bolt holes in the body flange. Since no net membrane stress is developed in the lid, the lid bolt holes (at the mid surface) will remain at the original location. Rotation of the edge of the lid will, however, produce radial movement of the outer surface of the lid at the bolt head location.

The hoop stress in the cask body cylinder is:

$$S_{hoop} = \frac{P\ R_i}{t}$$

Where P = 100 psi Design Pressure
R_i = 34.375 in. inside radius
T = 9.5 in. thickness

$$S_{hoop} = \frac{100 \times 34.375}{9.5} = 361.8 \text{ psi}$$

The radial deflection at the bolt circle is:

$$\delta_{b.c.} = R_{bc} \times \frac{S_{hoop}}{E}$$

R_{bc} = 38.03 in. bolt circle radius

$$\delta_{b.c.} = \frac{38.03 \times 361.8}{28 \times 10^6} = 0.0004914 \text{ inches outward}$$

When pressure is applied to the lid, the edge rotation can be calculated assuming the lid is simply supported. From Reference 4, Table 24 case 10:

$$\theta = \frac{3W(1-\nu)R^3}{2\ E\ t^3}$$

Where θ = edge rotation, radians
W = total applied load = 100 psi
v = Poisson's ratio = 0.3
R = 36.35 in. outer seal radius
T = 4.5 in. lid thickness
E = Young's modulus = 28 x 10^6

$$\theta = \frac{3 \times 100 \times (1-0.3) \times (36.35)^3}{2 \times 28 \times 10^6 \times (4.5)^3} = 0.00198 \text{ radians}$$

Figure 3A.3-1 shows the net movement of the threaded hole and the Point on the lid under the bolt head.

If it is assumed that the bolt head doesn't slide on the lid surface, the head will be forced from position a to a' as the lid deflects. Point a' under the bolt head moves outward 0.00445 in. while the threaded hole moves only 0.0004914 in. outward. The bolt head will be bent laterally by 0.00445 - 0.0004914 in. or 0.00396 in. from the threaded end.

The bending model of the bolt is shown in Figure 3A.3-2. The moment on the bolt is calculated assuming the bolt is

subjected to affect bending with the head and threaded end prevented from rotating. For a cantilevered bolt free to rotate at the head, the bending moment would be reduced by one half. Therefore the assumption of fixed ends is the most conservative and results in the highest stress.

The shear force, P, and bending moment, M, for a beam subjected to offset bending with ends prevented from rotating are:

$$P = \frac{12EI\,\delta}{L^3}$$

$$M = \frac{6EI\,\delta}{L^2}$$

Where

P	=	lateral load to deflect the bolt distance δ, lb.
δ	=	lateral displacement
	=	0.00396 in.
E	=	Young's modulus, 28 x 10^6 psi @300°F
L	=	bolt length in bending
	=	4.625 in. (including tapped hole chamfer)
I	=	$\pi r^4/4$
	=	0.177 in.^{4} (r=0.689 in. based on stress area of 1.492 in.^{2})

Therefore

$$M_B = \frac{6 \times 28 \times 10^6 \times 0.177 \times 0.00396}{4.625^2}$$
$$= 5{,}500 \text{ in - lb.}$$

$$P = \frac{12 \times 28 \times 10^6 \times 0.177 \times 0.00396}{4.625^3}$$
$$= 2{,}380 \text{ lb.}$$

The bending stress in the bolt is

$$\sigma_b = \frac{Mr}{I} = \frac{5{,}500 \times .689}{0.177}$$
$$= 21{,}430 \text{ psi}$$

The shear stress due to the lateral force is

$\tau_p = P/A = 2,380/1.492 = 1,595$ psi

3A.3.1.5 Combined Stresses

The total shear stress for the maximum preload stress (51,930 psi corresponding to a torque of 1230 ft-lbs.)is then equal to the residual torsional shear stress plus that due to force P.

$$\tau_{total} = \tau_{torsion} + \tau_p$$

$$= 16,160 + 1,595$$
$$= 17,755 \text{ psi}$$

The total shear stress for the minimum preload (torque of 880 ft-lbs.) equals:

$$\tau_{total} = 11,560 + 1,595 = 13,155$$

The average tensile stress is the bolt preload stress:

$$\sigma_{average} = 51,930 \text{ psi}$$

The maximum tensile stress at two locations in the bolt is the preload stress plus the bending stress.

$$\sigma_{max} = 51,930 + 21,430 = 73,360 \text{ psi}$$

Therefore, the average combined stress intensity is:

$$SI_{average} = (\sigma_{average}^2 + 4\ (\tau_{total})^2)^{1/2}$$

$$= (51,930^2 + 4 \text{ x } 17,755^2)^{1/2}$$

$$= 62,910 \text{ psi (62.9 ksi)} < 2S_m = 63.8 \text{ ksi}$$

The maximum combined stress intensity is:

$$SI_{max.} = (\sigma_{max}^2 + 4\ (\tau_{total})^2)^{1/2}$$

$$= (73,360^2 + 4 \text{ x } 17,755^2)^{1/2}$$

$$= 81,500 \text{ psi} = 81.5 \text{ ksi} < 3S_m = 95.7 \text{ ksi}$$

For Level A conditions, the average bolt stress is limited to 2 S_m or 2 x 31.9 = 63.8 ksi. The maximum bolt stress is limited to 3 S_m or 95.7 ksi. The analyzed stresses are within

these limits as well as the yield strength of the bolt material (also 95.7 ksi).

3A.3.2 Accident Conditions

The lid bolts are analyzed in this section under the loadings selected to bound those for the hypothetical bottom end drop and tipover onto the concrete storage pad.

3A.3.2.1 Bottom End Drop

The bottom end drop from a height of 5 feet onto the concrete storage pad is analyzed in Section 3A.2.3.2. That section indicates that the cask deceleration may reach 42 g. This analysis conservatively examines the effects (if any) of a 50 g quasistatic loading on the lid bolts.

During a bottom end drop, the rim of the lid is forced against the flange of the cask body. The lid is initially seated against the flange by preloading (torquing) the bolts. The bolt preload will not be affected if compressive yielding of the contact bearing area does not occur.

The contact force on the bearing area, conservatively neglecting internal pressure, is the bolt preload force less the seal compression force plus the 50 g inertial force of the lid system. The maximum preload force, from Section 3A.3.1, is 3,719,000 lb. The seal seating force is 631,570 lb. for the aluminum seal and 1,031,000 for the silver seal. The weight of the lid system (weight of lid plus weight of top neutron shield assembly, 11560 + 2960 = 14,480 lbs., the highest weight among TN-32, TN-32A and TN-32B casks) is 14,480 lb.

Therefore, during a 50 g deceleration in the axial direction, and conservatively assuming an aluminum seal is used (i.e., an aluminum seal has a lower seating force compared to the silver seal thus resulting in a greater contact force), the contact force between lid and cask body is:

$$F_{contact} = F_{Bolt\ Preload} - F_{seal\ seating} + 50\ (W_{lid\ system})$$

$$= 3{,}719{,}000 - 631{,}570 + 50(14{,}480)$$

$$= 3{,}811{,}000 \text{ lb.}$$

Figure 3A.3-3 illustrates the bearing interface between lid edge and body flange. The bearing area equals the area within the diameter of the lid raised section (74.0 in.) less the outside of the body chamber (70.22 in.) less the area of the seal groove.

$$A_{Bearing} = \frac{\pi}{4}(74^2 - 72.95^2 + 70.75^2 - 70.22^2) = 180 \text{ in}^2$$

The bearing stress during impact is then equal to:

$$S_{bearing} = 3{,}811{,}000 / 180 = 21{,}170 \text{ psi } (21.2 \text{ ksi})$$

This contact stress is below the 33.2 ksi yield strength of the lid and flange material at 300°F. The bolt preload will not be affected by the bottom drop. Therefore, this hypothetical accident case will not affect the bolt stresses.

3A.3.2.2 Tipover

The tipover onto the concrete storage pad is analyzed in Appendix 3A.2.3.2 of Revision 9A of the TN-32 TSAR and Appendix 3D.2.5 of Revision 0 of the TN-32 FSAR. The tipover scenario is summarized in Figure 3A.3-4. The peak deceleration occurs at the top of the cask. The deceleration is much less at the center of gravity and essentially zero at the bottom corner pivot point. For this analysis the lateral deceleration at the lid end of the body is conservatively taken as 67g and that at the pivot point is zero.

There are two dynamic loadings acting on the lid tending to push or throw it off of the cask body (i.e. producing tensile forces in the lid bolts). There is a small axial (parallel to cask longitudinal axis) centrifugal inertia load due to the internals acting on the lid and the lid weight itself while the cask is rotating.

For the evaluation of the lid bolts it is assumed herein that the cask impacts on the corner at the lid end. There is no accident condition postulated that would cause greater load on the lid bolts. The cask orientation for the analysis is shown in Figure 3A.3-5. The axis of the cask is 30° from horizontal with the lid down. Note that this orientation is well beyond that predicted in the tipover analyses. If the lateral load, G_L, is 67g, then the axial load used is 67 x tan 30° or 38.7g.

The loads acting on the cask are shown in Figure 3A.3-5. The loads acting on the lid are shown in Figure 3A.3-6. Also shown is the reaction load at the cask interface and the pivot point, O, for analysis of lid rotation. Figure 3A.3-7 shows the lid bolt loads resisting rotation of the lid about pivot point O. The increase in bolt load beyond the preload varies uniformly from pivot point, O, to the bolt farthest from O.

XI0029

The moment acting on the lid about pivot point, O, due to the inertia load is calculated as follows:

$M_I = W_I \times 38.7 \times R_T + W_L \times 38.7 \times R_T + W_L \times 67 \times a + P_A \times R_T$

Where M_I = Total moment about pivot point, in-lb.

W_I = Weight of internals

= 16,900 + 49,060 = 65,960 lb.

and W_L = Weight of lid system (including shield plate resin disk), 14,480 lb.

P_A = Internal pressure load

= 414,500 lb.

R_T = Distance from center of lid to pivot point

= 39.75 in.

a = Moment arm of lid inertia load, W_L, in.

= 2.25 in. (very conservative since shield weight effect moves CG toward 0)

Therefore:

M_I = (65,960 x 38.7 x 39.75) + (14,480 x 38.7 x 39.75)

+ (14,480 x 67 x 2.25) + (414,500 x 39.75)

= 142.4×10^6 in-lb.

This moment is resisted by the effect of the preload on the lid bolts. The moment due to preload is calculated as follows:

$M_p = N \times F_B \times R_T$

Where

M_p = moment due to bolt preload, in-lb.
N = number of bolts, 48
F_B = preload per bolt
= A_B x preload stress
= stress area bolt x preload stress

= 1.492 in.^{2} x 51,930 psi = 77,480 lb. with 1230 ft-lbs. torque

XI0030

= 1.492 in.^{2} x 37,150 psi = 55,430 lb. with 880 ft-lbs. torque

R_T = distance from center of lid to pivot point, 39.75 in.

Therefore:

Torque = 1230 ft-lbs.:
M_p = 48 x 77,480 x 39.75
= 147.8 x 10^6 in-lb. > 142.4 x 10^6 in-lb.

Torque = 880 ft-lbs.:
M_p = 48 x 55,430 x 39.75
= 105.8 x 10^6 in-lb. < 142.4 x 10^6 in-lb.

Since the bolt preload moment, M_p, corresponding to the maximum torque value is higher than the moment due to inertial load, M_T, there will not be any additional load due to a tipover accident. However, the bolt preload moment corresponding to the minimum torque value is less than the moment due to inertial load. In this case, the inertial moment is partially resisted by the effect of the preload on the bolts. The increase in bolt load beyond the preload due to the inertia g loads created by the difference between M_I and M_p which is:

$M_T = M_I - M_p$
$M_T = 142.4 \times 10^6 - 105.8 \times 10^6$
$M_T = 36.6 \times 10^6$ in-lbs.

The lid bolts resist the above moment which tends to unseat the lid. The increase in lid bolt load (ΔF_{BO}) beyond the preload is proportional to the distance from the pivot point O. The additional resisting moment, M_R, applied by the bolts about point O is obtained from the following expressions. Refer to Figure 3A.3-8 of Revision 9A of the TN-32 TSAR for terminology.

$$\Delta \text{BoltForce} = \Delta F_{BO} \times \frac{[R + R\cos(n\Theta) + B]}{2R + B}$$

$$\Delta \text{Moment for a given bolt} = \Delta \text{Bolt Force} \times [R + R\cos(n\Theta) + B]$$

Therefore, the additional moment, M_R, can be expressed in general terms as follows:

XI0031

$$M_R = \frac{\Delta F_{BO}\left[(2R+B)^2 + B^2 + 2\sum_{n=1}^{n=23}(R + R\cos(n\Theta) + B)^2\right]}{(2R+B)}$$

R = 39.75 in.
B = 1.72 in.
Θ = 7.5°

Substituting yields:

M_R = 1421.7 x ΔF_{B0}

Therefore, ΔF_{B0} = 36.6 x 10^6 / 1421.7 = 25,740 lb.

The increase in tensile stress in the bolt beyond the preload is

ΔF_{B0} / A_B where:

A_B = bolt area, 1.492 in.^{2}

The increased stress is then:

ΔF_{B0} / A_B = 25,740 / 1.492 = 17,250 psi.

The total tensile stress in Bolt 0 is 37,150 + 17,250 = 54,440 psi. Therefore, the total tensile stress for a bolt subjected to cask tipover is greater for a bolt torqued to 880 ft-lbs. than for a bolt torqued to 1230 ft-lbs. The remainder of this tipover analysis will evaluate both the 880 ft-lbs. torque and the 1230 ft-lbs. torque scenarios.

When the tensile stress is combined with the bolt bending stress due to the lid edge rotation under internal pressure calculated in Section 3A.3.1, the maximum tensile plus bending stress is 54,450 psi plus 21,430 psi (bending) = 75,830 psi and the minimum is 51,930 psi + 21,430 psi = 73,360 psi. The total shear stress due to torquing and lid deformation from Section 3A.3.1.5 is 17,755 psi for the 1230 ft-lbs. case and 13,155 psi for 880 ft-lbs. The combined stress intensity is then:

The average combined stress intensity is:

$SI_{average}$ = $(\sigma_{average}{}^2 + 4(\tau_{total})^2)^{1/2}$

1230 ft-lbs. torque: $SI_{average}$ = $(51{,}930^2 + 4 \times 17{,}755^2)^{1/2}$

XI0032

$SI_{average}$ = 62,910 psi < S_y = 95,700 psi

880 ft-lbs. torque: $SI_{average} = (54{,}400^2 + 4 \times 13{,}155^2)^{1/2}$

$SI_{average}$ = 60,430 psi < S_y = 95,700 psi

The maximum combined stress intensity is:

$$SI_{max} = (\sigma_{max}^2 + 4(\tau_{total})^2)^{1/2}$$

1230 ft-lbs. torque: $SI_{max} = (73{,}360^2 + 4 \times 17{,}755^2)^{1/2}$

SI_{max} = 81,500 psi < S_u = 113,930 psi

880 ft-lbs. torque: $SI_{max} = (75{,}830^2 + 4 \times 13{,}155^2)^{1/2}$

SI_{max} = 80,260 psi < S_u = 113,930 psi

In addition to the above calculations, the lid bolts are evaluated based on the interaction formula from Appendix F of the ASME Code (Reference 5) for tension and shear:

$$\frac{(f_t)^2}{(F_{tb})^2} + \frac{(f_v)^2}{(F_{vb})^2} \leq 1$$

Where:

f_t and f_v are the applied tensile and shear stresses

F_{tb} = allowable tensile stress, smaller of $(0.7S_u)$ or S_y(TSAR Table 3.1-3)= 0.7 x 113,930 = 79,750 psi

F_{vb} = allowable shear stress, smaller of $(0.42S_u)$ or $0.6S_y$(SAR Table 3.1-3)= 0.42 x 113,930 = 47,850 psi

1230 ft-lbs. torque:

$$\frac{51{,}930^2}{79{,}750^2} + \frac{17{,}755^2}{47{,}850^2} = 0.56 < 1$$

880 ft-lbs. torque:

$$\frac{54{,}400^2}{79{,}750^2} + \frac{13{,}155^2}{47{,}850^2} = 0.54 < 1$$

3A.3.3 Conclusions

Based on the above evaluation, it is concluded that:

1. The maximum normal and accident condition stresses in the lid bolts are acceptable.

2. A positive (compressive) load is maintained on seals during normal and accident condition loads as bolt preload is higher than the applied loads.

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1. Resilient Metal Seals and Gaskets, Helicoflex Catalog H.001.002, Helicoflex Co., Boonton, N.J., 1983 pp.5-7.

2. NUREG/CR-6007, Stress Analysis of Closure Bolts for Shipping Casks, April 1992.

3. Hopper, A.G. and Thompson, G.V. "Stress in Preloaded Bolts," Product Engineering, 1964.

4. Roark, R.J.: "Formulas for Stress & Strain", 6th Edition, McGraw-Hill Book Co.

5. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section III, Appendix F, 1992.

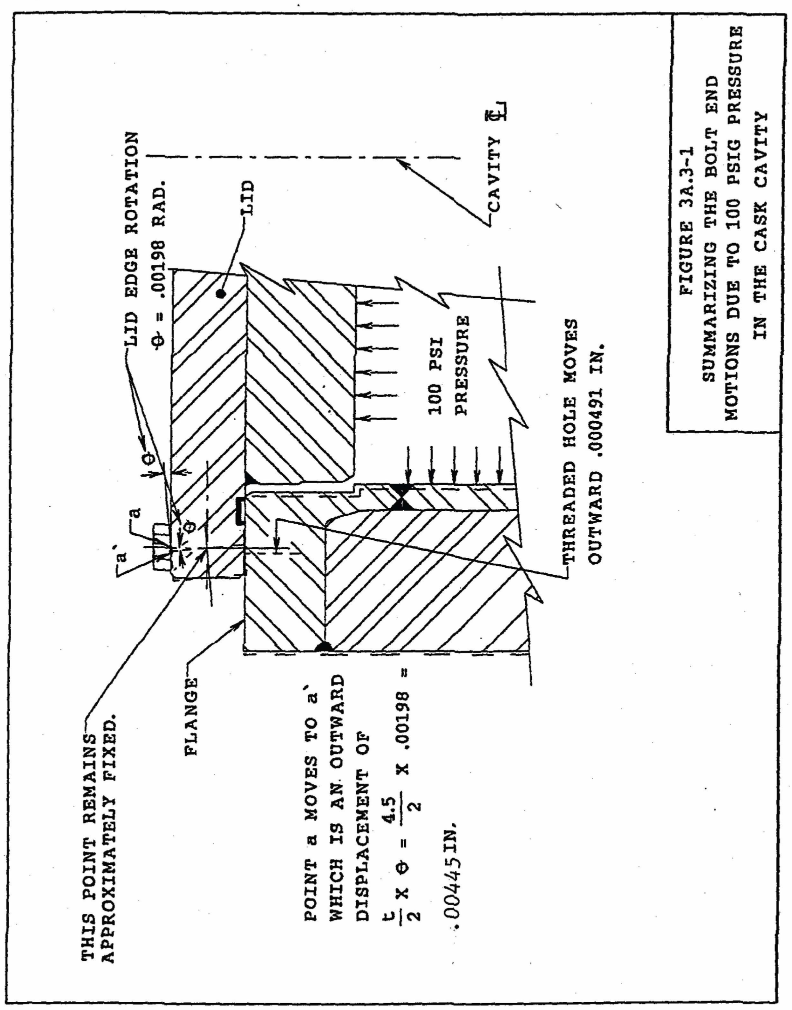


FIGURE 3A.3-1
SUMMARIZING THE BOLT END
MOTIONS DUE TO 100 PSIG PRESSURE
IN THE CASK CAVITY

REV. 9A 12/96

XI0036

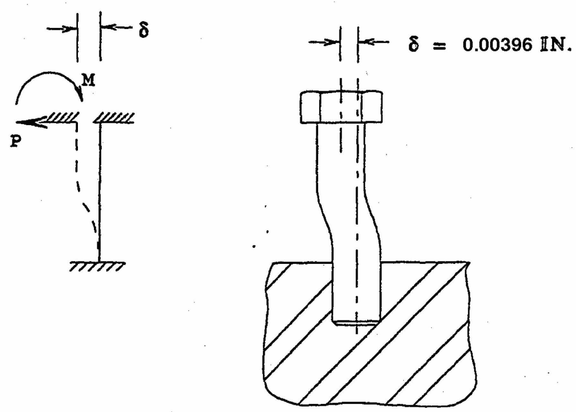
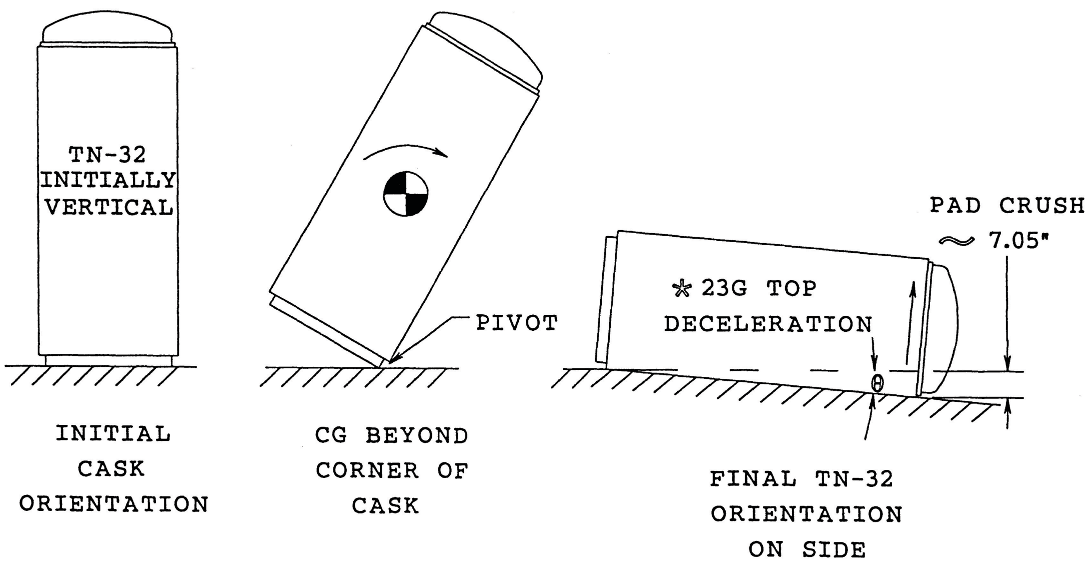


FIGURE 3A.3-2
LID BOLT BENDING DUE TO LID EDGE ROTATION UNDER INTERNAL PRESSURE

REV. 9A 12/96

XI0037

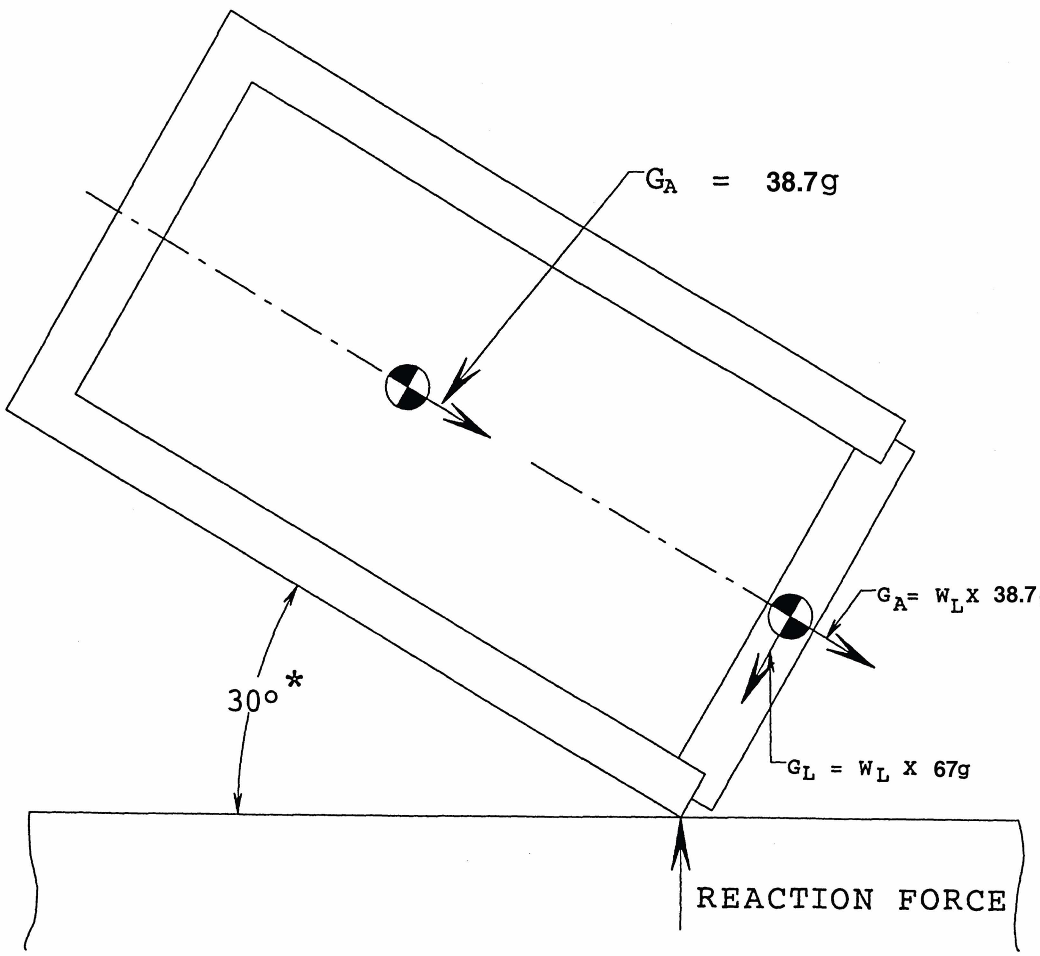


$$\theta = \sin^{-1}\frac{7.05"}{184} \approx 2.2^\circ$$

* 67g IS CONSERVATIVELY USED FOR LID BOLT ANALYSIS

FIGURE 3A.3-4
TIPOVER ONTO CONCRETE
STORAGE PAD

XI0014



* 30° WAS CONSERVATIVELY USED INSTEAD OF CALCULATED 2.2°

FIGURE 3A.3-5
SYSTEM OF INERTIA LOADS APPLIED ON CASK

XI0015

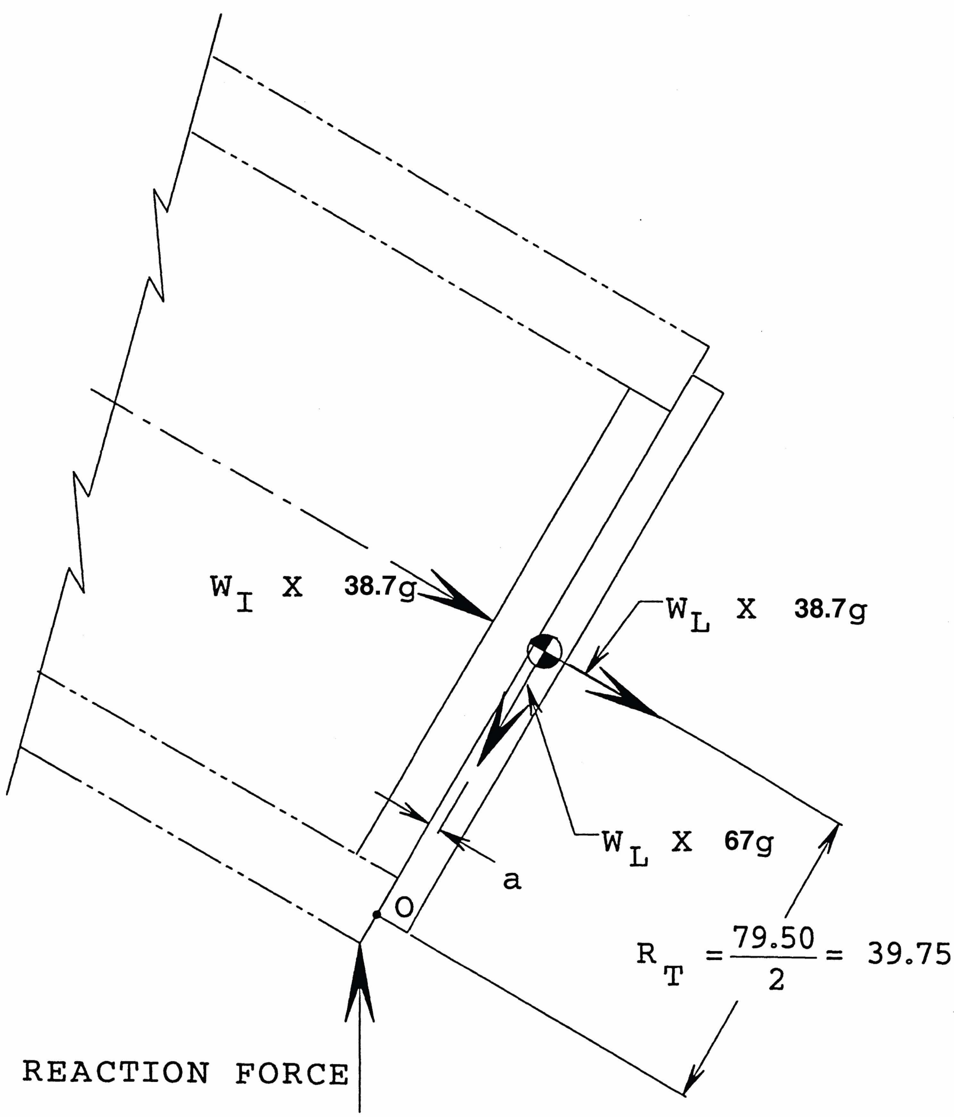


FIGURE 3A.3-6
INERTIA LOADS APPLIED ON LID

XI0016

Table 3.4-7
Summary of Maximum Stress Intensity and Allowable Stress Limits for Lid Bolts

STRESS CATEGORY	SERVICE CONDITION	MAXIMUM CALCULATED STRESS (psi)	ALLOWABLE STRESS (psi)	SAFETY FACTOR
Tensile	Level A	51,930	63,800 ($2S_m$)	0.23
	Level D	54,440	79,750 ($0.7\ S_u$)	0.46
Tensile + Bending	Level A	73,360	95,700 ($3\ S_m$)	0.30
	Level D	75,830	113,930 (S_u)	0.50
Shear	Level A	17,755	38,280 ($0.4\ S_y$)	1.16
	Level D	17,755	45,572 ($0.4\ S_u$)	1.57
Combined S.I.	Level A	62,910	95,700 ($3\ S_m$)	0.52
	Level D	81,500	113,930 (S_u)	0.40

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Appendix A.1

Attachment 2

Revision of Tornado Missile Analysis
Excerpt from TN-32 TSAR (Rev. 9A, 12/96)

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2.2.1.3.2 Missile B

Missile B (rigid) partially penetrates the cask wall. The loss in kinetic energy is dissipated as strain energy in the cask wall. The force F_b developed by an 8 in. diameter missile penetrating the cask body is calculated below. With a yield strength of 31,900 psi at 300°F, used for the gamma shield material:

$$F_b = S_y(\frac{\pi}{4})(8)^2 = 1.603 \times 10^6 \ lbs.$$

From conservation of energy:

$$F_b x = \frac{1}{2} m_b v_o^2$$

For a given puncture force F_b:

$$x = \frac{m_b v_o^2}{2 F_b}$$

where x is the penetration distance.

The penetration distance is found to be 1.10 in. for a perpendicular impact of the missile.

When the impact angle is not 90 degrees, the missile will rotate during the impact limiting the energy available for penetration (conservatively neglected), since part of the energy will be transformed into rotational kinetic energy.

When hitting the weather protective cover, Missile B deforms the dished head before penetration begins. This will decrease the penetration distance from the above value.

If the missile were to impact the top of the cask in the vertical orientation, the missile velocity would be 88.2 mph (70% of the horizontal impact velocity of 126 mph). The kinetic energy would be:

KE = 0.5 x (276 lbs/32.2ft/sec^2) x (88.2 mph x 5280ft/mi/3600 sec/hr)^2 x 12 in/ft
= 860,440 in lbs.

Ignoring the effect of the protective cover and the top neutron shield, the lid bending stresses under a top impact are evaluated. Reference 20 is used to evaluate the stresses for two boundary conditions:

1. Modeling the lid as a simply supported plate.
2. Modeling the lid as a plate with the edges fixed.

For edges simply supported, Table X, Case 2 of Reference 20 is used. The maximum stresses occur at the center, where the plate thickness is t = 10.5 in. The impact force, $F_b = 1.603 \times 10^6$ lbs.

The maximum stress at the center is calculated below:

$$S_r = S_t = 3W/(2\pi m t^2) \times [m + (m+1)\ln(a/r_o) - (m-1)r_o^2/4a^2]$$

Where r_o = uniform load radius = missile radius = 4 inches
$m = 3.33$
$t = 10.5$ inches
$a = 38.03$ inches(effective radius for a simply supported lid at the bolt circle)
$W = F_b = 1.603 \times 10^6$ lbs.

Therefore, $S_r = S_t \approx 28.5$ ksi

This is well below the Level D allowable stress of 63,000 psi.

For the second case, with the lid edges fixed, Table X, case 7 of Reference 20 is used.

The maximum stress occurs at the edge, where the plate thickness, t = 4.5 inches.

$$S_r = 3W/(2\pi t^2)\ (1 - r_o^2/2a^2)$$

Where $W = F_b = 1.603 \times 10^6$ lbs.
r_o = uniform load radius = missile radius = 4 inches
$t = 4.5$ inches
$a = 38.03$ inches

$S_r \approx 37.0$ ksi.

This is also well below the allowable Level D stress of 63,000 psi.

2.2.1.3.3 Missile C (steel sphere 1" diameter)

The impact of the steel sphere can result in a local dent by penetrating into the cask surface at the yield strength, S_y, for a penetration depth, d. The contact area on the cask surface is:

$$A = \pi(2Rd - d^2)$$

Where:

R is the radius of the sphere, 0.5 inches,
and d is the penetration depth

The kinetic energy of the steel sphere is dissipated by displacing the cask surface material:

$$KE = \tfrac{1}{2}(m_c v_o^2) = S_y \int_o^d \pi(2Rd-d^2)\partial_d$$

Where m_c = sphere mass

$KE = 0.5(4/3)(\pi)(0.5)^3(0.28)(1/32.2)(126 \times 5280/3600)^2 = 933$ in-lbs

$S_y \int_o^d \pi(2Rd-d^2)\partial_d = S_y\pi(Rd^2 - d^3/3) = KE = 933$ in-lbs

For a yield strength of 31,900 psi, by trial and error:

d = 0.14 in.

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The area, A, is therefore 0.38 sq. inches. A maximum impact force of 12.1×10^3 lb. ($A \times S_y$) will be developed. It can be concluded that only local denting of the cask will result.

If the impact is at the top of the cask (ignoring the protective cover and the neutron shielding), Reference 20, Table X, Case 4 is used to determine the stresses. The impact force is assumed to act at the center of the lid, where p = 0, r_o = 0.353 in. and a = 38.03 inches.

The maximum stress is:

$$S_r = S_t = 3W/(2\pi mt^2) \times [m + (m+1)\ln(a/r_o) - (m-1)r_o^2/4a^2] \approx 1.1 \text{ ksi.}$$

Since all penetrations are covered, the steel sphere will have a negligible effect on the cask.

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