1.6 VIBRATION - 10 CFR 71. APP. A (4)

The cask is mounted on supports which are structurally part of the transporting trailer. Thus, vibration of the cask itself is considered to be that of a long, essentially uniform beam, simply supported near the ends.

cask weight, W	48,000 LB.
cask bending stiffness, EI	295.54×10^9 LB-in^2
span between supports, ℓ	175 in.

The cask bending stiffness given does not include the water jacket shell (see section 2.3). The total cask weight is also used in determining vibration frequency, although part of the cask extends beyond the supports. thus, the calculated vibration frequency will tend to be less than the actual frequency. This is conservative, since the calculated frequency will be closer to vibration frequencies of the transport vehicle than is actually the case.

Lowest vibration frequency of the cask in a bending mode is

$$f_m = \frac{9.87}{2\pi}\sqrt{\frac{EIg}{W\ell^3}}\ Hz \qquad \text{Ref. 3}$$

$$f_m = \frac{9.87}{2\pi}\sqrt{\frac{(295.54\ (10^9)\ (386)}{48{,}000\ (175)^3}} = 33.08\ Hz$$

This natural frequency of the cask in bending is nearly optimum for truck transport, since it is well above the low frequency range of truck suspension systems (1-20 Hz) and also below the usual range of body structural vibrations (50-150) (see Ref. 3). High frequency vibrations of the truck body are not significent anyway, for they are readily absorbed by friction in the cask supports. Therefore, the cask will withstand vibrations normally incident to transport as required in 10 CFR 71, App. A(4).

1.7 TIE-DOWN DEVICES

The cask in its final position on the trailer is supported and anchored at both ends, resting as a simple beam between the front and the rear trailer supports. These supports are welded to the trailer and are not considered to form part of the design package of the cask itself. Each support is a major transverse bulkhead for the trailer. Particular attention has been given to the type and position of welds made to the heavily loaded main frame members of the trailer in order to prevent the development of service weld cracks.

The front support is a special saddle type configuration, extending for 120° under the cask and engaging it by a curved bar whose tapered edges fit into a circumferential groove around the cask, similar to the fit of a short length of V belting into its mating pulley configuraton. By this means the entire fore and aft design loads of the cask are taken on the engaged sloping faces of the saddle. The cask is bolted into the ends of the saddle by bolts aligned tangentially, thus providing a very heavy hoop force with the result that the cask is firmly held into the saddle by a preload greater than the vertical and lateral applied forces.

The rear support, which is relieved of fore and aft loadings, is designed to take the vertical loads of transit and of tilting to a vertical cask position for lift-off by crane, as well as lateral loads of transit.

Trunnion pins on the rear support structure are provided with square blocks, of dissimilar bearing material, which both rotate with the cask on the trunnion pins in lift-off, and slide in slots machined into the bottom of the cask to accomodate for relative motions caused by therm expansion or transit deflections.

As the cask is lowered vertically onto the rear trunnion pins it is guided, finally, in two directions by sloping faces on both the blocks and the cask recesses, with the weight of the cask coming to rest upon the square block. After tilting down to the horizontal, a gap must be established at this interface to allow for the necessary axial movements mentioned. This gap is created during the last few inches of the let down by the engagement of the tapered mating surfaces on the front supporting saddle, thus literally causing the cask to pull itself forward before finally sinking into complete annular lock.

In lowering the cask, as it pivots on the lower and rear trunnion pins, the first contact at the front support is at the two ends of the 120° arcuate saddle. Here the geometry of the sections reduces the nominal 30° slope to an effective 17°, approximately, thus providing an elongated contact at an increased mechanical advantage to pull the cask forward in the operation of final alignment of the cask groove over the saddle contour.

<u>1.7.1 DESIGN SPECIFICATIONS</u> require simultaneous loadings.

The several components of forces applied at the C.G. of the cask are as follows.

a) ± 10g longitudinal = 480000 lbs. - applied at front end only

b) ± 5g transversely = 240000 lbs. - applied front and rear

c) ± 2g vertically = 96000 lbs. - applied front and rear

In addition there are to be considered

d) +1g vertically = 48000 lbs., static wt. - applied front and rear horizontally or only at rear when being hoisted off vertically.

e) preload clamping against front saddle, effective as a multiple g weight force.

These loadings, in various combinations, apply at the interfaces between cask and trailer supports. The stress analysis of the trailer supports themselves is not part of the cask design package.

1.7.2 FRONT TIE-DOWN

1.7.2.1 SADDLE PRE-LOADING

The cask is bolted down upon the saddle by two (2) high strength socket head cap screws, $1\frac{3}{8}-6$, inclined at an angle of 30° to the vertical center line. The resultant of these two single bolts is a vertical force so great that the application of design loads vertically or transversly does not increase their stresses, but only varies the distribution of local contact stresses between cask and support.

$1\frac{3}{8}$-6 threads have effective tension area = 1.155 in.²

Resultant vertical force on cask, downward, is

$F = (2)(1.155\,in^2)(\cos 30°)(20980\,psi\ \text{torqueing stress})$

$= 2 \times 1.155 \times .866 \times 20980 = \underline{41970\ \text{lbs. preload}}$

Tension force on each bolt is

$T = 1.155 \times 20980 = \underline{24232\ lbs}$

Bolting bar welded into groove of cask, Figure 1.
Total load = 24232 lbs.
Maximum bending moment is approx.(conservatively)

$$M = \frac{24232}{2} \times 2 = \underline{24232 \text{ in.lbs.}}$$

Z of 2.5" deep bar, 3" wide with $1\frac{3}{4}$" ϕ hole

$$Z = \frac{1.25 \times 2.5^2}{6} = \underline{1.3 \text{ in.}^3}$$

$$\sigma_b = \frac{24232}{1.3} = \underline{18640 \text{ psi}} \qquad M.S. = \frac{30000}{18640} - 1 = \underline{.609}$$

Welding in shear - linear length of weld 2 × 9" = 18"

$$\frac{24232 \text{ lbs}}{18 \text{ in.}} = \underline{1346 \text{ lbs.}/\text{in.}}$$

Use minimum of $\frac{1}{2}$" penetration welds

$$\frac{1346}{1/2} = \underline{2692 \text{ psi}} \qquad M.S. = \frac{.6 \times 30000}{2692} - 1 = \underline{5.68}$$

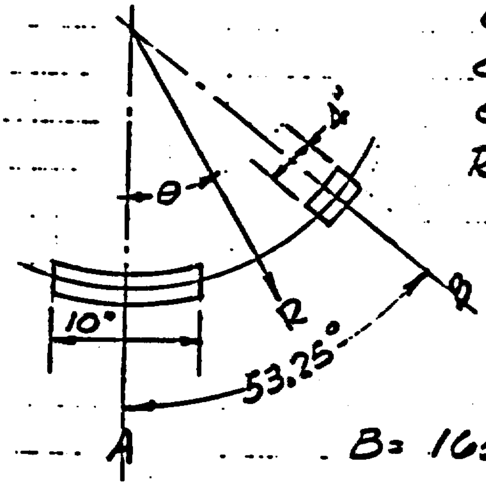
1.7.2.2 COMBINED LOADINGS ON SADDLE/CASK

Condition 1

b = 120000 lbs ←
c = 48000 lbs ↑
d = 24000 lbs ↑
e = 41970 lbs ↑

} 113970 lbs. ↓

Resultant load = 165497 lbs.



$$\theta = \text{arc tan}\ \frac{120000}{113970} = \text{arc tan}\ 1.0529$$

$$= 46.48° \text{ from vertical}$$

$$B = 165497 \left(\frac{46.48}{53.25}\right) = 71793 \text{ lbs.}$$

$$A = 165497 - B = 93704 \text{ lbs}$$

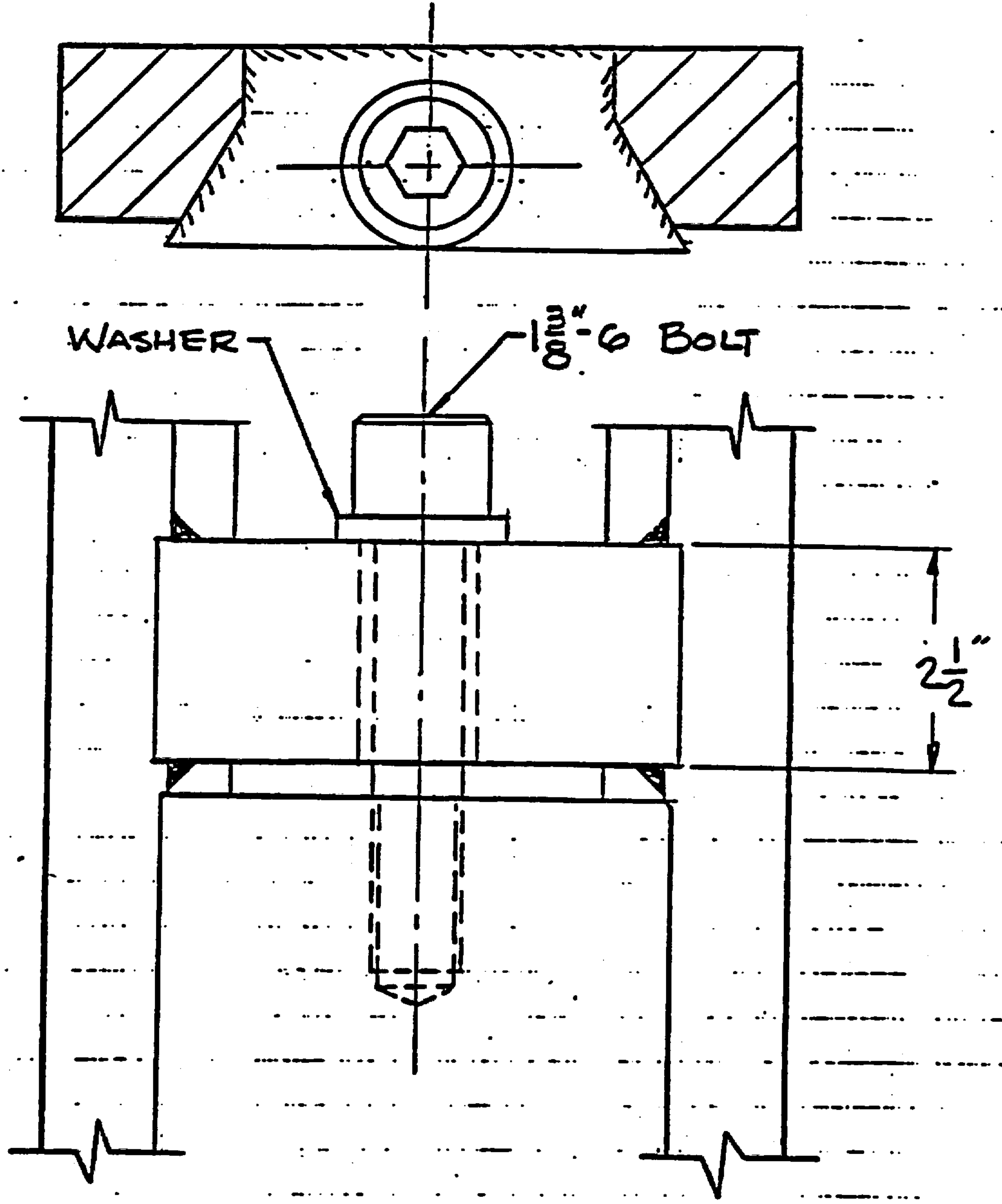
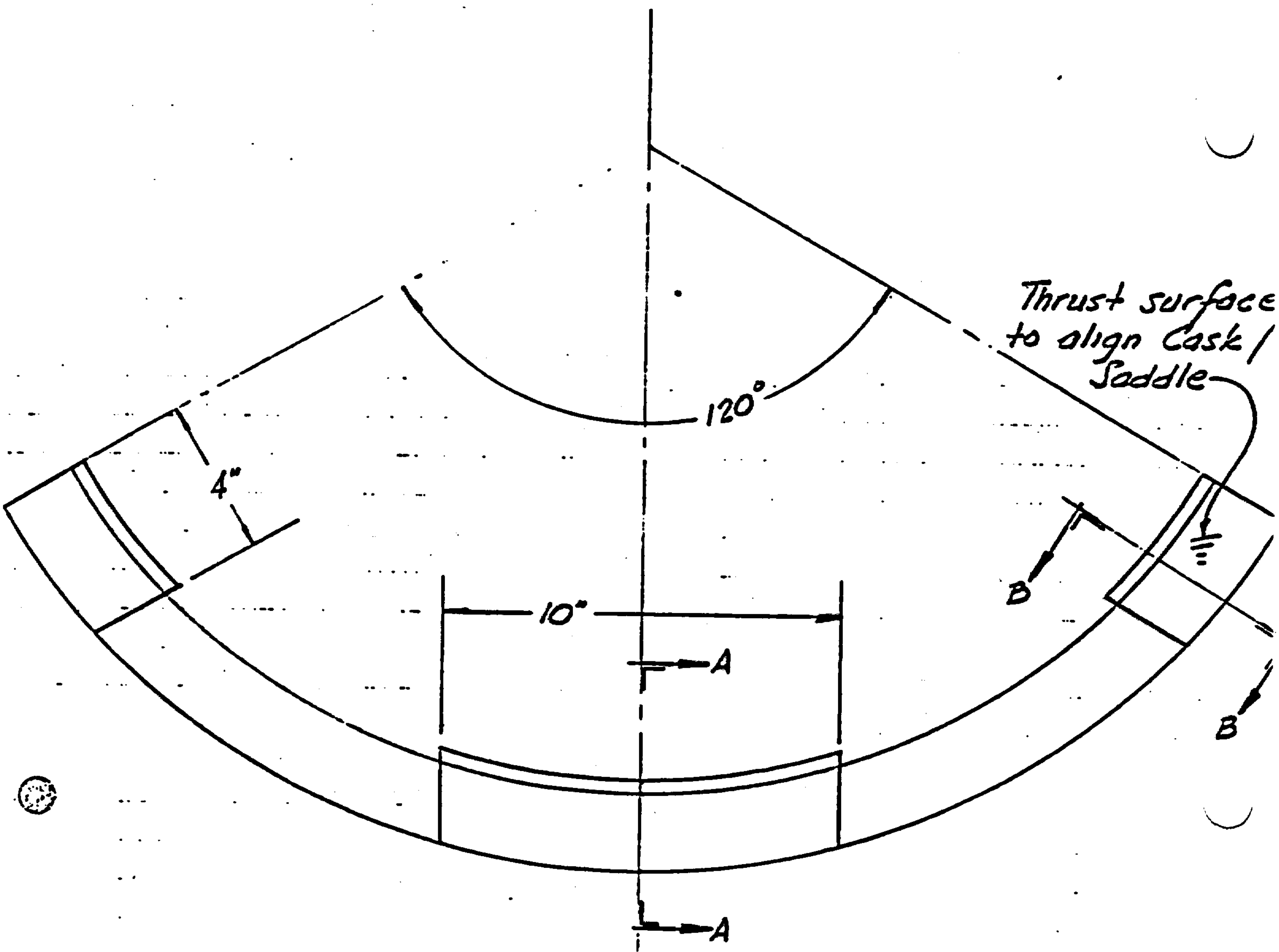
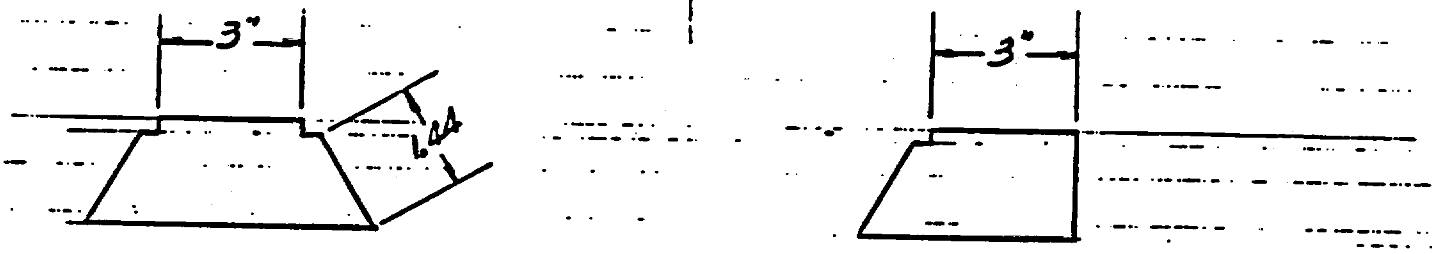


FIGURE 1



SADDLE



SECTION A-A

SECTION B-B

FIGURE 2

Allowable stress in bearing = 1.5 x 30000 = 45000

Bearing stress on "A" $\sigma_{BR} = \frac{93704}{10 \times 3} = \underline{3123 psi}$

Bearing stress on "B" $\sigma_{BR} = \frac{71793}{4 \times 3} = \underline{5983} psi$

$M.S. = \frac{45000}{5983} - 1 = 6.52$

__Condition 2__

b = 120000 lbs. $\leftarrow$

c = 48000 lbs. $\downarrow$

d = 24000 lbs. $\downarrow$

e = 41970 lbs. $\downarrow$

$\left.\right\}$ 17970 lbs. $\downarrow$

Resultant load = 121338 lbs.

$\theta = \text{arc tan} \frac{120000}{17970} = \text{arc tan } 6.678 = 81.48°$

Load at B = $121338\left(\frac{81.48}{53.25}\right) = 185664$ lbs.

Bearing stress on "B" $\sigma_{BR} = \frac{185664}{4 \times 3} = \underline{15474} psi$

$M.S. = \frac{45000}{15474} - 1 = 1.908$

__Condition 3__ includes 10g forward

a = 480000 lbs against one whole 60° face;
b, c, d, and e same as for condition 1

Normal force one face from $a = \frac{480000}{.866} = 555000$ l

Length of saddle at "A" = 10"

Bearing stress from "a" only

$$\sigma_{BR} = \frac{555000}{10 \times 1.44} = \underline{38{,}542 \text{ psi}}$$

$$M.S. = \frac{45000}{38542} - 1 = \underline{.167}$$

<u>Condition 4</u> Local thrust at ends of saddle, pulling the cask forward 1/4" over last drop of .91" (slope = .275). Area in contact initially is, for both corners, 2(3"×2") = 12 in.^{2}

$$\text{Thrust horizontally} = \frac{48000/2}{.275} = \underline{87000 \text{ lbs. max}}$$
(no friction)

Friction at rear support assumed
(.3) 24000 = 7200 lbs. horizontal

Friction at front support is approximately the same, but vertical, thus reducing the effective weight to 48000/2 − 7200 = 16800 lbs. and

$$\text{forward thrust to } \frac{16800}{.275} = \underline{61000 \text{ lbs. max.} > 7200 \text{ lbs}}$$

Crushing stress on contact areas at thrust of 7200 lbs. is

$$\sigma_c = \frac{7200}{12} = \underline{600 \text{ psi}}$$

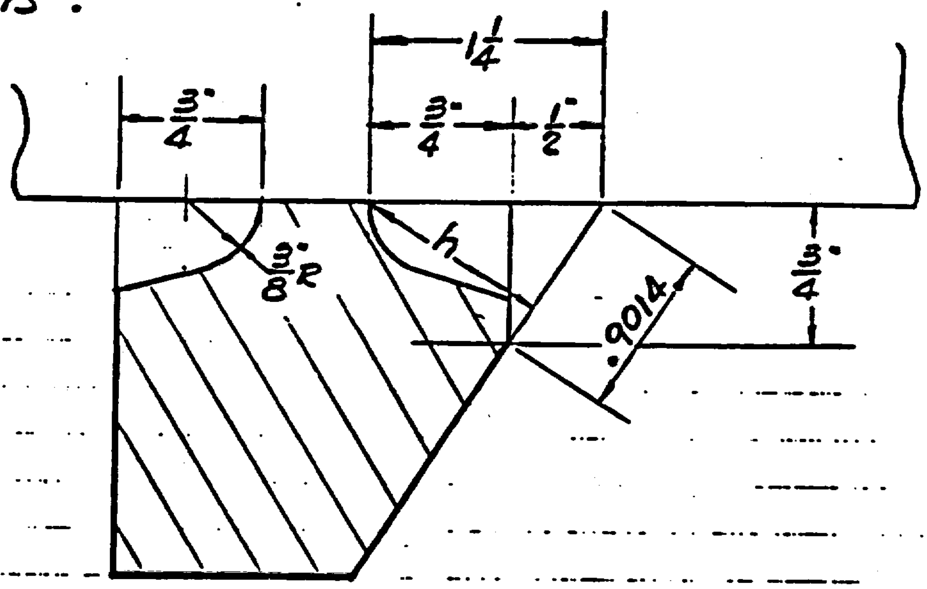
This stress is low enough to be immune to galling wear after a few initial "bedding in" operations.

It may be possible to supply a slight local lubricant or, if necessary, to hard face one of the contacting surfaces, preferably the trailer end areas.

1.7.2.3 Weld Strength of Taper Rings on Cask

The entire ± 10g longitudinal force, except for the frictional restraint at the rear support, is taken at the front support by either of the two taper rings welded to the cask by double 3/4" "J" welds. An additional 3/4" x 1/2" fillet weld is added for a distance of 15 inches circumferentially at the loading area:

The effective weld area over this minimum length is:



$$\text{Effective throat "}h\text{"} = \frac{.75 \times 1.25}{\sqrt{.75^2 + .5^2}} = 1.04"$$ Ref. 1 Pg. 4-1.

Total shear width = h + .75 = 1.79"

Total shear area = 1.79 × 15 = 26.85 in.^{2}

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The load is 480000 lbs - 7200 lbs. rear friction = 472800 lbs.

$$\text{Weld stress} = \frac{472800}{26.85} = \underline{17609 \text{ psi}}$$

$$M.S. = \frac{.6 \times 30000}{17609} - 1 = \underline{.0222}$$

1.7.3 REAR TIE-DOWN

1.7.3.1 LOADING CONDITIONS (Refer to 1.7.1)

a) does not apply
b) 120000 lbs $\rightarrow$
c) 48000 lbs $\updownarrow$
d) 24000 lbs $\updownarrow$
e) does not apply
f) 3×48000 lbs = 144000 lbs. A special case alone when cask is vertical on rear support for lift-off.

Lateral load (b) = 120000 lbs. is taken on one side of the cask extended weldment surrounding the cutout for the block. Eliminating the tapered guide area, the net area effective is

$$A = 2(5.25 \times 1.375) = 14.4 \text{ in}^2$$

$$C_c = \frac{120000}{14.4} = \underline{8350 \text{ psi}} \quad M.S. = \frac{30000}{8350} - 1 = \underline{2.5}$$

Effective area of single face of block - (for any direction of loading)

$$A = 4.25 \times 2 = 8.5 \text{ in}^2 \text{ each block}$$

Maximum load on single face of block is
(f) = 144000 lbs.

$$\sigma_c = \frac{144000}{2 \times 8.5} = \underline{8500 \text{ psi} \quad \text{static - no sliding}}$$

$$M.S. = \frac{30000}{8500} - 1 = \underline{2.53}$$

Sliding load on block in transit etc.

$$\sigma_b = \frac{24000}{2 \times 8.5} = \underline{1410 \text{ psi}}$$

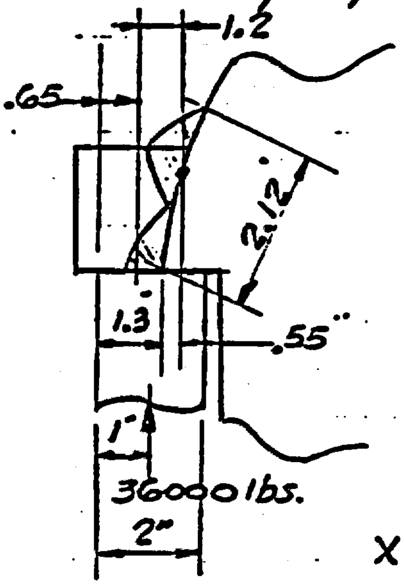
Block is much harder material than cask and so will prevent galling at such loading.

Stress analysis of socket weldment on cask for rear support.

<u>A.</u> Load on upper flange with cask in horizontal position = $\frac{3g}{2}$ Normal transport condition.

Load from trunnion block = $\frac{1.3}{2.0} \times \frac{72000}{2} = 23400$ l

Eccentricity of load to center of weld on cask=1.



Moment on weld:
M = 23400 × 1.2 = 28080 in lbs.
Length of weld = 2.12"
Width of weld = 7.25"

$$Z_{weld} = \frac{7.25 \times 2.12^2}{6} = 5.43 \text{ in}^3$$

Bending $S_b = \frac{28080}{5.43} = \underline{5171\ psi}$

Weld area = $7.25 \times 2.12 = 15.37\ in^2$

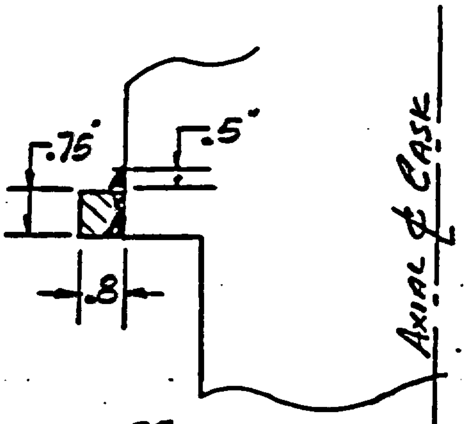
Shear $S_s = \frac{23400}{15.37} = \underline{1522\ psi}$

Combined $S_t = \frac{5171}{2} + \sqrt{2586^2 + 1522^2} = \underline{5586\ psi}$

$M.S. = \frac{30000}{5586} - 1 = \underline{4.37}$

B. Load on circumferential flange with cask in vertical position - Condition of loading cask onto trailer.

Load from trunnion block = $\frac{.8''}{2} \times 72000 = 28800$ l
Eccentricity of load to center of weld on cask = .4"



Moment on weld:
$M = 28800 \times .4 = 11520$ in.l

Thickness of plate at weld (less than weld) = .75"
Width of weld, conservative, neglecting end reinforcement = 6.25"

Bending $S_b = \frac{11520}{.586} = \underline{19659\ psi}$

$Z_{weld} = \frac{6.25 \times .75^2}{6} = .586\ in^3$

Shear area = $6.25 \times .75 = 4.69\ in^2$, neglecting end supports

$S_s = \frac{28800}{4.69} = \underline{6141\ psi}$

Combined $S_t = \frac{19659}{2} + \sqrt{9830^2 + 6141^2} = \underline{21420\ psi}$

$M.S. = \frac{30000}{21420} - 1 = \underline{.4}$

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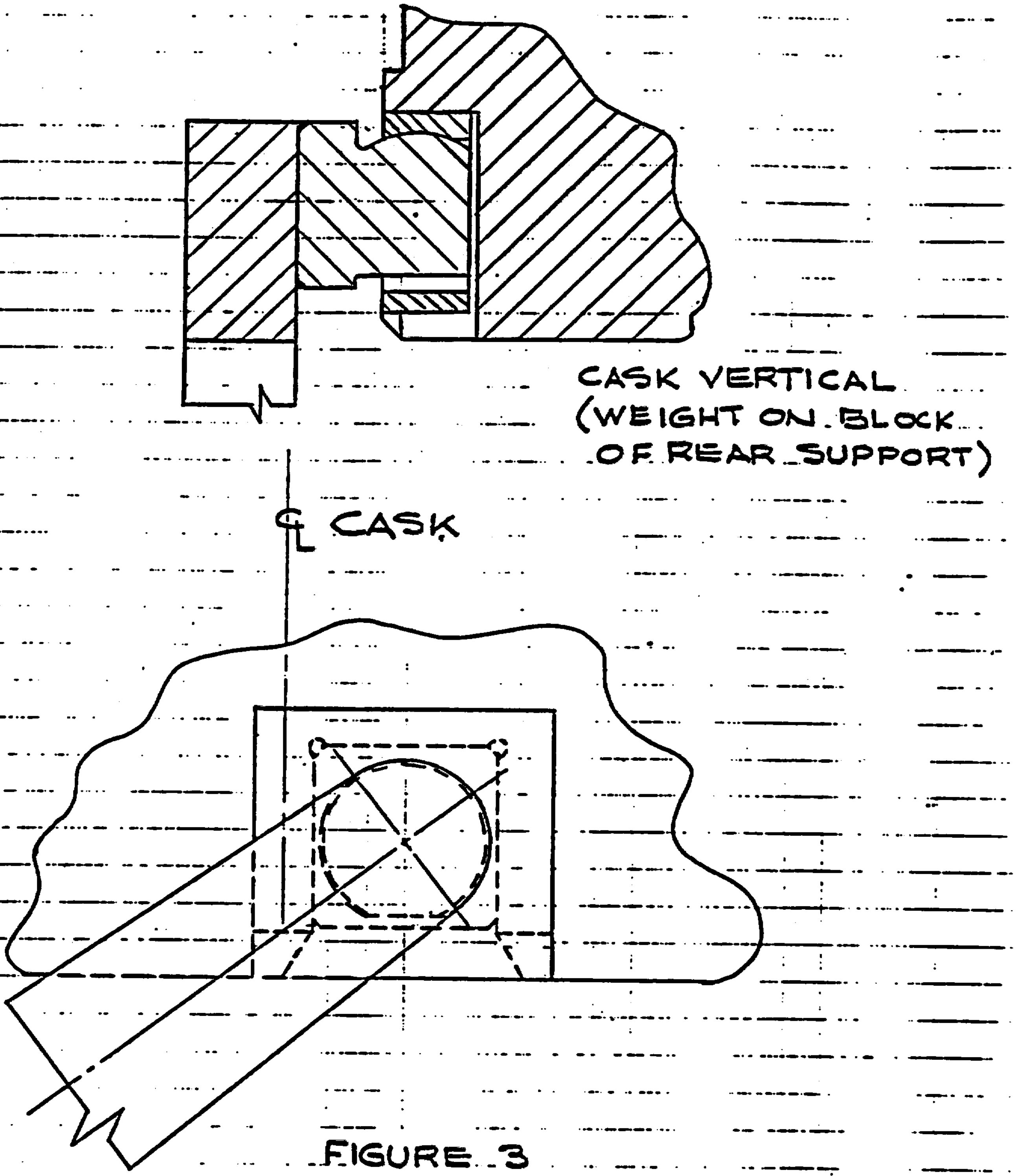


FIGURE 3

1.7.4 Longitudinal Bracing

The tie-down structure supporting the cask on the trailer bed consis of a transverse structure near each end of the cask, together with a sloping member for each structure to provide longitudinal bracing. There are two longitudinal rails which interconnect the four members of the support structures.

This structure, with the cask in place, can be moved and adjusted in position relative to the trailer chassis so that a precise correction for C.G. placement can be secured. It is then welded to the top flanges of the trailer at specific locations and with a definite weld area and strength. By this procedure the welds between the structure and the trailer frame are so designed that in the event of a high "g" longitudinal impact these welds will shear and tear away, allowing the structure to be intact and maintain its relationship to the cask as they both leave the trailer.

The cask has its integral support detail members designed for 10 g longitudinal with stresses less than material y.p. The individual front and rear support members are designed for 5 g longitudinal on their y.p. In both cases, there is a margin of safety on the y.p. and the further ability to go to an ultimate stress limit if developed under accident conditions.

It is reasonable, therefore, to design the welds to the trailer on the basis of 6 g ultimate shear stress, to insure progressive failure in longitudinal impact. Since the trailer and cask will overturn at less than .6 g, a lateral strength of 2 g ultimate is assigned to these welds Welds to trailer are made at Station A,B,C,D – Fig. 4. Stations A and D support vertical g loadings in compression and resist lateral 1 g in welds. Station B trasmits the entire longitudinal load to the frame in its welds. Station C, being the opposite end of a beam on which the cask is initially lowered and tilted down, must be welded to the frame for stability.

All welds are 60,000 psi ultimate tensile strength.

At each of the four locations for A and D on the two rails, the lateral 2 g loading required 1-1/2 inches of 3/8 fillet weld.

$$\frac{(48000)\ \text{lbs.} \times 2}{4 \times 1.5 \times 3/8 \times .7} = 60{,}000\ \text{psi}$$

At B, the resultant inclined load is taken in shear at each side rail by 9" of 3/8" fillet weld, both inboard and outboard of the rail

$$2xx\ \ 9 \times 3/8 \times .707 \times .6 \times 60000 = 171800\ \text{lbs.}$$

$$g's = \frac{171800}{24000} = 7.15\ g\ \text{shear}$$

At C, the resultant load is essentially verticall, being a weld tension of 24,000 lbs. for each side rail.

This requires only 1-1/2 inches of 3/8 fillet weld minimum to stabilize the total cask weight when vertical on the rear support.

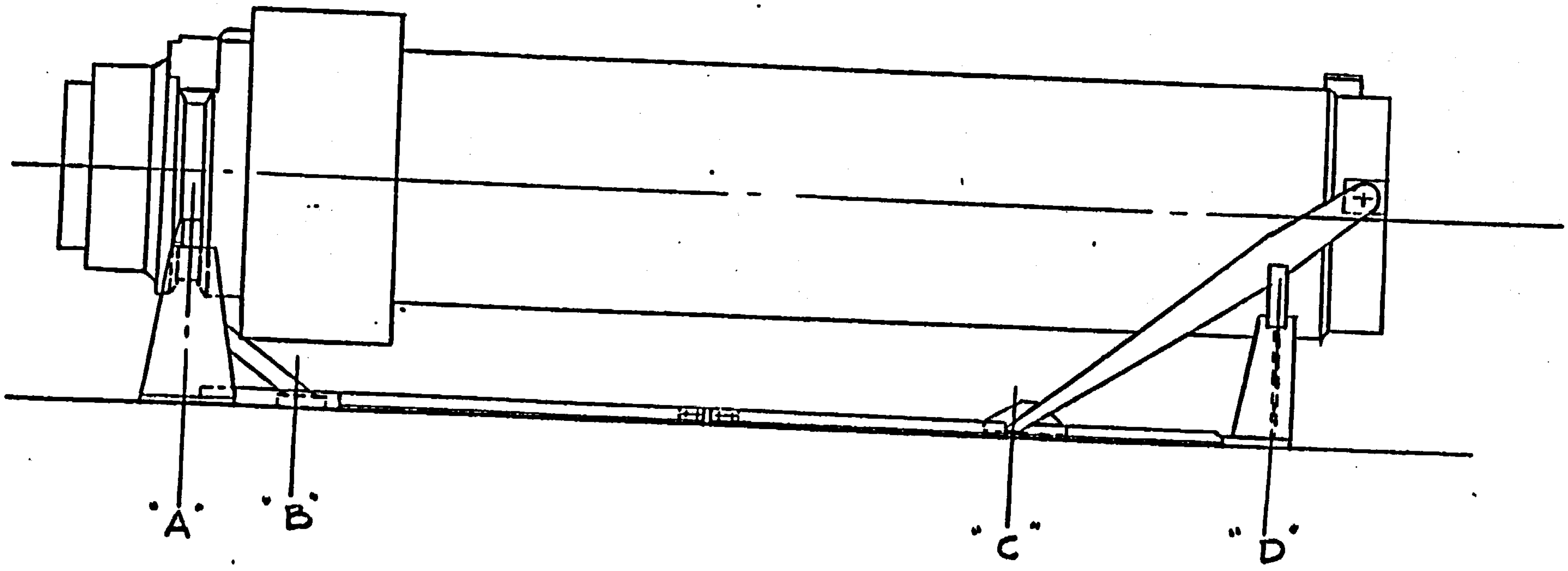


FIGURE 4

This is increased to give a reasonable margin of safety, since it will not interfere with allowing breakaway at B under a critical total load of 171800 lbs. and a load at A of 14400 lbs.

The bending of the trailer frame under 1 g weight, with perhaps + 1g additional due to transport vibration, would tend to shear the welds at B and C unless a sliding connection is provided near the center of the trailer length. This allows independent anchorages of the two supports to the trailer, yet relieving them of beam flexure shear loads, and insuring proper sequence of weld releases under accident impact at substantially 7 g. See drawing 70514F, Sheets 1 and 2.

For tension load along the 3 x 2 x 3/8 angle, these welds would be in shear. Such a tension load would be exerted after the welds at B had been broken loose in an accident impact. The shear load thus developed by the welds on one side rail is

@ C. 6 x 5/16 x .707 x .6 x 60,000 = 47723 lbs.

@ D 1½ x 3/8 x .707 x .6 x 60,000 = 14317 lbs.

Total 62040

The angle strength is 1.73 in.^{2} x 60,000 = 103,800 lbs. O.K. 62040 lb

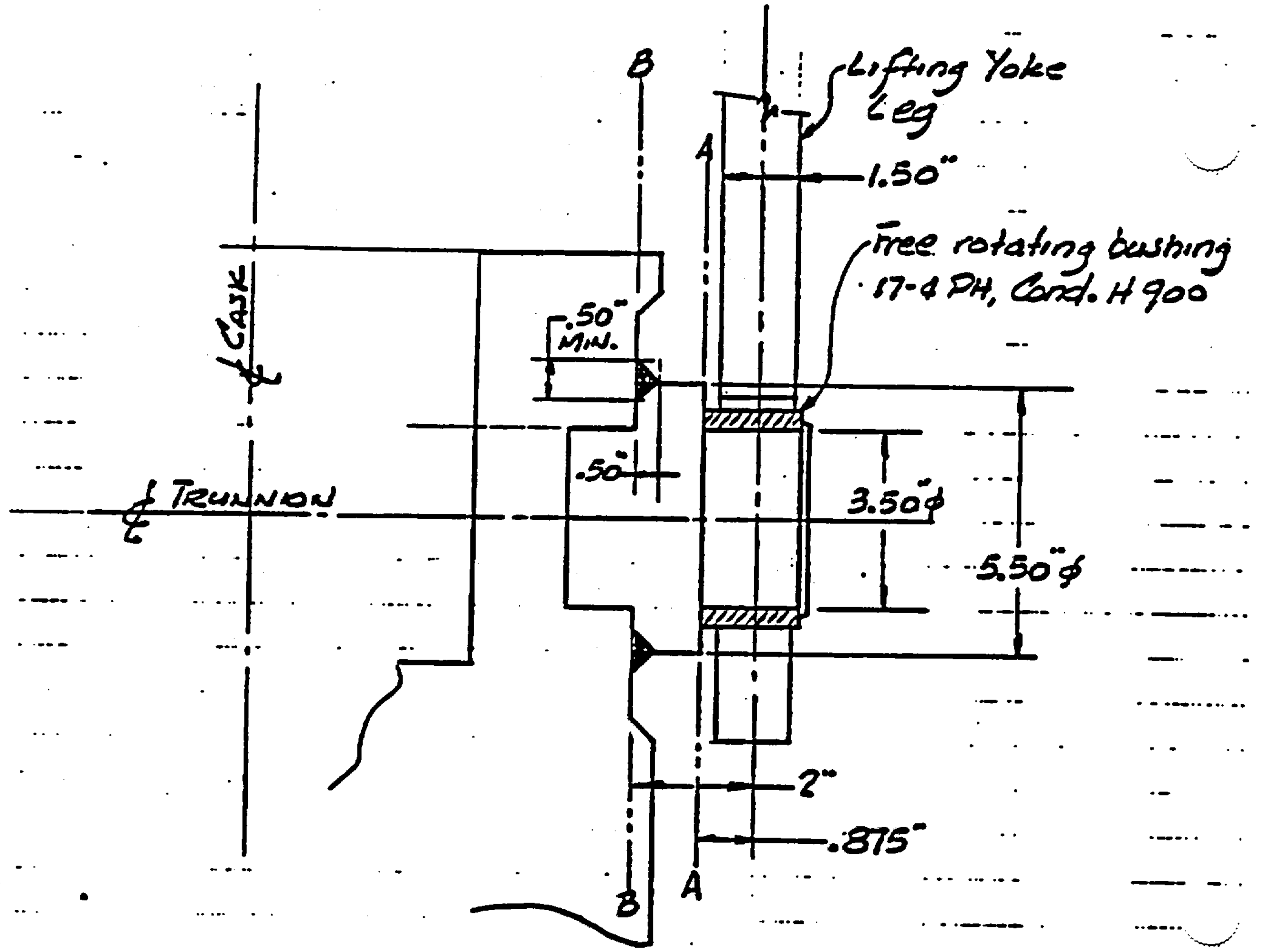
The shear strength of 2 bolts (5/8-11) at the midlength joint is 2 x 33,100 = 66,200 lbs. O.K. 62040

1.8 LIFTING DEVICES   10 CFR 71.31c

Requirement: " the system shall be capable of supporting three times the weight of the loaded package without generating stress in any material of the packaging in excess of its yield strength."

1.8.1 CASK TRUNNIONS

Loaded package weight 48000 lbs.
Design load = 3 × 48000 = 144000 lbs.
Cask is provided with four trunnions. Only two trunnions will be used at any one time to lift the cask. Therefore the design load per trunnion is 72000 lbs.



Section B-B Weld

Throat area of weld $= \pi D t = \pi(5.50)(.50) = 8.643 \text{ in.}^2$

Z of thin ring of 5.50" Dia. $= \frac{\pi R^3 t}{R} = \pi R^2 t$

$Z = \pi(2.75)^2(.50) = 11.83 \text{ in.}^2$

Bending moment $M_B = 72000(2) = 144000$ in. lbs.

$\sigma_B = \frac{M_B}{Z} = \frac{144000}{11.83} = \underline{12172 \text{ psi}}$

$\sigma_S = \frac{72000}{8.643} = \underline{8334 \text{ psi}}$ neglecting assistance of 3.50"φ center.

Combined shear stress $= \sigma_C = \sqrt{\left(\frac{\sigma_B}{2}\right)^2 + \sigma_S^2}$

$$\sigma_C = \sqrt{\left(\frac{12172}{2}\right)^2 + (8334)^2} = \underline{10320 \text{ psi}}$$

$$M.S. = \frac{(.6)(36000)}{10320} - 1 = \underline{.74}$$

Section A-A 3.50" Dia. trunnion section

Area of 3.5" dia $= 9.621 \text{ in}^2$

$Z = .098(3.5)^3 = 4.202 \text{ in}^3$

Bending moment $M_B = 72000(.875) = 63000$ in. lbs.

$\sigma_B = \frac{M_B}{Z} = \frac{63000}{4.202} = \underline{14993 \text{ psi}}$

Shear stress $\sigma_s = \frac{P}{A} = \frac{72000}{9.621} = 7484$ psi

Combined stress $= \sigma_c = \sigma_s + \sqrt{\left(\frac{\sigma_B}{2}\right)^2 + \sigma_s^2}$

$\sigma_c = 7497 + \sqrt{(7497)^2 + (7484)^2} = 18090$ psi

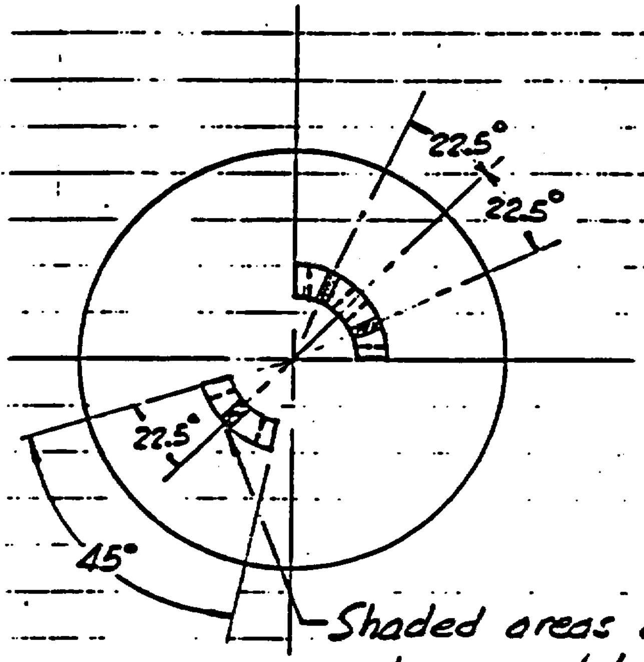
$M.S. = \frac{30000}{18090} - 1 = 0.658$

1.8.2 OUTER CLOSURE LIFTING LUGS

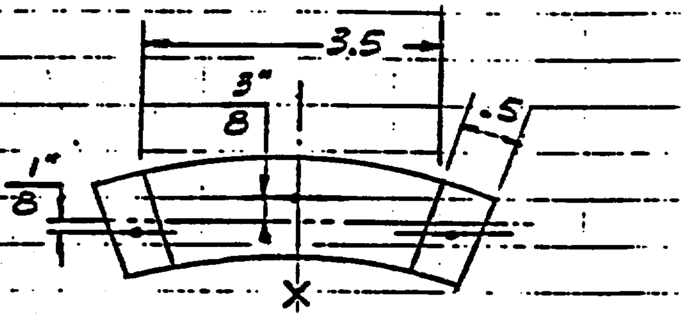
See page XI-88

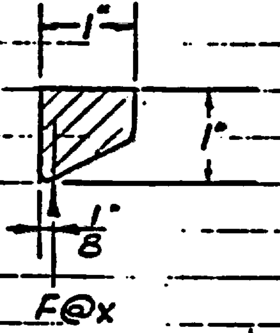
1.8.3 INNER CLOSURE HEAD LIFTING LUGS - CONFIGURATION "A"

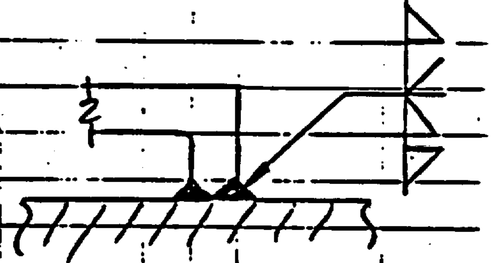
The inner closure head is provided with two arcuat bridge-like lugs which receive the retractible hooks of the lifting tool. Of the two lugs used, the one whic receives the single hook (instead of the double hook) is most highly stressed, even tho it receives half th total load.



Shaded areas are pick up point locations







Inner closure head weight = 666 lbs.
Design load = 3 x 666 = 1998 lbs.
Vertical load on lug at "X" is

$$F = \frac{1998}{2} = 999 \text{ lbs.}$$

It may be assumed, very conservatively, that the bridge section is reprented by a simply supported beam, 3.5 in. long, 1" wide and .75" deep, with $\frac{3}{8}$ in. eccentricity of the center load point.

Bending stress at center

$$M = \frac{Fl}{4} = \frac{999 \times 3.5}{4} = 875 \text{ in.lbs.}$$

$$Z = \frac{1 \times .75^2}{6} = 0.09375 \text{ in.}^3$$

$$S_b = \frac{875}{0.09375} = 9333 \text{ psi}$$

Torsion stress at center

$$T = \frac{F}{2} \times .375 = 188 \text{ in.lbs.}$$

$$I_p = \frac{1 \times .75^3}{12} + \frac{.75 \times 1^3}{12} = 0.0976 \text{ in}^4$$

$$S_s = \frac{188}{0.0976} = 1926 \text{ psi}$$

Combined stress

$$S_t = \frac{9333}{2} + \sqrt{4666^2 + 1926^2} = 9714 \text{ psi}$$

$$M.S. = \frac{30000}{9714} - 1 = 2.08$$

Weld strength

A single 1/4" full penetration weld plus 1/4" fille
has a throat of:

$$\frac{.25}{.707} = .353 \text{ in.}$$

Total weld area of lug is:

$$2(2 \times .353 \times 1) = 1.412 \text{ in}^2$$

$$\text{Tension stress} = \frac{999}{1.412} = 707 \text{ psi}$$

Bending due to 1/8 in. offset of load

$$M = \frac{999}{2} \times .125 = 62.4 \text{ in lbs}$$

$$Z_{weld} = (\text{approx.}) \frac{.5 \times 1^2}{6} = 0.0833$$

$$S_b = \frac{62.4}{.0833} = 749 \text{ psi}$$

Total tension stress

$$S_t = 707 + 749 = 1456 \text{ psi}$$

$$M_S = \frac{30000}{1456} - 1 = 19.6$$

Tension on each end support

$$S_t = \frac{999}{1 \times .5} = 1998 \text{ psi}$$

$$M.S. = \frac{30000}{1998} - 1 = 14$$

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1.9 Pressure Vessel Design - Inner Container

Design Conditions

The normal operating pressure under 10CFR71 Normal Conditions of Transport requirements is determined as follows.

The normal operating pressure is based on a maximum fuel rod temperature, under normal conditions of transport, of 1000 F. The inner container is initially filled with helium at an assumed gas temperature of 68°F and one atmosphere pressure. Assuming the gas temperature reaches the fuel assembly metal temperature, the normal operating pressure would be:

$$\frac{(1000 + 460)\ 14.7}{68 + 460} = 40.65 \text{ psia} = 25.95 \text{ psig}$$

NORMAL OPERATING PRESSURE 26 psig

If it is assumed that all the fuel rods fail, which is a highly unlikely event, the pressure in the inner container would increase as a result of the fuel rod gas pressures being released to the inner container environment. Data on initial rod fill pressure is not specifically reported, but based on available information, an upper limit of 500 psig* is assumed for PWR type fuels and 200 psig for BWR type fuels. It is realized that these rod fill pressures are extreme but the object is to establish an inner container design pressure that would include the hypothetical condition of a gas release from all rods in a fuel assembly and also accommodate future fuel designs which could involve increased fuel rod fill pressures.

*See Appendix C.

Table I shows design parameters of the four common types of fuel rods. The values shown except for V_1 were obtained or calculated from data in the fuel PSARs. V_1 was calculated from the following equation:

$$V_1 = \frac{P_0 T_1 V_0}{T_o P_1} + \frac{1.36 E f M R T_1}{P_1} \qquad \text{(1) Ref. 11}$$

It is possible that future elements will be pressurized to a higher pressure than that shown in Table I (P_o). Therefore a new value of P_1 will be calculated using equation (1) with an initial fill pressure of 515 PSIA for PWR type fuel. This assumes that the initial geomtry of the rods and the volume V_1 does not change. The total amount of gas in the rods will then be calculated. It will then be assumed that all the rods release the gas to the inner container. The pressure of the inner container will be calc

for normal conditions of transport and for the fire accident condition.

From equation (1)

$$P_1 = \frac{P_o T_1 V_o}{T_o V_1} + \frac{1.36 E f M R T_1}{V_1}$$

Using the B & W element data from Table I since this results in slightly higher cask pressures, P_1 is calculated as follows:

$$P_1 = \frac{515\ (1100)\ 1.534}{530(.889)} + \frac{1.36\ (42.2)\ (.23)\ 2.19 \times 10^{-3}(40.84)(1100)}{.889}$$

P_1 = 3305 PSIA

Table I

FUEL ROD GAS PRESSURE DESIGN PARAMETERS

	CE	B&W	Westinghouse	GE
Internal Rod Pressure (P_1), psia	2620	2765	2115	1100
Gas Temperature in Reactor (T_1), R	1100	1100	1100	1010
Initial Fill Temperature (T_O), R	530	530	530	530
Initial Fill Pressure (P_O), psia	15	365	365	15
Initial Fill Volume (V_O), in^3	1.73	1.534	1.58	4.482
Fraction Released (f)	0.31	0.23	0.21	0.35
Burnup, (E) GWD/MTU	37.3	42.2	42.0	37.4
Uranium Loading (M) MTU	2.25×10^{-3}	2.19×10^{-3}	2.21×10^{-3}	4.04×10^{-3}
Free Gas Volume in Reactor (V_1), in^3	0.627	0.889	1.14	2.76

This is the pressure that will develope in the fuel pins at the end of life condition if the initial fill pressure is 515 PSIA. From this pressure and the temperature and volume from Table I, the number of Mols of gas in the fuel assembly can be calculated.

$$PV = NRT$$

$$\frac{3305(.889)}{12(1545)1100} = N \quad \text{For 1 fuel pin}$$

$$1.44 \times 10^{-4}(208 \text{ rods}) = .0299 \text{ mols} = N \text{ total}$$

The fuel cavity will be filled with helium before shipment if it is assumed that the helium fill gas temperature is 100 F and 14.7 PSIA. The amount of helium in the fuel cavity would be -

For the PWR basket

$$N_H = \frac{14.7(144)\ 5.6}{560(1545)}$$

$$= .0137$$

The total number of mols in the fuel cavity if all the fuel rods rupture equals NT

$$NT = .0137 + .0299$$

$$= .0436 \text{ mols}$$

The pressure in the inner container can now be calculated for the fire accident and normal conditions of transport.

Fuel temperature at fire accident = 1102 °F

$$P = \frac{.0436(1545)\ (1102 + 460)}{5.6(144)}$$

$$= \quad 131 \text{ psia}$$

Fuel temperature at normal conditions = 1000° F Avg.

$$P = \frac{.0436\ (1545)\ (1000 + 460)}{5.6\ (144)}$$

$$= \quad 122 \text{ psia}$$

Inner container pressure based on BWR type fuel assemblies.

Assume an inital fill pressure of 200 psig. The fuel pin pressure at End of Life Conditions would be:

$$P_1 = \frac{P_o\ T_1\ V_o}{T_o\ V_1} + \frac{1.36EfMRT_1}{V_1}$$

$$= \frac{215(1010)4.482}{530(2.76)} + \frac{1.36(37.4)\ (.35)\ (4.04 \times 10^{-3})\ 40.84(1010)}{2.76}$$

$$P_1 = 1740 \text{ psia}$$

Number of mols of gas in one fuel pin is:

$$N = \frac{1740(144)2.76}{1545(1010)1728} = 2.56 \times 10^{-4}$$

One BWR fuel assembly has 49 rods and there are two fuel assemblies in the cask.

$$N_f = 2.56 \times 10^{-4}\ (49)\ 2 = 2.51 \times 10^{-2} \text{ mols}$$

Number of mols of helium in inner container

$$N_h = \frac{14.7(144)\ 4.876}{560(1545)} = 1.193 \times 10^{-2} \text{ mols}$$

Total number of mols in inner container

$$N_t = .01193 + .0251 = .03703 \text{ mols}$$

The pressure in the inner container can now be calculated for the free accident and normal conditions of transport.

Fuel temperature at fire accident = 1102 F

$$P = \frac{.03703(1545)(1102 + 460)}{4.876\ (144)} = 127.27 \text{ psia}$$

Fuel temperature at normal conditions - 1000 F

$$P = \frac{.03703(1545)(1000 + 460)}{4.876(144)} = 118.96 \text{ psia}$$

Summary of internal pressure conditions

Normal Operating Pressure	- 26 psig
Normal Operating Pressure - Fuel Failure	- 108 psig
Fire Accident Pressures - Fuel Failure	- 117 psig

Design Conditions

Pressure	- 120 psig	
Metal temperature	- 850 F	
Material	- 304 stainless steel	
Yield Point @850 F	-16500 psi	Ref. 4

1.9.1 Inner Container Shell

Ref. 1 Table XIII, Case 1

$$p = \frac{tSy}{R}$$

$t = .25$

$Sy = 16500$ psi

$R = 6.44$

$$p = \frac{.25 \times 16500}{6.44} = \underline{640.5 \text{ psi} \quad 120 \text{ psi design}}$$

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1.9.2 INNER CONTAINER BOTTOM HEAD

Reference 2.

$$T = d\sqrt{\frac{cp}{S}} \quad ; \quad p = \frac{T^2 S}{d^2 c}$$

$$p = \frac{(1.25)^2 \times 16500}{(12.625)^2 \times .3}$$

$T = 1.25$
$d = 12.625$
$S = 16500$
$c = .3$

$p = \underline{543}$ psi >120 psi design

1.9.3 INNER CONTAINER CLOSURE HEAD

Very conservatively, consider only the inner plate of the closure head taking the full pressure load as an edge supported plate.

Reference 1, Table x, Case 1

$$\text{Max. } s_r = s_t = \frac{3W}{8\pi m t^2}(3m+1)$$

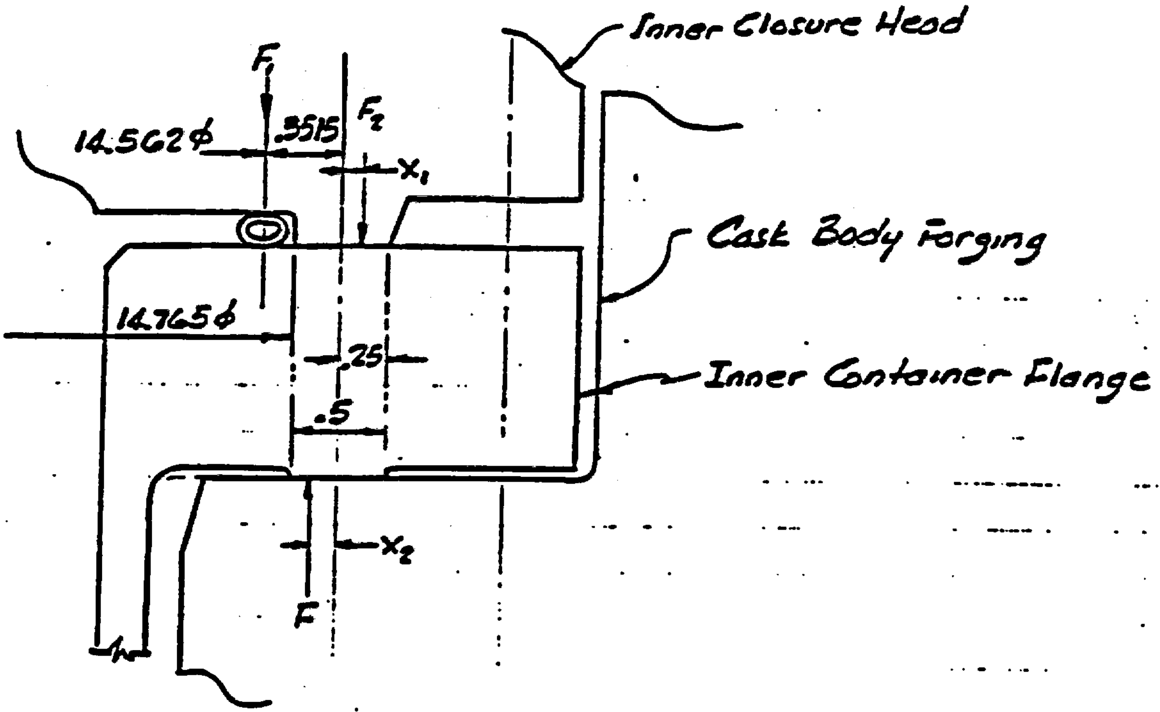
$m = \frac{10}{3}$
$d = 14.$
$W = .785 \times 14^2 p$
$t = 1.5$
$s_r = 16500$ Ref. 4

$$s_r = \frac{(3)\left(\frac{\pi}{4}\right)(14^2)p}{8\pi\left(\frac{10}{3}\right)1.5^2}(11)$$

$$16500 = 26.95\,p$$

$$p = \frac{16500}{26.95} = \underline{612 \text{ psi}} \quad \underline{>120 \text{ psig design}}$$

1.9.4 INNER CONTIANER FLANGE



The maximum loading is imposed by the clamping force of the inner closure head studs and nuts (2.1.4.2)
F = 194125 lbs.

The required clamping force on the gasket is –

$$F_1 = (1000\ lbs/in.)\pi\ 14.562 = 45739\ lbs.$$

$$F_2 = F - F_1 = 148386$$

If there were no gasket load F_1, then F_2 and F would both be 194125 lbs. and be concentrated on the ℄ of the 0.5" wide opposed contact bands shown. Now, assume F_1 added and F_2 reduced as shown. This would result in non-uniform stress distributions across the two 0.5" contact bands, with F_2 moving a distance x_1 and F moving in an opposite direction by a distance x_2.

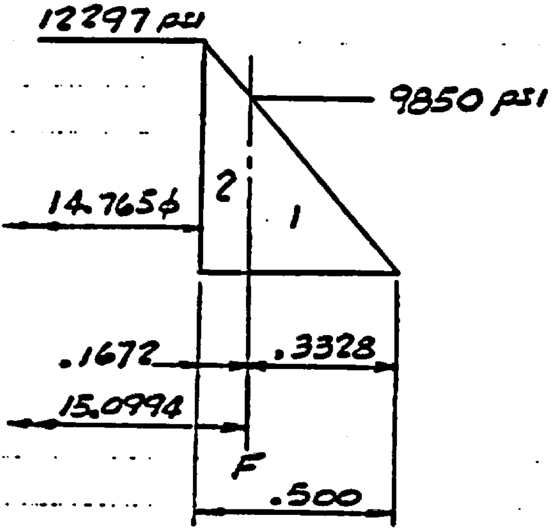
The minimum disturbance shift would let $x_1 = 0$ and:

$$x_2 = \frac{F_1 \times .3515}{F} = \frac{45739 \times .3515}{194125} = .0828$$

$$.25 - .0828 = .1672$$
$$.50 - .1672 = .3328$$

The mean stress associated with force F is:

$$\frac{194125}{(\pi 15.0994 \times .5) - (8 \times 1).5} = 9850 \text{ psi}$$



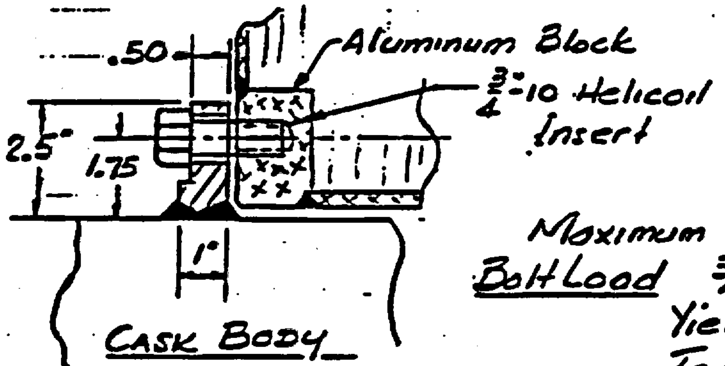
Assume triangular loading as shown.
Peak stress = 12297 psi
Moment of area 2 about F = 152.727
Moment of area 1 about F = 151.089
Small differential acceptable

$$M.S. = \frac{30000}{12297} - 1 = \underline{1.44}$$

This load distribution would result from a combination of gasket and bolting loads and would not impose any rotating moment on the flange.

1.10 IMPACT STRUCTURE – CASK ATTACHMENT

The impact structures are bolted to the cask body as shown below. The bolted attachments are designed to secure the impact structure to the cask under the loadings imposed by the normal conditions of shipment as specified by 10 CFR 71.31(d). Vertical and lateral loads are transmitted directly to the cask itself by the cylindrical inner surface of the balsa end caps. The center of gravity of the caps is within the cask outline, thus there is no overturning moment. The 10 g loading in the direction of travel, which is an axial load relative to the cask, is taken in tension by 4 bolts 3/4"-10 parallel to the cask axis.



Weight of Top Impact Structure = 1000 lbs

Weight of Bottom Impact Structure = 500 lbs.

Maximum axial force = 10(1000) = 10000

Bolt Load 3/4"-10 stainless steel bolt

Yield point = 30 000 psi

Tension area = .334 in^2

Strength at Y.P. = 10020 lbs.

Load on each bolt = $\frac{10000}{4}$ = 2500 l

Margin of Safety = $\frac{10020}{2500} - 1 = \underline{3.0}$

Stainless Steel lug on cask - Weld Stress

Moment on penetration weld at base of lug

$M = 2500 \times 1.75 = 4375$ in. lbs.

$$Z_{lug} = \frac{2(1)^2}{6} = .333 \text{ in}^3$$

$S_b = \frac{4375}{.333} = \underline{13138}$ psi $S_s = \frac{2500}{2(1.)} = \underline{1250 \text{ psi}}$

Combined $S_T = \frac{13138}{2} + \sqrt{6569^2 + 1250^2} = 13256$ psi

Lug M.S. $= \frac{30000}{13256} - 1 = \underline{1.26}$

<u>Aluminum Block</u> 6061 Alloy

Yield point = 37000 psi , Shear strength = 27000 psi
Internal threads 3/4"-10 x 1" deep

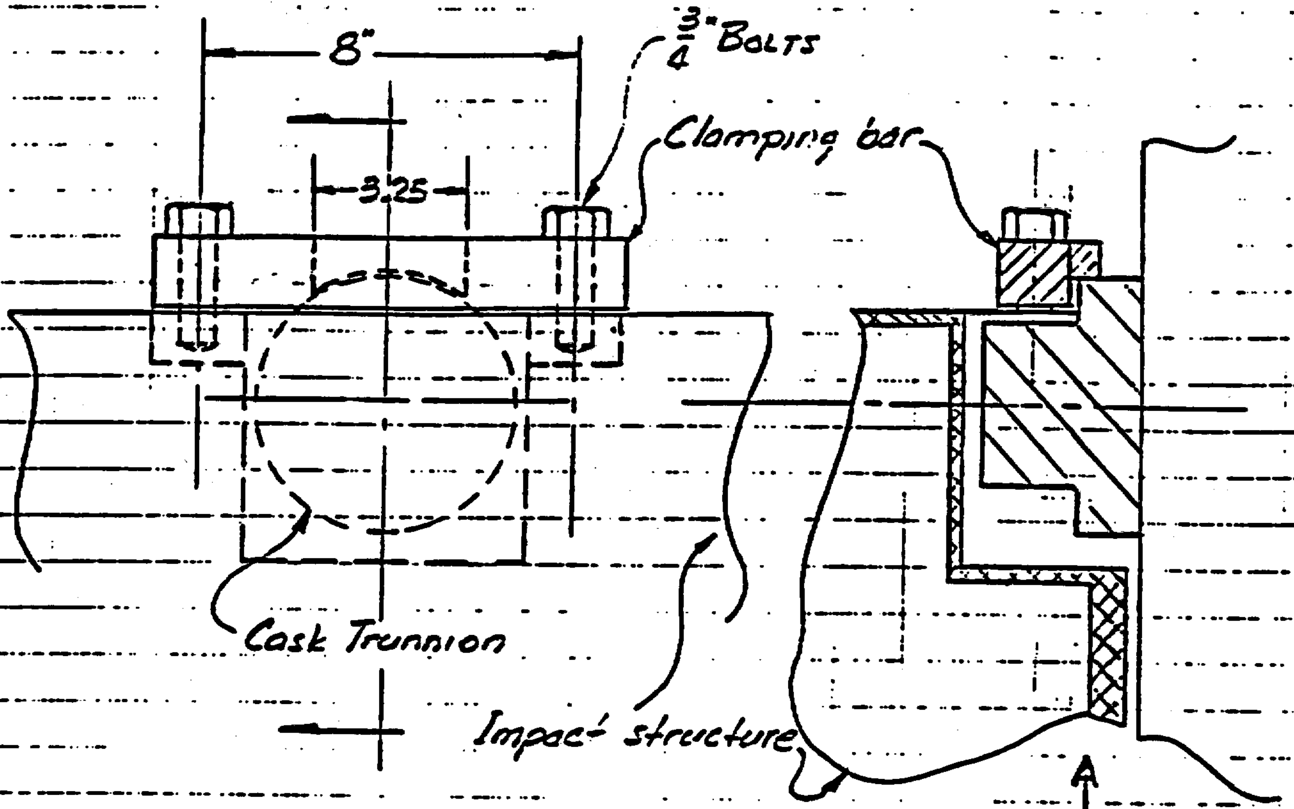
Shear area at Pitch. Dia. $= \frac{1(\pi \cdot .685)}{2} = 1.076 \text{ in}^2$, conservative

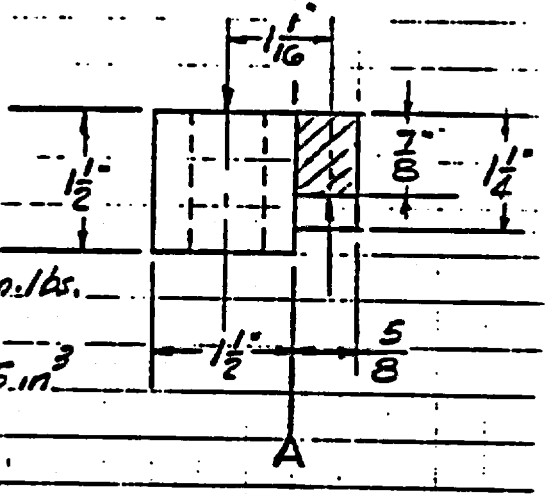
Shear force, ult. = 27000 (1.076) = 29052 lbs.

M.S. $= \frac{29052}{2500} - 1 = \underline{10.6}$ No credit taken for the additonal shear strength provid by the Helicoil Insert

Attachment of closure head end impact structure.

For the top impact structure only, 2 of the 4 bolts are anchored to the cask by a bar which clamps the structure against the cask lifting trunnion. See figures on following page





Stress in bar at center neglecting 5/8" x 1 1/4" side plate

$$\text{Bending } M = \frac{5000 \times 8"}{4} = 10000 \text{ in.lbs.}$$

$$Z = \frac{1.5 \times 1.5^2}{6} = .5625 \text{ in}^3$$

$$S_b = \frac{10000}{.5625} = 17778 \text{ psi}$$

Torsion due to $1\frac{1}{16}''$ eccentricity

$$T = \frac{5000}{2} \times 1.0625 = 2656 \text{ in lbs}$$

$$I_p = 2 \times \frac{1.5 \times 1.5^3}{12} = .84375 \text{ in}^4$$

$$S_s = \frac{2656}{.84375} = 3148 \text{ psi}$$

Combined stress

$$S_t = \frac{17778}{2} + \sqrt{8889^2 + 3148^2} = 18319 \text{ psi}$$

$$M.S. = \frac{30000}{18319} - 1 = .63$$

Weld attaching side plate to bar plane A-A

Total weld = $2 \times 1\frac{1}{4} \times \frac{1}{4}$ fillet weld + $2 \times 10 \times \frac{1}{8}$ seal we

Weld area = $2 \times 1\frac{1}{4} \times \frac{1}{4} \times .707 + 2 \times 10 \times \frac{1}{8} = 2.94 \text{ in}^2$

$$\text{Shear stress} = \frac{5000}{2.94} = 1700 \text{ psi}$$

Bending stress -

$M = 5000 \times .3125 = 1563$ in lbs.

Two seal welds resist this as a force couple

$$\text{Force in each weld} = \frac{1563}{1.25} = 1250 \text{ lbs}$$

$$S_t = \frac{1250}{10 \times .125} = 1000 \text{ psi}$$

Combined stress

$$S_t = \frac{1000}{2} + \sqrt{500^2 + 1700^2} = 2272 \text{ psi}$$

$$M.S. = \frac{30000}{2272} - 1 = 12.2$$

Stress in bolts (3/4"-10)

Tensile stress area = .334 in^2/bolt.

$$\text{Load in each bolt} = \frac{5000}{2} \times \left(\frac{1.0625 + .625}{.625}\right) =$$

$$= 6042 \text{ lbs}$$

$$S_t = \frac{6042}{.334} = 18090 \text{ psi}$$

$$M.S. = \frac{30000}{18090} - 1 = .65$$

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1.11 One Foot Drop Requirement Normal Conditions of Transport

The extent of damage to the impact limiters, as calculated in the following paragraphs, may in certain situations be considered objectionable. In those cases where damage exceeds that which is predicted in the following paragraphs corrective action will be taken to replace the damaged impact limiter.

1.11.1 One Foot End Drop

From 2.1.1.2, the initial and peak load is 1,891,541 lbs. to start crushing the aluminum shell and the balsa.

The deformation for a 1 foot drop is

$$\frac{48,000 \times 12.31 \text{ in. lbs.}}{1,891,541 \text{ lbs.}} = .31 \text{ in.}$$

The effect of a flat end drop of 1 foot is to indent the whole end balsa assembly 5/16 inch.

$$\frac{1,891,541}{48,000} = 39.4 \text{ g's}$$

The 39.4 g valve is the same as for the 30 foot end drop for which the various elements of the cask were analyzed in Section 2.1. The results of these analyses showed the various cask elements did not suffer permanent deformation or damage. There is no reduction in the effectiveness of the packaging as a result of the one foot end drop.

1.11.2 One Foot Side Drop

See Section 2.3.1.4, 2.3.1.5, and 2.3.1.5.3

As reported in referenced sections, the crush distance is 1.20 inches and the peak "g's" are 16.38.

In the analysis of the 30 foot side drop, it was shown that the various cask elements did not suffer permanent deformation or damage. Therefore, the lower "g" load developed by the one foot side drop will not reduce the effectiveness of the package.

1.11.3 One Foot Drop - Corner Impact

From 2.2, corner impact, the full area along line d-a-e is only reached when the mean deformation is the distance $\overline{ac} = \frac{4.875}{2} = 2.4375"$

However, the 1 foot end drop effects a compression of only 5/16 inch, compared to 2.4375" above.

Therefore, a 1 foot corner drop will damage less than the full area, and will leave the balsa assembly physically capable of sustaining a further 30 foot drop.

1.11.4 One Foot Drop - 45° Oblique Impact

$$\text{Vol. a} = 547\ \text{in}^3$$

$$\text{Vol b} = \underline{359\ \text{in}^3}$$
$$906\ \text{in}^3$$

$$KE = (48000)\ (12 + 5.65) = 847{,}200\ \text{in. lbs.}$$

$$\frac{KE}{\text{Vol.}} = \frac{847{,}200\ \text{in. lbs.}}{906\ \text{in.}^3} = 935\ \text{psi}$$

The estimate effective 45° crushing strength is 1850 psi x .7 = 1,295 psi

The estimated deformation is therefore considered more than conservative since the energy absorbed by the covering deformation of the aluminum sheet is not included.

Such small and localized damage is thus expected that a further drop of 30 feet in any altitude could be absorbed.

The final impact areas are estimated as:

$$a = 8 \times 14\ 1/4 = 14\ \text{in}^2$$

$$b = 65 \times 21 = \underline{136.5\ \text{in}^2}$$
$$250.5\ \text{in}^2$$

Max force = 1295 x 250 .5 = 324,398 lbs.

$$\text{Accel} = \frac{324{,}398}{48{,}077} = 6.75\ g\text{'s}$$

1.11.5 Accelerations in One Foot Drop

39.4 g's	Flat End Drop
16.38 g's	Side Drop
6.75 g's	45° Oblique Drop
< 39.4 g's	Corner Drop

1.11.6 Anchorage Security of Each Balsa Unit in 1 Foot Drop

	Top	Bottom
Weight of End Balsa Unit	145.56	122.71
Weight of Side Balsa Unit	299.18	308.81

1.11.6.1 End Balsa Unit - Side Drop

Max. Wt. = 145.56

Max. Lateral g's = 16.38

Max. Lateral Shear Force = 145.66 x 16.38 = 2,385 lbs.

Max. Moment = 2,385 x 8.5" = 20,272 in. lbs.

The stresses of shear and bending are taken by 1/8 fillet weld around perimeter of cylindrical shell 34" O. D.

Area of Weld - $\pi 34(1/8 \times 0.707) = 9.44 \text{in}^2$

$$S_s = \frac{2385}{9.44} = 252.6 \text{ psi}$$

I of Ring Weld $= \pi R^3 t = \pi(17)^3(1/8 \times 0.707) = 1,364 \text{ in}^4$ about $\mathrm{C\!\!\!L}$

I Tangent to Edge $= 1,364 \text{ in}^4 + Ar^2 = 1,364 + 9.44(17^2) = 4,092 \text{ in}^4$

$$\text{Edge} = \frac{4092}{34\text{dia.}} = 120.3 \text{ in}^3$$

$$S_e = \frac{20,272}{120.3} = 168.5 \text{ psi}$$

$S_e + S_s = 421.1$ psi in weld OK < 27,000 U.T.S. modified 6061

In the top end balsa unit lateral shear is also imposed on four (4) retaining bolts which connect the end impact limiter to the rest of the impact structure. See Figure 4 a. Assume both shear and bending is taken on two bolts.

Max. lateral shear = 2385 lbs. (1.11.6.1)

Bolts are 1/2 - 13, A 354 Gr. BD with stress area through threads = 0.1419 $in.^2$

125,000 psi Y.P.

$$S_s = \frac{2385}{(2)(0.1419)} = 8404 \text{ psi}$$

M = 2385 x 0.5 = 1193 in. lbs.

$$S_b = \frac{M}{Z} = \frac{1193}{(2)(0.098)(0.5^3)} = 48,694 \text{ psi}$$

$$S_t = \frac{48694}{2} + \left(24347^2 + 8404^2\right)^{0.5} = 50104 \text{ psi}$$

$$M.S. = \frac{125000}{50104} - 1 = 1.50$$

1.11.6.2 Side Balsa Unit - End Drop

Max. Wt = 308.81 lbs.

Max. Axial g's = 39.4

Max. Axial Shear Force - 308.81 x 39.4 = 12,167 lbs.

The side balsa units are welded to a cylindrical aluminum shell which provides a common anchorage for both the side and end impact limiters, See Figure 4 a.

Throat width = (1/8 x 0.707)   Area of Welds = 30.25 (0.125 x 0.707) = 8.4 in^2

$$S_s = \frac{12167}{8.4} = 1449 \text{ psi}$$

OK

27,000 x 0.6
= 16,200 pr.
ult. shear stress
6061

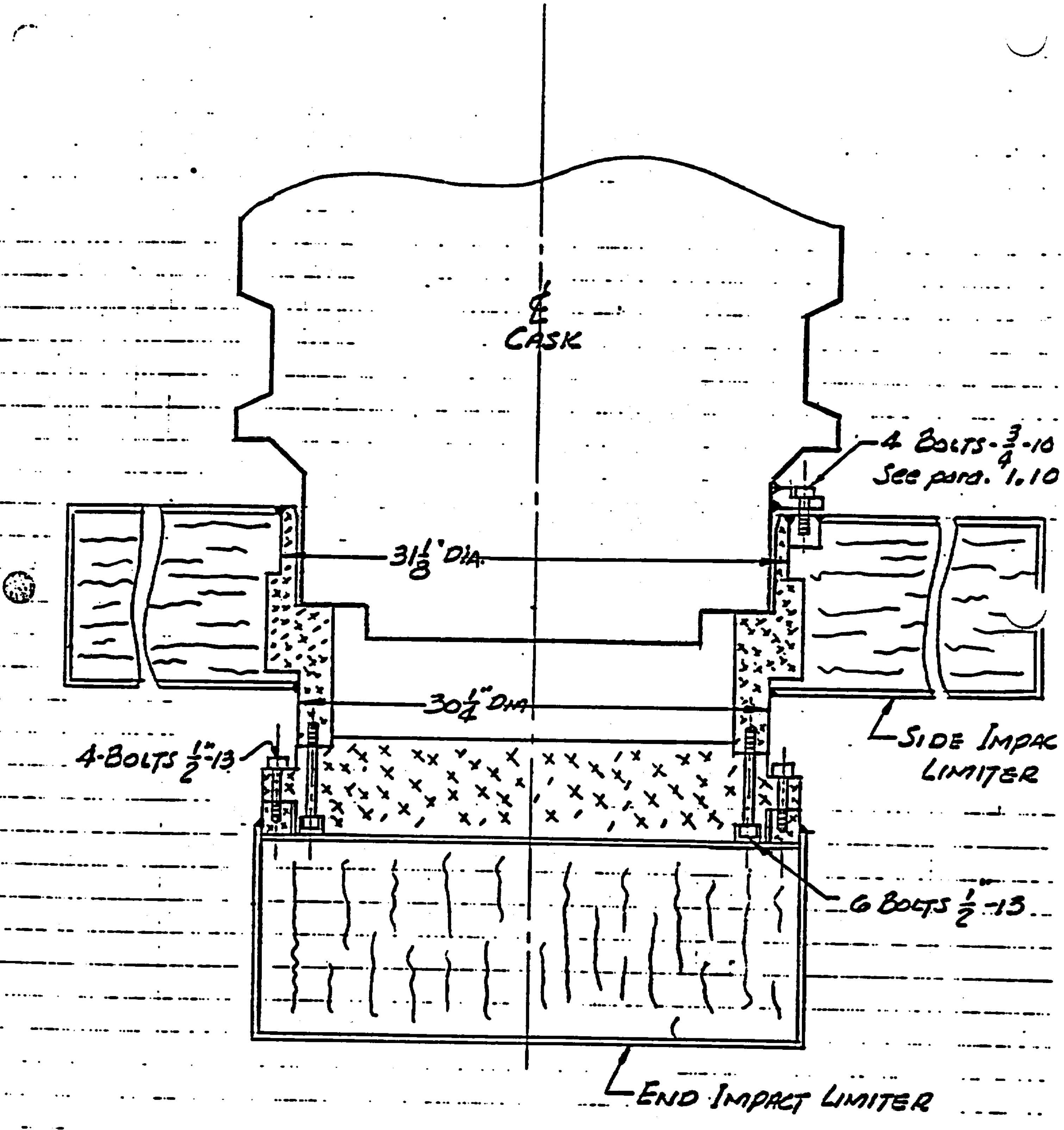
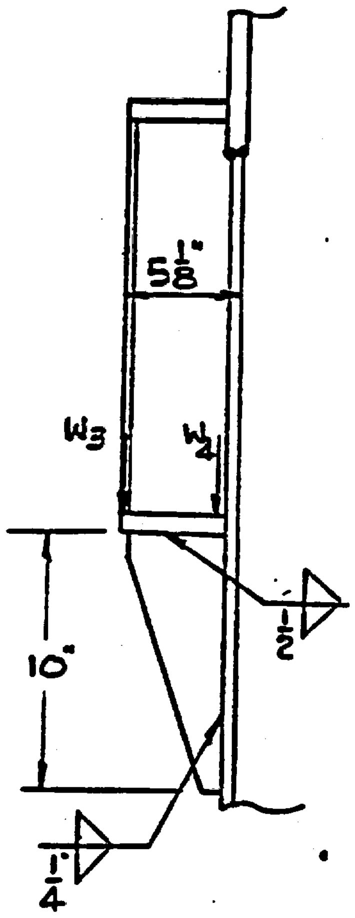

℄
CASK
4 BOLTS-$\frac{3}{4}$-10
See para. 1.10
$31\frac{1}{8}$" DIA.
$30\frac{1}{4}$" DIA.
4-BOLTS $\frac{1}{2}$"-13
SIDE IMPAC LIMITER
6 BOLTS $\frac{1}{2}$"-13
END IMPACT LIMITER

FIGURE 4a

1.11.7 One Foot Drop - Expansion Tank

The maximum loading is developed by the flat end drop with a peak value of 39.4 g's (XI-1-46)



Load due to weight of outer shell;

$W_1 = 340 \text{ lbs.} \times 39.4 \text{ g's} = 13{,}396 \text{ lbs.}$

Load due to weight of 2 end plates and solution

$W_2 = (246 + 547)\ 39.4 \text{ g's} = 31{,}244 \text{ lbs.}$

The distribution over the four (4) gussets is, for each,

$$W_3 = \frac{W_1 + \frac{W_2}{2}}{4} = \frac{13396 + \frac{31{,}244}{2}}{4} = \frac{29018}{4}$$

$$= 7254.5 \text{ lbs.}$$

$$W_4 = \frac{W_2}{2. \times 4} = 3905.5 \text{ lbs.}$$

Very conservatively, the plates are not assumed to support these loads by their attachments to the cask shell, but the total load is supported by the four (4) gussets.

Weld for each gusset to the 1/4" 216 SS shell.
2 welds, 1/4 inch fillets, 10 in. long

Bending $Z = \frac{(2 \times 1/4 \times .707)\ 10^2}{6} = 5.89 \text{ in.}^3$

$M_1 = 5.125\ W_3 = 37{,}179.3 \text{ in. lbs.}$

$S_b = M/Z = 6312 \text{ psi}$

Shear $S_s = \frac{W_3 + W_4}{2 \times 1/4 \times .707 \times 10} = \frac{11,160}{3.535} = 3157$ p

Combined stress $= \frac{6312}{2} + \quad 3156^2 + 3157^2$

$= 3156 + 4463 = 7619$ psi

M.S. weld $= \frac{.6 \times 30,000}{7619} -1 = 1.36$

For reverse direction of loading, moment on 5-1/8 welds of lower gussets to 5/8 lower plate is

$M_2 = W_3 \times 5.125 = 37179$

2 welds, 1/2 in. fillets, 5-1/8 long

Bending $Z = \frac{(2 \times \frac{1}{2} \times .707) \; 5.125^2}{6} = 3.09$ in. ^{3}

$S_b = \frac{M_2}{Z} = 12,032$ psi

M.S. weld $\frac{.6 \times 30,000}{12032} - 1 = .496$

Hoop stress in outer shell

Dynamic head = 20.5" x 39.4 = 807.7 in.

Dynamic pressure $= 807.7 \times \frac{68.453}{1728} = 32$ psi

Design pressure = 250 psi

Total pressure at end of tank = 250 + 32 = 282 psi

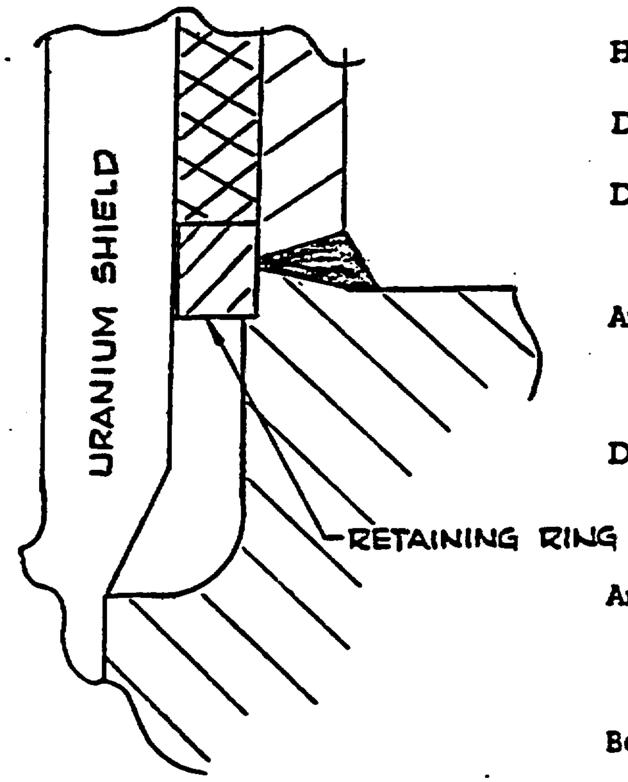
Hoop Stress+ $\frac{pR}{t} = \frac{282}{.375} \times \frac{46.75}{2} = 17,578$ psi

M.S. $= \frac{30,000}{17,578} -1 = .70$

1.11.8 Retaining Ring - Lead Shielding - End Drop

With impact on the bottom end, the load from the lead mass assuming no bond to the retaining walls, is impressed on the 2 inch wide annulus of the retaining ring. This, in turn, is supported on a $\frac{1}{4}$ inch wide lip of the bottom forging.

Max. impact force is 39.4 g's
Weight of lead - 9489.71 lbs.



Total force on 2 inch annulus is less than developed from total weight of lead.

Height of lead column is 160 inches

Dynamic head = 160 in. x 39.4 g's = 6304 in.

Dynamic pressure = 6304 x .41 lbs/in.^{3}
=2585 psi

Area of 2 in. ammulus is

$$\pi/_4\ (23.31^2 - 20.053^2) = 110.924 \text{ in}$$

Dynamic load on face is

$$(2585)\ 110.924 = 286{,}739 \text{ lbs.}$$

Area of supporting land is

$$\pi/_4\ (23.31^2 - 22.81^2) = 18.11 \text{ in.}^2$$

Bearing stress on land is

$$\frac{286{,}739 \text{ lbs.}}{18.11 \text{ in.}^2} = 15{,}833 \text{ psi}$$

OK 30,000 psi
for 304 SS

The annulus is enclosed by rigid cask walls and incapable of twisting due to offset of this land relative to of load.

Sheer stress on annulus

$$S_s = \frac{286{,}739}{(22.81)\ 2} = 2000.7 \text{ psi}$$

$$\text{M.S.} = \frac{.6 \times 30{,}000}{2000.7} - 1 = 8.$$

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2.0 HYPOTHETICAL ACCIDENT CONDITIONS

The cask package consists of the cask proper with the addition of two detachable energy absorbing caps at the ends. These serve to protect the cask in the several attitudes of the required 30 foot free fall. The bulk of the crushable materia is great enough to allow complete energy absorption without striking thru into contact with the cask. The average and maximum stresses are low enough to insure smaller than usual g-loac accelerations in the various drop conditions. The puncture test alone develops no higher-than-yield point stresses, and even then satisfys the specified conditions.

2.0.1 STRUCTURE OF THE END CAPS

Reference: NLI drawing 70510 F

Each end cap is structurally two independent balsa wood filled cylinders, joned mechanically by one commom cylindrical member. The essential dimensions are the same for top and bottom caps, the differences being largely in the internal adaptations to the contours of the cask.

The outer shells are $\frac{1}{8}$ inch aluminum alloy and are considered in the evaluation of the major loads and deformations and g's.

The internal energy absorbing material is balsa, so oriented in each compartment (end or side annulus) as to present nominal end grain to the direction of crushing.

The end cylindrical compartments are filled with blocks and slabs of balsa glued together in a pattern similar to that of quarter sawn oak.

The pattern for the side impact annulus is that of a large number of radial keystones, thus presenting end grain at all positions around the circumference.

2.0.2 ATTACHMENT OF BALSA END CAPS TO CASK

The end caps are assembled and disassembled by sliding axially along the cask ends into an extreme position where the flat bottom plate of the central balsa cylinder contacts either the bottom of the cask or the top of the closure head.

In these positions the caps are retained from axial motion by bolts which pull a threaded block inside the balsa shell toward a radial lug on the cask, the bolts being parallel to the cask axis.

These bolts have only a nominal function, since they are not subjected to dynamic loadings in any drop attitude. Consequently, convenience and operational considerations govern the design.

In each end cap there are four (4) 3/4 inch bolts used for such attachment.

2.0.3 Specification Strength of Balsa

Balsa wood, when subjected to end grain compression testing, displays an early maximum stress, after a slight initial "bedding in". Further compression, to an average maximum of 85% of its initial height, shows practically a straight line fall-off in stress thru a point representing 30% reduction in stress at 75% penetration, extrapolating to 40% reduction at nominally 100% penetration.

The relation between density and peak with-the-grain compression is well expressed by a graph showing published data from Balsa Ecuador Lumber Company, plotted on Fig. 5. Additional test data on a very wide range of Balsa test values, from samples of lumber used in manufacture of the impact limiters, shows excellent conformity with this straight line specification, also plotted on Fig. 5.

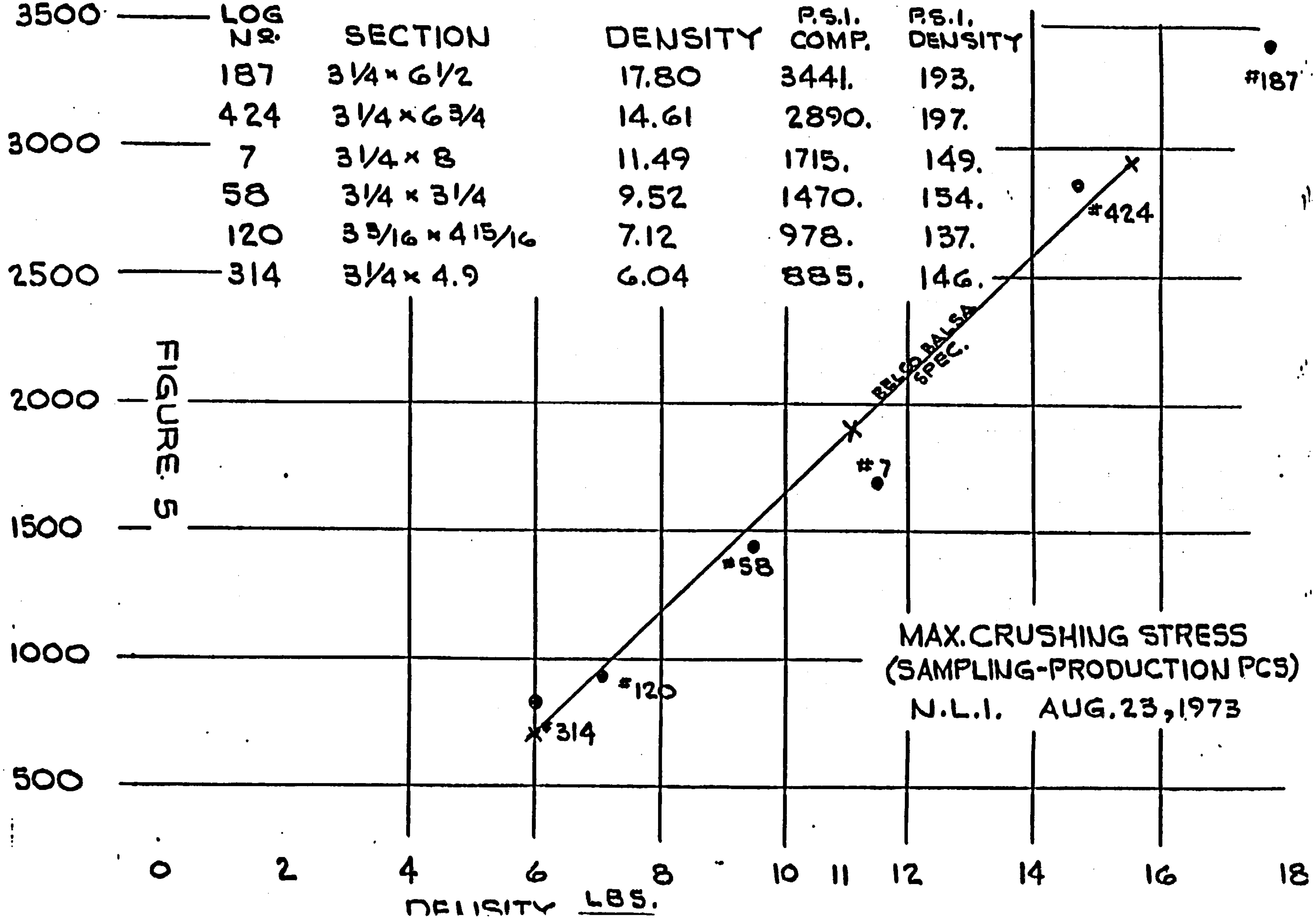
Accordingly, weighted analyses of the total lot of material used in the actual construction gave a mean density and a peak compression stress as shown on Fig. 1 of 10.777 lbs./ft.^{3}, 1850 psi. This is curve B on Fig. 6, which converges with curve A at a point representing 250% penetration and zero stress. Curve A quite accurately is that for "moist" condition, while Curve B, which represents the actual lot of lumber used, seems to correspond well to "dry" lumber conditions.

As indicated in Fig. 6 the moisture content has some effect on the mechanical properties of the balsa wood. The indication is that with an incarease in moisture there is a decrease in the peak compression stress.

The initial condition of the balsa wood impact limiters is maintained by treating the wood surface with Woodlife , a wood preservative manufactured by U.S. Plywood. This particular product also meets the Federal Standard TTW-572B which deals with the requirements of wood preservatives. There is no indication that balsa wood is effected by aging. The wood preservative which is applied to the surfaces of the wood impact limiters will provide an effective barrier to aging. Additional protection against moisture and aging is the encapsulation of the balsa wood in aluminum cans which are welded closed.

N.L.I. TESTS

LOG Nº	SECTION	DENSITY	P.S.I. COMP.	P.S.I. DENSITY
187	3 1/4 × 6 1/2	17.80	3441.	193.
424	3 1/4 × 6 3/4	14.61	2890.	197.
7	3 1/4 × 8	11.49	1715.	149.
58	3 1/4 × 3 1/4	9.52	1470.	154.
120	3 3/16 × 4 15/16	7.12	978.	137.
314	3 1/4 × 4.9	6.04	885.	146.



MAX.CRUSHING STRESS
(SAMPLING-PRODUCTION PCS)
N.L.I. AUG.23,1973

FIGURE 5

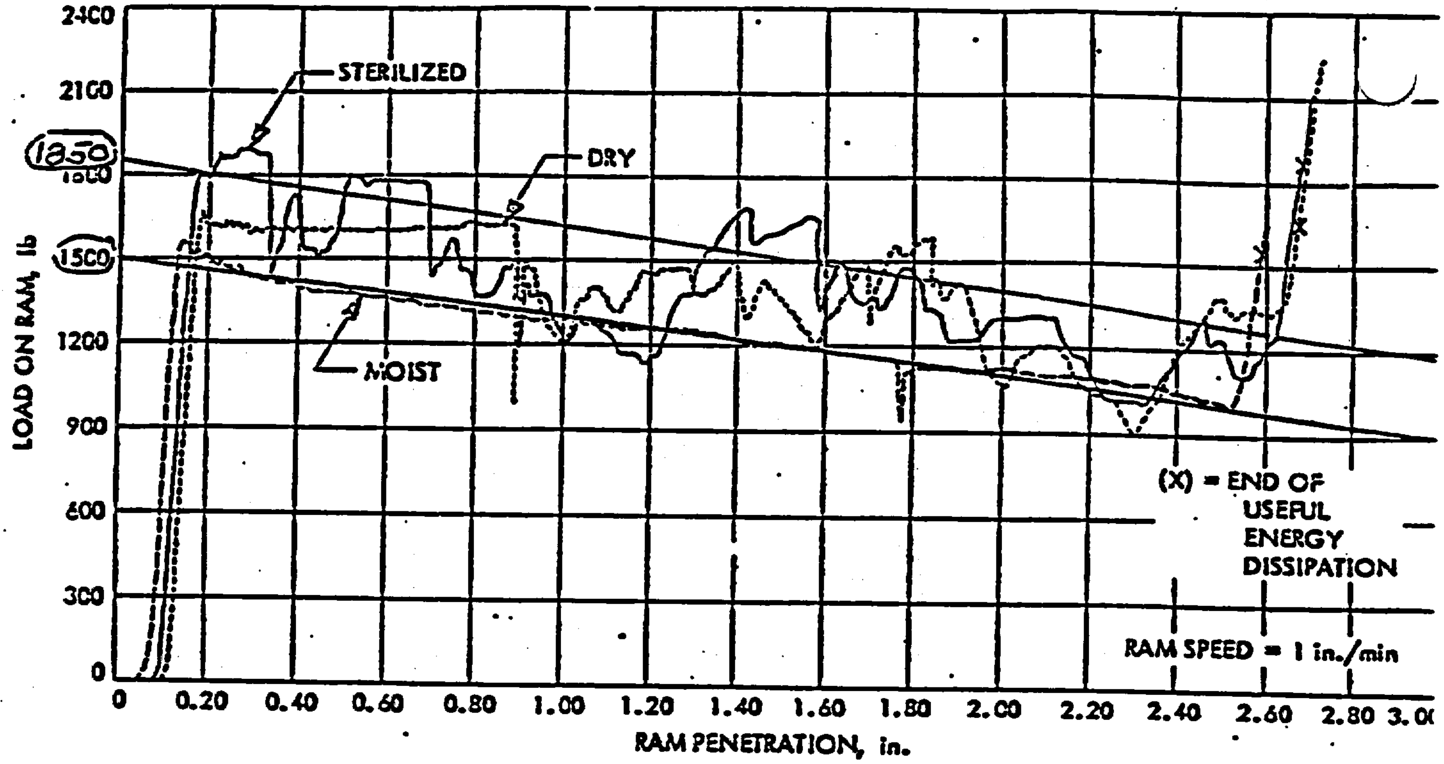


Fig. 6. Typical trace of crush test ram vs depth of ram penetration for balsa in each condition from a single stick

Table 1. Summary of data derived from 1,241 crush tests as a function of condition

Condition	Mean specific energy / Standard deviation, ft-lb/lb	Mean stress / Standard deviation, psi	Mean thickness efficiency / Standard deviation	Mean density, lb/ft³
Moist	20,400/3,229	1,342/337	0.839/0.013	7.93
Dry	22,900/3,369	1,383/379	0.885/0.024	7.75
Sterilized	23,097/3,674	1,454/354	0.849/0.027	7.63

Fig. 6

Reference - JPL Technical Report 32-1295

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2.1 End Impact Balsa Wood and Aluminum Cylinder Analysis

2.1.1 Buckling of 1/8" Aluminum Alloy Cylinder in Compression

Material : 6061-T6 sheet, 37,000 psi yp

Roark XVI - Case M - Stability of circular tube under compression

"Applicable if length is several times as great as $1.72 \sqrt{r\,t}$, which is a half wave in buckling. Tests indicate an actual buckling strength of $S^1 = .3\,E\,{}^{t}/_{r}$ ".

$t = 1/8"$ $r = 16.688"$ $E = 10{,}500{,}000$ psi

$1.72 \sqrt{16.688 \times .125} = 1.72\,(1.444) = 2.48"$ O.K. < 17."

$S^1 = .3\,(10{,}500{,}000)\,\dfrac{.125}{16.688} = 23{,}595$ psi O.K.<37,000 psi

Area cylinder, A = $33.375 \times .125 = 13.106$ in.^{2}

Buckling force = 23,595 (13.106) = <u>309,236 lbs.</u>

Equiv. g's = $\dfrac{309{,}236}{48{,}000} = 6.44$ g's for al. cyl. above

2.1.1.1 Balsa Cylinder

Balsa deformation 10.6"

KE = (360 + 10.6) 48,000 = 17,788,800 in.lb

Aluminum alloy shell absorbs (309,236) 10.6= 3,277,902 in.lbs

Remainder to be absorbed = 14,510,898 in.lbs

Percent of balsa crush $\dfrac{10.6"}{16.625} = 64\%$

Peak stress = 1850 psi

Final stress = 1850 - (.64) 740 - 1850 - 474 = 1376 psi

Mean stress = $\dfrac{1850 + 1370}{2} = 1613$ psi

Area of balsa = $\pi/4 \times 33^2$ = 855.3 in.^{2}

Energy absorbed = (1613) (855.3) (10.6) = 14,623,748 in. lt

2.1.1.2 Peak Deceleration

Force on aluminum shell	309,236 lbs.
Balsa peak = (1850) (855.3)=	1,582,305 lbs.
	1,891,541 lbs.

$$g\text{'s} = \frac{1,891,541}{48,000} = 39.4 \text{ g's}$$

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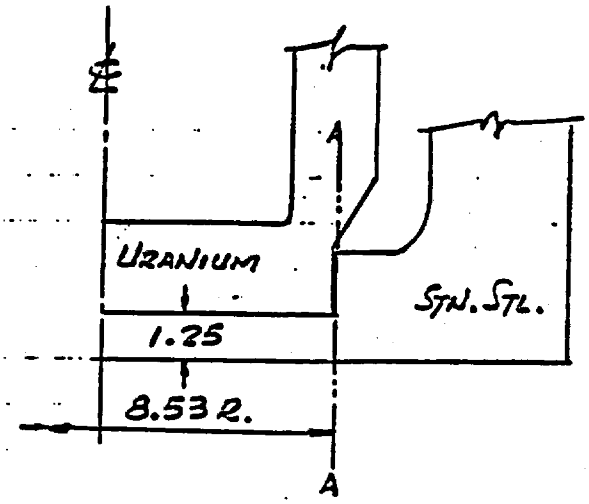
2.1.2 SUPPORTS FOR INNER SURFACES OF END IMPACT CYLINDER

At the bottom of the cask, the 33 inch diameter base plate is directly supported by the cask bottom itself.

At the top of the cask, direct contact is made on a 4 inch thick aluminum alloy support plate, which in turn is mounted on the upper end of the 30 inch diameter cylindrical extension of the cask proper. The adequacy of this arrangement is analyzed in PART 2.4.2.

2.1.3. BOTTOM END IMPACT ON CASK PARTS

2.1.3.1 URANIUM SHIELD IMPACTING BOTTOM STN. STL. PLATE



Weight of Uranium shield = 17,940

Max. g's = 39.4

$\downarrow F_1 = (17{,}940)(39.40) = 706836$ lbs.

The direct support within R = 8.53 inches provided by the balsa is

$\downarrow F_2 = \pi(8.53)^2(1850 psi) = 422{,}882$

Net force $\downarrow F_3$ remaining to be transmitted in shear thru section A-A:

$$F_3 = F_1 - F_2 = 706836 - 422882 = 283954 \text{ lbs.}$$

Shear area of A-A in Stn. Stl.

$$A_s = 2\pi(8.53)(1.25) = 66.98 \text{ in}^2$$

$$f_s = \frac{283954}{66.98} = 4239 \text{ psi}$$

$$M.S. = \frac{18000}{4239} - 1 = 3.14$$

Therefore the bottom Stn. Stl. plate will withstand the full stresses induced by the uranium shielding.

2.1.3.2 INNER AND OUTER CLOSURE HEADS

More critical stress values are obtained on head imp

2.1.3.3 INNER CONTAINER

The maximum stress is developed in tension of the upper neck section.

Total weights are 650.4 lbs. Inner Container
793.51 lbs. PWR Basket
1600 lbs. PWR Element
3043.91 lbs.

$$F_{impact} = (3043.91)(39.40) = 119\,930 \text{ lbs}$$

$$\text{Area of neck section} = 0.785(13.125^2 - 12.625^2) = 10.1 \text{ in}^2$$

$$S_t = \frac{119\,930}{10.1} = 11{,}874 \text{ psi} \quad M.S. = \frac{27500}{11874} - 1 = \underline{1.31}$$

Dynamic stress at temperature is –

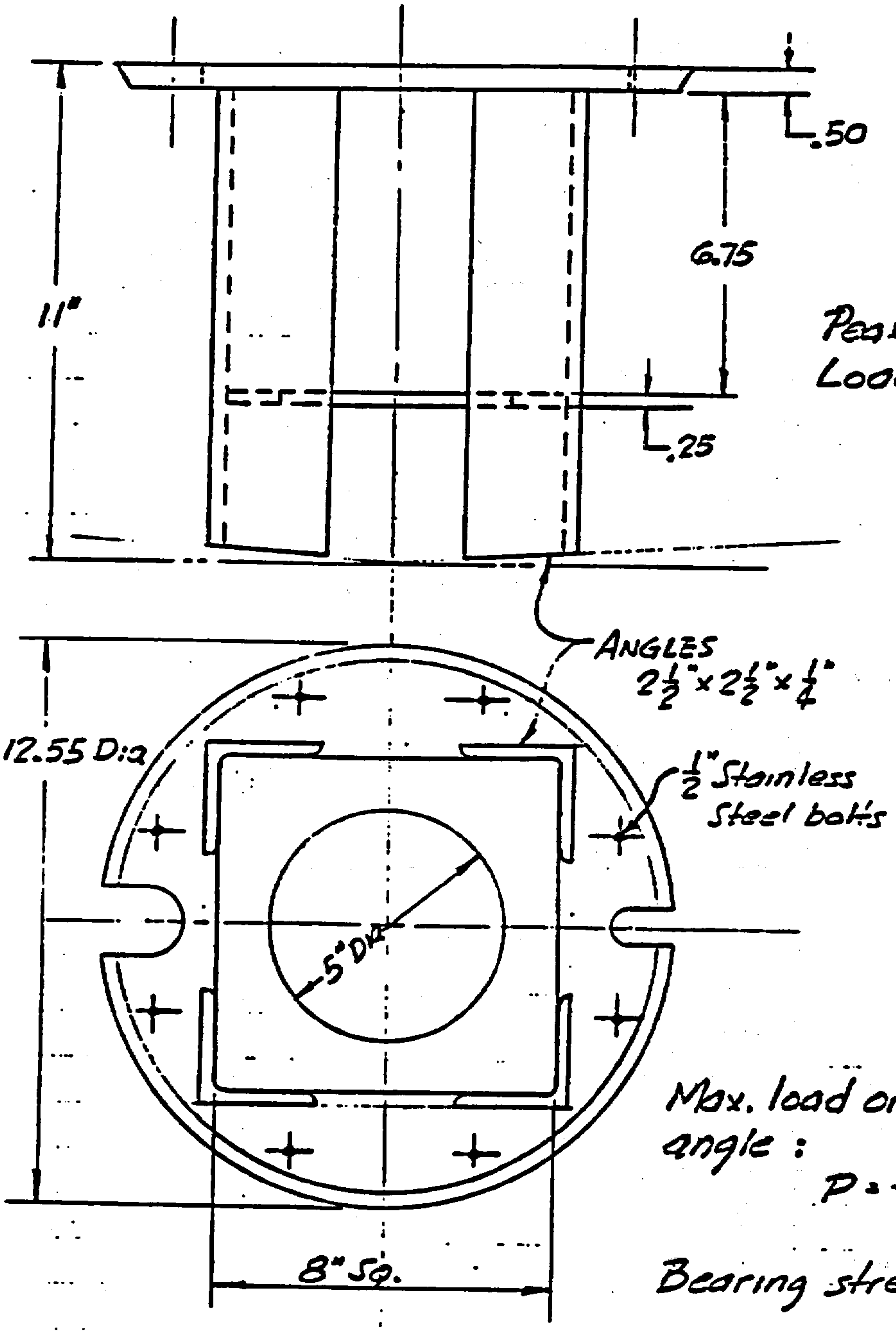
$$\frac{50000}{30000} \times 16500 = 27500 \text{ psi for 304 Stn. Stl.}$$

2.1.3.4 PWR AND BWR FUEL BASKETS

Since the fuel baskets are of different geometries separate analyses are required. In both cases the baskets will not fail nor will there be a change in the relationship between the fuel assembly and the cask internals.

The worst conditions will be defined as the dynamic loads generated during the impact from a 30 foot end drop.

2.1.3.4.1 PWR LOWER SUPPORT



Support material is type 304 stainless steel. Support is bolted to bottom of aluminum basket.

Peak g loading is 39.40 g

Load = 1600 lbs. fuel
794 lbs. basket
2394 lbs.

Max. load on single supporting angle :

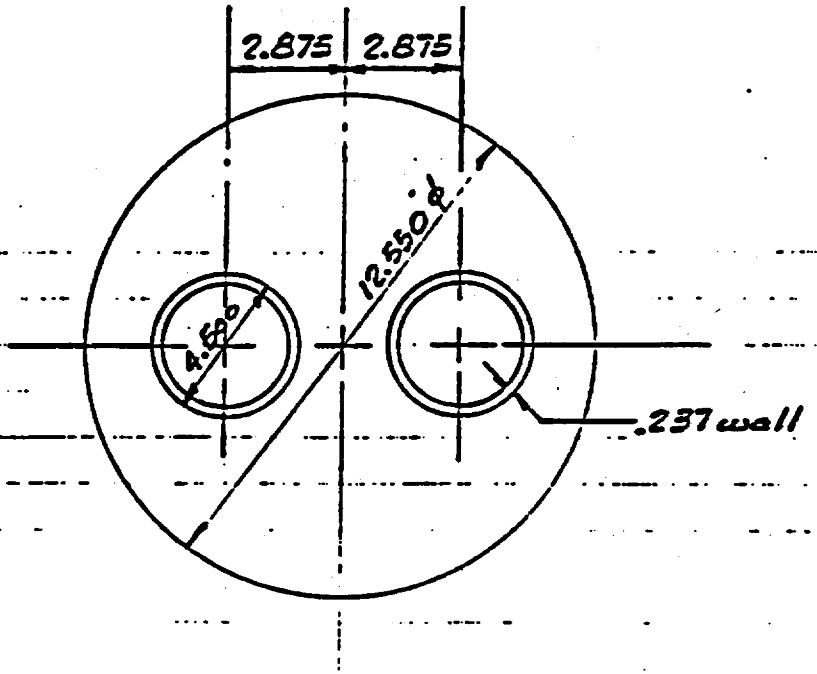
$$P = \frac{2394}{4}(39.40) = \underline{23581\text{ lb}}$$

Bearing stress on end of angle

$$\sigma_{BR} = \frac{23581}{.25(2.25 + 2.50)} = \underline{19858\text{ psi}}$$

$$M.S. = \frac{30000}{19858} - 1 = \underline{.51}$$

2.1.3.4.2 BWR LOWER SUPPORT



Peak G loading is 39.40 g's
Load = 1380 fuel
1000 basket
2380 lbs.

Maximum load per support cylinder

$$\frac{2380}{2} \times 39.40 = 46886 \text{ lbs.}$$

Bearing area of support cylinder

$$.785(4.5^2 - 4.026^2) = 3.173 \text{ in}^2$$

$$S_{br} = \frac{46886}{3.173} = \underline{14777 \text{ psi}}$$

$$M.S. = \frac{30000}{14777} - 1 = \underline{1.03}$$

.2.1.4. TOP END IMPACT ON CASK PARTS

2.1.4.1 URANIUM SHIELDING BEARING ON UPPER FLAT END

$F=(17,940)(39.40) = 706,836$ lbs.

End area $= .785(19.98^2-17.88^2) = 62$ in.^{2}

$S_c = \frac{706836}{62} = \underline{11400 \text{ psi}}$ on Stn. Stl. forging $\quad M.S. = \frac{30000}{11400} - 1 = \underline{1.63}$

2.1.4.2 INNER CLOSURE - STUDS & CLOSURE PLATE

CONFIGURATION "A"

Total load on studs 3041.83 Inner Container
665.69 Inner Closure
3707.52 lbs.

$F=(3707.52)39.40 = 146,076$ lbs
This load is taken by 8 bolts - 1"-8 @ 105000 psi Y.P.
Stress area $= 8 \times .6051 = 4.8408$ in^2

$S_t = \frac{146076}{4.8408} = \underline{30176 \text{ psi}} \quad M.S. = \frac{105000}{30176} - 1 = \underline{2.48}$

Bolts are SA-193-B7 ; 105000 psi Y.P. At 1100°F min. tempering temperature; strength = 63535 lbs.

Shear on closure plate. Weight of Uranium = 390.67 lbs

$F = (390.67)\ 39.40 = 15,392$ lbs.

Shear area .5" stn. stl. plate $= A_s = \pi(14.625).5 = 23$ in^2

$S_s = \frac{15392}{23} = \underline{669 \text{ psi}} \ \underline{< 18000 \text{ psi}}$

2.1.4.3 INNER CLOSURE HEAD SEAL-CONFIGURATION "A"

The hollow metal "O"-ring seal on the inner closure head will not be damaged by shock or temperature effects during the accident damage conditions. Excessive deflection of the seal is prevented by the metal to metal contact of the flange faces. The hollow metal "O"-ring material is Inconel X which has a tensile strength of 120000 psi at the fire accident temperatures.

The bolts retaining the inner closure are torqued to a clamping force equivalent to 127 "g's", assuring an effective seal under impact.

Twelve bolts at 80000 psi stress (1-8thds) provide a clamping force of 12 x .5608 x 80000 = 538368 The load required to compress the metal "O" ring is 67720 lbs. The net clamping force = 470648 lbs. For a total weight of container and closure of 3707.52 lbs., this represents a restraint of

$$\frac{470648}{3707.52} = \underline{126.9 \text{ g's}}$$

2.1.4.4 INNER CLOSURE HEAD VALVES AND PIPING CONFIGURATION "A"

The valves and piping located on the inner closure head can be subject to (1) peak dynamic loading of 39.40 g on top impact, and (2) vibrational stress of normal transport.

The root section of the pipe welded into the closure head is .840 O.D. x .622 I.D., with section modulus of

$$Z = \frac{.049(.840^4 - .622^4)}{.42} = .0406 \text{ in}^3$$

A gross moment is conservatively taken as 1.5 lbs x 7.5 in = 11.25 in. lbs.

The bending stress is $\frac{39.40 \times 11.25}{.0406}$ = 10 917 psi

While this is satisfactory as a fatigue stress under possible, but very high frequency, vibration conditions, a supporting plate near the end connection of the pipe assembly prevents such repeated stress.

2.1.4.3 OUTER CLOSURE HEAD

See page XI- B17

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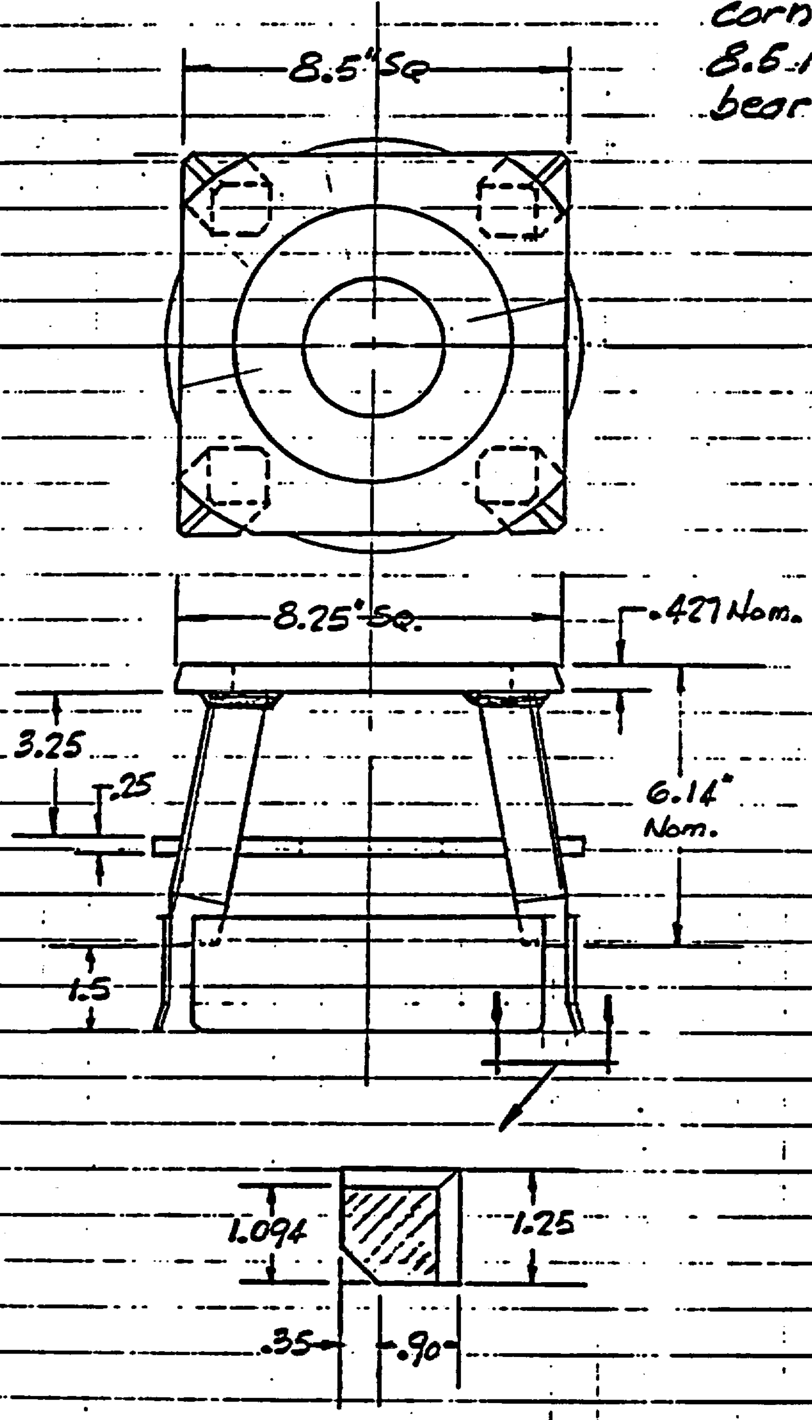
2.1.4.4 PWR TOP SUPPORT

Westinghouse Fuel

Support material is type 304 stainless steel. The support is basically four 1.25 inch square bars arranged in a slightly sloping 12½° attitude as corner posts based on an 8.6 inch square. Net end bearing area is 1.136 in^2/ba



Load = 1600 lbs. fuel
794 lbs. basket
2394 lbs.

Peak g loading is 39.40 g

Max. load on one bar
(corner post)

$$P = \frac{2394}{4}(39.40) = 23581\,l$$

Analysis of the stress conditions are now made by two concepts of the system behavior.

CASE "A" Even though inclined, each post is stabilized by frictional loads developed at its ends and so takes a direct compression load.

tan $12\frac{1}{2}°$ inclination = .22169

coefficient of friction, metal to metal, grease free, in air = .39

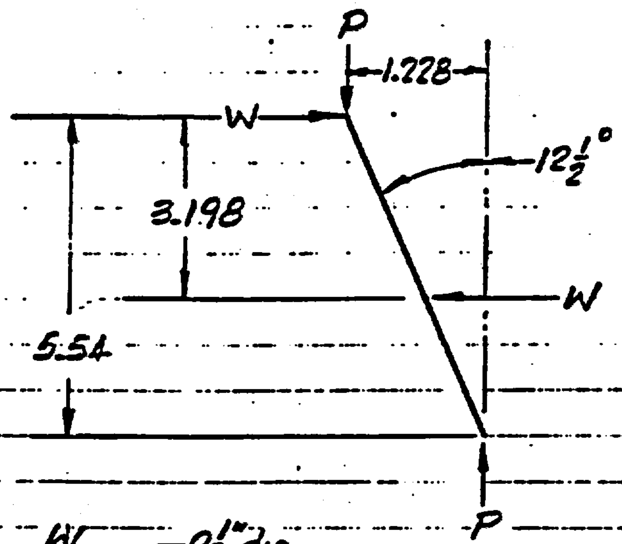
$$M.S. = \frac{.39}{.22169} - 1 = .759 \text{ for stability}$$

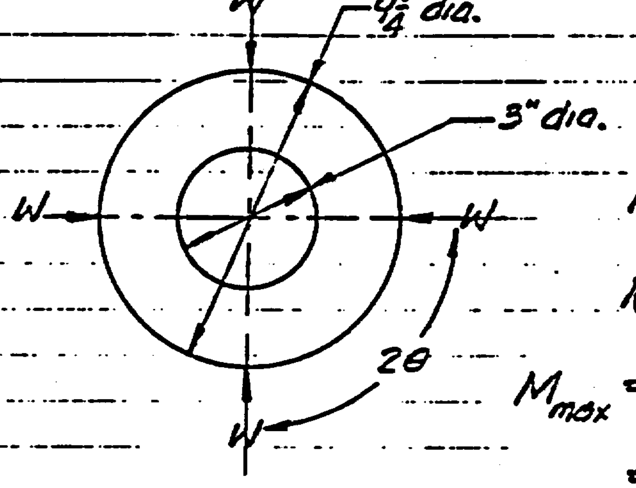
Bearing area = 1.136 in^2/bar

$$S_c = \frac{23581}{1.136} = 20758 \text{ psi}$$

$$M.S. = \frac{30000}{20758} - 1 = .445$$

CASE "B" Neglecting the benefits of end frictio, assume the moment created by vertica end loads is resisted by a horizontal couple present in the upper end ring and in the intermediate .25 inch thick tie-plate. The latter is essentially a ring loaded at four (4) points by radial tension loads at each inclined post, thus stabilizing the four posts, and allowing the direct stress calculated in Case A. The .25 inch plate is most heavily stressed





Formulas taken from Ref. 1
Table VIII, Case 9

Ring section = .25 x 3.125

$\theta = 45°$

Ring radius R = 3.0625" mean

$$M_{max} = \frac{WR}{2}\left(\frac{1}{\theta} - \cot\theta\right) = W\frac{3.0625}{2}\left(\frac{1}{.7854} - 1\right.$$

$$= W(.4183)$$

$$Z = \frac{.25\,(3.125)^2}{6} = .4069\ in^3$$

$$S_b = \frac{W(.4183)}{.4069} = W(1.028), \quad W = \frac{30000}{1.028} = 29183\ lbs.\ max.$$

Resisting moment = 3.198 W = 93327 in.lbs.

Applied moment = 1.228 P = 1.228(23581) = 28357 in.lbs.

$$M.S. = \frac{93327}{28357} - 1 = 2.29$$

Combustion Engineering Fuel - Top Support

The Combustion Engineering fuel top support is bolted to the inner closure head and extends down to support the fuel at the top surface of the top nozzle. The control rods when shipped with the fuel will extend up between 1 3/4 x 2 inch bars of the top support. The support material is type 304 stainless steel.

During top end impact the top support is loaded in compression.

Fuel wt. = 1381 lbs.
Basket wt. = 794 lbs.
Total = 2175 lbs.

Peak g loading is 39.4 g's

$$\text{Max load on one bar} = \frac{39.4(2175)}{4}$$

$$= 21423 \text{ lbs.}$$

$$\text{Bar area} = 2 \times 1.75 = 3.5 \text{ in.}^2$$

$$\text{Compression stress} = \frac{21423.75}{3.5}$$

$$= 6120 \text{ psi}$$

$$\text{M.S.} = \frac{30000}{6120} - 1 = 3.9$$

The length of the spacer plug is 17 inches when used with the inner container. For the alternate configuration without the inner container, the length will be 22 5/8 inches.

-J

FIGURE WITHHELD UNDER 10 CFR 2.390

Combustion Engineering Fuel Top Support

X[-2-13d

2.1.4.5 BWR FUEL

No top support is required for shipment of the design basis BWR fuel elements (nominal overall length of 176.16 inch) in cask. Under normal conditions of transport there will be an axial clearance of approximately 1 7/8 inches between the fuel and the cask cavity. Under the accident condition of a top end impact the fuel could displace axially about 6 inches if the fuel handle should crush. Movement of the fuel through the maximum distance of nearly 8 inches will not affect safety of the package with respect to either criticality or shielding. As shown in the criticality analysis of Section X, two BWR fuel elements either dry or water moderated have a sufficiently low multiplication factor that criticality can not occur and no posion material is required. Hence, axial position of the fuel is not significant for criticality control. Under either normal or accident conditions the active fuel zone will remain well within the confines of the aluminum basket and also within the cask region containing gamma and neutron shielding. Shielding is thus unaffected by the maximum possible axial fuel movement.

FIGURE WITHHELD UNDER 10 CFR 2.390

along line d-e, where the mean stress is found from deformation c-a, or

$$\frac{2.4375}{16.625} = 14.6\%$$, which gives a stress of $S = 1850 - (.146)740 = 1742$ psi peak

Peak "g" value $= \dfrac{1742(855.3) + 309236}{4800} = \underline{37.48}$ $\underline{g\text{'s}}$

The extreme compaction occurs at point "q" on line b-q

$$10.6 + \frac{4.875}{2} = 13.0375"$$

$$\frac{13.0375"}{16.625} = .7836 = 78.36\%$$

This is less than the 85% at which the balsa begins to go solid, so the corner impact condition is without any non-linear behavior.

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2.3 Side Impact

Side impacts in both the 30 foot drop and the 1 foot drop are absorbed by the balsa wood rings encased in 1/8 inch aluminum shells which are assumed to buckle easily and so present no significant crushing force. Both balsa rings are 75 in. outside diameter.

The bottom annulus is 34.75 in. inside diameter and 10.44 in. wide, with 14.81" crush distance for 30 foot drop.

The top annulus is 32.5 in. inside diameter and 11.75 in. wide, with 13.4" crush distance for 30 foot drop.

The one foot drop is assumed, as a worst case to be absorbed only by the bottom annulus, with a 2.25" crush distance. Analysis of the top annulus would be closely similiar.

2.3.1 Balsa stress analysis

The balsa wood within each annulus is oriented to present end grain to radial forces, modified only by the 19° angularity of alternate layers of balsa planks. The peak (initial) end grain stress is determined to be 1850 psi, resulting in an effective radial stress of 1850 (cos 9.5°) = 1850 (.987) = 1826 psi for each layer

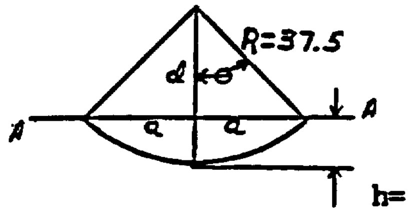
The effective stress expressed in relation to the percentage of deformation is:

S_c = 1826 - (% deform) 740 for central radial loading only

For off center loading as the penetration progresses, a cosine correction is made, reducing the 1826 psi value proportionately.

2.3.1.1 Calculation of crush distance, areas and volumes

using mean strengths of balsa as determined in 2.3.1.5 , obtained by trial and error methods between the two calculations for any single condition of side impact.



2.3.1.2 Top balsa annulus - 11.74" thick - 30 ft. drop

h = 13.4" d = 24.1" $\cos \Theta = \frac{24.1}{37.5} = .6428$ $\Theta = 50^\circ$

$\sin\Theta = .766$ $a = R \sin\Theta = 28.72$"

Side area $= \frac{2 \times 50^\circ}{360^\circ} (\pi 37.5^2) - 24.1 \times 28.72 = 535.03$ in.

Vol. $= 11.75 \times 535.03 = 6286.6$ in.^{3} "available"

Vol. req'd $= \frac{(360 + 13.4)\frac{48000}{2}}{1425 \text{ psi mean}} = 6288$ in.^{3} req'd

Contact area A-A $= 2 \times 28.72 \times 11.75 = 674.92$ in.^{2}

Final force on A-A = (1290 psi mean) x 674.92 = 870,647 lbs.

Final and peak g's $= \frac{870,647}{24000} = 36.28$ g's

2.3.1.3 Bottom balsa annulus - 10.44" thick - 30 ft. drop

h = 14.81" d= 22.69" $\cos\Theta = \frac{22.69}{37.5} = .60506$

$\Theta = 52^\circ 46'$

$\sin\Theta = .79618$ $a = R \sin\Theta = 29.857$"

Side area $= \frac{2 \times 52.767^\circ}{360} (\pi\, 37.5^2) - (22.69 \times 29.857)$

$= 1295.10 - 677.46 = 617.64$ in.^{2}

Vol. = 10.44 (617.64) = 6448.16 in.^{3} "available"

$$\text{Vol. Req'd} = \frac{(360 + 14.81)\ \frac{48000}{2}}{1397.6 \text{ mean}} = \frac{8,995,440}{1397.6} =$$

6436.345 in.^{3}

Contact Area = 2 x 29.857 x 10.44 = 623.414 in.^{2}

Final Force = (1235 psi mean) x 623.414 = 769,916 lbs.

$$\text{Peak g's} = \frac{769,916}{24000} = 32.08 \text{ g's}$$

2.3.1.4 Top balsa annulus = 11.75" thick - 1 foot drop

$h = 1.2"$ $d = 36.30"$ $\cos\theta = \frac{36.30}{37.5} = 0.94$ $\theta = 14.533°$

$\sin\theta = 0.25095$ $a = R\sin\theta = 9.4102"$

$$\text{side area} = 2 \times \frac{14.533}{360} \times (\pi\ 37.5^2\ ) - (9.4102 \times 36.30)$$

$$= 356.694 - 341.59 = 15.104 \text{ in.}^2$$

Vol. 11.75 (15.104) = 177.472 in.^{3} "available"

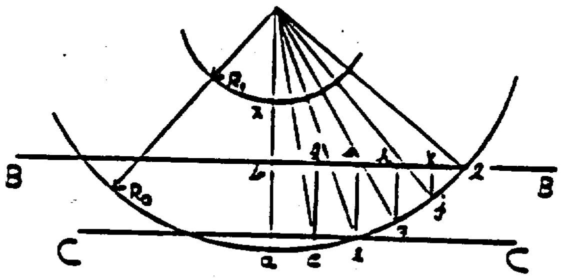
$$\text{Vol. Req'd} = \frac{(12 + 1.20)\ (24000)}{1789.44} = 177.039 \text{ in.}^3$$

Contact area = 2 x 9.4102 x 11.75 = 221.14 in.^{2}

Final force = (1777.27 psi mean) 221.14 = 393,026 lbs.

$$\text{Peak g's} = \frac{461,920}{24000} = 16.38 \text{ g's}$$

2.3.1.5 Calculation of mean psi for balsa side crushing



a-x = radial 100% thickness of balsa

a-b = radial crush dist.(actual max.)

$$\text{\% of crush on center radius} = \frac{a-b}{a-x}$$

(for point b)

Revised
Apr. 1988
Oct. 1990

% of crush on 10° impact point $= \frac{c-d}{a-x}$

etc., etc.

Stress at any point a, c, e, g, j, and l is $S = (1826 \text{ psi}) \cos\Theta$ for that point.

Stress at any point b, d, f, h, k is derived as shown now for point d:

$$S_d = (\text{stress at c}) - \frac{c-d}{a-x} \quad (740)$$

2.3.1.5.1 Top balsa annulus - $37.5 R_o$ - $16.25 R_1$ - 30 ft. drop

a-x = 21.25 a-b = 13.4" crush

at point a, stress = 1826 psi

b, $1826 - \frac{13.4}{21.25} \times 740 =$ 1360

mean a b = 1593

c, $(1826) \cos 10^{\circ} = 1798$

d, $1798 - \frac{12.6}{21.25} \times 740 =$ 1359

mean c d = 1578.

e, $(1826) \cos 20^{\circ} = 1716$

f, $1716 - \frac{11}{21.25} \times 740 =$ 1333

mean e f = 1524.

g, $(1826) \cos 30^{\circ} = 1581$

h, $1581 - \frac{8}{21.25} \times 740 =$ 1303

mean g h = 1442

j, $(1826) \cos 40^{\circ} = 1399$

k, $1399 - \frac{4.4}{21.25} \times 740 =$ 1246

mean j k = 1322.

l, $(1826) \cos 50^{\circ} = 1174$

Mean at impact line BB is $\frac{b + 2d + 2f + 2h + 2k + 21}{11}$

$= 1290$ psi

Mean of whole vol. crushed is =

$$\frac{ab + 2cd + 2\,ef + 2gh + 2jk + 21}{11}$$

$= 1425$ psi

2.3.1.5.2 Bottom of balsa annulus 37.5 R_o - 17.375R_1-30 ft.dr

a-x = 20.12 a-b=14.81" crush

a 1826 psi

mean
a b = 1553.

b $1826 - \frac{14.81}{20.12} \times 740 = 1281$

c 1798

mean
c d =1540.5

d $1798 - \frac{14}{20.12} \times 740 = 1283$

e 1716

mean
e f =14

f $1716 - \frac{12.2}{20.12} \times 740 = 1267$

g 1581

mean
g h = 1408

h $1581 - \frac{9.4}{20.12} \times 740 = 1235$

j 1399

mean
j k =1296

k $1399 - \frac{5.6}{20.12} \times 740=1193$

l 1174

Mean at impact = 1235 psi

Mean for whole vol. 1397.6 psi.

2.3.1.5.3- Top balsa annulus 37.5 R_o - 16.25 R_1 - 1 ft. drop

a-x = 21.25"   a-b =1.20" crush

		Mean
a	1826	
b	$1826 - \frac{1.20}{21.25} \times 740 = 1784.2$	a b= 1805.1
c	1822.3	
d	$1822.3 - \frac{1.12}{21.25} \times 740 = 1783.2$	c d= 1802.8
e	1811.3	
f	$1811.3 - \frac{0.90}{21.25} \times 740 = 1780$	e f= 1795.7
g	1793.1	
h	$1793.1 - \frac{0.523}{21.25} \times 740 = 1774.8$	g h= 1783.9
i	1767.6	
j	1767	i j= 1767.6

Mean at impact line CC is :

$$\frac{1784.2 + 2\ 1783.2 + 1780 + 1774.8 + 1767.6)}{9} =$$

1777.27 psi

Mean of whole volume is :

$$\frac{1805.1 + 2(1802.8 + 1795.7 + 1783.9 + 1767.6)}{9} =$$

1789.44 psi

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2.3.2 CASK BENDING STRESSES

Due to the side impact being absorbed by widely spaced end caps, there are developed large stresses in the various cylindrical components. These stresses, however, do not exceed the yield points (static values) of the materials used and therefore the cask is not at all damaged in side drop.

The construction to be analyzed consists of two bodies which behave somewhat differently but which contribute to the total moment resisting ability of the total cask.

The water jacket is the extreme outer cylindrical member which can be made to absorb elastic energy. This has been insured by so welding it to the massive end forgings of the cask body proper that proportionate resisting moments can be developed within this shell.

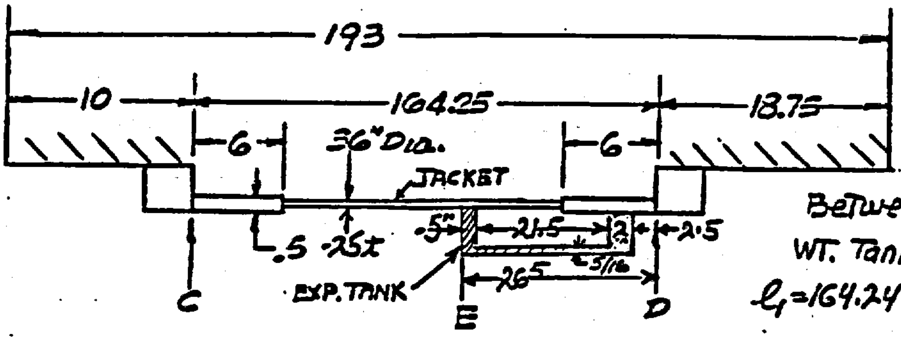
The water jacket is loaded transversely by its water and the weights of the jacket shell and the expansion tank shell. Consequently its elastic curve and its end slopes will not be the same as those for the cask proper, when considered as free bodies.

The various shells constituting the cask proper are all constrained to develope the same elastic curves, the deflections of nested cylinders being the same at any point for all members. Therefore it is possible to divide a total moment impressed on the cask among the various nested cylinders in proportion to their individual EI values.

The analysis is made for 1g loading and at the end, conversion is made to actual stresses at 36.28g's

The water jacket analysis produces an end slope different from that of the independent cask analysis. Since both are constrained to have the same slope, both being welded to the end forgings, corrective moments are applied to each to equalize these slopes.

2.3.2.1 WATER JACKET ANALYSIS - Independent - 1g



WT. of jacket solution (X1-1-10) = 3112

Wt. of jacket = 1438.

Between C & D, $W_1 = 4550$

WT. Tank conc. at E, $W_2 = 313$

$\ell_1 = 164.24$ $\ell_2 = 26.5$

Jacket - .25 thk. stainless steel type 216 ; 55000 psi Y.P.

$EI = 135.384 \times 10^9$

$Z = 255.8 \text{ in}^3$ Area $= 28.4 \text{ in}^2$

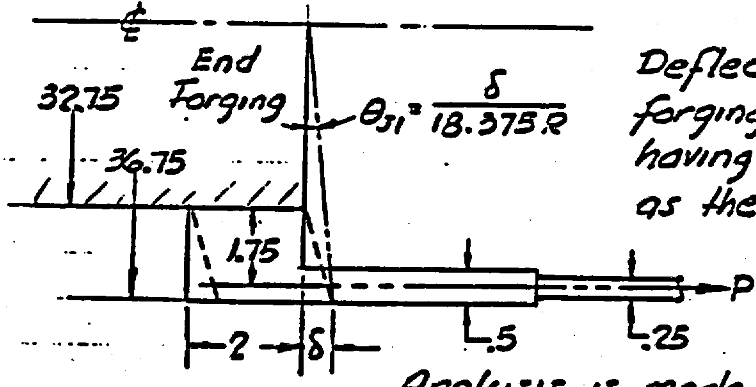
Assume simple beam between C-D, with 4550 lbs. distributed and 313 conc. load at E. Then max. moment at center of jacket

$$M_{MAX} = \frac{W_1 \ell_1}{8} + W_2\left(\frac{\ell_2}{\ell_1}\frac{\ell_1}{2}\right) = \frac{4550(164.24)}{8} + \frac{313(26.5)}{2} = 93412 + 4147$$

$M_{c_1} = 97,559$ in. lbs. For convenience, let $W_3 = \frac{97559 \times 8}{164.25} = \underline{\underline{4752 \text{ lbs}}}$

$$\theta_J \text{ at ends} = \frac{1}{24}\frac{W\ell^2}{EI} = \frac{1}{24}\frac{(4752)(164.25)^2}{135.384 \times 10^9} = .000,039,456$$

Correction for θ_J due to flexibility of end anchorages



Deflection of the end rings on the forgings prevents the jacket from having exactly the same end slope as the other cask cylinders. The differential angle is

$$\theta_{J1} = \frac{\delta}{18.375R}$$

Analysis is made for an element 1" wide at the position of max. bending stress

(Ref. 5, pg. 175)

$$\delta = \frac{P\ell^3}{3EI}\left[1 + .71\left(\frac{h}{\ell}\right)^2 - .10\left(\frac{h}{\ell}\right)^3\right]$$

$h = 2"$, $\ell = 1.75$, $h/\ell = 8/7$, a final evaluation (X1-2-21) gives $P = 1000$ psi

$$= \frac{250(5.359)}{60 \times 10^6}\left[1 + .71\left(\frac{64}{49}\right) - .10\left(\frac{512}{343}\right)\right]$$

$P = (.25 \times 1)1000 = 250$ lbs

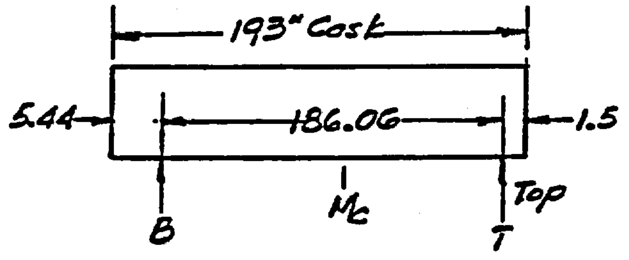
$3EI = 3(30 \times 10^6)\left(\frac{1 \times 2^3}{12}\right) = 60 \times 10$

$\delta = .000\,038,908"$

$\theta_{J1} = \frac{\delta}{18.375R} = .000\,003117$

$\theta_{J2} = \theta_J - \theta_{J1} = \underline{.000,037,3385}$

2.3.2.2 CASK ANALYSIS - Without jacket or impact limiters



Design weight = 48000 gross
− 6133 for jacket & impact limiters
41867 lbs.

EI × 10^9 (for rigidity)
15.626 - 1/2" inner shell
136.209 - 2 3/4 uranium
18.598 - 2 1/8 lead
157.957 - 7/8 outer shell
328.39 total for cask

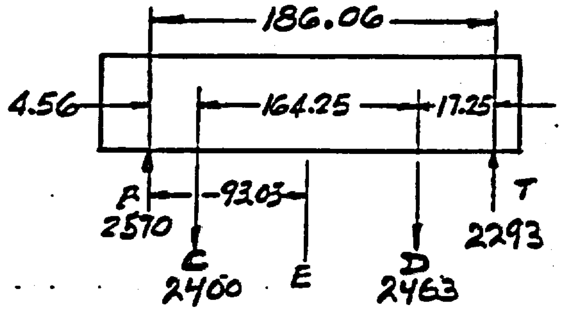
$M_c = \frac{WL}{8} = \frac{41867\,(186.06)}{8} = \underline{973722}$ in

Slope at B and T as simple beam

$\theta_1 = \frac{1}{24}\frac{WL^2}{EI} = \frac{1}{24}\frac{41867\,(186.06)^2}{328.39\times 10^9}$

$= \underline{.000\,1839}$ _ Ref. 1, Table III, Case 13

2.3.2.3 ADDITIONS OF SHEAR LOAD FROM JACKET TO CASK



Jacket & Tank = 4863 at C & D

Reaction at T = $\frac{2400(4.56) + 2463(168.81)}{186.06}$

T = 2293.

B = 4863 − 2293 = 2570

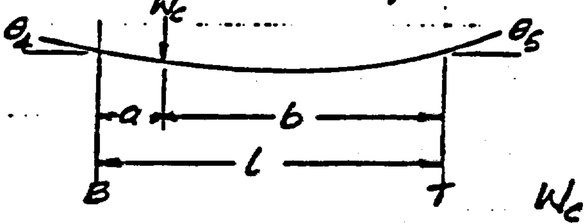
M_c = 2570 (4.56) = (11719. in. lbs.

M_D = 2293 (17.25) =)39,554. in. lbs.

Moment midway between B and T, at E

M_E = 2570 (93.03) − 2400 (88.47) = 26,159 in. lbs.

Increase in slope from load C . Ref. 1, Table III, Case 12



$\theta_4 = -\frac{1}{6}\frac{W}{EI}\left(bl - b^3/l\right)$ at B

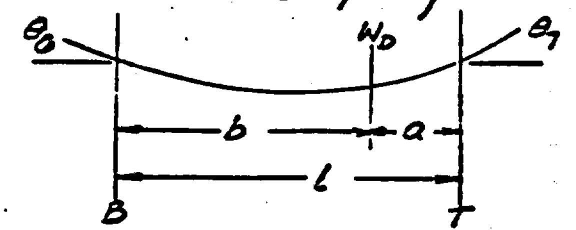
$\theta_5 = \frac{1}{6}\frac{W}{EI}\left(2bl + b^3/l - 3b^2\right)$ at T

$W_c = 2400$, $a = 4.56$, $b = 181.50$, $l = 186.06$

$\theta_4 = \frac{1.218}{10^9}(1635) = .000,00199$ at B

$\theta_5 = \frac{1.218}{10^9}(847.92) = .000\,001,033$ at T

Increase in slope from load D



$W_D = 2463$, $a = 17.25$, $b = 168.81$, $l = 186.0$

$\theta_7 = \frac{1.25}{10^9}(5553.97) = .000{,}006{,}942$ at

$\theta_6 = \frac{1.25}{10^9}(3181.95) = .000\ 003977$ at

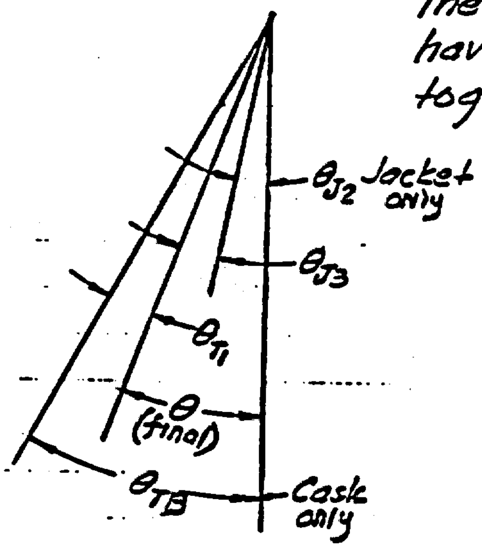
Total Slope at B:

$$\begin{array}{rl} \theta_1 = & .000\ 183\ 900 \\ +\ \theta_4 = & .000\ 001\ 990 \\ +\ \theta_6 = & .000\ 003\ 977 \\ \hline \theta_B = & .000\ 189{,}867 \text{ Cask} \end{array}$$

Total Slope at T:

$$\begin{array}{rl} \theta_1 = & .000\ 183\ 900 \\ +\ \theta_5 = & .000\ 001\ 033 \\ +\ \theta_7 = & .000\ 006\ 942 \\ \hline \theta_T = & .000\ 191\ 875 \text{ Cask} \end{array}$$

Slopes at B and T are close; use mean $\theta_{TB} = .000{,}190{,}871$



The separate end slopes for cask and jacket have been calculated. Now, by welding all together to the end forgings, slope θ will result; θ_{TB} being decreased by θ_{T1}, and θ_{J2} being increased by θ_{J3}.

The moments to equalize at θ are equal.

For end moments $\theta = \frac{Ml}{2EI}$ or $M = \frac{2\theta EI}{l}$, $l = 186.0$

M = equalizing moment = Mcask = Mjacket

$$M_{cask} = 2(\theta_{T1})\frac{EI_{cask}}{l} = \frac{2\theta_{T1}(328.39 \times 10^9)}{l}$$

$$M_{jacket} = 2(\theta_{J3})\frac{EI_{jacket}}{l} = \frac{2\theta_{J3}(135.384 \times 10^9)}{l}$$

$$\theta_{T1} = \theta_{J3}\frac{135.384}{328.39} = .41226\,\theta_{J3}$$

$$\theta_{T1} + \theta_{J3} = 1.41226\,\theta_{J3} = \theta_{TB} - \theta_{J2} = .000{,}190{,}871 - .000{,}037{,}3385 = .000{,}15$$

$$\theta_{J3} = \frac{.000{,}153{,}5325}{1.41226} = .000{,}108{,}714$$

$$M = \frac{2\theta_B(135.384\times10^9)}{186.06} = \frac{2(.000,108,714)\,.(135.384)10^9}{186.06} =$$

M= 158,208 in.lbs. This is constant from B to T

Resulting moments at center of cask

For cask proper M_c = 973,722 from 2.3.2.2

$-M$ = 158,208

net M_c = 815,514 in.lbs.

For jacket M_{c_1} = 97,559. from 2.3.2.1

$+M$ = 158,208

net M_c = 255,767 in.lbs.

Resultant max. bending stresses at midlength of cask & jack:

For maximum <u>stress</u> determination, the lead is neglected and the minimum section of the uranium is the 2 3/8" weld.

	EI×10⁹	% M	M	Z	σ_b (1g)	σ_b 36.28g	Y.P.
1/2 inner shell	15.626	.052873	43119.	74.8375	576.17	20903	30000
2 3/8 U weld	121.955	.412654	336525.5	511.343	658.12	23877	35000
7/8 outer shell	157.957	.534472	435869.5	418.984	1040.3	37742	55000
	295.538	1.0	815,514.				
1/4 Jacket			255767 (as above)	255.8	999.9	36276	55000
			1,071,281				

2.3.3 DEFLECTION AT CASK CENTER

The slope of the cask ends at 36.28 g's is 36.28 θ_{TB}

$\theta_{40} = (36.28)(.000\,190\,871) = .006925$

For a uniformly loaded beam

$$\text{Max. } y = \frac{5}{384}\frac{WL^3}{EI} \quad \text{and} \quad \theta = \frac{1}{24}\frac{WL^2}{EI}$$

$$\text{or } y = \frac{120}{384}\theta L = \frac{120}{384}(.006925)\,186.06 = \underline{.403"}$$

2.3.4 SIDE IMPACT ON CLOSURE HEADS

2.3.4.1 INNER CLOSURE HEAD

The rim of the container and the inner closure head both have a small clearance within the cask head forging and are directly supported on it for lateral loa

The contact area for the inner closure is the more heavily loaded.

Projected contact area = 1.75(18.125) = 31.72 in^2

Weight of inner closure head = 665.69 lbs.

Side drop "g" force = 36.28 (665.69) = 24151 lbs.

Mean lateral bearing stress =

$$\sigma_{BR} = \frac{24151}{31.72} = \underline{761.38} \ \underline{psi}$$

$$M.S. = \frac{30000}{761.38} - 1 = 38.4$$

2.3.4.2 OUTER CLOSURE HEAD

See page XI-B21

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2.4 PUNCTURE

2.4.1 ANALYSIS PER CASK DESIGNER'S GUIDE

Formula 2.1

$$t_{min.} = \left(\frac{1.3W}{S}\right)^{.71}$$

W = 48000 lbs
S = 100000 psi for Type 216 S/S

$$t_{min.} = \left(\frac{1.3\times48000}{100000}\right)^{.71} = (.624)^{.71} = \underline{.715" \ min.}$$

Actual shell thickness = .875" outer shell of cask only.

2.4.2 ANALYSIS OF BENDING OF CASK ON 40" PIN DROP

The greatest damage from a drop of 40" upon a 6" diameter vertical pin would be caused by impact at cask mid-length, which would possibly cause the cask to take a very slight permanent set as the result of a plastic hinge.

The total kinetic energy of the fall is

48000(40) = 1,920,000 in. lbs.

This energy will be absorbed in three sucessive steps:

1) Indenture of the water jacket until contact is made with the body of the cask outer shell proper, with absorption of energy elastically by hydraulic pressure which stretches the cylindrical water jacket radially.

2) Elastic bending of the cask as a double cantilever upon the 6" diameter pin.

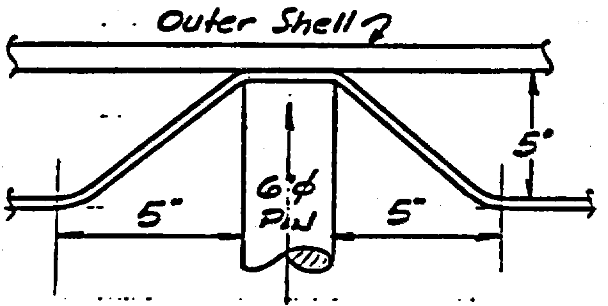
3) Plastic hinge absorption of the remaining K.E.

STEP 1 INDENTURE OF WATER JACKET

Material - Stainless Steel Type 216
Yield Point 55000 psi
Ultimate Tensile 100000 psi

Assume only 160 inches of the 164.25 inches of water jacket is stretched radially. The wall is .25 inches thick with a 36 inch I.D. No allowance is made for the dynamic increase in physical properties. This is conservative, since with higher physicals more energy would be absorbed here, with less available for plastic hinge formation.

Assume the 6 inch diameter pin pushes the jacket wall 5 inches to contact the outer cask shell. Very conservatively, a volume of water, equal to 506 in^3, is displaced as shown. Allowing for 6 in^3 release, the net displacement is 500 in^3.



To check the volume passed by two orifices, the velocity of fall is: $v = (2gh)^{\frac{1}{2}} = (2 \times 386 \times 40)^{\frac{1}{2}} = 176$ in./sec.
5 inches is traversed in:

$t = 5/176$ sec $= .0284$ sec.

The two orifices have an area of $2(.028\,in^2) = .056\,in^2$
Flow rate is 2×10 gpm @ 900 psi pressure drop.

$$20\,gpm = \frac{20 \times 231\,in^3/gal.}{60\,sec.} = 77\,in^3/sec$$

In time 't' = .0284 sec, volume passed is:
$(.0284)(77) = 2.187\,in^3$ ($6\,in^3$ allowed)
The volume of 500 in^3 will distend the jacket.

$$\Delta_r = \frac{500\,in^3}{\pi\,36\,(160)} = .0276 \text{ inches}$$

The strain due to hoop stress is $2\pi(.0276) = .173$ inches.

The hoop stress is $\frac{.173}{\pi 36} \times 29 \times 10^6 = .0443 \times 10^6 = \underline{44300\ psi}$
OK < 55000 psi

The elastic energy stored in the jacket is:
$U_E = 44300 \times 160 \times .25 \times .173 = 306000$ in. lbs.

Remaining K.E. = 1,920,000 − 306000 = 1614000 in. lbs.
Hydraulic pressure developed is

$$S_2 = p\frac{R}{t} \quad ; \quad 44300 = p\frac{18}{.25} \quad ; \quad p = \frac{44300}{72} = \underline{615\ psi}$$

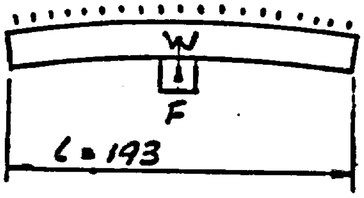
Bursting pressure is $P_u = 2S_u\frac{b-a}{b+a}$ $a = 18$; $b = 18.25$

$$P_u = 2(100000)\frac{.25}{36.25} = \underline{1380\ psi}$$

Force on pin required to shear .25 inch jacket
Area of 6 inch diameter pin = 28.72 in.^{2}
Area of jacket in shear = $.25\pi 6$ = 4.72 in^2
Shearing stress = .6(100000 UTS) = 60000 psi
Force required on pin to shear =
$4.72\ in^2(60000) = \underline{283200\ lbs.}$
Maximum force on pin from hydraulic pressure is −
Area 16 inch annulus (max.) = 172.73 in^3
$F_{max} = (172.73)(615\ psi) = \underline{106300\ lbs.}$
The jacket does not shear, but deforms.

STEP 2 ELASTIC BENDING OF CASK

The pin is now in contact with the cask proper, the water jacket having locally collapsed, but remaining unruptured, and there remains 1614000 in. lbs. to be dissipated.



The energy absorbed by the cask is

$$U_E = \frac{W^2 l^3}{640EI} \qquad l = 193$$

	$EI \times 10^9$	I	Z	Y.P.	$M_{elastic} = Z$ (Y.P.)
.5" inner shell	15.626	540 in^4	74.8375	30000	2265125
2.75" uranium	136.209	5680	568.	35000	19880000
.875" outer shell	157.957	5440	418.984	55000	23044000
	309.792×10^9	11670 in^4			45169125 in.

For 1g static loading

$$M_{center} = \frac{WL}{8} = \frac{48000(193)}{8} = 1158.000 \text{ in/lbs.}$$

At the Y.P., therefore the effective weight is increased by the ratio: $\frac{45169125}{1158000} = 39$

$$\text{Therefore } U_E = \frac{(39 \times 48000)^2 (193)^3}{640 (309.792 \times 10^9)} = 127068 \text{ in.lbs.}$$

STEP 3 PLASTIC HINGE FORMATION

Net remaining K.E. = 1614000 − 127068 = 1486932 in.lbs.
The plastic moment is 1.5 times the elastic moment at Y.P.

$M_{plastic} = 45169125(1.5) = 67753688$ in. lbs.

Energy absorbed by plastic hinge is $U_H = M_{P\ell}\,\theta$

$$\theta = \frac{1486932}{67753688} = 0.0219 \text{ radians}$$

Estimating the percent elongation at the local plastic hinge, the extreme fibers of the outer shell will be stretched very locally

$\theta R = .0219(13) = .2853$

Assume this occurs over a 2" gage length

$$\text{Strain} = \frac{.2853}{2} = \underline{.1426 \text{ in./in. or } 14.3\% \text{ elongation}}$$

Since the material has 45% elongation, there is no rupture, and the cask integrity is maintained.

2.4.3 Puncture - Top End - Center Impact

2.4.3.1 The top of the cask and the outer closure head in particular are completely protected in both the 30-foot free fall and the 40-inch pin drop by a 4-inch thick plate of 7075-T651 al. alloy plate, having 70,000 psi U.T.S. and 60,000 psi Y.P. and 48,000 psi shear. This plate retains its original strength properties because it is bolted rather than welded into the composite impact structure protecting the top end.

2.4.3.1.1 It is assumed that the top end has already been deformed 10.6" by flat end drop of 30 feet on the balsa, and that now the 6" pin pierces to 85% of balsa thickness during its initial energy absorption travel.

$$Y_1 = (.85 \times 16.375) - 10.6 = 3.32"$$

Balsa stress at 10.6" penetration is

$$\left.\begin{array}{l} 1850 - \dfrac{10.6}{16.375} \times 740 = 1368 \text{ psi} \\ \text{Balsa stress at 85\% penetration is} \\ 1850 - (.85)\ 740 = 1221 \text{ psi} \end{array}\right\} \text{mean} = 1295 \text{ psi}$$

Area end of 6" dia. bar is 28.274 in^2

KE_1 absorbed by penetration of 6" bar is

$KE_1 = (28.274)\ 1295$ psi $(3.32") = 121,561$ in lbs.

The loads and center deflection of the plate in absorbing this KE are calculated in three (3) stages, from the elastic to the fully plastic. The plate is 4 in. thick and 29 in. diameter at bolt holes.

2.4.3.1.2. Elastic phase

Roark X - Case 2

$$\text{Max } S_r = S_t = \frac{3W}{2\pi m t^2}\left[m + (m+1)\log\frac{a}{r_c} - (m-1)\frac{r_c{}^2}{4a^2}\right]$$

$$60{,}000 \text{ psi} = \frac{3W}{2\pi\, 10/3 \times 16}\left[\frac{10}{3} + \frac{13}{3}\log\frac{14.5}{3} - \frac{7}{3} \times \frac{3^2}{4 \times 14.5^2}\right]$$

$$= W(.0907)$$

$$W = \frac{60{,}000}{.0907} = 661{,}000 \text{ lbs.}$$

Elastic deformation $y_2 = \frac{Wa^2}{Et^3}(.5515)$

$$Y_2 = \frac{661{,}000\,(14.5)^2\,(.5515)}{10.5 \times 10^6 \times 4^3} = .114''$$

Energy of elastic deformation

$$KE_2 = \frac{661{,}000}{2} \times .114 = 37{,}700 \text{ in. lbs.}$$

2.4.3.1.3 Elastic to collapse phase

Collapse load = $W = Mp\left(\frac{6\pi}{3 - \frac{2a}{R}}\right)$ where $Mp = \frac{t^2}{4}$ (y.p.) Ref. 8 pg. 126

$$W = \frac{4^2}{4}(60{,}000)\left(\frac{6\pi}{3 - \frac{6}{14.5}}\right) = 1{,}745{,}000 \text{ lbs.}$$

assume load limit at 1,745,000 lbs. and 2% dia. = .02(29.) = .58"

$$\text{mean load} = \frac{1{,}745{,}000 + 661{,}000}{2} = 1{,}203{,}000 \text{ lbs.}$$

$$Y_3 = .58'' - .114'' = .466''$$

KE_3absorbed = 1,203,000 (.466) = 560,600 in. lbs.

2.4.3.1.4 Collapse Phase

This continues at W = 1,745,000 lbs. load until KE is exhausted, with a deformation of Y_4, determined as follows :

Total kinetic energy KE = 48,000 $(40" + 3.32 + .114 + .466 + Y_4)$ in lbs.

Successive depletions have been

$$121,561 + 37,700 + 560,600 = 719,861 \text{ in lbs.}$$

Remaining $KE_4 = 1,745,000 (Y_4)$

Equating, $48,000 (43.9" + Y_4) - 719,861 = 1,745,000 Y_4$

$Y_4 = .81752"$ giving

$KE_4 = 48,000 (44.71752") - 719,861 = 1,426,573$ in lbs.

2.4.3.1.5 Total Deformation of 4" Plate

$Y_2 + Y_3 + Y_4 = .114 + .466 + .81752 = \underline{1.39752 \text{ in}}$

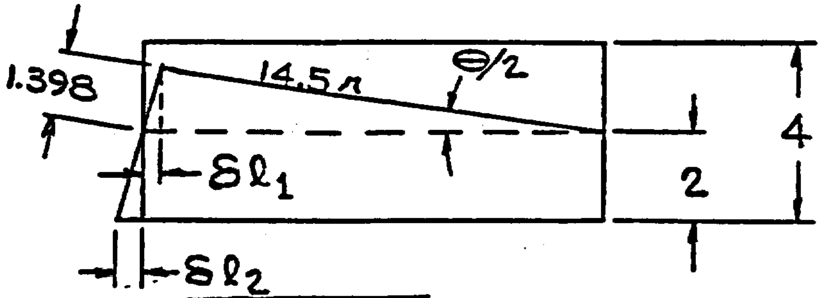
1.50 in. air clearance space available
before touching outer closure head.

2.4.3.2 Ductility Evaluations

The extension of the extreme fiber on the bottom of the plate and at its center can be represented by the sum of two separate extensions:-

$\delta\ell_1$ = elongation of neutral axis as plate is bent. This can be called a "membrane stretch" and assumed to be also applied to the bottom extreme fiber as well as the neutral axis.

$\delta\ell_2$ = elongation of the extreme bottom fiber beyond that calculated as $\delta\ell_1$ for the neutral axis, due to plastic hinge.



$$\delta\ell_1 = \sqrt{14.5^2 + 1.398^2} - 14.5 = .067$$

$$\delta\ell_2 = \frac{1.398}{14.5} \times 2" = .193$$

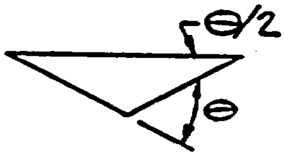
$\delta\ell_1 + \delta\ell_2$ = .260 on each side of the ℄ of disk.

For these dimensions (29" dia. x 4" thickness) an 8" gauge length is appropriate. On this basis, the % elongation is

$$\frac{2 \times .26"}{8" \text{ g.l.}} = 6.5 \text{ elongation}$$

The angle of bend of the plastic hinge which results is tan $\theta/2$ =

$$\frac{1.398}{14.5} = .0964 \qquad \theta/2 = 5.5$$



$\theta = 11°$ of bend which is satisfactory for this material

Examining the ductility ratio.

r '

The available "ductility ratio" $= \frac{\mathbf{11\%}}{.2\%} = 55.$

where 11% elongation is min. for 7075-T651

The used "ductility ratio" $\frac{6.j}{.2\%} = 32.5$

M.S. $= \frac{55}{32.5}$ **-1** $= .69$

Considering the ratio of $\frac{\text{final deflection}}{\text{elastic deflection}}$ $\frac{1.398'}{.114"}$ 12.26

This may be roughly related to the above ratio of 55. "available".

2.4.3.3 Shear in plate at 6" pin contact

Perimeter of Pin = 6 T7inches

Area of plate in shear = 4(6 7r) = 24 **7** in^2

Collapse load = 1,745,000 lbs.

Ss = 1,745,000 =23,166psi O.K. <48,000psi

FIGURE WITHHELD UNDER 10 CFR 2.390

Bearing on cylinder area = $29\text{-}3/8\pi\ 7/8 = 80.75$ in.^{2}

$$S_{BR} = \frac{1,745,000}{80.75} = 21,600 \text{ psi.}$$

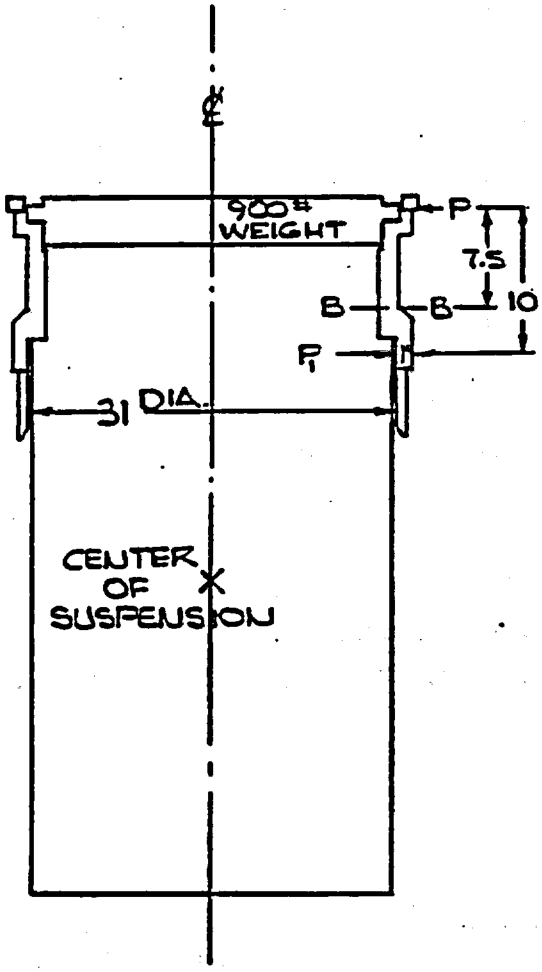
$$MS = \frac{42,000}{21,600} - 1 = .94$$

Cylinder Bearing at Cask End Area = $29\pi 3/4 = 68.33$ in.^{2}

$$S_{BR} = \frac{1,745,000}{68.33} = 25,500 \text{ psi.}$$

$$MS = \frac{42,000}{25,500} - 1 = .64$$

2.4.4 Puncture - Top End - Side Impact



The 6" dia. pin is considered impacting against the 4" plate at point P.

The cylinder between P and P_1 is maintained in position on the cask by frictional loading at P_1.

For a coef. friction= .61 between al. alloy and S.S cask the M.S. is

$$M.S - \frac{.61\ P_1\ (31)}{P\ (10)} - 1 = .79$$

The cask is now analyzed as a free body with a "center of percussion" at its extreme top end. The corresponding "center of suspension" is found, about which the whole body rotates instantaneously. The inertia loads, shears and max. moment and value of P are derived according to the following analyses.

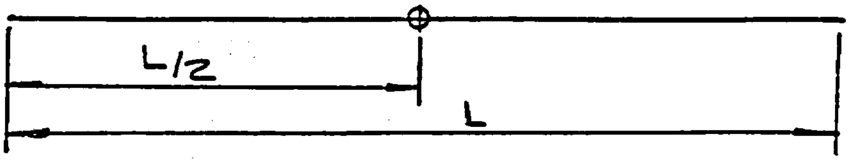


FIG. 1

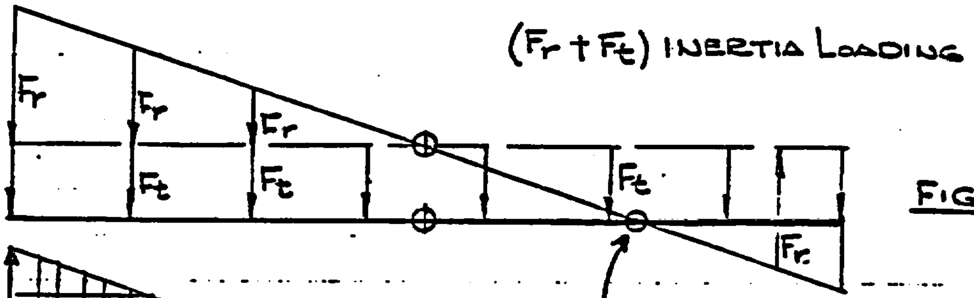


FIG. 2

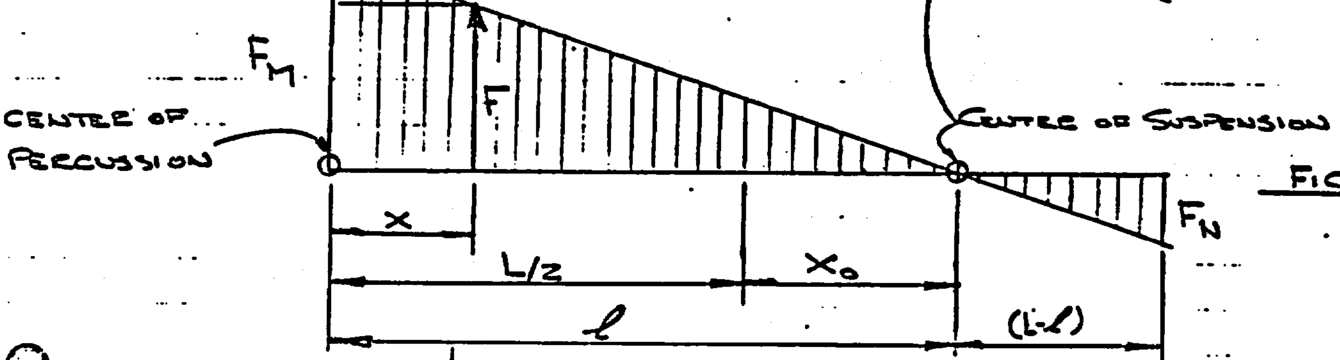
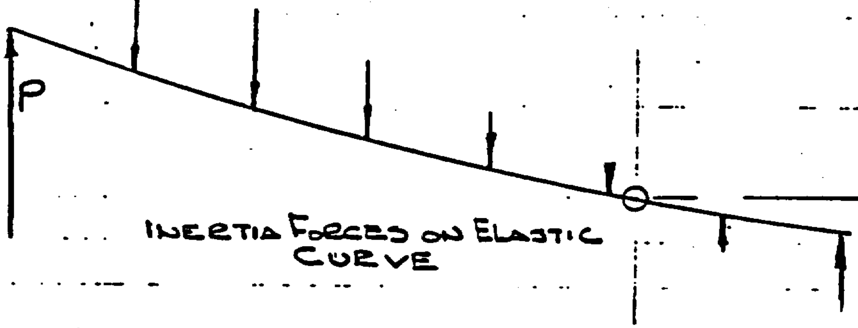


FIG 3



INERTIA FORCES ON ELASTIC CURVE

FIG 4

THE SHEAR DIAGRAM MUST SHOW P BEING PROGRESSIVELY REDUCED BY OPPOSING INERTIA FORCES, INCLUDING A REVERSAL OF SIGN TOWARD THE OPPOSITE END.

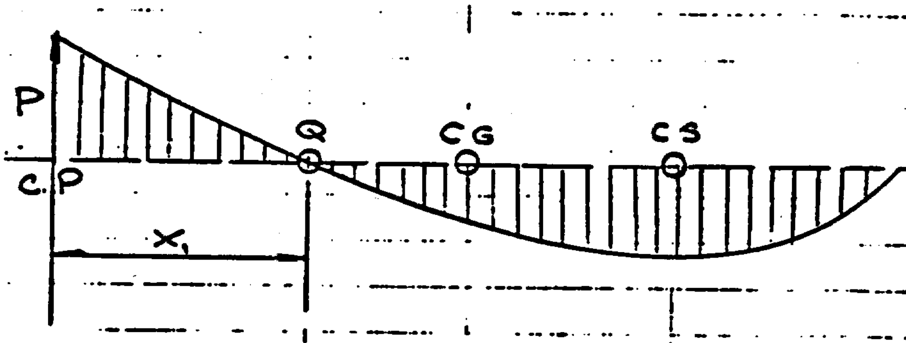


FIG 5
SHEAR DIAGRAM

MAX. MOMENT AT Q WHERE SHEAR = 0

2.4.4.1 Analysis of Center of Percusion at End of Cask Structure.

For this analysis a uniform section cask is assumed, with overall length = 196", and outside radius = 13".

By definition point "0" is the"point of suspension "about which the cask would rotate instantaneously if impacted by force P at its other extreme end.

$$\ell = \frac{I}{m x_o}$$

where m = mass of body r-radius of cylinder

L = length

ℓ = distance from center of percussion to "point

suspension "0" ".

I = moment of inertia about axis of suspension

X_o = distance from C.G. to axis of suspension.

$$I = I_{CG} + m x_o^2$$

$$I = \frac{m}{12}(3r^2 + L^2) + m x_o^2$$

$$\ell = \frac{\frac{m}{12}(3r^2 + L^2) + m x_o^2}{m x_o} = \frac{3r^2 + L^2}{12 x_o} + x_o$$

but $\ell = L/2 + x_o$

$$\frac{L}{2} = \frac{3r^2 + L^2}{12 x_o}$$

$$12 L x_o = 6r^2 + 2L^2$$

$$x_o = \frac{6r^2 + 2L^2}{12 L} = \frac{3r^2 + L^2}{6L}$$

general formula for cylinder

let $r = 13"$ $L = 193" + 3" = 196"$

then $X_o = \frac{507 + 38416}{1176} = 33.1$

and $\ell = \frac{L}{2} + X_o = 98 + 33.1 = 131.1$

From the mechanics of Fig. 4 it is evident that P is the difference between the total shears on the left and right of center of percussion.

Then from Fig. 3

$$P = \frac{Fm\ell}{2} - \frac{Fn\,(L-\ell)}{2} = \frac{Fm\ell}{2} - \left[\frac{(L-\ell)}{\ell}\,Fm\right] \times \frac{L-\ell}{2}$$

$$= Fm\left[\frac{\ell}{2} - \frac{(L-\ell)^2}{2}\right] = Fm\left[L - \frac{L^2}{2\ell}\right] \quad \text{(general)}$$

$$\text{So } P = Fm\left[196 - \frac{196^2}{2 \times 131.1}\right] = Fm\,(196 - 146.5) = Fm\,(49.5)$$

To find point Q, the point of max. moment,

let F = inertia load at distance X from from impact P

then shear at X is zero when

$$S = P - Fx - (Fm - F)\,\frac{X}{2} = 0 \qquad \text{and } F = \frac{\ell - X}{\ell}\,Fm$$

$$P = Fx + \frac{Fm\,X}{2} - \frac{F\,X}{2} = \frac{X}{2}\,(Fm + F)$$

Substituting and clearing

$$Fm\,(49.5) = \frac{X}{2}\left(Fm + \frac{\ell - x}{\ell}\,Fm\right)$$

$$49.5 = \frac{X}{2}\left(\frac{\ell + \ell - x}{\ell}\right) = X - \frac{X^2}{2\ell}$$

$$X^2 - 2\ell X + 99\ell = 0$$

$$X^2 - 262.2\,X + 12{,}978.9 = 0$$

$$X = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} = \frac{+262.2 \pm \sqrt{16833.24}}{2}$$

$$X = \frac{+262.2 \pm 129.7}{2} = \frac{132.5}{2} = 66.25''$$

To find max. moment at point of $X = 66.25$

$$M_X = PX - \frac{FX^2}{2} - (Fm - F)\frac{X}{2}\frac{2X}{3} = PX - \frac{FX^2}{6} - \frac{Fm\,X^2}{3}$$

$$= X\left(P - \frac{FX}{6} + \frac{Fm\,X}{3}\right)$$

but $F = \frac{\ell - x}{\ell}\,Fm = \frac{131.1 - 66.25}{131.1}\,Fm = \frac{64.85}{131.1}\,Fm = .4946\,Fm$

$$Fm = \frac{.4946\,P}{49.5}$$

then $M_x = 66.25\left(P - \frac{66.25\,(.4946)}{6}\,\frac{P}{49.5} - \frac{66.25}{3}\,\frac{P}{49.5}\right)$

$$= 66.25\,P\,(1 - .1103 - .4461) = 66.25\,P\,(.4436) = 29.388$$

and max. allowable $P = \frac{\text{max. allowable } M}{29.388}$

From 2.3.2.2 $M = 1{,}071{,}281 \times 36.28 = 38{,}866{,}074$ in. lbs.

$$P \text{ max.} = \frac{38{,}866{,}074}{29.388} = 1{,}322{,}575 \text{ lbs.}$$

$\frac{P\ max}{48{,}000}$ = 27.55 g's at one end only gives the equivalent of 36.28 g's on the cask proper with impact at both ends.

2.4.4.2 Elastic energy of cask deflection as a beam.

It is now required to prove that this impact force of P=1,322,515 lbs (which causes the same safe bending stress in the cask as the 30 foot side drop) will result in the absorption of the K.E. of the 40 inch drop by the cask in elastic deformation.

The strain energy of flexure is (Roark - pg. 98 - 4th edition)

$$U = \int_O^L \frac{M^2 \, dx}{2\,EI}$$ where M represents the bending moment equation

$$M = Px - \frac{Fx^2}{6} - \frac{Fm\,x^2}{3} \qquad F = \frac{.4946P}{49.5} = .009992P$$

$$F_M = \frac{P}{49.5} = .020202\ P$$

$$M = Px - \frac{.009992Px^2}{6} - \frac{.020202Px^2}{3}$$

$$= Px - .001666Px^2 - .\ 006734\ Px^2$$

$$= Px - .0084\ Px^2 = P\ (x - .0084\ x^2\ )$$

$$M^2 = P^2\ (\ x^2 - .0168\ x^3 + .00007056x^4)$$

$$U = \frac{P^2}{2\ EI} \int_O^L (.00007056\ x^4 - .0168x^3 + X^2\ )\ d\,X$$

$$U = \frac{P^2}{2\ EI} \left(\frac{.00007056L^5}{5} - \frac{.0168\,L^4}{4} + \frac{L^3}{3} \right)$$

Where $\frac{P^2}{2\ EI} = \frac{(1,322,515)^2}{2x\ 328.39\ x10^9} = 2.663$ (EI from page XI-2-19)

$L = 196$ $\qquad L^5 = 289,255.\ x\ 10^6$

$L^4 = 1475.789\ x\ 10^6$

$L^3 = 7.5295\ x\ 10^6$

$$U = 2.663\left(\frac{70.56}{5} \times 289{,}255 - .0042 \times 1475.789 \times 10^{6} + \frac{7.5295 \times 10^{6}}{3}\right.$$

$$= 2.663(4{,}081{,}967 - 6{,}198{,}314 + 2{,}509{,}848)$$

$$= 2.663(393{,}498) = 1{,}047{,}895 \text{ in. lbs.}$$

The energy to be absorbed is (42.25) 48,000 lbs. = 2,028,000 in. lbs.

which is $\frac{2{,}028{,}000}{1{,}071{,}281}$ = 1.893 times the moment analyzed on pg. XI-2-21.

Extrapolation of the stresses in the cask numbers shows

	pg. XI-2-21 Bending stress @36.28 g's side drop	XI.893	Dynamic yp	Material	M.S.
1/2" inner shell	20,903	39,568	50,000	304	.26
2-3/8" U weld	23,877	45,199	50,000+	U	.106
7/8" outer shell	37,742	71,466	75,000	216	.049
1/4" jacket	36,276	68,670	75,000	216	.092

The picture presented here is that of an elastic cylinder impacting at one end and being instantaneously stationary, but with the full K.E. of the 40 in. drop converted into internal P.E. of the still dynamically elastic cylinder structure.

This is conservative because no account has been taken of; (1) any deformation of the aluminum end structure, (2) any deformation of the pin itself and (3) inelastic energy absorption by the lead in the cask.

The force P would now be required to be 1.893 (1,322,515) = 2,505,521

corresponding to $\frac{2,503,521}{48,000}$ =52.16 g's

Even this force is academic since the pin itself with a dynamic y p = 50,000 psi. would support only a load of $P_1 = \pi/4\ 6^2\ (50,000) =$

1,413,720 lbs. = 29.45 g's

Thus,damage would be restricted to the pin itself and the local aluminum structure, which forms a completely enveloping cylinder about the whole cask end.

2.4.4.3 Cylinder Supporting 4" end plate - impact structure

This cylinder is machined in contour to fit over the end of the cask and be held to it be axial screws. It also is bolted to the 4 inch alum. protection plate which is the supporting base for the end cylinder of balsa wood. It is also welded to the containing plates for the side annulus of balsa woo

The cylinder material is 6061-T6 plate, 2 in. thick, rolled and welded.

with typical properties of 45,000 psi U.T.S., 40,000 psi y.p. and 30,000 psi shear strength.

Shear at outer end, 7/8 in. thickness

P_1 = 1,413,720 lbs. (dynamic strength of pin)

$$S_s = \frac{1,413,720}{\pi\ 29.25 \times 7/8} = 17,582 \text{ psi}$$

$$\text{M.S.} = \frac{40,000}{17,582} -1 = 1.27$$

Shear at section below this, 1" thickness

$$S_s = \frac{1,413,720}{\pi\, 29.25 \times 1} = 15,385 \text{ psi}$$

$$\text{M.S.} = \frac{30,000}{15,385} -1 = .94$$

max. bending moment occurs at plane of end of cask.

$M_1 = 1,413,720\ (9") = 12,723,480$ in. lbs.

$$Z = \frac{.049\ (30.25^4 - 28.25^4)}{15.125} = 649.345 \text{ in.}^3$$

$S_b = M/_Z = 19,594$ psi

$$\text{M.S.} = \frac{40,000}{19,594} - 1 = .1.04$$

Combined shear and bending stress

$$S_t = \frac{19,594}{2} + \sqrt{9797^2 + 15,385^2}$$

$$= 9797 + 18,240 = 28,037 \text{ psi}$$

$$\text{M.S.} \ \frac{45,000}{28,037} - 1 = .60$$

The cylinder - under such lateral loading - is not restrained at the cask end by the four bolts which hold it in tension against the 10 g loads forward, but by the frictional grip at the edge of the cask itself. The coef. of friction between aluminum and mild steel is .61 (Marks 3-40)

The frictional grip on the cask is therefore:

$F = .61\ P_1 = 864,200$ lbs.

The resisting couple across the diameter of the cask is :

$M_2 = (864,200)\ 30" = 25,925,976$ in. lbs.

which is 2.04 times the moment M_1 imposed .

At the extreme lower end of the cylinder, the circular contour in interrupted to fit over the four trunnions, and provided with heavy walled boxing contours. These regions are protected on side drop since they are beyond the terminal plane of impact.

The cylinder also supports in compression the maximum end drop force on the balsa cylinder of

$$P = 1{,}891{,}541 \text{ lbs. } (39.4 \text{ g's}) \quad (2.1.1.3)$$

$$\text{Min. area} = \pi 29.25 \times 7/8 = 80.4 \text{ in.}^2$$

$$S_c = \frac{P}{80.4} = 23{,}527 \text{ psi}$$

$$\text{M.S.} = \frac{45{,}000}{23{,}527} - 1 = .91$$

2.4.5 Puncture - top end - oblique impact

Considering any angle between 0° and 90° from the cask centerline, the components of any such impact angle would each be less than the 0° and 90° cases already analyzed. It is considered that any such glancing blow striking the side of the support cylinder would quite obviously be less drastic than the orthogonal cases stated.

The general conclusion to top impact, whether from free fall of 30 feet or puncture from 40 inches, is that the aluminum cylinder and its 4 inch end plate form an impenetrable shield over the cask upper end, and support th the energy absorbing end cylinder and side annulus most effectively and conveniently.

As a result both of the cask seals are always kept intact because their contacting surfaces are not subjected to deformation of drop or puncture.

2.4.6 Puncture- bottom end - center impact

The most severe condition assumes for the bottom, as for the top, impact, that the balsa has already been compacted 10.6" by the 30 ft. free fall, and only 3.32 is now to be pierced (see 2.4.3.1.1) The remaining energy of the 40 in. drop is then expended in plastic deformation of the bottom cask structure itself. This consists of (a) 1-1/4 in. thick outer shell end plate, (b) 3-1/8 in. of uranium, an air gap of 1/4", and (c) 1-1/4 in. thick inner shell end plate. Between the inner shell and the container bottom is a 1-1/2 inch air gap, sufficient to allow for cask deformation without touching the fuel container.

The collapse load $W_u = \frac{t^2}{4} \text{ (y.p.) } \left(\frac{6\pi}{3-\frac{2a}{R}} \right)$ eq. (1)
Ref. 8
pg. 72 & 126

y.p. = 50,000 psi

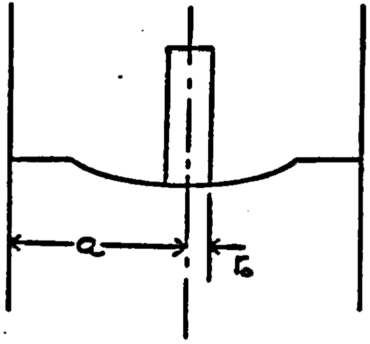
2a = 6" $R = \frac{1706}{2}$ dia. = 8.53"

All 3 discs have 50,000 psi y.p. and 17.06" dia. and are fixed.

$$W_u = t^2 \frac{50,000}{4} \frac{6\pi}{3-\frac{6}{8.53}} = 102,595\ t^2$$

For (a)	1-1/4 outer shell	W= 160,305 lbs.
For (b)	3-1/8" uranium	W= 1,001,903 lbs.
For (c)	1-1/4 inner shell	W= 160,305 lbs.

The elastic deflection at the center of a fixed edge disc is from Ref. 1 Table X, Case 7



$$y \text{ max.} = \frac{3\ W\ (m^2-1)}{16\pi E m^2\ t^3} \left(4a^2 - 4r_o^2 \log \frac{a}{r_o} - 3\ r^2_o\right)$$

r_o = 3" a= 8.53"

r^2_o = 9" a^2 = 72.76

$\log \frac{a}{r_o} = \log \frac{8.53}{3}$ log 2.8433 = 1.045

$$y \text{ max.} = \frac{W}{Et^3}\left[\frac{3\ (.91)}{16\ \pi}\quad (4 \times 72.76 - 4 \times 9 \times 1.045 - 27.)\right]$$

$$= \frac{W}{Et^3}\ (12.297) \qquad \text{eq. (2)}$$

$E = 29 \times 10^6$ S.S. Sy dynamic for S.S.=50,000 psi

$E = 24 \times 10^6$ U Sy dynamic for U = 50,000 psi

The load W at which the material turns partly plastic is, for max. stress at the center,

$$S_r = S_t = \frac{3W}{2\pi m t^2}\left[(m+1) \log \frac{a}{r_o} + (m+1)\frac{r^2_o}{4 a^2}\right]$$

$$= \frac{W}{t^2}\left[\frac{3}{2 \quad {}^{10}/_{3}}\left(\frac{13 \times 1.045}{3} + \frac{13}{.3} \times \frac{9}{4 \times 72.76}\right)\right]$$

$$\frac{W}{t_2}\left[.14324\ (4.5283 + .134)\right]$$

$$= \frac{W}{t^2}\ (.6678)$$

$$W = Syt^2/.6678 = 1.4975\ Syt^2 \qquad \text{eq. (3)}$$

The procedure involves the following consecutive steps, which progressively increase the deflections and reduce the available K.E. from impact.

Step 1 The K.E. reduced by pin penetration thru the balsa.

From 2.4.3.1.1 Y_1 = 3.32" and $\bar{KE}_1$ = 121,561 in lbs.

Total drop KE = (48,000) (40" + 4.8985") = 2,155,128 in lbs.

Remaining KE = 2,155,128 - 121,561 = 2,033,567 in lbs.

Step 2 Calculate the elastic and plastic properties of plates (a) and (b) to the point where the deflection of (b) equals 1/4 inch and contact is made with plate (c)

For SS plate (a) by eq. (3)

$$W_1 = 50{,}000\ (1.4975)\ (1.25)^2$$

$$= 116{,}989 \text{ lbs.}$$

$$y_2 = \frac{116{,}989\ (12.297)}{29 \times 10^6\ (1.25)^3} = .0254 \text{ in. (Elastic)}$$ by eq. (2)

The plastic collapse load is by eq. (1)

$W_u = 160{,}305$ lbs. or,

about double the true elastic load. Assume, conservatively that the deformation has increased proportionately. Then the K.E. absorbed up to the point where full plastic stress developes

is $$\frac{W_u\, y_3}{2} = \frac{160{,}305}{2} \times .0348 = 2790 \text{ in. lbs.} = KE_2$$

where $$y_3 = \frac{160{,}305}{116{,}989} \times .0254 = .0348$$

The remaining distance $y_4 = .250 - .0348 = .2152$ is traversed under collapse load, with K.E. absorption equal to $KE_3 =$ $(160{,}305)\ .2152 = 34{,}500$ in. lbs.

Total remaining K.E. $= 2{,}033{,}567 - 2790 - 34{,}500 = 1{,}996{,}277$ in. ll

For the uranium plate (b) a similar analysis shows:

$$W_1 = (50{,}000)\ (3.125)^2\ (1.4975) = 731{,}201 \text{ lbs.}$$

$$y_1 = \frac{731{,}201\ (12.297)}{24 \times 10^6\ (3.125)^3} = .0122765$$

$$W_u = 1{,}001{,}903 \text{ lbs.}$$

$$y_2 = \frac{1{,}001{,}903}{731{,}201} \times .0122765 = .0168 \text{ in.}$$

$$\frac{W_u \times y_2}{2} = 8{,}416 \text{ in. lbs.}$$

The remaining distance y_3 = .250 - .0168 = .2332

K.E. absorbed = W_u y_3 = 233,644 in. lbs. plastically

Remaining K.E. = 1,996,277 - 8,416 - 233,644 = 1,754,217 in. lbs.

Step 3 Plate (c) is now about to be loaded, similarly to plate (1)

For the next y_2 = .0348 we have K.E. absorbed by

Plate (a)	160,305 (.0348)=	5,578 in. lbs.
Plate (b)	1,001,903 (.0348) =	34,866 in. lbs.
Plate (c)	160,305/2 (.0348)=	2,789 in. lbs.
	=	43,233 in. lbs.

The remaining K.E. = 1,754,217 - 43,233 = 1,710,984 in. lbs;

Step 4 All plates are now plastic; with total resisting force of

Plate (a)	160,305 lbs.
Plate (b)	1,001,903 lbs.
Plate (c)	160,305 lbs.

max load P = 1,322,513 lbs. = 27.55 g's for the cask

This now moves thru distance y_4 to absorb the remaining energy

$$y_4 = \frac{1,710,984}{1,322,513} = 1.2937$$

Total deflection of the inner plate (c) is 1.2937+ .0348 = 1.3285"

The total deflection of plate (c) is accommodated by the air gap distance between plate (c) and the inner container bottom head without making contact with the inner container head.

This justifies initial assumption of 44.8985" drop,

= 40 + 3.32 + .250 + .0348 + 1.2937

Plates (a) and (b) deform .250 + .0348 + 1.2937= 1.3785 in.

Plate (c) deforms only 1.3285"

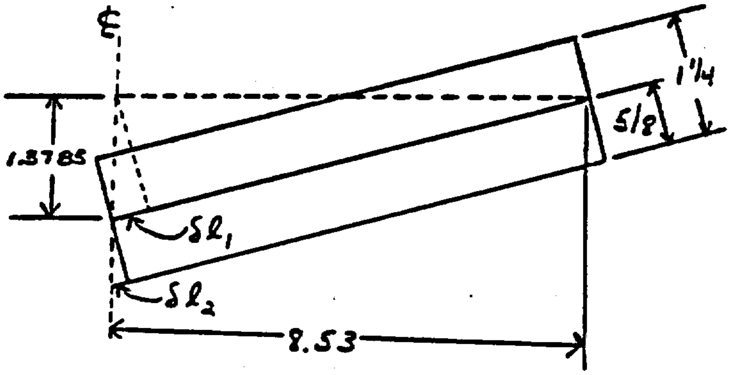
Ratio of $\frac{\text{total deflection}}{\text{deflection at y.p.}}$ for each plate

Plate (a) Ratio = $\frac{1.3785}{.0245}$ = 56.3

Plate (b) Ratio = $\frac{1.3785}{0.0168}$ = 82.05

Plate (c) Ratio = $\frac{1.3285}{0.0254}$ = 52.3

Plate (a) 304 SS



$$\delta\ell_1 = .1107$$
$$\delta\ell_2 = \underline{.101}$$
$$.2117"$$

$$\delta\ell_2 = \frac{1.3785}{8.53} \times 5/8" = .101$$

on 4" gauge length $\frac{2 \times .2117}{4"}$

= 10.58% elo

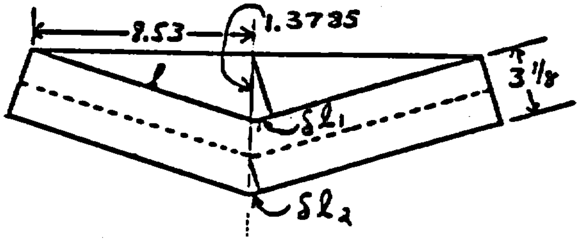
on 2"gauge length $\frac{2 \times .2117}{2"}$

=21.17 % elon

Plate (c) 304 SS is subjected to even less distortion

The composite "sandwich" of plates (a) (b) and (c) constitute a system in which, even with a fracture of the uranium plate (b), protection is provided by the effective puncture resistance and the extraordinary ductility of the two layers of 304 SS themselves.

Plate (b) Uranium



$\ell = 8.6407$

$\underline{\quad 8.53}$

$\delta\ell_1 = .1107$ (called "membrar stretch" of who plate)

$$\delta\ell_2 = \frac{3\text{-}1/8}{2} \times \frac{1.3785}{8.53} = .253$$ (plastic stretch of tension fibers)

total extreme fiber elongation at plastic hinge at ℄ is

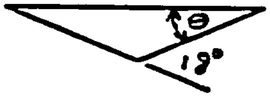
$$\delta\ell_1 + \delta\ell_2 = .1107 + .253 = .3637$$

' Due to dimension involved, considering the 17.06 dia. as representative of a "test piece", a guage length for measuring elongation of 8" is appropriate.

$$\%\ \text{elongation} = \frac{2(.3637)}{8} = 9.09\%$$

This is within the nominal elongation values.

Considered as a bent bar, the angle of 18° is O.K. as evaluated from specimens bent in a vise. Ref. 9



$$\tan\theta = \frac{1.3785}{8.53} = .1616 \qquad \theta = 9.3^\circ$$

It is considered that the uranium plate will not fracture.

2.4.7 Puncture - bottom end- off center impact

2.4.7.1 A side or corner impact at any angle to the bottom edge of the cask would be analyzed substantially the same as that previously calculated for the top end, with simillar and largely equal results, and therefore acceptable. Any deformation of the edge of the cask itself, if it is impacted directly by the 6" pin, would be of no serious interest, considering what happens to the pin, and the fact that the cask will bounce elastically.

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3.0 References

1. Roark, R.J. "Formulas for Stress and Strain" Fourth Edition
2. "ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, Unfired Pressure Vessels," The American Society of Mechanical Engineers.
3. Harris and Crede, "Shock and Vibration Handbook" McGraw-Hill, 1961.
4. "ASME Boiler and Pressure Vessel Code, Section III," The American Society of Mechanical Engineers.
5. Timoshenko: Strength of Materials Part I
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7. Marks Mech. Engineers Handbook 6th Edition
8. Plastic Analysis and Design of Plates, Shells and Discs Save and Massonnet
9. GGA Shipping Cask Safety Analysis Report - Docket No. 70-1165, NLI work on uranium properties.
10. AISC Manual of Steel Construction, 7th Edition
11. Calvert Cliffs, Unit 1 and 2 FSAR, Baltimore Gas and Electric Co., Docket 50-317.
12. Properties of Ethylene Glycol and its Aqueous Solutions - C.S. Cragoe ,National Bureau of Standards.

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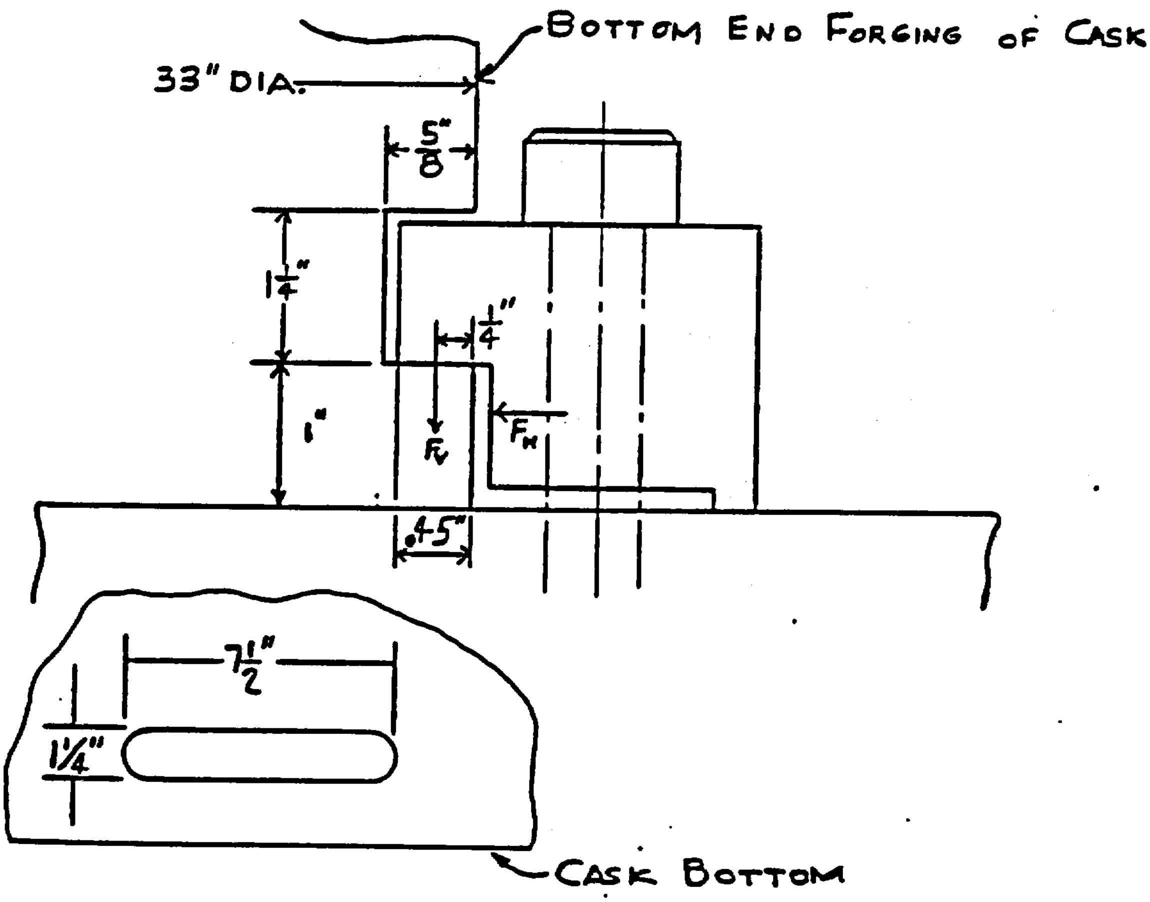
SECTION XI

APPENDIX "A"

In order to facilitate the attachment of a large diameter aluminum plate to the bottom of the cask, notches are machined into the wall of the bottom end forging. The large diameter aluminum plate is part of a dash pot system which is being used by some utilities to protect the spent fuel storage pool in the event of a cask drop accident. The aluminum plate is attached to the cask by eight (8) lugs which are bolted to the plate and fit into the notches on the cask essentially acting as a clamp. The design loads which could be imposed on the cask by the dash pot system design are 42000 pounds vertical and 50000 pounds horizontal the most highly loaded lug location. The effects of this loading on the cask are analyzed as follows:



Vertical Load

F_v is uniformly applied to cask lower lip along a length of 6" and at a distance of ¼ inch in from the O.D. of the forging. This loading is substantially unaffected by slight tilting errors of the clamping lug itself. The lip is 1 inch thick.

Stress thru root of lip :

M= 42,000 lbs. x ¼ in. =10500 in. lbs.

$$Z = \frac{6 \times 1^2}{6} = 1 \text{ in.}^3 \text{ conservative,}$$

$$S_b = \frac{10500}{1} = 10500 \text{ psi.}$$

$$S_s = \frac{42,000}{6 \times 1} = 7000 \text{ psi}$$

$$\text{Combined stress } S_t = \frac{10500}{2} + 5250^2 + 7000^2 = 14,000 \text{ psi}$$

$$\text{M.S.} = \frac{30,000}{14,000} -1 = 1.14$$

$$\text{Bearing stress} = \frac{30,000}{6 \times .45} = 11,111 \text{ psi}$$

$$\text{M.S.} = \frac{30,000}{16,000} -1 = .875$$

Horizontal load is taken by direct compression against lip

$$S_c = \frac{50,000}{6 \times 7/8} = 9534 \text{ psi}$$

$$\text{M.S.} = \frac{30,000}{9524} -1 = 1.7$$

SECTION XI

APPENDIX B

The calculations of Appendix B are addressed to only those features of the cask affected by the alternate shipping configuration of the cask. The paragraph numbers refer to the paragraphs of the main body of Section XI.

B.1.0 Normal Conditions of Transport

B.1.2 Requirement: The containment vessel will suffer no loss of contents if subjected to an external pressure of 25 psig.

With the inner container removed, the inner shell is the containment vessel. No credit is taken for the support of the containment shell provided by the fuel basket.

Containment Vessel (Inner Shell)

Formulas taken from Ref. 1, Table XIII, Case 1 Shell Material - SA-351, CF8M or SA-240, Type 304 or A358 Class 1.

Minimum yield point is 30000 psi for these materials.

$$p' = \frac{t}{R}\left(\frac{S_y}{1 + 4\frac{S_y}{E}\left(\frac{R}{t}\right)^2}\right)$$

t = .513 in. - inner shell thickness

R = 6.94 in.

S_y = 30000 psi

E = 29×10^6 psi

$$p' = \frac{.513}{6.94}\left(\frac{30000}{1 + 4 \quad \frac{30000}{29x10^6}\left(\frac{6.94}{.513}\right)^2}\right)$$

$$p' = 1262 \text{ psi}$$

Inner shell is more than adequate to withstand 25 psig external pressure.

The outer shell calculations are unaffected by the removal of the inner container.

B.1.3 Pressure Vessel Design - Cask Cavity (See Fig. XI-B1, Pg. XI-B3)

Design Pressure - Section 1.9, page XI-1-37 calculates the design pressure in the inner container if all the fuel pins fail and release fission gas to the fuel cavity. The design pressure calculated is 120 psig and this pressure will be used as the cask cavity pressure for the alternate configuration without the inner container. The design temperature of the cask cavity will be 850° F.

Ref. 1, Table XIII, Case 1

$$p = \frac{t\sigma_Y}{R}$$

t = .513 in. - inner shell thickness

σ_Y = 16500 psi Ref. 2

R = 6.94 in.

$$p = \frac{.513\ (16500)}{6.94} = 1220 \text{ psi} > 120$$

B.1.3.1 External Pressure - Inner Container @ 850° F.

Delete for Alternate configuration.

Figure XI-B1 CASK CAVITY

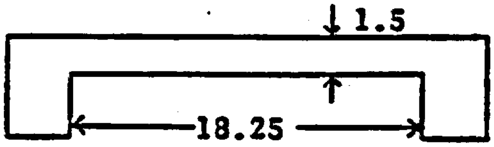
FIGURE WITHHELD UNDER 10 CFR 2.390

B.1.3.2 Bottom Head - Cask Cavity

The cask cavity design pressure is 120 psig. The bottom head as shown in Section 1.3.2 will withstand 461 psi which is greater than 120 psi.

B.1.3.3 Outer Closure Head

By inspection the weakest element is the flat end plate.



Ref. 1, Table X, Case 6
edges fixed

$w = .785d^2 \ p$

$d = 18.25$

$p =$ pressure

$t = 1.5"$

$S_r = 16500$ psi ref 2

$$\text{Max } S_r = \frac{3 \ w}{4 \pi t^2}$$

$$S_r = \frac{3 \ (.785) \ (18.25^2) \ p}{4 \ \pi \ (1.5)^2}$$

$$16500 = 27.74 \ p$$

$$p = 594.8 \text{ psi} > 120 \text{ psi}$$

B.1.3.4 Outer Closure Head Bolts

Bolt load due to internal pressure $W_p = .785G^2 \ p$;

where $G =$ mean gasket dia. $= 20.3"$; $p = 120$ psi

$W_p = .785 \ (20.3^2) \ (120) = 38818$ lbs.

Bolt load required to compress gasket $W_G = \pi Gy$;

$G =$ mean gasket dia $= 20.3"$; $y = 100$ #/lineal inch.

$$W_G = \pi (20.3)\ 100 = 6377 \text{ lbs.}$$

Closure head bolts are 1" - 8 thd.

Yield point 30000 psi

Effective stress area = .6051 in^2

The load on each bolt is $\frac{38818 + 6377}{8} = 5649$ lbs

$$\sigma_c = \frac{5649}{.6051} = 9335 \text{ psi} < 30000 \text{ psi}$$

B.1.5.1 Axial Expansion Inner Container/Inner Shell

Delete for Alternate configuration.

B.1.5.4 Radial Expansion Aluminum Basket/Inner Shell (See Fig. XI-B2)

Even though the inner container is removed and the basket is now slightly cooler, the basket temperature will be as shown on page XI-1-13. The largest temperature gradient occurs at initial steady state conditions.

Aluminum Basket 661° F $\quad T = 661 - 68 = 593^\circ$ F

$$L_t = L_o \left[1 + c\ (12.19\ T + .003115\ t^2)\ 10^{-6}\right]$$

Reference: Alcoa Aluminum Handbook

L_t = diameter at temperature

L_o = diameter at 68° F

c = alloy constant (from table)

T = 593° F

Figure X1-32 RADIAL EXPANSION ALUMINUM BASKET/INNER SHELL

11,;

FIGURE WITHHELD UNDER 10 CFR 2.390

$$L_t = 13.256\left[1 + .990\ (12.19 \times 593 + .003115 \times 593^2)\ 10^{-6}\right]$$

$$= 13.365$$

Inner Shell 440°F $\Delta T = 440-68 = 372° F$

$$\epsilon_T = 9.9 \times 10^{-6}\ (372)\ 13.375 = .049$$

∴ ID of inner shell at temperature

$$= 13.375 + .049 = 13.424$$

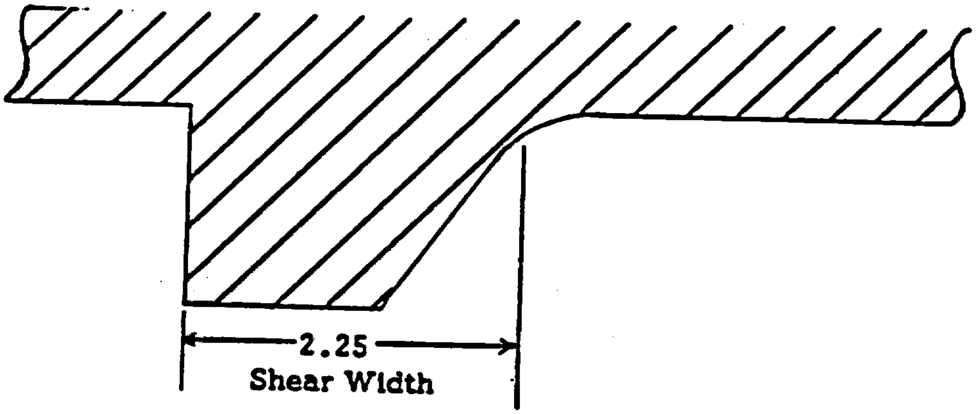
No radial interference between basket and inner shell

B.1.5.5 Radial Expansion - Inner Container/Inner Shell

Delete for alternate configuration.

B.1.7.2.3 Strength of Taper Rings on Cask

The entire ± 10 g longitudinal force, except for the functional restraint at the rear support, is taken at the front support by either of the two taper rings which are machined from the cask body forging -



The loaded area is considered to extend for a distance of 15 inches circumferentially.

Total shear area = 2.25 x 15 = 33.75 in^2

The load is 480000-7200 (rear friction) = 472800 lbs.

$$\text{Shear stress} = \frac{472800}{33.75} = 14009 \text{ psi}$$

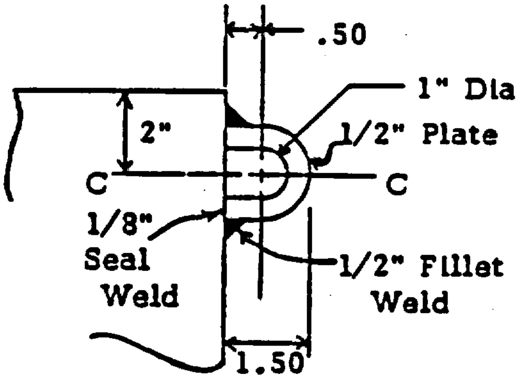
$$\text{M.S.} = \frac{.6 \times 30000}{14009} - 1 = .285$$

B.1.8.2 Outer Closure Lifting Lugs

Outer Closure Head weight is 341 lbs.

Design Load = 3 x 341 = 1023

The outer closure head is provided with two lugs welded to the side of the closure head.



$$\text{Plate tension c-c} = \frac{1023}{4(.5x.5)} = 501 \text{ PSI}$$

$$\text{M.S.} = \frac{30000}{1023} - 1 = 28.3$$

$$\text{1/2" fillet weld shear} \quad \frac{1023}{4(.5 \times .35)} = 1461 \text{ PSI}$$

$$\text{Bending} \quad \frac{1023\ (.5)}{4\left(\frac{.5 \times .35^2}{6}\right)} = 12526 \ \text{PSI}$$

$$\text{Combined stress} = \frac{12526}{2} + \sqrt{\left(\frac{12526}{2}\right)^2 + (1461)^2}$$

$$= 12694 \ \text{PSI}$$

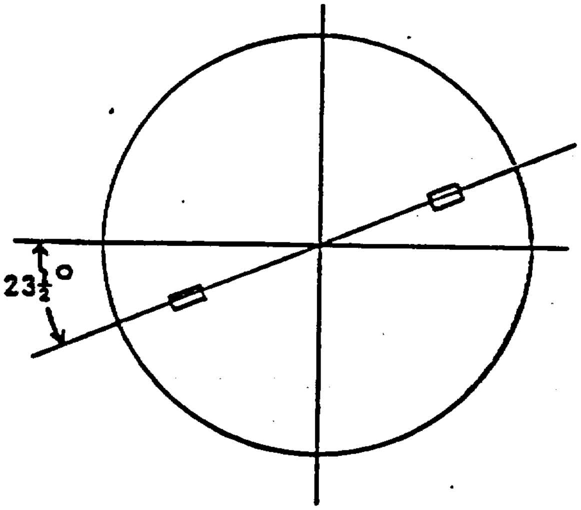
$$\text{M.S.} = \frac{30000}{12694} - 1 = 1.36$$

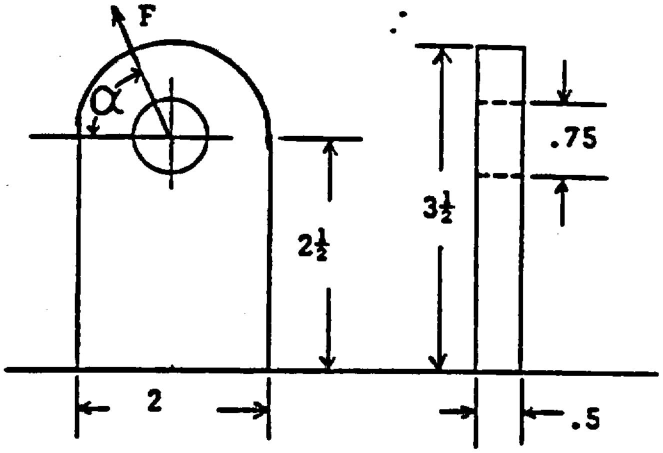
This is conservative since 1/8 seal weld is not included.

B.1.8.3 Inner Closure Head Lifting Lugs

Lift lugs have been added to the inner closure head to facilitate handling of the closure head. The closure head calculated weight is 744 lbs.

F = Design Load = 3 x 744 = 2232 lbs.





$\alpha = 83.1°$

The closure head is connected to the cask lift rig by means of wire rope slings attached to the lift lugs. The angle α is the angle that the sling makes with the closure head.

$F_1 = F \cos 83.1$ = horizontal component

$F_2 = F \sin 83.1$ = vertical component

$F_1 = 268.1$ lbs $\qquad F_2 = 2216.$ lbs.

Bearing in hole:

$$S_{BR} = \frac{2232}{.5(.75)} = 5952 \text{ psi}$$

$$\text{M.S.} = \frac{30000}{5952} - 1 = 4.0$$

Shear in lug:

$$S_s = \frac{2232}{(2 - .75).5} = 3571.2 \text{ psi}$$

$$\text{M.S.} = \frac{.6(30000)}{3571.2} - 1 = 4.0$$

Tension in lug:

$$S_t = \frac{2232}{(2 - .75).5} = 3571.2 \text{ psi}$$

$$\text{M.S.} = \frac{30000}{3571.2} - 1 = 7.4$$

Because of the more than adequate margins of safety and large value of α, the effects of breaking the force into components is not considered in the lug.

Tension in weld:

$$S_t = \frac{F2}{\text{Area}} = \frac{2216}{2(.5)} = 2216 \text{ psi}$$

Shear in weld:.

$$S_s = \frac{F_1}{Area} = \frac{268.1}{1} = 268.1 \text{ psi}$$

Bending in weld:

$$I = (1/12)(.5)(2^3) = .333$$

$$S_B = \frac{MC}{I} = \frac{268.1(2.5)(2)}{.333} = 4026 \text{ psi}$$

Maximum Combined Stress in Weld:

$$S = \sqrt{(4026 + 2216)^2 + 268.1^2} = 6248 \text{ psi}$$

$$M.S. = \frac{30000}{6248} - 1 = 3.8$$

B.1.9 Pressure Vessel Design - Inner Container

For the alternate configuration, the inner container is removed. But the calculation for the fuel cavity pressure shown in Section 1.9, page XI-1-37 is still valid. The volume for fission gas expansion without the inner container remains the same since the fuel basket diameter is larger. The design pressure calculated in Section 1:9 is 120 psig. This is the cavity design pressure that is used for the alternate configuration.

B.1.9.1 Inner Container Shell

Delete for alternate configuration

B.1.9.2 Inner Container Bottom Head

Delete for alternate configuration

B.1.9.3 Inner Closure Head

The inner closure head for the alternate configuration seals against the cask cavity. The thickness of the inner head has changed due to an increase in uranium thickness. The inner plate of the head remains the same. Since the inner plate only is considered the pressure boundary, the analysis of Section 1.9.3 will not be repeated.

B.1.9.4 Inner Container Flange

Delete for alternate configuration

B.2.0 Hypothetical Accident Conditions

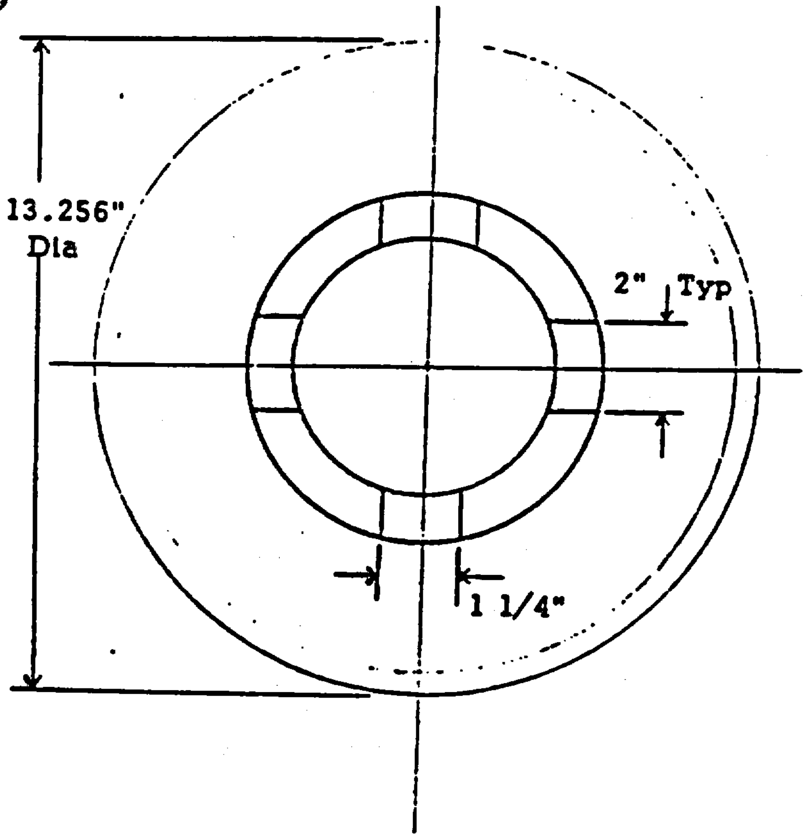
B.2.1.3.3 Inner Container

Delete for alternate configuration

B.2.1.3.4 PWR and BWR Fuel Baskets

The fuel baskets for the alternate configuration have been increased in diameter and increased in weight. The bottom supports have been redesigned and are analyzed for the 30 ft. drop. In both cases, the baskets will not fail and there will be no change in the relationship between the fuel assembly and the cask internals.

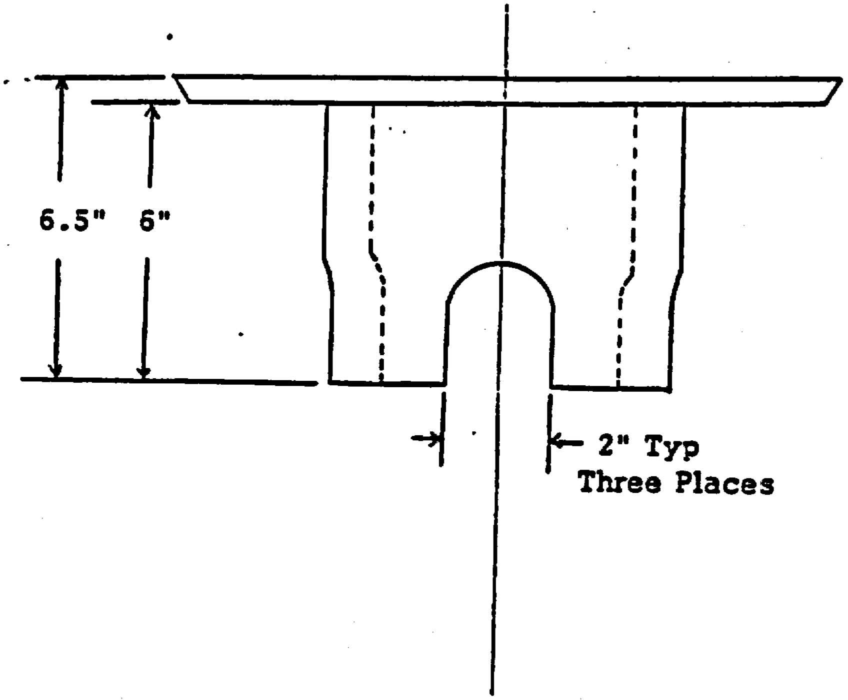
B.2.1.3.4.1 PWR Lower Support



The support material is type 304 stainless steel and the support is bolted to the bottom of the aluminum basket.

Peak g loading is 39.4 g's

Load = 1600# fuel
1012.9# basket
2612.9#



Max. load on 6" SCH 80

= 2612.9 (39.4) = 102948#

Pipe area

$= (\pi D_o - 7.25)\ .432$

$= (\pi \times 6.625 - 7.25)\ .432$

$= 5.86\ in^2$

Bearing stress on end of pipe

$$\sigma_{BR} = \frac{102948}{5.86} = 17568$$

$$M.S. = \frac{30000}{17568} - 1 = .707$$

B.2.1.3.4.2 BWR Lower Support

The load carrying members are the same. The only difference in the sketch shown on page X-1-2-9 is that 12.550 in dia. is 13.256 in . which has no affect on the structural integrity of the component.

B.2.1.4.2 Inner Closure - Studs and Closure Plate

Total load on studs -	fuel	1600
	PWR basket	1012.91
	PWR spacer	17.66
	closure head	743.27
		3373.84 lbs.

$F = 3373.84\ (39.4\ G) = 132929$ lbs.

This load is taken by 12 bolts 1"-8 @ 105,000 psi Y.P.

Stress area = 12 x .6051 = 7.26 in^2

$S_t = \frac{132929}{7.26\ in^2} = 18309$ psi $\qquad$ $M.S. = \frac{105000}{18309} - 1 = 4.73$

Bolts are SA-193-B7; 105000 psi Y.P. at 1100° F min.

tempering temperature, strength = 63535 lbs.

Shear on closure plate. Weight of Uranium = 472.25 lbs.

$F = 472.25\ (39.4) = 18606$ lbs.

Shear area .5" stn. stl. plate = $A_s = \pi (13.875).5 = 21.8\ in^2$

$$S_s = \frac{18606}{21.8} = 853 \text{ psi}$$

$$M.S. = \frac{18000}{853} - 1 = 20.1$$

B.2.1.4.3 Inner Closure Head Seal (page XI-2-10a)

The hollow metal "O"-ring seal on the inner closure head will not be damaged by shock or temperature effects during the accident damage conditions. Excessive deflection of the seal is prevented by the metal to metal contact of the flange faces. The hollow metal "O"-ring material is Inconel X which has a tensile strength of 120,000 psi at the fire accident temperatures.

The bolts retaining the inner closure are torqued to a clamping force equivalent to 63 g's, assuring an effective seal under impact.

Twelve bolts at 39663 psi stress (1-8ths) provide a clamping force of 12 (.6051) 39663 = 288000 lbs. The load required to compress the metal "O"-ring is 67,720 lbs. The net clamping force = 220280. For a total weight of basket, spacer plug, fuel and closure head of 3478.59 lbs. This represents a restraint of 220280/3478.59 = 63 .g

B.2.1.4.4 Inner Closure Head Valve Assembly

Each of the two penetrations into the cask cavity through the inner closure is provided with a valved quick-disconnect fitting and a bolted-on cover (see drawing 70884F). A metal O-ring contained in the cover seals the cavity under both normal and accident conditions of transport. In order to keep the joint sealed the cover bolts must be preloaded to maintain a seal load of 700 lb/in. as specified by the manufacturer. The maximum loading on the joint occurs in the top end impact accident.

The required bolt preload is determined as follows:

Temperature - 400°F (conservative, see Sect. VIII).

Internal Pressure - 120 psig. (Cavity design value).

Seal load - 700 lb/in.

Cover weight - 12.1 lb.

Bolts - 3 of 1/2-13 UNC, 304 S/S.

Stress area = 0.1419 in.^{2}

Yield strength - 20700 psi @ 400°F.

Seal diameter - 2.375 in.

Impact acceleration - 39.4 g.

Seal load/bolt = $2.375\pi(700)/3 = 1741$ lb.

Pressure load/bolt = $(\pi/4)(2.375^2)(120)/3 = 177$ lb.

Inertia load/bolt = $(12.1)(39.4)/3 = 159$ lb.

Total load/bolt = $1741 + 177 + 159 = \underline{2077}$ lb.

With a bolt preload of 2400 lb., the margin of safety for joint sealing is MS = (2400/2077)-1 = <u>0.156</u>
Bolt preload stress = 2400/0.1419 = 16913 psi.

MS = (20700/16913) - 1 = <u>0.224</u>

In the fire accident condition there is no inertia loading on the joint, but the maximum temperature is taken as 600°F (conservative, see Sect. VIII). Under these conditions the joint is evaluated as follows:

Bolt yield strength @ 600°F = 18300 psi.

Total load/bolt = 1741 + 177 = <u>1918</u> lb.

Joint sealing margin of safety is

MS = (2400/1918) - 1 = <u>0.251</u>

Bolt stress MS = (18300/16913) - 1 = <u>0.082</u>

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B.2.1.4.3 Outer Closure Head

Weight = 341

$$F = 341 \cdot (39.4) = 13435$$

The closure head is clamped against its seat by eight (8) bolts 1" - 8 thds.

yield point = 30000 psi min.

effective stress area = .6051 in^2

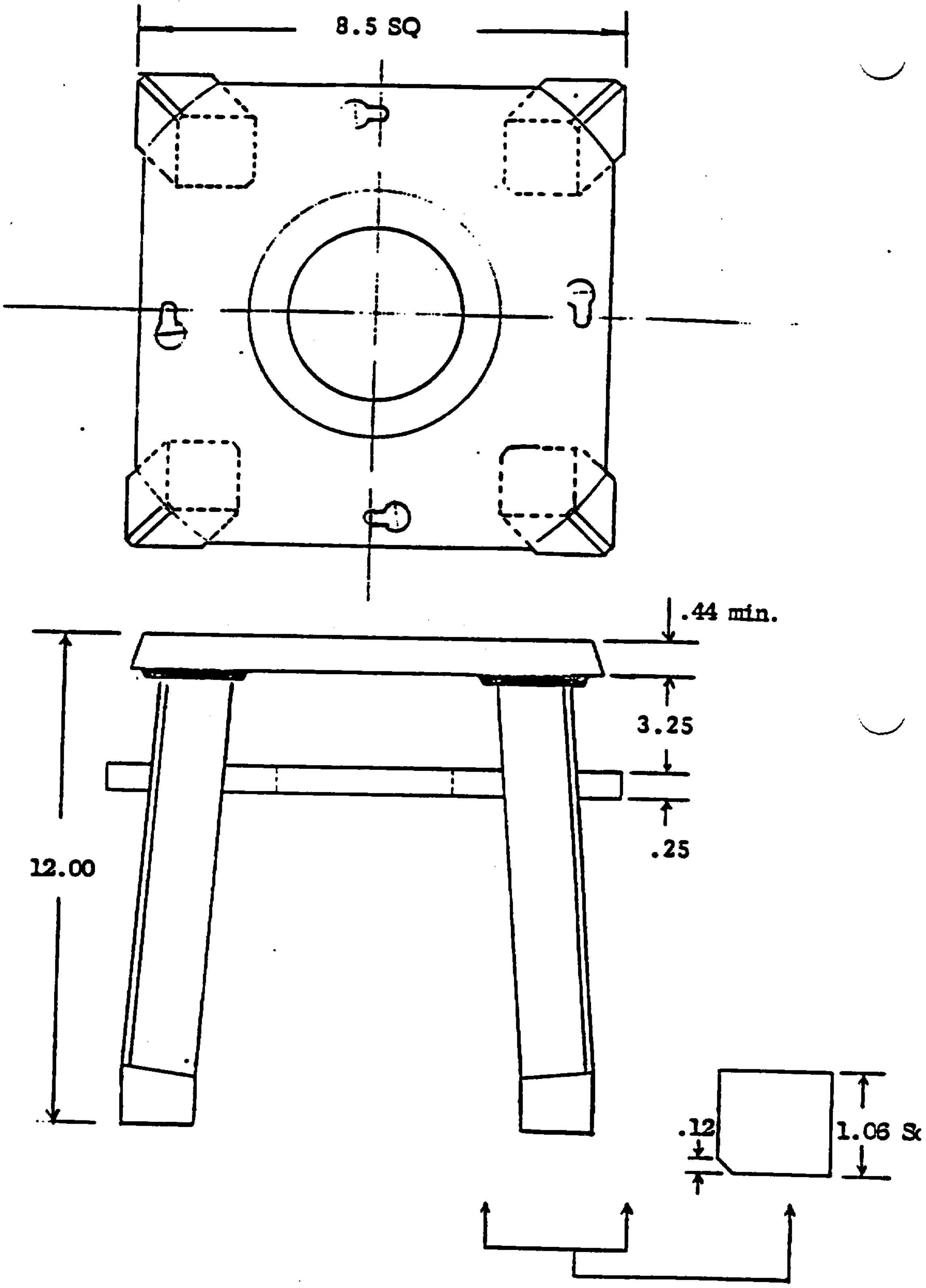
The load on each bolt is $\frac{13435}{8} = 1679$ lbs.

$$\sigma_c = \frac{1679}{.6051} = 2775 \text{ psi}$$

$$\text{M.S.} = \frac{30000}{2775} - 1 = 9.8$$

B.2.1.4.4 PWR Top Support (page XI-2-13)

Support material is type 304 stainless steel. The support is basically four 1.25 inch square bars arranged in a slightly sloping 6.5° attitude as corner posts based on an 8.5 inch square. Net end bearing area is 1.116 in^2/bar.


8.5 SQ
.44 min.
3.25
.25
12.00
.12
1.06 Sq

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Load - 1600 lbs. Fuel
1017.4 lbs. Basket

2617.4 lbs.

Peak g loading is 39.4 g's. Max. load on one bar (corner post) $P = \frac{2617.4}{4} (39.4) = 25780$ lbs. Analysis of the stress conditions are now made by two concepts of the system behavior.

Case A - Even though inclined, each post is stabilized by frictional loads developed at its ends and so takes a direct compression load.

tan 6.5 inclination = .1139

Coefficient of friction, metal to metal, grease free, in air = .39

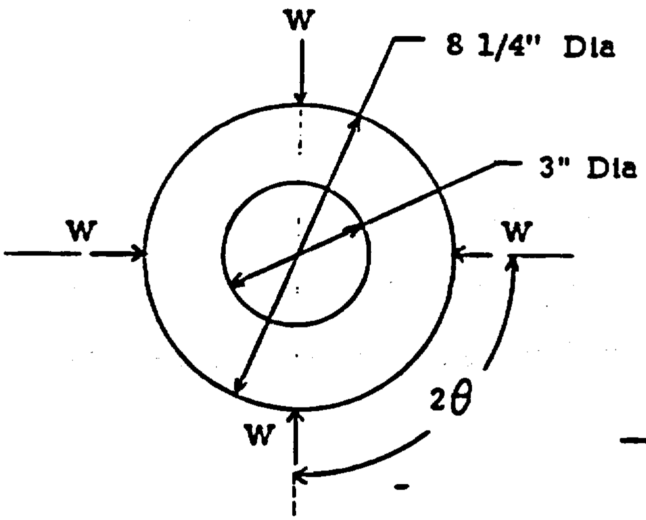
$$M.S. = \frac{.39}{.1139} - 1 = 2.42 \text{ for stability}$$

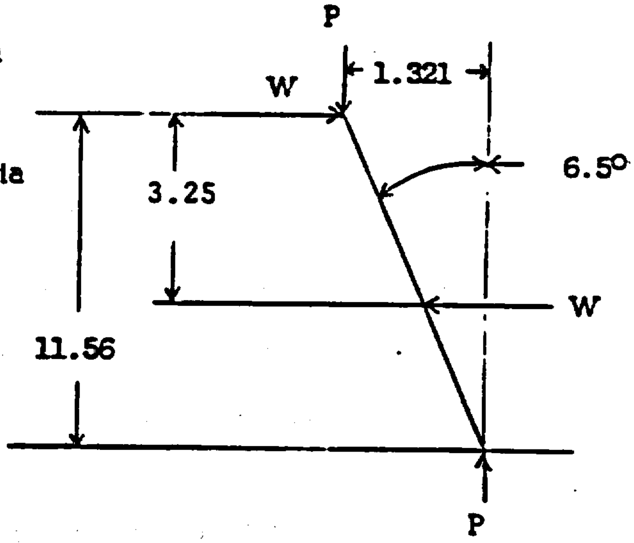
Bearing area $= 1.116$ in^2/bar

$$S_c = \frac{25780}{1.116} = 23100$$

$$M.S. = \frac{30000}{23100} - 1 = .299$$

Case B - Neglecting the benefits of end friction, assume the moment created by vertical end loads is resisted by a horizontal couple present in the upper end ring and in the intermediate .25 inch thick tie plate. The latter is essentially a ring loaded at four (4) points by radial tension loads at each inclined post, thus stabilizing and allowing the direct stress calculated in Case A. The .25 inch plate is most heavily stressed.





Formulas taken from Ref. 1, Table VIII, Case 9

Ring section = .25 x 2.625

$$\theta = 45^\circ$$

Ring radius R = 2.8125 mean

$$M_{max} = \frac{WR}{2}\left(\frac{1}{\theta} - \mathrm{Cot}\,\theta\right) = w\,\frac{(2.8125)}{2}\left(\frac{1}{.7854} - 1\right)$$

$$= w\,(\,.3842\,)$$

$$Z = \frac{.25\,(\,2.625\,)^2}{2} = .2871 \text{ in}^3$$

$$S_B = \frac{w\,(.3842)}{.2871} = w\,(1.338)$$

$$w = \frac{30000}{1.338} = 22418 \text{ lbs. max.}$$

Resisting moment = 3.25 w = 94844

Applied moment = 1.321 (25780) = 34055

$$\text{M.S.} = \frac{94844}{34055} - 1 = 1.785$$

Babcock and Wilcox Fuel - Top Support

The B & W fuel top support is also bolted to the inner closure head and extends down to support the fuel at the top end of the top nozzle. The support material is type 304 stainless steel.

B & W fuel will require a top support only when shipped in the alternate configuration cask, since with the inner container, the fuel fills the shipping cavity.

FIGURE WITHHELD UNDER 10 CFR 2.390

Babcock and Wilcox Fuel Top Support

Fuel weight	=	1550 lbs.
Control rod weight	=	132 lbs.
Basket weight	=	794 lbs.
		2476 lbs.

peak g loading = 39.4 g's

$$\text{Maximum load on one bar} = \frac{2476\ (39.4)}{4}$$

$$= 24388.6 \text{ lbs.}$$

$$\text{Bar area} = 4\ (.75) - .75^2$$
$$= 2.4375 \text{ in.}^2$$

$$\text{Compression stress} = \frac{24388.6}{2.4375}$$

$$= 10000 \text{ psi}$$

$$\text{M.S.} = \frac{30000}{10000} - 1 = 2.0$$

Combustion Engineering Fuel

For use in the alternate cask configuration, the C-E fuel top spacer will be 22 5/8 inches long. See section 2.1.4.4 for calculations.

B.2.1.4.5 BWR Fuel

See Page XI-2-14

B.2.3.4 Side Impact on Closure Heads

B.2.3.4.1 Inner Closure Head

The rim of the inner closure head has a small clearance within the cask head forging and Is directly supported on it' for lateral loads.

FIGURE WITHHELD UNDER 10 CFR 2.390

bolt area 8 (.6051) - 4.84 in^2

In order to calculate the section modulus of the bolts, they will be replaced with a ring of the same area and an outer diameter equal to the bolt circle diameter.

$$A = \frac{\pi(D_o^2 - D_i^2)}{4}$$

$$\frac{4.84(4)}{\pi} = 22.25^2 - D_i^2$$

$$22.111 = D_i$$

$$\text{ring thickness} = \frac{22.25 - 22.111}{2} = .0695$$

$$\text{Area moment of inertia} = I_{oo} = \pi r^3_o t$$

$$= \pi(11.125^3)\ .0695$$

$$I_{oo} = 300.63$$

Area moment of inertia transferred to axis A-A

$$I_{AA} = I_{oo} + A r_o^2$$

$$= 300.63 + 4.84(11.125^2)$$

$$= 899.65$$

Moment due to weight of closure head

$$M = 3.12(341)\ 36.28$$

$$= 38599 \text{ in-lbs.}$$

Max. tension stress on bolts due to side impact

$$S = \frac{MC}{I}$$

$$= \frac{38599\ (22.25)}{899.65}$$

$$= 954.6 \text{ psi}$$

$$M.S. = \frac{30000}{954.6} - 1 = 30.4$$

B.2.5. ALTERNATE CONSTRUCTION - HEAD END FORGING CASK BODY

FIGURE WITHHELD UNDER 10 CFR 2.390

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SECTION XI

APPENDIX C

Initial fuel rod fill pressure has been increased from 500 psig (XI-1-37) to 550 psig. This increase has the following effect on the calculations shown on pages XI-1-37A through XI-1-37E.

Internal fuel rod pressure:

$$P_1 = \frac{565\ (1101)\ 1.534}{530\ (.889)} + \frac{1.36\ (42.2)(.23)\ 2.14 \text{ X } 10^{-3}\ (40.84)(1100)}{.889}$$

$$P_1 = 3438 \text{ psia}$$

Mols of gas in fuel assembly (208 rods):

$$N = \frac{(3438)(.889)}{12(1545)(1100)}\ (208) = .0131$$

Total number of mols of gas in the cask cavity will be:

$$NT = .0131 + .0316 = .0447$$

Pressure in cask cavity at fire accident temperature

$$= \frac{(.0447)(1545)(1102 + 460)}{(5.6)(144)} = 133.8 \text{ psia or } 119.1 \text{ psig}$$

There is no effect on the structural analysis previously presented since the increase in the cask cavity pressure is less than the design pressure of 120 psig.

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SECTION XI - APPENDIX D
SUPPLEMENTAL ANALYSIS FOR FISSION GAS RELEASE

1.0 Summary

The NLI-1/2 cask is designed to safely contain irradiated fuel under a variety of normal and accident conditions. To verify the design for shipment of a high burn-up fuel assembly (having a burnup of 58,600 MWD/MTU), analyses of leak rates and total release were performed in accordance with ANSI - 14.5-1977 and 10 CFR 71.36 (a) (2).

The permitted leak rate is found to be 1.24×10^{-4} cm^3/sec. and the post accident condition leak rate is found to be 1.79×10^{-5} cm^3/sec. The quantity of material released is much less than the 1000-curie limit imposed by IAEA Safety Series 6. This analysis shows that the transport of high burn-up fuel with four extra rods is not limited by fission gas release.

The results of this analysis are a bounding condition. PWR fuel assemblies having a burnup of 40,000 MWD/MTU without extra rods will develop less internal pressure and have lower leak rates. The fission product inventory of 25 PWR rods with a burnup of 60,000 MWD/MTU*, or 25 BWR rods with a maximum burnup of 75,000 MWD/MTU will be lower than a high burn-up fuel assembly with 208 rods. Metallic fuel does not develop internal pressure as oxide fuel does, and fission product inventories are much lower because of the lower burnup and long cool time. The total fission gas inventory of 21 rods of metallic fuel cooled 1 year is 609 curies, which is much less than the 4950 curies present in the intact PWR fuel. The radioactive inventory of severely failed metallic fuel in filters is totally available for release. The containment analysis for these contents is presented in Section 2 of Appendix H of this Chapter.

The fission gas activity of the Mark 42 fuel assembly is much lower than that of the design basis PWR fuel assembly as shown in Table IX-24. The percentage of fission gas that is released from the fuel assembly is also much lower for the Mark 42 fuel because it has a metal form and is not an oxide. Therefore, the activity released during normal operations and during an accident will be bounded by the design PWR fuel assembly analysis.

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* Includes up to two PWR rods at 65,000 MWD/MTU.

Revised
Oct. 1986
Feb. 1987
Aug. 1988
Jan. 1990
Feb. 1991
Oct. 1991

2.0 Description of the Reference Fuels

Analysis of fission gas release is based on the quantity of gas released to the cask cavity assuming the rupture of all of the rods in the cavity, and on the type and quantity of the radioactive gas released. The analyzed high burn-up fuel assembly is a Westinghouse 15 x 15 PWR. The assembly, as shipped, is modified to contain 208 rods from various assemblies having different burn-up. The make-up of the analyzed assembly is described further in Section 3.3. Data provided by the vendor ORIGEN run shows that each rod contains 0.49 moles of gas at STP for which 0.13 mole per rod is fission gas.

The rupture of rods during the hypothetical accident increased the cask cavity pressure and makes fission gas available for release to the environment.

3.0 Hypothetical Accident Conditions

3.1 Cask Cavity Temperature

The maximum cavity temperature occurs shortly after the fire accident and is estimated in Section VIII of the SAR to be 860°F. The estimated temperature of the hottest fuel pin is given in the same section as 1102°F. For conservatism, a temperature of 1102°F is used as the cavity temperature.

3.2 Cask Cavity Temperature

During the loading operation, the cavity is charged with helium to one atmosphere at 68°F. This helium expands, increasing internal pressure as the cavity temperature increases to 1102°F. The free volume of the NLI-1/2 is estimated, from the Operating Manual, to be 115.2 liters (9471 in^3) for a cavity containing a PWR assembly. The cavity, therefore, contains 6.93 moles (115.2 1/22.4 moles per 1) of gas at STP conditions.

Since the cask is not at STP conditions (32°F and 14.7 psia), the actual number of moles in the cask is found as:

#m = 6.93 (492°R/530°R) = 6.43 moles

For the modified assembly, plus four rods, the quantity of gas released during the complete rod rupture is:

204 + 4 rods x .049 moles per rod = 10.192 moles

This gas is added to the cavity.

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The total pressure in the cask cavity at 1102°F is calculated, using the ideal gas law, as:

$$P = \frac{nRT}{V} \text{ or } \frac{(6.43 + 10.192)\ (40.77)\ (1562)}{9471} = 111.8 \text{ psia}$$

where T is in Rankin and V is in in^3.

This pressure represents the driving force for disposal of fission gas from the cavity.

3.3 Cask Cavity Radionuclide Concentration

The cask cavity concentration or cask activity is found as: Total curies released from rods divided by the cavity free volume.

The analyzed assembly has been disassembled and then reconstituted with rods having various burn-up. The curie content of each rod group is determined from an ORIGEN code using the appropriate burn-up (MWD/MTU) and cool time.

# rods	Average Burn-Up (MWD/MTU)
196	58,600
4	47,380
2	41,500
2	27,650
2	18,930
2	8,290
208	

ORIGEN Code Average Burn-Up MWD/MTU	Number of Rods	Cool Time (yrs.)	Curies Per Rod		
			Krypton	Tritium	Xenon
55,000	196	1.25	33.2	2.06	-*
46,930	4	3	21.3	1.49	-
39,439	2	1.23	22.5	1.43	-
31,313	2	1.23	19.5	1.13	-
18,550	4	1.23	13.1	.66	-

*After 150 days, Xenon exists only in trace amounts.

NRC Safety Guide (Regulatory Guide 1.25) provides that calculations for gaseous releases from irradiated rods be based on the assumption that ten percent of the Tritium and Xenon and 30 percent of the Krypton contained in each rod is released (to the cavity free volume). Consequently, the activity of the released gas for the assembly and extra rods is:

Krypton: $(.3)(4 \times 13.1 + 2 \times 19.5 + 2 \times 22.5 + 4 \times 21.3 + 196 \times 33.2) =$ 2019 curies

Tritium: $(.1)\ (4 \times .66 + 2 \times 1.13 + 2 \times 1.43 + 4 \times 1.49 + 196 \times 2.06) =$ 42 curies

Xenon: There are only trace quantities of Xenon (less than 1×10^{-7} curies

The cavity concentration is then:

$(2019 + 42)/.1552\ m^3 = 13280$ curies/m^3 or 1.33×10^{-2} curies/cm^3

3.4 Cask Leak Criteria

The NLI-1/2 cask is tested before first use and thereafter annually to ensure that it remains leak-tight. The closure seals for the vent, drain and inner head are tested by helium leak detectors. The acceptance leak rate is 1 x 10^{-6} atm. cm^3/sec. Leak rates greater than this value are cause for rejection of the seal.

In the performance of this test, the cask cavity is pressurized to 20 psig with helium.

4.0 Package Containment Requirements

The NLI-1/2 truck cask is of the type B(u). ANSI-14.5-1977 sets a containment requirements of $Ra = A_2 \times 1.65 \times 10^{-9}$ curies/sec. for accident conditions for this type cask. The values of A_2 for each radionuclide released are specified in Table VII, Regulations for Safe Transport of Radiactive Materials, Safety Series 6, International Atomic Energy Agency, 1973 edition.

In accordance with Safety Series 6, IAEA, a weignted ratio method is used to calculate the A_2 value of the cavity gas mixture. Specifically, for asembly the A_2 value is calculated as follows:

A_2 = Total curies/sum of ratios, therefore:

Product	Curies	A_2 Value[3]	Ratio
Kr	2019.	1000	2.02
H_3	42.0	1000	.042
Xe	trace	100	--

and $A_2 = 2.061 \times 10^3/2.062 = 999$ curies.

(A_2 values are from Table VII of Safety Guide 6.)

Therefore, $Ra = 999 \times 1.65 \times 10^{-9} = 1.65 \times 10^{-6}$ ci/sec.

In accordance with Section 5.32 of Safety Guide 6, the permissible leakage rate for the gas in the cavity is given by La = Ra/Ça. The Permitted Leak Rate (La) is then 1.65×10^{-6} ci $sec^{-1}/1.33 \times 10^{-2}$ ci cm^{-3} or $La = 1.24 \times 10^{-4}$ cm^3/sec.

5.0 Determination of Package Leak Rate

Gas leak rates can be shown to be proportional to the driving pressure (the pressure differential across the leak) and inversely proportional to the gas viscosity. The correlation between a measured leak rate and equivalent leak rate is given in ANSI - 14.5, Equation B5 as:

$$Lx = \frac{Ly\ my\ (Pu^2 - Pd^2)x}{mx\ (Pu^2 - Pd^2)y}$$

In this equation, Lx is the leak rate at condition "x", and mx and $(Pu^2-Pd^2)x$ are the gas viscosity and up steam and down stream pressures, respectively, at condition "x". Similarly, Ly, my, (Pu^2-Pd^2) are the leak rate, viscosity, and up stream and down stream pressures, in atmosphere at condition "y".

In this calculation, the condition "y" is taken as the cask test condition given in Section 3.4 above. (Pressure: 20 psig, leak rate: 1 x 10^{-6} atm cm^3/sec). The viscosity of helium at the test condition is 1.941 cp.

The vicosity of helium at 1100°F (the accident condition) is 4.042 cp. Helium is considered to be the escaping medium for conservatism. It has the lowest viscosity of gases in the cavity and therefore has a viscosity lower than the weighted average of the mixture.

In the discussion of pressure in the cavity, it was found that the fuel assembly rupture, combined with the fire accident temperature, caused a cavity pressure of 111.8 psia (7.6 atm). During the cavity qualification test, the cavity pressure is 20 psig + 14.7 psi = 34.7 psia (2.36 atm). These are the up-stream pressures. In each case, the down stream pressure is 14.7 psia (1 atm).

Substituting these values in the equation given above:

$$L = 1 \times 10^{-6} \frac{\text{atm cm}^3}{\text{sec}} \frac{(1.941 \text{ cp})(7.6a^2 - 1a^2)}{(4.042 \text{ cp})(2.36a^2 - 1a)}$$

$$L = 5.96 \times 10^{-6} \text{ atm cm}^3/\text{sec}$$

The construction of the cask is such that there are three (3) possible leakage paths in the normal configuration, the inner head seal and two (2) valve seals (inlet and drain). Each of these seals is tested to 1 x 10^{-6} atm cm^3/sec leak rate (or less). If each of these were to leak, then the total leak rate would be 3 x La or 1.79 x 10^{-5} atm cm^3/sec.

This leak rate is less than the permissible leak rate calculated in Section 4.0.

This shows that leak rate is not a constraint for this fuel. For fuels cooled longer than 150 days, the permitted leak rate is unchanged. The calculated leak rate is reduced because cayity concentration is reduced due to radioactive materials decay.

6.0 Activity and Release Rate for Leaked Gas

The activity of the gas in the cavity has been calculated and was presented in Section 3.3. From this value and the calculated release rate presented in Section 5, the curie release rate can be found. From this value, the number of curies released in a week can be determined.

The curie release rate and curies released in one week of 6.05 x 10^5 seconds is found as:

1.79×10^{-5} cm^3/sec x 1.33×10^{-2} curies/cm^3 =
2.4×10^{-7} curies/sec (Curie Release Rate)

The curies released in one week are:

2.4×10^{-7} ci/sec x 6.05×10^5 sec = .15 curies.

This value is insignificant compared to the 1000 curies/week limit established by IAEA Safety Guide 6.

7.0 Conservatism of Analysis

This analysis is conservative because: 1) the pressure calculation is based on the temperature of the hottest fuel pin and not the cavity average temperature, 2) the viscosity of helium is used for the accident condition leakage rate, resulting in a higher rate of leakage, and 3) no credit is taken for the double containment of the NLI-1/2 inner and outer head arrangement that would cause a very slow release to the environment.

Section XI
APPENDIX E - SUPPLEMENTAL ANALYSIS OF CONSOLIDATED FUEL

1.0 INTRODUCTION

Consolidation is the process of packing the rods from up to two PWR assemblies into a single canister for shipment. Non-fuel bearing components of the assemblies are handled separately as waste.

Analyses of the effects of the shipment of consolidated fuel in the NLI-1/2 cask have been performed for criticality, shielding, structural and thermal effects. The consolidated fuel modeled in the criticality, shielding, and structural analyses is W15 x 15 fuel cooled for two years, with an initial enrichment of 3.7 w/o U-235 and a burnup of 40,000 MWD/MTU. These values are considered to be representative of consolidated PWR spent fuel shipments. The thermal analysis has been performed for W14 x 14 fuel cooled 12 years to add an additional margin of conservatism because the thermal behavior of consolidated spent fuel is currently being investigated.

2.0 SUMMARY

The NLI-1/2 cask design weight used in the Safety Analysis Report (SAR) is 48,000 pounds.

The weight of the NLI-1/2 cask loaded with a consolidated fuel canister containing 408 PWR fuel rods is calculated to be 49,250 pounds (Configuration A).

The effect of the increased weight of the cask contents on the structural adequacy of the NLI-1/2 cask (based on SAR calculations), wit the inner container (Configuration A of the NLI-1/2 cask) considered to be the primary containment system, was evaluated. The evaluation shows that the increased weight has a minimal effect on impact stresses. It does result in slightly increased deformation of the impact limiters; however, the limiters are adequately sized to accommodate the increased deformation. Thus, the structural adequacy of the NLI-1/2 cask is not affected by the increased weight of the contents covered by this application.

A summary of the results of the detailed analysis appear in Table E1. The specific weights, material proprietries and allowable stresses used in the detailed analysis are summarized in Tables E-2 through E-4, respectively.

3.0 Effect of Metallic Fuel

The NLI-1/2 cask design weight used in the Safety Analysis Report (SAR) is 48,000 pounds.

The weight of the NLI-1/2 with 21 sound metallic fuel rods is 47,500 pounds. The weight of the NLI-1/2 with 6 failed metallic fuel rods is 45,800 pounds. Thus the metallic fuel does not cause the cask to exceed the design weight. It should be noted also that the cask neutron shield tank may be drained if metallic fuel is shipped. This results in a further weight reduction of 2860 pounds. The calculated weight of 47,500 pounds or 45,800 pounds, includes the weight of a filled neutron shield tank.

Fuel and Basket Weight Combinations

21 Rods - Metallic	2,500 pounds	2,612 pounds
Basket (Sound Fuel)	112	
6 Rods - Metallic Failed Fuel	720	903 pounds
Basket (Failed Fuel)	183	
Control Assembly	150	150 pounds

Maximum Weight (Metallic Fuel Only)

Fuel and Basket	2,612
Cask Body	42,500
Outer Closure Head	340
Inner Closure Head	665
Top Impact Structure	865
Bottom Impact Structure	460
Total	47,442

TABLE E1
NLI-1/2 CASK
CONSOLIDATED FUEL TABLE E1
STRUCTURAL EVALUATION

Summary of Results

Analysis	Comment/Margin of Safety
Calculation of Design Weight	≈ +2.6% (increase)
Normal Transport Conditions	
Simple Beam Vibration Tie Down Lifting Devices	No significant change. Margins of safety reduced 2.6%.
One Foot Drops	Impact load on cask increases a maximum of 1.0%. Impact limiter depth of crush increases a maximum of 3.0% but remains much less than maximum crush depth.
Accident Conditions	
Bottom End Impact	Impact load is unchanged. Depth of crush = 66%. M.S. = +0.65 (tension at neck section).
Top End Impact	M.S. = +1.62 (tension in innner closure bolts). M.S. = +Large (shear on closure plate) Clamping Force = 91.35 g's > 38.4 g's for the inner closure head seal. Clamping force exceeds the impact force maintaining the seal.
Corner Impact	Impact Load - No significant change Depth of crush - 80.2%
Side Impact Top Limiter (30' drop) Bottom Limiter (30' drop) Bottom Limiter (1' drop)	Impact load increases 1.0% max. Depth of crush is 75.3% max.
Puncture	
Outer Shell Thickness	$t_{min} = 0.729" < t_{act} = 0.875"$
Impact at Mid-Length (Cask Bending)	Strain-Outer Shell = 14.7% <45%
Top & Bottom End Impacts	No significant change. Margins of Safety reduced 2.6%

TABLE E2
WEIGHT CALCULATIONS

(Cask body) From NLI-1/2 Interface Manual	W_{cb} = 42,505 lbs. (1)
(Outer closure head) From NLI-1/2 Interface Manual	W_{och} = 340 lbs. (2)
(Inner closure head) From NLI-1/2 Interface Manual	W_{ich} = 665 lbs. (3)
(Top impact structure) From NLI-1/2 Interface Manual	W_{tis} = 865 lbs. (4)
(Bottom impact structure) From NLI-1/2 Interface Manual	W_{bis} = 460 lbs. (5)
(Configuration A inner container) From drawing 70562F, Rev. 8, sheet 1	W_{ic} = 650 lbs. (6)
(PWR fuel basket) From drawing 70562F, Rev. 8, sheet 1	W_{pwrfb} = 800 lbs. (7)
Spacer plug - Estimated	Use W_{sp} = 20 lbs. (8)
Consolidated fuel canister (w/408* fuel rods) *Two W 15 x 15 PWR assemblies Fuel (408)(6.75) = 2,754 lbs. Canister $(8.78)(160)(.09)(4)(.285) + (2)(9.0)^2(0.75)(.285)$ = 180 lbs.	W_{cf} = 2,934 lbs. (9)
Total	W_c = 49,239 lbs.

Use $(W_{des})_{consol}$ = 49,250 lbs.

SAR cask design weight W_d = 48,000 lbs.

TABLE E3
MATERIAL PROPERTIES

(Type 304 stainless steel)

$S_u = 75$ ksi $S_y = 30$ ksi   At $T_{des} = 659°F$**, $S_y = 17.8$ ksi

Ref. ASME Section III, Division 1, Appendix I.

(SA-193, Gr B7 low alloy bolting material)

$S_u = 125$ ksi $S_y = 105$ ksi   At $T_{des} = 659°F$**, $S_y = 82.2$* ksi

Ref. ASME Section III, Division 1, Appendix I.

(6061-T6 aluminum alloy)

SU = 45 ksi $S_y = 40$ ksi   At $T_{des} = 659°F$**, $S_y = 5.0$ ksi

Ref. "Metals Handbook, Ninth Edition, Volume 2, Properties and Selection: Nonferrous Alloys and Pure Metals," American Society for Metals, Metals Park, Ohio.

*Extrapolated and ratioed from Sm values.
**Ref. No. 3, SAR, Page VIII-5.

TABLE E4
ALLOWABLE STRESSES

The allowable stresses used in this evaluation are chosen to be the same as those used in the NLI-1/2 Cask Safety Analysis Report (Ref. No. 3).

S_t = Tensile stress

S_c = Compressive stress

S_{br} = Bearing stress

S_b = Bending stress

S_s = Shear stress

$$S_t = S_c = S_{br} = S_b \leq S_y = 30{,}000 \text{ psi at } -20°F$$
$$= 17{,}800 \text{ psi at } 659°F$$

$$S_s = 0.6\ S_y = 18{,}000 \text{ psi at } -20°F$$
$$= 10{,}680 \text{ psi at } 659°F$$

3.0 DETAILED ANALYSIS

3.1 Structural Evaluation--Cask Components

3.1.1 Increased Design Weight

The weight of the NLI 1/2 Cask loaded with a consolidated fuel canister containing 408 PWR fuel rods is calculated to be:

$$(W_{consol})_{cask-calc} = 49{,}239 \text{ lbs.}$$

Then, the design weight of the cask for this evaluation is:

$$(W_{consol})_{cask-des} = 49{,}250 \text{ lbs.}$$

The design weight of the cask that is used in the SAR is:

$$(W_{sar})_{cask-des} = 48{,}000 \text{ lbs.}$$

Thus, the percentage increase in weight of the loaded cask is:

$$\Delta W = \frac{49250 - 48000}{48000} (100) = 2.6\%$$

3.1.2 Normal Transport Conditions

The margins of safety for the following Normal Transport Conditions are inversely proportional to the increase in weight of the cask:

1. Simple beam with 5g load
2. Vibration
3. Tie Down
4. Lifting devices

The NLI 1/2 Cask remains structurally adequate for these Normal Transport Conditions when the small increase in weight of the loaded cask is considered in the stress evaluations. (Ref. No. 3, SAR).

(One Foot Drops)

End Drop Refer. page XI-1-43, Ref. No. 3

Initial & Peak Load $F_p = 1{,}891{,}541$ lbs.

1' Drop Deformation $d = \frac{(49250)(12.0 + 0.32)}{1{,}891{,}541} = 0.32$ in.

. Peak g's $= \frac{1{,}891{,}541}{49250} = 38.4$ g's $<$ 39.4 g's (SAR Calc.)

Structural Evaluation--Cask Components

Normal Transport Conditions (Cont'd.)

Side Drop Refer. page X1-1-44, Ref. No. 3

1' Drop Deformation: d = 2.30 in.

$g's = 9.47\ g's < 9.62\ g's$ (SAR)

Corner Drop Refer. page X1-1-44, Ref. No. 3

As discussed in the SAR, a corner drop produces localized deformation of the impact limiter and a lower peak g-force than an end drop. The impact limiter remains capable of sustaining a subsequent 30 foot drop event.

45° Oblique Drop Refer. page X1-1-45, Ref. No. 3

Estimated Final Impact Areas

$a = (8)(14.25) = 114\ in.^2$

$b = (6.5)\ (21) = \underline{136.5\ in.^2}$

$A_i = 250.5\ in.^2$

Balsa's effective 45° crushing strength is 1850 psi X 0.7 = 1295 psi

Max. Impact Force $F_i = (250.5)(1295) = 324,400$ lbs.

Peak $g's = \frac{324,400}{49250} = 6.6\ g's < 6.75\ g's$ (SAR)

(One Foot Drops)

Drop Orientation	Cask SAR Weight	SAR Peak g-factor	SAR Max. Impact Load	Cask Consol. Fuel Weight	Consol. Fuel Peak g-factor	Consol. Fuel Impact Load
End	48000	39.4	1,891,200	49250	38.4	1,891,200
Side	48000	9.62	461,760	49250	9.47	466,398
Corner	48000	--	--	49250	--	--
45° Oblique	48000	6.75	324,000	49250	6.6	325,050

Thus; the impact force on the cask: (1) remains unchanged for the end drop; (2) remains essentially unchanged for the corner drop; (3) increases by [(466,398 - 461,760)/(461,760)] (100) = 1.00% for the side drop; and (4) increases by [(325,050 - 324,000)/(324,000)] (100) = 0.32% for the 45° oblique drop.

Normal Transport Conditions (Cont'd)

The depth of deformation of the impact limiter increases slightly (maximum of 3%) over the values determined in the SAR for the one foot drops, but does not approach the maximum crush depth of the limiter.

These evaluations lead to the conclusion that the NLI 1/2 cask can accommodate canisters of consolidated fuel without significantly affecting safety margins.

3.1.3 Accident Conditions

(End Impacts) Ref. No. 3, page X1-2-4

The buckling force on the aluminum cylinder in the impact limiter is:

$$(P_{cyl})_b = 309{,}236 \text{ lbs.}$$

The equivalent g's = $(P_{cyl})_b/49{,}250 = 6.28$ g's

For the balsa cylinder,

Balsa deformation = 10.9 in.

Kinetic Energy = K.E. = (360 + 10.9)(49,250) = 18,266,825 in.-lbs.
Aluminum Cylinder absorbs (309,236)(10.9) = 3,370,672 in.-lbs.

Remainder to be absorbed by balsa wood = 14,896,153 in.-lbs.

$$\% \text{ of balsa crush} = \frac{10.9}{16.625}(100) = 65.56\%$$

Peak stress = 1850 psi (balsa wood)

Final stress = 1850 - (0.6556)(740) = 1365 psi

$$\text{Mean stress} = \frac{1850 + 1365}{2} = 1607.5 \text{ psi}$$

Area of balsa = 855.3 in.^{2}

Energy absorbed = (1607.5)(855.3)(10.9) = 14,986,353 in.-lbs.

Accident Conditions, Cont'd

Peak Deceleration

Force on aluminum cylinder = 309,236 lbs.
Peak force on balsa wood = (1850)(855.3) = 1,582,305 lbs.

1,891,541 lbs.

$$\text{Peak g's} = \frac{1,891,541}{49250} = 38.4 \text{ g's}$$

(Bottom End Impact) Refer to Sections 2.1.3.1;2.1.3.2;2.1.3.3; and 2.1.3.4 of Ref. No. 3

Limiter Impact Force: (SAR) $F_1 = (39.4)(48000) = 1,891,200$ lbs.

(Consol. Fuel) $F_1 = (38.4)(49250) = 1,891,200$ lbs.

Limiter Deformation: (SAR) d_s = 10.6 in.;(Consol.Fuel) d_{cf}= 10.9 in.

(Max. allowable) $d_{ma} = (0.85)(16.625) = 14.13$ in.

Thus, there is no significant change in the cask evaluation. The Inner Container is evaluated in tension at the upper neck section.

Total weights are:	Inner Container	650 lbs.
	PWR Canister Support	800 lbs.
	PWR Consolidated Fuel Canister	2934 lbs.
		4384 lbs.

The impact force is: $F_{impact} = (4384)(38.4) = 168,346$ lbs.

X-Section area = A $(\pi/4)(13.125^2 - 12.625^2) = 10.1$ in.^{2}

$$S_t = \frac{168346}{10.1} = 16,648 \text{ psi} \qquad \text{M.S.} = \frac{27,500^*}{16,648} - 1 = +0.65$$

*Dynamic strength at temperature for Type 304 Stainless Steel. Ref. No. 3, page X1-2-7.

(Top End Impact)

Based on the above comparison of limiter impact forces and deformations and on Section 2.1.4 of Ref. No. 3 (SAR), only the Inner Closure--Studs & Closure Plate and the Inner Closure Head Seal must be evaluated for the increased loads due to the Consolidated Fuel.

Accident Conditions, Cont'd

The total load on the studs is:

Inner Container	650 lbs.
PWR Canister Support	800 lbs.
PWR Consolidated Fuel Canister	2934 lbs.
Inner Closure Head	665 lbs.
F_t =	5049 lbs.

The Impact Load is: $(F_t)_i = (5049)(38.4) = 193,882$ lbs.

The impact load is carried by 8 bolts (1-8UNC at 105,000* psi Y.P.)

Bolt Stress Area $= A_b = (8)(0.606) = 4.85$ in.^{2}

The Bolt Tensile Stress is: $S_t = \frac{193,882}{4.85} = 39,992$ psi

$$M.S. = \frac{105,000^*}{39992} - 1 = \underline{\underline{+1.62}}$$

*Bolts are SA-193, Gr.-87; $S_y = 105,000$ psi. At the minimum tempering temperature of 1100° F., $S_t = 63,535$ lbs.

Shear on Closure Plate: Weight of Uranium = 390.67 lbs.

$F_s = (390.67)(38.4) = 15,002$ lbs.

Shear area of 0.5 in. St. Stl. plate $= A_s = (\pi)(14.625)(0.5) = 23.0$ in.^{2}

$S_s = \frac{15002}{23.0} = 653$ psi $\quad (0.6)(30,000) = 18,000$ psi

$$M.S. = \frac{18,000}{653} = \underline{\underline{26.5}}$$

Inner Closure Head Seal:
(Ref. No. 3, page X1-2-10a)

Preload Clamping Force of 12 bolts is:

$F_{pl} = (12)(0.551)(80,000) = 528,960$ lbs.

The O-ring Compression Load is: $F_{o-r} = 67,720$ lbs.

Net Clamping Force: $(F_{net})_{cl} = F_{pl} - F_{o-r} = 461,240$ lbs.

Accident Conditions, Cont'd

For a total Container & Closure Weight of 5049 lbs., this represents a net restraint on the lid seal of (461,240)/(5049)= 91.35 g's >> 38.4 g's Impact Load.

Therefore, the cask is structurally adequate for end drop conditions

(Corner Impact)

Refer to pages X1-2-15 & X1-2-15a of Ref. No. 3 (SAR):

FIGURE WITHHELD UNDER 10 CFR 2.390

Thus, there is no significant change in the cask loading and the additional impact limiter deformation remains within acceptable limits.

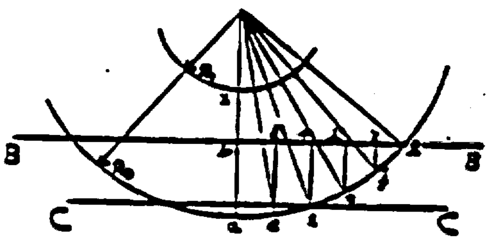
(Side Impact)

Balsa Stress Analysis (Pages XI-2-16 thru XI-2-16e, Ref. No. 3,5

From Section 2.3.1.5.1 on page XI-2-16c of Ref. No. 3, th mean strengths of the balsa wood are calculated as follows:

(Top Balsa Annulus) $R_o = 37.5$ in.; $R_1 = 16.25$ in.; Drop = 30

$a - x = 21.25$ in. $a - b = 13.7$ in. crush



At Point

a $S_c = 1826$ psi

b $S_c = 1826 - \frac{13.7}{21.25}(740) = 1349$ psi

$(S_c)_{mean} = 1588.$

c $S_c = (1826) \cos 10° = 1798$ psi

d $S_c = (1798) - \frac{12.9}{21.25}(740) = 1349$ psi

$(S_c)_{mean} = 1574$ p

e $S_c = (1826) \cos 20° = 1716$ psi

f $S_c = (1716) - \frac{11.3}{21.25}(740) = 1322$ psi

$(S_c)_{mean} = 1519$ p

g $S_c = (1826) \cos 30° = 1581$ psi

h $S_c = (1581) - \frac{8.3}{21.25}(740) = 1292$ psi

$(S_c)_{mean} = 1437$ p

j $S_c = (1826) \cos 40° = 1399$ psi

k $S_c = (1399) - \frac{4.7}{21.25}(740) = 1235$ psi

$(S_c)_{mean} = 1317$ p

l $S_c = (1826) \cos 50° = 1174$ psi

Mean at impact line BB $S_c = \frac{b + 2d + 2f + 2h + 2k + 2l}{11} = 12$

Mean of whole volume Crushed $S_c = \frac{ab + 2cd + 2ef + 2gh + 2jk + 2l}{11} = 1421$ p

Accident Conditions

(Side Impact)

Balse Stress Analysis (Cont'd.)

(Bottom Balsa Annulus) $R_o = 37.5$ in.; $R_1 = 17.375$ in.; Drop=

$a - x = 20.12$ in. $a - b = 15.16$ in. crush

At Point a $S_c = 1826$ psi

b $S_c = (1826) - \frac{15.16}{20.12}(740) = 1268$ psi $\Big\}$ $(S_c)_{mean} = 1547$ psi

c $S_c = (1826)\cos 10° = 1798$ psi

d $S_c = (1798) - \frac{14.35}{20.12}(740) = 1270$ psi $\Big\}$ $(S_c)_{mean} = 1534$ psi

e $S_c = (1826)\cos 20° = 1716$ psi

f $S_c = (1716) - \frac{12.55}{20.12}(740) = 1254$ psi $\Big\}$ $(S_c)_{mean} = 1485$ psi

g $S_c = (1826)\cos 30° = 1581$ psi

h $S_c = (1581) - \frac{9.75}{20.12}(740) = 1222$ psi $\Big\}$ $(S_c)_{mean} = 1402$ psi

j $S_c = (1826)\cos 40° = 1399$ psi

k $S_c = (1399) - \frac{5.95}{20.12}(740) = 1180$ psi $\Big\}$ $(S_c)_{mean} = 1290$ psi

l $S_c = (1826)\cos 50° = 1174$ psi

Mean at Impact Line $(S_c)_{mean} = 1224$ psi

Mean for Whole Volume $(S_c)_{mean} = 1392$ psi

Accident Conditions

(Side Impact)

Balsa Stress Analysis (Cont'd.)

(Bottom Balsa Analysis) $R_o = 37.5$ in.; $R_1 = 12.375$ in.; Drop=1,

$a - z = 20.12$ in. $\quad a - b = 2.30$ in. Crush

$$\left.\begin{array}{ll} \text{At Point } a & S_c = 1826 \text{ psi} \\ b & S_c = (1826) - \frac{2.30}{20.12}(740) \\ & \quad = 1741 \text{ psi} \end{array}\right\} (S_c)_{mean} = 1784 \text{ psi}$$

$$\left.\begin{array}{ll} c & S_c = (1826) \cos 10^\circ = 1798 \text{ psi} \\ d & S_c = (1798) - \frac{1.75}{20.12}(740) \\ & \quad = 1734 \text{ psi} \end{array}\right\} (S_c)_{mean} = 1766 \text{ psi}$$

$$e \quad S_c = (1826) \cos 20^\circ = 1716 \text{ psi}$$

Mean at Impact Line cc: $(S_c)_{mean} = 1728$ psi

Mean for Whole Volume: $(S_c)_{mean} = 1750$ psi

Accident Conditions

(Side Impact)

Balsa Stress Analysis (Cont'd.)

(Calculation of Crush Distances, Areas & Volumes)

Top Balsa Annulus

Refer to Section 2.3.1.2, page XI-2-16a, Ref. No. 3:

Drop = 30 ft.

$h = 13.7$ in. $d = 23.8$ in. $\cos\theta = 23.8/37.5 = 0.635 \rightarrow \theta = 50$

$\sin\theta = 0.773$

$a = R\sin\theta = 28.98$ in.

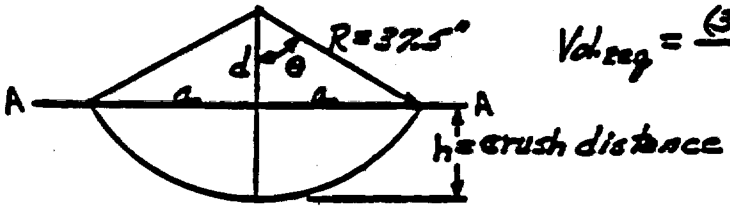
$t = 11.74$ in.

$\% \text{Crush} = (100)\frac{13.7}{21.25} = 64.5\%$

$A_{side} = \frac{2 \times 50.6^\circ}{360^\circ}(\pi)(37.5)^2 - (23.8 \times 28.98)$
$= 552.2 \text{ in.}^2$

$Vol_{AVAIL} = (11.74)(552.2) = 6483 \text{ in.}^3$

$Vol_{req} = \frac{(360+13.7)\left(\frac{49250}{2}\right)}{1421} = 6476$



$\text{Contact Area A-A} = (2)(28.98)(11.74) = 680.5 \text{ in.}^2$

$\text{Final Force on A-A} = (1281)(680.5) = 871657 \text{ lbs.}$

$\text{Final \& Peak g's} = \frac{871657}{49250/2} = 35.4 \text{ g's}$

Bottom Balsa Annulus

Refer to 2.3.1.3, pg. XI-2-16a, Ref. No.

Drop = 30 ft.; $t = 10.44$ in.; $h = 15.16$ in.; $d = 22.34$ in.

$\cos\theta = \frac{22.34}{37.5} = 0.5957 \longrightarrow \theta = 53.4^\circ$

$\sin\theta = 0.803$ $a = R\sin\theta = 30.12$ in.

$A_{side} = \frac{(2)(53.4)}{(360)}(\pi)(37.5)^2 - (22.34)(30.12) = 637.75 \text{ in.}^2$

$\text{Avail. } V_A = (10.44)(637.75) = 6658.1 \text{ in.}^3$

$\% \text{Crush} = (100)\frac{15.}{20.}$
$= 75.3;$

$\text{Req. } V_R = \frac{(360+15.16)(49250/2)}{(1392)} = 6636.7 \text{ in.}^3$

$A_{contact} = (2)(30.12)(10.44) = 628.9 \text{ in.}^2$ $F_{Final} = (628.9)(1224) = 769{,}780.$

$\text{Peak g's} = \frac{769{,}780}{49250/2} = 31.3 \text{ g's.}$

Accident Conditions

(Side Impact)

Balsa Stress Analysis (Cont'd.)

(Calulation of Crush Distances, Areas & Volumes)

Bottom Balsa Annulus Refer to 2.3.1.4, pg. XI-2-16b, Ref. No.

Drop = 1 ft.; $t = 10.44$ in.; $h = 2.30$ in.; $d = 35.20$ in.

$$\cos\theta = \frac{35.20}{37.5} = 0.9387 \longrightarrow \theta = 20.17^\circ$$

$\sin\theta = 0.3448$ $a = R\sin\theta = 12.93$ in.

$$A_{side} = \frac{(2)(20.17)}{(360)}(\pi)(37.5)^2 - (12.93)(35.20) = 39.91 \text{ in.}^2$$

$$\text{Avail. } V_A = (10.44)(39.91) = 416.7 \text{ in.}^3$$

$$\text{Req. } V_R = \frac{(12+2.30)(49250)}{(1750)} = 402.4 \text{ in.}^3$$

$$A_{contact} = (2)(12.93)(10.44) = 270.0 \text{ in.}^2$$

$$F_{final} = (270.0)(1728) = 466,560 \text{ lbs.}$$

$$\text{Peak } g\text{'s} = \frac{466560}{49250} = 9.47 \text{ } g\text{'s}.$$

$$\%\text{ Crush} = (100)\frac{2.30}{20.12} = 11.4\%$$

Cask Evaluation

Limiter Location	Drop Height	SAR Cask Wgt.	SAR Peak g-factor	SAR Max. Impact Load	Consol. Fuel. Wgt.	Consol. Fuel Peak g-factor	Consol. Fuel Max Imp Load	% Change in Load	Consol. i Imp. Lin Crush
Top	30 ft.	48,000	36.28	1,741,440	49250	35.4	1,743,150	+0.12	64.
Bottom	30 ft.	48,000	32.08	1,539,840	49250	31.3	1,541,525	+0.11	75.3
Bottom	1 ft.	48,000	9.62	461,760	49250	9.47	466,398	+1.00	11.4

Consideration of consolidated fuel causes no significant chan in the previously analyzed impact loads on the cask; th impact limiter deformations are well within the 85% cr limit.

Accident Conditions, Cont'd

(Side Impact)

Cask Evaluation

Based on the calculations above, the cask bending stress analysis in Section 2.32 of the SAR and the cask center deflection calculation in Section 2.3.3 of the SAR remain essentially unchanged.

Since the Inner Container is supported by the cask Inner Shell along its entire length, the Inner Container is structurally adequate to contain the consolidated fuel canister.

The Inner Closure Head has an extremely large margin of safety for the side drop condition (Ref. Section 2.3.4, SAR).

(Puncture - 40 in. drop onto 6 inch Diameter Pin)

Outer Shell Thickness--Cask Designer's Guide

Referring to page X1-2-26, Section 2.4.1 of Ref. No. 3, the Cask Designers Guide analysis of the Outer Shell is:

$$t_{min} = \left(\frac{1.3 * W}{S}\right)^{0.71} = \underline{0.729 \text{ in. minimum} \quad t_{act} = 0.875 \text{ inches}}$$

where,

W = 49,250 lbs.
S = 100,000 psi for Type 216 Stainless Steel
1.3 = Factor for casks less than 30 inches in diameter

Analysis of Cask Bending (Ref. No. 3, Section 2.4.2, page X1-2-26)

The greatest damage to the cask from the pin puncture event will occur for an impact at mid-length.

The total kinetic energy of the drop is:

$$K.E. = (49250)(40) = 1.97 \times 10^6 \text{ in.-lbs.}$$

Accident Conditions, Cont'd

This energy is absorbed in three steps:

(1) Deformation of the Water Jacket (Shield Tank) wall.
(2) Elastic bending of the cask as a double cantilever upon the 6 inch diameter pin.
(3) Plastic hinge absorption of the remaining K.E.

U_{wj} = Elastic energy absorbed by water jacket

= 306,000 in.-lbs. (Ref. page X1-2-28).

For 1 g static bending of the cask, (Ref. page X1-2-29):

$$M_{center} = \frac{WL}{8} = \frac{(49250)(193)}{8} = 1.188 \text{ X } 10^6 \text{ in.-lbs.}$$

Then, at the yield point the effective weight increase is:

$$R_w = \frac{45169125}{1.188 \text{ X } 10^6} = 38.0$$

Therefore, the elastic energy absorbed in cask bending is:

$$U_{cb} = \frac{(38.0 \text{ X } 49250)^2(193)^3}{(640)(309.792 \text{ X } 10^9)} = 127{,}141 \text{ in.-lbs.}$$ (Ref. No. 3, page X1-2-29)

The remaining kinetic energy which must be absorbed by a plastic hinge is:

$$U_{ph} = 1.97 \text{ X } 10^6 - 306{,}000 - 127{,}141$$

$$= 1{,}536{,}859 \text{ in.-lbs}$$

The maximum plastic moment is 67,753,688 in.-lbs. (pg. X1-2-29).

The energy absorbed by the plastic hinge is:

$$U_{ph} = M_{ph}\,\theta$$

so, $\theta = \frac{1{,}536{,}859}{67{,}753{,}688} = 0.022683$ radians

Accident Conditions, Cont'd

Thus, the extreme fibers of the Outer Shell are stretched very locally:

$$d = \theta R = (0.022683)(13) = 0.2949 \text{ in.}$$

Assuming this occurs over a 2 inch gage length,

$$\text{Strain} = ep = \frac{0.2949}{2} = 0.1474 \text{ in./in. or } 14.7\% \text{ elongation}$$

This strain value is not significantly different than the existing SAR calculated value on page X1-2-29 and is much less than the material's 45% elongation at rupture.

The margins of safety for the following Puncture events are inversely proportional to the increase in weight of the cask:

(1) Top End--Center Impact (Page X1-2-30, Ref. No. 3)
(2) Top End--Side Impact (Page X1-2-33, Ref. No. 3)
(3) Top End--Oblique Impact (Page X1-2-42, Ref. No. 3)
(4) Bottom End--Center Impact (Page X1-2-43, Ref. No. 3)
(5) Bottom End--Off-Center Impact (Page X1-2-49, Ref. No. 3)

As previously calculated in this evaluation, the increase in weight of the cask is $\Delta W = 2.6\%$. Therefore, the margins of safety for the five puncture events are reduced by 2.6%. The margins as listed in the SAR (Ref. 3) are sufficiently large to tolerate this reduction without infringing the adequacy of the cask.

4.0 REFERENCES

1. "Interface Manual, NLI-1/2 Legal Weight Truck (LWT) Spent Nuclear Fuel Shipping Casks," NL Industries, Inc., 1130 Central Avenue, Albany, New York, 12205, Rev. 11/01/79.

2. "NLI Drawing No. 70562F, Rev. 8," National Lead Company, Nuclear Division.

3. "Safety Analysis Report, NLI-1/2 Legal Weight Truck (LWT) Spent Nuclear Fuel Shipping Cask," NL Industries, Inc., 1130 Central Avenue, Albany, New York, 12205.

4. "ASME Boiler and Pressure Vessel Code, Section III, Division 1, Appendices," The American Society of Mechanical Engineers, 1983.

5. "Metals Handbook, Ninth Edition, Volume 2, Properties and Selection: Nonferrous Alloys and Pure Metals," American Society for Metals, Metals Park, Ohio.

6. "Formulas for Stress and Strain," R. J. Roark, McGraw-Hill Book Company, 4th Edition, 1965.

7. "Structural Analysis of Shells," E. H. Baker, L. Kovalevsky and F. L. Rish, McGraw-Hill, 1972.

8. "Cask Designers Guide," ORNL-NSIC-68, L. B. Shappert, Oak Ridge National Laboratory, February, 1970.

SECTION XI

APPENDIX F

Analysis of Configuration C for the Fermi-1 and EBR-II Fuels
LICENSE AMENDMENT - FERMI FUEL, EBR-II BLANKET FUEL

STRUCTURAL EVALUATION

Discussion

The NLI 1/2 cask design weight is 48,000 pounds.

The weight of the NLI 1/2 cask loaded with its maximum possible weight of Fermi Fuel, EBR-II Blanket Fuel or EBR-II Scrap Cladding is calculated to be 48,625 pounds including the basket (Fuel Basket and four EBR-II Blanket Fuel Canisters).

The effect of the increased weight of the cask and contents on the structural adequacy of the NLI 1/2 cask is evaluated. The evaluation shows that the increased weight has a minimal effect on impact stresses. It does result in slightly increased deformation of the impact limiters; however, the limiters are adequately designed to accommodate the increased deformation. Thus, the structural adequacy of the NLI 1/2 cask is not significantly affected by the slight increase in design weight covered by this application. This evaluation is based on the detailed analysis of the NLI 1/2 Cask Consolidation Fuel License Amendment (Appendix E) which envelopes the calculated cask weight for this amendment.

The decay heat of the Fermi-1 and EBR-II fuels are less than the design basis decay heat load of 10.6 kW. Therefore, no additional thermal analyses are required.

The detailed analyses, which follow, demonstrate the structural adequacy of the fuel basket, the Fermi fuel tubes, and the waste basket for the loads associated with the cask contents covered by this amendment request. To assure that the Boral neutron poison remains in position, no inelastic deformation is permitted in the fuel basket for normal operation or hypothetical accident conditions. For criticality safety, the Fermi Fuel Tubes must retain the Fermi Fuel for all loading conditions; thus, inelastic deformation of the tubes is permitted for accident conditions, but ultimate failure is not. The analysis results are summarized on the following page.

Conclusion

The NLI 1/2 Cask, the fuel basket and the waste basket evaluated in this amendment request are structurally adequate to meet all of the applicable regulatory requirements.

NLI 1/2 CASK
LICENSE AMENDMENT-FERMI FUEL, EBR-II BLANKET FUEL
STRUCTURAL EVALUATION

Summary of Results

Analysis	Comment / Margin of Safety
Calculation of Design Weight	+1.3% (Increase); no further analysis required based on a previous amendment for the consolidated fuel canister.
Fuel Basket	
Section A-A, Cruciform Side Arm	*M.S. = +0.44 (Bending *M.S. = +Large (Shear)
Section B-B, Cruciform Lower Arm	*M.S. = +Large (Compression) *M.S. = +Large (Stability)
Section C-C, Drain Structure Cover	*M.S. = +1.03 (Bending)
Section D-D, Weld at End Plugs of Cruciform Arms	*M.S. = +Large (Shear)
Section E-E, Weld-Cruciform Arm to Drain Cover	*M.S. = +Large (Shear & Bending)
Section F-F, Drain Structure Leg	*M.S. = +Large (Bending)
45° Basket Orientation	Not Critical
Bolt-Bottom Attachment of Basket to Drain Assembly	M.S. = +1.18 (Tension for Handling Condition)
Drain Assembly	
Bolt-Attachment to Cask	M.S. = +Large (Shear for Handling Condition)
Longerons	M.S. = +Large (Compression for Handling Condition)
Base Plate	*M.S. = +0.28 (Bending)
Gussets (Six, Radial)	*M.S. = +0.18 (Compression)
Fermi Fuel Tube	M.S. = +0.94 (Bending)
Waste Basket	
Cruciform	*M.S. = +Large (Bending-Normal Operation) *M.S. = +Large (Bending-30 ft. side Drop)

*Design Criteria is Material Yield Strength for both Normal Operation and Accident Load Conditions.

Page Added
Oct. 1990

STRUCTURAL EVALUATION

MATERIAL TO BE SHIPPED

EBR-II BLANKET FUEL

60 SHIPPING CANISTERS DESCRIBED BY DRAWING 089000010
GROSS CANISTER WEIGHT - 715 LBS NOMINAL
CONTENTS - 287.9 KGS DEPLETED URANIUM (0.21% U-235)
- 3.876 KGS Pu, FUEL BURNUP, 0-2400 MWD/MTU
RADIATION LEVELS - 2-5 R/HR AT 6 IN.
HEAT GENERATION - 69 WATTS PER CAN MAXIMUM
TO BE SHIPPED TO ROCKWELL HANFORD OPERATIONS

EBR-II SCRAP CLADDING

145 SHIPPING CANISTERS DESCRIBED BY DRAWING 089000026
GROSS CANISTER WEIGHT - 135 LBS NOMINAL
CONTENTS - STAINLESS STEEL CLADDING AND BUNDLE HARDWARE
RADIATION LEVELS - 4.7 - 187 R/HR AT 1 FT.
HEAT GENERATION - 1.9 WATTS/CAN MAXIMUM
PRIMARY RADIONUCLIDES: Cs^{137}, Sr^{90}, Y^{90} (.05-7.5 Ci), Co^{60} (.084-132.4 Ci)
TO BE SHIPPED TO THE NEVADA TEST SITE.

FERMI FUEL

205 FUEL ASSEMBLIES, APPROXIMATELY 2.7 IN. SQUARE X 35 IN. LONG.
WEIGHT EQUALS 60.6 LBS. EACH.

8 FUEL ASSEMBLIES IN WELDED STAINLESS STEEL CANS, APPROXIMATELY 2.93 INCHES SQUARE X 26.6 IN. LONG. WEIGHT EQUALS 74.0 LBS. EACH.
FUEL FORM IS U-10% Mo 25.6% ENRICHED.
EACH ASSEMBLY CONTAINS A NOMINAL 4.8 Kg U-235
FUEL BURNUP - 410-2840 MWD/MTU
RADIATION LEVELS - 12R/HR MAXIMUM AT 3 FT.
HEAT GENERATION - 0.6-1.2 WATTS/ASSEMBLY
TO BE SHIPPED TO ROCKETDYNE

FIGURE WITHHELD UNDER 10 CFR 2.390

CONTAINER-FERMI FUEL

(

NLI-1/2 CASK
LICENSE AMENDMENT - FERMI FUEL, EBR-II BLANKET FUEL

STRUCTURAL EVALUATION

Weight Calculations

(Cask body) From NLI-1/2 Interface Manual	W_{cb} = 42,505 lbs. (1)
(Outer closure head) From NLI-1/2 Interface Manual	W_{och} = 340 lbs. (2)
(Inner closure head) From NLI-1/2 Interface Manual	W_{ich} = 745 lbs. (3)
(Top impact structure) From NLI-1/2 Interface Manual	W_{tis} = 865 lbs. (4)
(Bottom impact structure) From NLI-1/2 Interface Manual	W_{bis} = 460 lbs. (5)
	=====================
(Empty Cask Weight)	$(We)_t$ = 44,915 lbs. (6)
(Rockwell Fuel Basket and Drain Assy.) From Drawings 460052 - D1, D2, F3, D5 & D6	W_{fbd} = 850 lbs. (7)
(Rockwell Waste Basket and Drain From Drawings 460052 - D1, D2, D4, D5 & D6	W_{wbd} = 330 lbs. (8)
(EBR-II Blanket Fuel)	
One Canister (From Rockwell Data)	$(W_{ebr})_1$ = 715 lbs. (9)
Four Canisters	$(W_{ebr})_4$ = 2860 lbs.(10)
(EBR-II Scrap Cladding)	
One Canister (From Rockwell Data)	$(W_{sc})_1$ = 135 lbs.(11)
Four Canisters	$(W_{sc})_4$ = 540 lbs.(12)
(Fermi Fuel) From Rockwell Data	
One Assembly	$(W_a)_1$ = 61 lbs.(13)
Sixteen Assemblies	$(W_a)_{16}$ = 976 lbs.(14)
One Canned Assembly	$(W_c)_1$ = 74 lbs.(15)
Eight Canned Assemblies	$(W_c)_8$ = 592 lbs.(16)
	=====================
Maximum Loaded Cask Weight	W_{max} = 48,625 lbs.

SAR Cask Design Weight, W_d = 48,000 lbs.

STRUCTURAL EVALUATION

Material Properties

(Type 304 stainless steel)

Su = 75 ksi Sy = 30 ksi

Ref. ASME Section III, Division 1, Appendix I.

(SA-193, Gr B7 low alloy bolting material)

Su = 125 ksi Sy = 105 ksi

Ref. ASME Section III, Division 1, Appendix I.

(6061-T6 aluminum alloy)

Su = 45 ksi Sy = 40 ksi

Ref. "Metals Handbook, Ninth Edition, Volume 2, Properties and Selection: Nonferrous Alloys and Pure Metals," American Society for Metals, Metals Park, Ohio.

Design G-Loads

1 foot end drop	39.4g
1 foot side drop	9.62g
1 foot 45° oblique drop	6.75g
1 foot corner drop	<39.4g
30 foot end drop	39.4g
30 foot side drop	36.28g
30 foot corner drop	37.48g

NLI-1/2 CASK
LICENSE AMENDMENT - FERMI FUEL, EBR-II BLANKET FUEL

STRUCTURAL EVALUATION

Allowable Stresses

The allowable stresses used in this evaluation are chosen to be the same as those used in the NLI-1/2 Cask Safety Analysis Report.

S_t = Tensile stress

S_c = Compressive stress

S_{br} = Bearing stress

S_b = Bending stress

S_s = Shear stress

$S_t = S_c = S_{br} = S_b \leqq Sy$

$S_s = 0.5\ Sy$

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STRUCTURAL EVALUATION

References

1. "Interface Manual, NLI 1/2 Legal Weight Truck (LWT) Spent Nuclear Fuel Shipping Casks," NL Industries, Inc., 1130 Central Avenue, Albany, New York, 12205, Rev. 11/01/79.

2. "NLI Drawing No. 70562F, Rev. 8," National Lead Company, Nuclear Division.

3. "ASME Boiler and Pressure Vessel Code, Section III, Division 1, Appendices," The American Society of Mechanical Engineers, 1983.

4. "Metals Handbook, Ninth Edition, Volume 2, Properties and Selection: Nonferrous Alloys and Pure Metals," American Society for Metals, Metals Park, Ohio.

5. "Formulas for Stress and Strain," R.J. Roark, McGraw-Hill Book Company, 4th Edition, 1965.

6. "Structural Analysis of Shells", E.H. Baker, L. Kovalevsky and F.L. Rish, McGraw-Hill, 1972.

7. "Cask Designers Guide," ORNL-NSIC-68, L.B. Shappert, Oak Ridge National Laboratory, February, 1970.

NLI 1/2 Cask
License Amendment - Fermi Fuel, EBR-II Fuel & Cladding

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Structural Evaluation

DETAILED ANALYSIS

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NLI 1/2 Cask
License Amendment – Fermi Fuel, EBR-II Fuel & Cladding

Structural Evaluation

Cask Components

Increased Design Weight

The weight of the NLI 1/2 Cask loaded with its maximum possible weight of Fermi Fuel, EBR-II Blanket Fuel or EBR-II Scrap Cladding is calculated to be:

W_{max} = 48,625 lbs. (Fuel Basket and four EBR-
Blanket Fuel Canisters)

Then, the design weight of the cask for this evaluation is:

$(W_{EBR})_{DES}$ = 48,625 lbs.

The design weight of the cask that is used in the SAR is:

$(W_{SAR})_{CASK-DES}$ = 48,000 lbs.

Thus, the percentage increase in weight of the loadec cask is:

$$\Delta W = \frac{48625 - 48000}{48000}(100) = +1.30\%\ \text{increase}$$

Based on the structural evaluation of the NLI 1/2 Ca performed for the Consolidated Fuel License Amendme submitted August 9, 1985, and approved as Revision 16 Certificate of Compliance No. 71-9010 on January 22, 1986 the structural adequacy of the NLI 1/2 Cask is not significantly affected by this slight increase in de weight.

Refer to Appendix E for the structural evaluation for the Consolidated Fuel License Amendment which evaluated a 2.6% increase in design weight.

Thus, no further structural analysis of the NLI 1/2 cask components is required.

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NLI 1/2 Cask
License Amendment - Fermi Fuel, EBR-II Fuel & Cladding

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Structural Evaluation

Fuel Basket Geometry

FIGURE WITHHELD UNDER 10 CFR 2.390

NLI 1/2 Cask

License Amendment - Fermi Fuel, EBR-II Fuel & Cladding

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Structural Evaluation

Bending Analysis - Side Arm (Section A-A)

Load - Side Drop with Drain Structure Down

Since the design criteria requires no inelastic deformation for either normal operation or accident conditions, the 30 foot side drop (36.28g) is critica

EBR-II Fuel: $(W_{ebr})_1 = 715$ lbs. ; $D = 4.875$ in. ; $L = 165.13$ in

Maximum Moment Arm $d = 2.90$ in. (graphical determination

Moment $M = (2.90)(715)(36.28g) = 75{,}237$ in-lbs.

Shear $V = (715)(36.28) = 25{,}940$ lbs.

Arm Properties

The arm is constructed of a Boral plate sandwic between two stainless steel plates; only the stainle. steel plates are effective and they are assumed to act independently.

$L = 174.15$ in.

$t = 0.25$ in.

$c = 0.125$ in.

$$I = \frac{(174.15)(0.25)^3}{12} = 0.227\ \text{in.}^4$$

$I/c = 1.814$ in.^{3} $\quad A = Lt = 13.5$ in.^{2}

Allowable Stress (Type 304 Stainless Steel)

$S'_B = S_y = 30.0$ ksi $\qquad S'_s = 0.5\,S_y = 15.0$ ksi

Calculated Stress

$$S_B = \frac{M}{2(I/c)} = 20{,}735\ \text{psi}$$

$$M.S = \frac{S'_B}{S_B} - 1 = \underline{\underline{+0.44}}$$

$$S_s = \frac{V}{2A} = 298\ \text{psi}$$

$$M.S = \frac{S'_s}{S_s} - 1 = \underline{\underline{+\ \text{Large}}}$$

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NLI 1/2 Cask
License Amendment-Fermi Fuel, EBR-II Fuel & Cladding
Structural Evaluation

Compression Analysis - Lower Arm (Section B-B)

Load - Side Drop, Drain Structure Not Down

$(W_{ebr})_1 = 715$ lbs. (One fuel Canister)
$W_{fbd} = 850$ lbs. (basket & drain assy.)
$g = 36.28$ (30 foot side drop)

$$P_c = [(850-100) + (715)(2)](36.28) = 79,090 \text{ lbs.}$$

Section Properties

$t_p = 0.25$ in.
$L_p = 174.15$ in.
$A_p = 2 L_p t_p = 87.1 \text{ in.}^2$

$E = 28.0 \times 10^6$ psi
$\nu = 0.275$
$a = 6.25$ in.

$K = 2.0$ (free - fixed column)

Allowable Stress (Type 304 Stainless Steel)

$S_c' = S_y = 30.0$ ksi

$$S_s' = \frac{\pi^2 E}{(12)(1-\nu)^2 (Ka/t)^2}^{*} = 9965 \text{ psi}$$

Calculated Stress

$$S_c = \frac{P_c}{A_p} = 908 \text{ psi}$$

$$M.S. = \frac{S_c'}{S_c} - 1 = \underline{\underline{+\text{Large}}}$$

For Stability

$$M.S. = \frac{S_s'}{S_c} - 1 = \underline{\underline{+\text{Large}}}$$

---

* Ref. Guide to Stability Design Criteria for Metal Structures, B.G. Johnston, 3rd. Ed., 1976, page 85, Eqn. 4.3.

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NLI 1/2 Cask
License Amendment-Fermi Fuel, EBR-II Fuel & Cladding

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Structural Evaluation

Drain Protection Cover (Section C-C)

Load - Side Drop with Drain Structure Down

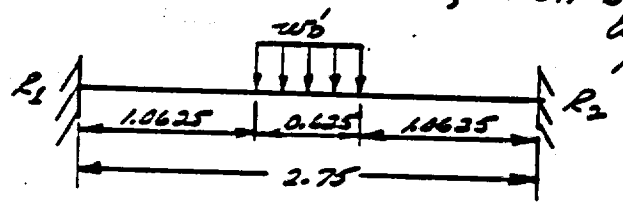
$(W_{ebr})_1 = 715$ lbs. (One fuel Canister)
$W_{fbd} = 850$ lbs. (Basket & Drain Assy)
$g = 36.28$ (30 foot Side Drop)

$$P_D = [(850-100) + (715)(2)](36.28) = 79,090 \text{ lbs.}$$

$$w_D = \frac{P_D}{L} = \frac{79090}{174.15} = 454 \text{ lbs/in. of length}$$ which is distributed over a width of 0.625 in.

$$w_D' = \frac{w_D}{0.625} = 726.6 \text{ lbs/in.}$$

For a unit width beam, Ref. Blodgett, "Design of Welded Structures", page 8.1-14, Case 42.



$$R_1 = \frac{(726.6)(0.625)}{(4)(2.75)^3}\left[(4)(1.375)^2(2.75+2.75) - (0.625)^2(\overset{0}{\cancel{1.0625}}-1.0625)\right]$$
$$= 227.0 \text{ lbs.}$$

$$R_2 = (726.6)(0.625) - 227.0 = 227.0 \text{ lbs.}$$

$$M_1 = \frac{(726.6)(0.625)}{(24)(2.75)^2}\left[(0.625)^2\left(2.75 + (3)\overset{0}{\cancel{0}}\right) - (24)(1.375)^2(1.375)\right]$$
$$= -153.4 \text{ in-lbs.}$$

$$M_2 = (227.0)(2.75) - (726.6)(0.625)(1.375) + (-153.4)$$
$$= -153.4 \text{ in-lbs.}$$

$$M_{max(+)} = -153.4 + (227.0)\left(1.0625 + \frac{227.0}{(2)(726.6)}\right) = 123.2 \text{ in-lbs.}$$

Beam Properties. (Assumed Unit Width)

$w = 1.00$ in.
$t = 0.25$ in.
$c = 0.125$ in.

$$I = \frac{(1.00)(0.25)^3}{12} = 0.0013 \text{ in.}^4$$
$$I/c = 0.0104 \text{ in.}^3$$

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NLI 1/2 Cask
License Amendment - Fermi Fuel, EBR-II Fuel & Cladding

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Structural Evaluation

Drain Protection Cover (Cont'd.) (Section C-C)

Allowable Stress (Type 304 Stainless Steel)

$S_B' = S_y = 30.0$ ksi

Calculated Stress

$S_B = \frac{M_L}{I/c} = 14,750$ psi $\qquad$ M.S. $= \frac{S_B'}{S_B} - 1 = \underline{\underline{+1.03}}$

Note: The corners of the drain protection struct
are chamfered so that the fuel canister can
come into contact with them. A minimum thick
of 0.25 in. will remain at the corner.

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Structural Evaluation

Weld at End Plugs (Section D-D)

Load - Side Drop with Drain Structure Up

For a side drop loading the fuel canisters load the side arms of the cruciform in bending which tends to separate the two plates compos. the upper arm. The weld at the end plug of the upper arm is assumed to resist this separation.

From the previous bending analysis the moment on the cruciform is:

$$M = 75{,}227 \text{ in-lbs.}$$

The distance of the end plug weld from the side arm is a minimum for the drain protection structure arm which is the shortest one and equals 4.75 in.

Then, the load on the weld is: $P_w = \frac{75227}{4.75} = 15837 \text{ l}$

Weld Properties (1/8" Fillet - conservatively assumed)

$w = 0.125 \text{ in.}$ $L = 174.15 \text{ in.}$ $A = .707 wL = 15.39 \text{ in.}^2$

Allowable Stress (Equivalent to Type 304 Stainless Steel)

$S_s' = 0.5 S_y = 15.0 \text{ ksi}$

Calculated Stress

$S_s = \frac{P_w}{A} = 1029 \text{ psi}$ M.S. $= \frac{S_s'}{S_s} - 1 =$ <u>+Large</u>

NLI 1/2 Cask
License Amendment – Fermi Fuel, EBR-II Fuel & Cladding

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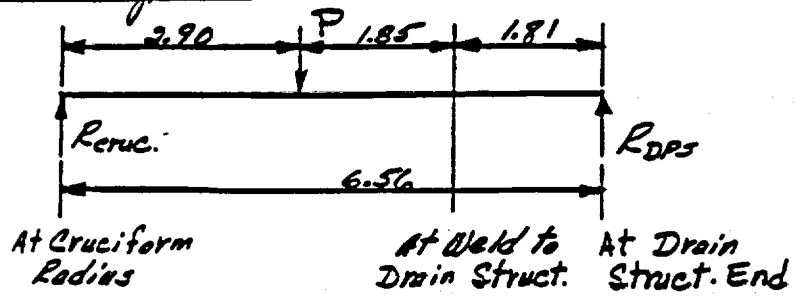
7/16/...

Structural Evaluation

Arm With Drain Protection Structure

Geometry



Loads

$(W_{ESR})_1 = 715$ lbs. ; $P = (715)(36.28g) = 25,940$ lbs.

$R_{cruc.} = \frac{(25,940)(3.66)}{6.56} = 14,473$ lbs. ; $R_{DPS} = \frac{(25940)(2.90)}{6.56} = 11,46$

At P, $M_{max} = \frac{(25940)(2.90)(3.66)}{6.56} = 41,971$ in-lbs.

(This moment is smaller than that previously used in the arm analysis; thus, arm bending is not evaluated here.)

At Weld $M_w = (1.81)(11467) = 20,755$ in-lbs.

Weld Properties (1/8" Fillet Each Side)

$^*A_w = (2)(170.5) = 341$ in.

$^*S_w = (0.75)(170.5) = 127.9$ in.^{2}

Allowable Load (Equivalent to 30k st. stl)

$^*f_a = (.707)(.5)(30000)(1/8) = 1325$ lbs/in.

Calculated Weld Load

$$\left.\begin{aligned} f_s &= \frac{R_{DPS}}{A_w} = 34 \text{ lbs/in.} \\ f_b &= \frac{M_w}{S_w} = 162 \text{ lbs/in.} \end{aligned}\right\} f_e = (f_s^2 + f_b^2)^{1/2} = 166 \text{ lbs/in.}$$

$$M.S. = \frac{f_a}{f_e} - 1 = \underline{+ \text{Large}}$$

* Blodgett, "Welded Structures", pg. 7.4-7.

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NLI 1/2 Cask
License Amendment-Fermi Fuel, EBR-II Fuel & Cladding

Structural Evaluation

Leg Bending Analysis-Drain Structure (Section F-F)

Loads (Ref. Previous Page)

$R_{DPS} = 11{,}467$ lbs. ; $M = (1.10)(R_{DPS}) = 12{,}614$ in-lbs.

Section Properties

$t = 0.25$ in. $\qquad I = \frac{bt^3}{12} = 0.222$ in.^{4}

$b = 17.05$ in.

$c = 0.125$ in. $\qquad I/c = 1.776$ in.^{3}

Allowable Stress (Type 304 Stainless Steel)

$S'_B = S_y = 30.0$ ksi

Calculated Stress

$S_B = \frac{M}{I/c} = 7102$ psi

$M.S. = \frac{S'_B}{S_B} - 1 = \underline{\underline{+\text{Large}}}$

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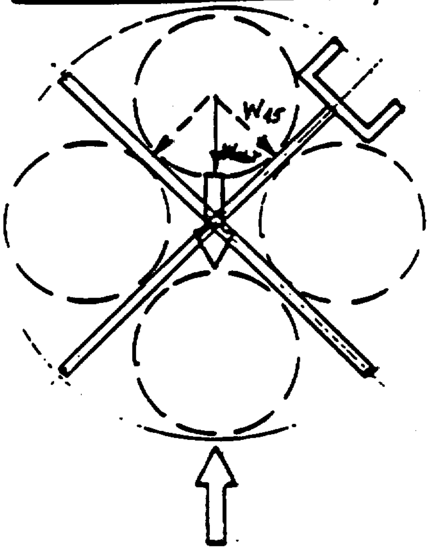
7/2/8

NLI 1/2 Cask
License Amendment - Fermi Fuel, EBR-II Fuel & Cladding

Structural Evaluation

Arm Bending Analysis

Load - Side Drop with Basket Rotated 45°



$(W_{ebr})_1 = 715$ lbs.

$W_{45} = (W_{ebr})_1 \cos 45° = 505.6$ lbs.

$g = 36.28$ (30 foot side drop)

$d = 2.90$ in. (Moment Arm)

$M = 2.90\ W_{45}(36.28) = 53,193$ in-lb on each of the upper arms (and each of the lower arms)

The cruciform loads for this basket orientation for a 30 foot side drop are lower than those previously evaluated for the basket oriented with the cruciform arms vertical and horizontal. Therefore, no further analysis is required.

NLI 1/2 Cask
License Amendment-Fermi Fuel, EBR-II Fuel & Cladding

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Structural Evaluation

Bolt-Bottom Attachment of Basket to Drain Assy.

Load

$(P_H)_f = 136 \text{ lbs.}^{*}$ (Friction due to Canister Removal)

Friction resistence force on the exterior of the basket is conservatively neglected.

Thread Properties (1/4-20 UNC)

$A_t = 0.0318 \text{ in.}^2$

Allowable Stress (304 St. Steel)

$S_t' = S_y = 30.0 \text{ ksi}$

Calculated Stress

$$S_t = \frac{(P_H)_f}{A_t} = 13{,}711 \text{ psi}$$

$$M.S. = \frac{S_t'}{S_t} - 1 = \underline{\underline{+1.18}}$$

---

* $(W_{EBR})_t = 715 \text{ lbs.}$
Coefficient of friction - Alum. Canister on St. Stl. Cruciform
$f_{static} = 0.61$; $f_{sliding} = 0.47$ (Ref. Mark's Hdbk., pg. 3-35, Table 1
$(P_H)_f = (0.61)(715) = 436 \text{ lbs.}$

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NLI 1/2 Cask
License Amendment - Fermi Fuel, EBR-II Fuel & Cladding

TCT
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Structural Evaluation

Bolt - Drain Attachment to Cask

Load (Ref. Previous Page)

$P_B = (P_H)_{\text{¢}} = 436 \text{ lbs.}$

Bolt Properties (3/8-16 UNC x 2 Long)

$D = 0.28 \text{ in.} \qquad A_S = 0.0616 \text{ in.}^2$

Allowable Stress (18-8 Stainless Steel)

$S'_S = 0.5 S_y = 15.0 \text{ ksi}$

Calculated Stress

$$S_S = \frac{P_B}{2A_S} = 3539 \text{ psi}$$

$$M.S. = \frac{S'_S}{S_S} - 1 = \underline{\underline{+\text{Large}}}$$

Longeron(s) - Drain Assembly Support

Load

$P_L = P_B = 436 \text{ lbs.}$

Angle Properties (1 x 1 x 1/8 Angle)

$A_L = 0.234 \text{ in.}^2$ (Each)

Allowable Stress (304 St. Steel)

$S'_L = S_y = 30.0 \text{ ksi}$

Calculated Stress

$$S_L = \frac{P_L}{2A_L} = 932 \text{ psi}$$

$$M.S. = \frac{S'_L}{S_L} - 1 = \underline{\underline{+\text{Large}}}$$

NLI 1/2 Cask
License Amendment-Fermi Fuel, EBR-II Fuel & Cladding

TCT
7/11/8.

Structural Evaluation

Base Plate - Drain Assy.

Load (Normal Operation - 39.4g One foot End Drop)

$W_{LB} = 2860 + 850 = 3710$ lbs.

$P_{BPN} = (39.4)(3710) = 146,174$ lbs. $\quad p_{BPN} = P_{BPN}/A_{PL} = 1101.5$ psi

Plate Properties

$D = 13.00$ in. $\quad t = 0.375$ in. $\quad A_{PL} = \pi/4\, D^2 = 132.7$ in.^{2}
$a = 1/2(D - 4.00) = 4.50$ in. (center tube outward)

Allowable Stress (Type 304 Stainless Steel)

$S_b' = S_y = 30.0$ ksi $\qquad S_u = 75.0$ ksi

Calculated Stress (Roark, 4th Ed., Page 234, Case 67)

(60° Circular Sector - $\beta = 0.147$)

$$S_b = \frac{\beta p_{BPN} a^2}{t^2} = 22317 \text{ psi}$$

$$M.S. = \frac{S_b'}{S_b} - 1 = \underline{\underline{+0.28}}$$

(Accident Condition)

The g-load for the 30 foot end drop accident condition is 39.4g, the same as for the one foot dr. Therefore, no further analysis is required.

NLI 1/2 Cask
License Amendment - Fermi Fuel, EBR-II Fuel & Cladding

TCT
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Structural Evaluation

Gussets - Drain Assy. Base Plate

Load (Normal Operation - 39.4g One Foot End Drop)

Loaded Basket, $W_{LB} = 2860 + 850 = 3710$ lbs.

$P_{GN} = (39.4)(3710) = 146,174$ lbs.

Compression Area (Center Tube & 6 Radial Gussets)

$A_c = (\pi/4)(4.00^2 - 3.548^2) + (6)(0.25)(2.50) - (0.75)(0.226)(4) = 5.75$ in^2.

Allowable Stress (Type 304 Stainless Steel)

$S_c' = S_y = 30.0$ ksi

Calculated Stress

$S_c = \frac{P_{GN}}{A_c} = 25,413$ psi

$M.S. = \frac{S_c'}{S_c} - 1 = +0.18$

(Accident Condition)

The g-load for the 30 foot end drop accident condition is 39.4g, the same as for the one foot drop. Therefore, no further analysis is required.

7/15/86 CWH

NLI 1/2 CASK
LICENSE AMMENDMENT - "FERMI FUEL LOADING"
STRUCTURAL EVALUATION "FUEL TUBE"

The fuel tube provided is 5" O.D. x .25" wall comprised of aluminum alloy 6061-T6511 having mechanical properties* of 38 ksi ultimate and 35 ksi yield strength (σ_y). The most severe loading on the tube occurs when 6 Fermi fuel (second fuel assembly described by material shipping section, "Fermi Fuel") canisters having a weight of 74 lbs. each are loaded in the tube under the accident condition of a 30' side drop which induces an acceleration load of 36.28 g's on the fuel tube. In the analysis of this condition it is assumed that the tube is supported by the cruciform and the cask and therefore the tube is supported at locations 135° apart. It is conservatively assumed that the Fermi fuel will act as a point load in the center of the 135° span.

* Mil Handbook 5C.

7/15/86 Coller

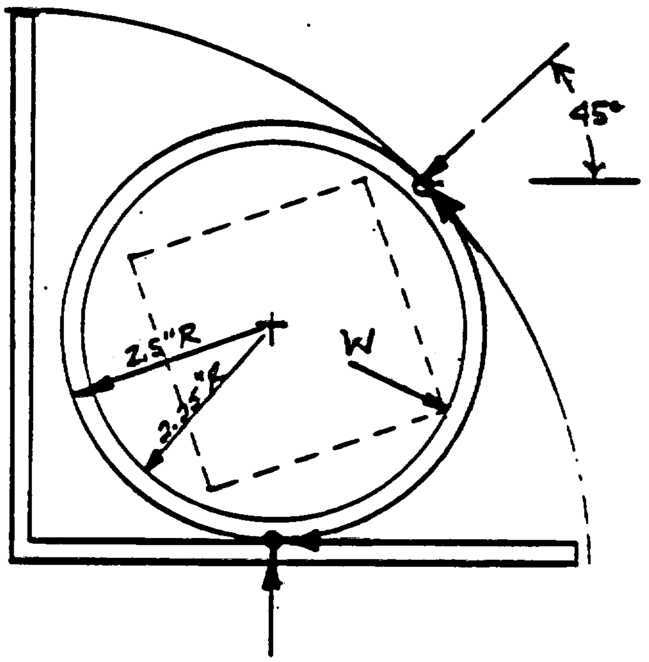
NLI 1/2 CASK
LICENSE AMENDMENT "FOUR FUEL LOADING"
STRUCTURAL EVALUATION "FUEL TUBE"

THE LOAD PER UNIT LENGTH OF THE TUBE IS:

W = (6 cans) x (7.4 Lbs/can) x (36.28 g's) ÷ 170" length

W = 94.76 Lb/INCH

THE FREE BODY DIAGRAM AND SKETCH OF THE LOADING IS AS SHOWN BELOW:



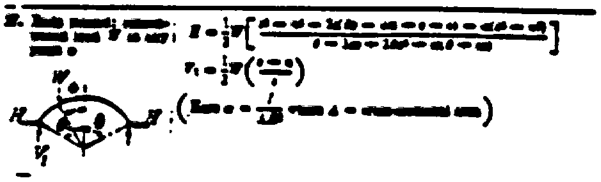
TO DETERMINE THE MOST ADVERSE BENDING STRESS IN THE TUBE THE REACTIONS MUST BE DETERMINED THEN THE MAXIMUM BENDING MOMENT CAN BE FOUND AT THE LOCATION OF THE LOAD POINT. THIS IS conservatively computed by Analysis OF THE 135° SEGMENT Assumed. As A Circular Arch.

7/15/86 Collier

NLI 1/2 CASK

LICENCE AMMENDMENT "FERMI FUEL LOADING"

STRUCTURAL EVALUATION "FUEL TUBE"



*

$s = \sin\theta = \sin 67.5^\circ = .9239$

$n = \sin\phi = \sin(0) = 0$

$c = \cos\theta = \cos 67.5^\circ = .3827$

$\theta = 67.5^\circ = 1.178$ RAD's

$\phi = 0$

$e = \cos\phi = 1$

$$\left.\begin{array}{l} I = \frac{(1)(.25)^3}{12} = .0013\,\text{in}^4 \\ A = (1)(.25) = .25\,\text{in}^2 \\ R = 2.50'' \end{array}\right\} \therefore \alpha = \frac{I}{AR^2} = \frac{.0013}{(.25)(2.50)^2} = .000333$$

$W = 94.76$ Lbs.

* Reference is Given To Roarke 4th Edition Case # 27; Page 179

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NLI 1/2 CASK

LICENSE AGREEMENT "FERMI FUEL LOADING"

STRUCTURAL EVALUATION "FUEL TUBE"

$$H = \frac{W}{2}\left[\frac{s^2 - \cancelto{0}{n^2} - 2c(\theta s - \phi \cancelto{0}{n} + c - e) - \alpha(s^2 - \cancelto{0}{n^2})}{\theta - 3sc + 2\theta c^2 + \alpha(\theta + sc)}\right]$$

$$= 47.38\left[\frac{.9239^2 - 2(.3827)\left[(1.178)(.9239) + .3827 - 1\right] - .000833\,(.9239)}{1.178 - 3(.9239).3827 + 2(1.178).3827^2 + .00128}\right.$$

$\leftarrow = \alpha(\theta$

$H = 50.32$ Lbs.

$$V_1 = \tfrac{1}{2}W\left(\frac{s + \cancelto{0}{n}}{s}\right) = \frac{W}{2}$$

$$V_1 = \frac{94.76}{2}$$

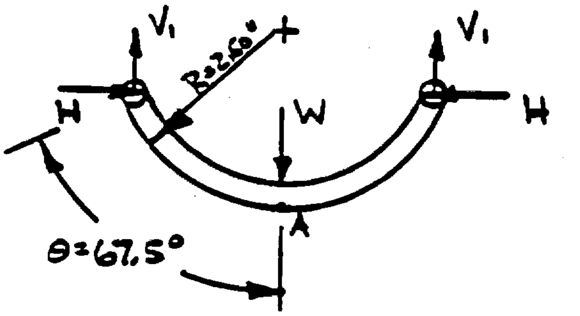
$V_1 = 47.38$ Lbs.

7/15/86 Coller

NLI 1/2 CASK

LICENSE AGREEMENT "FERMI FUEL LOADING"

STRUCTURAL EVALUATION "FUEL TUBE"



The Bending Moment (M_A) at Point A is the Maximum on The Tu

$$M_A = H(R - R\cos\theta) + V_1 R\sin\theta$$
$$= HR(1-\cos\theta) + V_1 R\sin\theta$$
$$= 50.32(2.5)(1-\cos 67.5) + 47.38(2.50)\sin 67.5^\circ$$
$$\underline{M_A = 187.09 \text{ INCH-LBS.}}$$

The Bending Stress At Point A (σ_A):

$$\sigma_A = \frac{M_A\, c}{I} \quad ; \quad c = .25/2 = .125"$$
$$I = (1)(.25)^3/12 = .0013 \text{ in}^4$$

$$\therefore \sigma_A = 17,989 \text{ Psi} = 17.99 \text{ Ksi} \approx 18 \text{ Ksi}$$

The Yield Margin of Safety is $\underline{M.S.} = \frac{\sigma_y}{\sigma_A} - 1 = \frac{35}{18} - 1 = \underline{.94}$

∴ The Tube Maintains The Fuel in Position under Most Adverse Cond

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TCT
7/1/86

NLI 1/2 Cask
License Amendment-Fermi Fuel, EBR-II Fuel & Cladding

Structural Evaluation

Waste Basket - Cruciform Bending

Load

$(W_{sc})_L = 135$ lbs. One Waste Canister

Moment $= M_{wc} = (135)(2.95) = 398$ in-lbs.

Section Properties

$t = 0.25$ in. $I = {}^{Lt^3}/_{12} = 0.227$ in.^{4} $I/c = 1.814$ in.^{3}
$L = 174.15$ in. $c = 0.125$ in.

Allowable Stress (304 Stainless Steel)

$S_B' = S_y = 30.0$ ksi

Calculated Stress

$$S_B = \frac{\overset{*}{[illegible]} M_{wc}}{I/c} = 2111 \text{ psi}$$

$$M.S. = \frac{S_B'}{S_B} - 1 = \underline{\underline{+ \text{Large}}}$$

* One foot side Drop G-Load

30 foot Side Drop Accident

$$S_B = \frac{\overset{**}{36.3} M_{wc}}{I/c} = 7964 \text{ psi}$$

$$M.S. = \frac{S_u}{S_B^u} - 1 = \underline{\underline{+ \text{Large}}}$$

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4.3 Mechanical Properties of Materials

6061-T6 Aluminum Alloy (Reference No. 1)

Temperature (^{o}F)	Ultimate Strength S_u (ksi)	Yield Strength S_y (ksi)
-112	49	42
-18	47	41
75	45	40
212	42	38
300	34	31
400	19	15
500	7.5	5

At the Design Temperature of $260^{o}F$: $S_u = 37.6$ ksi; $S_y = 34.2$ ksi.
Density = 0.098 lbs/in^3.
Modulus of Elasticity: E = 10.0E6 psi.
Poisson's Ratio = 0.33 .

4.4 Allowable Stresses

(Tension) $S'_t = S_y$

(Compression) $S'_c = S_y$

(Bending) $S'_b = S_y$

(Shear) $S'_s = 0.5S_y$

(Bearing) $S'_{br} = S_y$

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4.5 References

1. "Metals Handbook Ninth Edition, Volume 2, Properties and Selection: Nonferrous Alloys and Pure Metals," American Society for Metals, Metals Park, Ohio.

2. "Formulas for Stress and Strain," Roark, R.J., 4th Edition, McGraw-Hill, 1965.

3. NAC Drawing 347-291-F3, Liner - 3 Element, NLI 1/2 Cask, Fuel Movement Project, Assy of.

SECTION XI

APPENDIX G

LINER - 3 ELEMENT LICENSING ANALYSIS

1.0 Structural Evaluation of the Liner - 3 Element (Basket)

This appendix documents the structural adequacy of the Liner - 3 Element (Basket) for the NLI-1/2 Spent Fuel Shipping Cask. The design temperature is 260 ^{o}F. The conservatively calculated minimum margin of safety for any component is +0.43. The Liner - 3 Element is structurally adequate to satisfy all regulatory requirements..

2.0 Discussion

The purpose of the Liner - 3 Element (basket) for the NLI-1/2 Cask is to safely, efficiently, and effectively retain and support intact irradiated fuel rods during normal operation handling and during transport in the cask. The basket has no function related to containment because the fuel rods are demonstrated to be intact by leak detection testing ("sipping"). Also, the basket does not have a criticality control function, since the fuel is not enriched and cannot sustain a nuclear chain reaction in any geometric arrangement.

The Liner - 3 Element consists of three 5.625-inch outside diameter tubes which are restrained and supported radially in the cask cavity by plate segments at the top, the bottom, and three intermediate locations along the length of the basket. Longitudinal restraint is provided by an integral spacer on the bottom of the liner. Each of the three tubes can contain up to seven fuel rods which are contained in a transfer basket.

3.0 Method of Analysis

The design of the liners provides simple, clearly defined load paths. The detailed structural analysis utilizes classical, hand calculation methods of stress analysis. Conservative loads and loading assumptions are used throughout the analyses. The material properties used in these analyses are conservatively based on the minimum specified tensile strength of the material, rather than on the actual tensile strength, which always considerably exceeds the specified minimum. The allowable stresses utilized in these analyses are the material yield strengths, which are obtained from Reference 1.

4.0 Analysis Input

4.1 Load Conditions

(A) The design temperature is 260°F.

(B) The design loads are based on the weight of the liners and their contents subjected to the appropriate g-load factor for the critical normal operation or handling condition:

Handling	2.0g (Assumed)
1-Foot End Drop	39.4g
1-Foot Side Drop	16.4g

4.2 Weights

Fuel Rod	130 lbs
Liner - 3 Element (Empty)	115 lbs
Ruptured Rod Liner (Emtpy)	200 lbs
Intact Fuel Rod Basket (Empty)	30 lbs
Failed Fuel Rod Canister (Empty)	15 lbs
Liner - 3 Element (Fully Loaded with 21 Fuel Rods)	2935 lbs
Ruptured Rod Liner (Fully Loaded with 10 Fuel Rods)	1650 lbs
Intact Fuel Rod Basket (Fully Loaded with 7 Fuel Rods)	940 lbs
Failed Fuel Rod Canister (Fully Loaded with 1 Fuel Rod)	145 lbs

5.0 Detailed Analysis

Liner - 3 Element (Drwg. No. 347-291-F3)
NLI 1/2 Cask

-5 Tube - Contact Stress
Load (1-Foot Side Drop)

$P_{IFRB} = (16.4)(940) = 15,416 \text{ lbs.}$

$p_{IFRB} = P_{IFRB}/L = 15416/157.0 = 98.2 \text{ lbs/in.}$

Tube Properties

$O.D. = 5.625 \text{ in.}$ $\nu = 0.33$ $L = 157.0 \text{ in.}$

$I.D. = 5.375 \text{ in.}$ $E = 10.0 \times 10^6 \text{ psi}$

$t = 0.125 \text{ in.}$ $R = 2.8125 \text{ in.}$

$$b^* \approx 2.15 \left(\frac{p_{IFRB}}{E} \times \frac{D_1 D_2}{D_1 + D_2} \right)^{1/2} = 0.011 \text{ in.}$$

Allowable Stress (6061-T6 Alum. Alloy)

$S'_c = S_y = 34.2 \text{ ksi}$ $S'_b = S_y = 34.2 \text{ ksi}$

Calculated Stress

$$S_c = 0.591 \left(\frac{pE(D_1 + D_2)}{D_1 D_2} \right)^{1/2} = 11,507 \text{ psi}$$

Margin of Safety

$$M.S. = \frac{S'_c}{S_c} - 1 = \underline{\underline{+1.97}}$$

* Ref. No. 2, page 320, Case 5.

Liner - 3 Element (Drwg. No. 347-291-F3) NLI 1/2 Cask

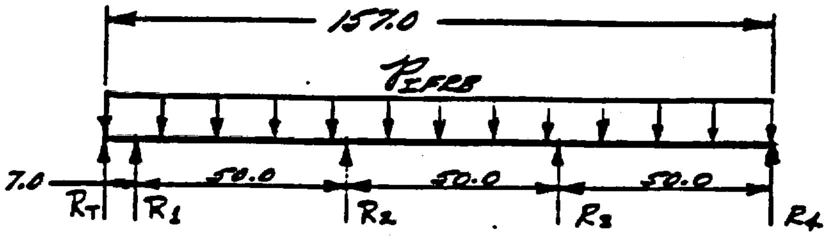
-5 Tube - Beam Analysis

Load (1-Foot Side Drop)

Since one tube rests directly on a second one, the weight of two loaded tubes is considered to be a distributed load on one tube acting as a beam.

$$P_{IFRB} = (2)(16.4)(940) = 30{,}832 \text{ lbs.}$$

$$p_{IFRB} = \frac{P_{IFRB}}{L} = 30832/157.0 = 196.4 \text{ lbs/in.}$$



The reactions are determined based on the span which they support:

$$R_T = (7.0/2)(196.4) = 687.3 \text{ lbs.}$$

$$R_1 = (7.0/2 + 50.0/2)(196.4) = 5{,}596.9 \text{ lbs.}$$

$$R_2 = R_3 = (50.0)(196.4) = 9{,}819.1 \text{ lbs.}$$

$$R_4 = (50.0/2)(196.4) = 4{,}909.6 \text{ lbs.}$$

$M_{max.} = 61{,}369.4$ in-lbs (at center of each 50 ft. span)

$V_{max.} = 4{,}909.6$ lbs.

Liner - 3 Element (Drwg. No. 347-291-F3)
NLI 1/2 Cask

-5 Tube - Beam Analysis (cont'd.)

Tube Properties

O.D. = 5.625 in. $t = 0.125$ in. $A = 2.16\ \text{in.}^2$
I.D. = 5.375 in. $L = 157.0$ in. $I = 8.17\ \text{in.}^4$
$c = 2.8125$ in.

Allowable Stresses (6061-T6 Aluminum Alloy)

$S'_b = S_y = 34.2$ ksi $S'_s = 0.5 S_y = 17.1$ ksi

Calculated Stresses

$$S_b = \frac{Mc}{I} = \frac{(61369.4)(2.8125)}{8.17} = 21{,}126 \text{ psi}$$

$$S_s = \frac{V}{A} = \frac{4{,}909.6}{2.16} = 2{,}273 \text{ psi}$$

Margin of Safety

$$M.S. = \frac{1}{\left[\left(\frac{21126}{34200}\right)^2 + (4)\left(\frac{2273}{17100}\right)^2\right]^{1/2}} - 1 = \underline{\underline{+0.48}}$$

Liner - 3 Element (Drwg. No. 347-291-F3)
NLI 1/2 Cask

-5 Tube - Bearing on Support Segment Load (1-Foot Side Drop)

$R_{ss} = R_2$ or $R_3 = 9819.1$ lbs (Ref. page XI-G8)

Bearing Area

$A_{br} = Dt = (1.90)(0.25) = 0.475$ in.^{2}

where,

$t = 0.25$ in. (support segment thickness)

$D = 1.90$ in. (Chord of 30° arc of segment beneath lower tube)

Allowable Stress (6061-T6 Alum. Alloy)

$S'_{br} = S_y = 34.2$ ksi

Calculated Stress

$$S_{br} = \frac{R_{ss}}{A_{br}} = \frac{9819.1}{0.475} = 20{,}672 \text{ psi}$$

Margin of Safety

$$M.S. = \frac{S'_{br}}{S_{br}} - 1 = \underline{\underline{+0.65}}$$

Liner - 3 Element (Drwg. No. 347-291-F3)
NLI 1/2 Cask

Lifting Holes

Load (Handling)

$$P_L = (2.0)(2935) = 5870 \text{ lbs.}$$

Section Properties (One Hole conservatively assumed)

$$d_s = 1.25 - (.75/2) \cos 15^\circ = 0.983 \text{ in.}$$

$$A_s = (2)(d_s)(0.125)(2) = 0.492 \text{ in.}^2$$

Allowable Stress (6061-T6 Alum. Alloy)

$$S_s' = 0.5\,S_y = 17.100 \text{ ksi}$$

Calculated Stress

$$S_s = \frac{P_L}{(A_s)_{total}} = 11942 \text{ psi}$$

Margin of Safety

$$M.S. = \frac{S_s'}{S_s} - 1 = \underline{\underline{+0.43}}$$

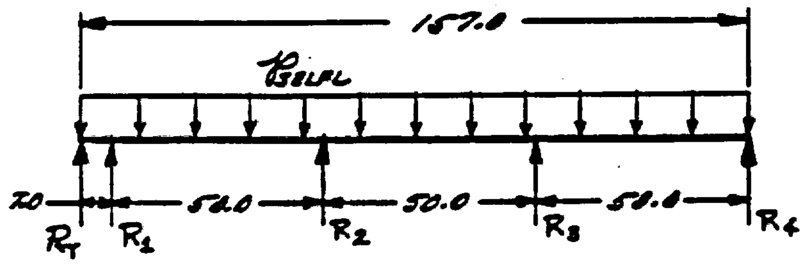
Liner - 3 Element (Drwg. No. 347-291-F3)
NLI 1/2 Cask

Support Segment Bearing on Cask Load (1-Foot Side Drop)

The weight of the liner and contents is assumed to be uniformly distributed on a beam which is supported by the five upper segments (similar to page XI-G8).

$P_{3ELFL} = (16.4)(2935) = 48{,}134$ lbs.

$p_{3ELFL} = P_{3ELFL}/L = 48{,}134/157.0 = 306.6$ lbs/in.



The reactions are determined based on the span which they support:

$R_T = (7.0/2)(306.6) = 1{,}073.1$ lbs.

$R_1 = (7.0/2 + 50.0/2)(306.6) = 8{,}737.7$ lbs.

$R_2 = R_3 = (50.0)(306.6) = 15{,}329.3$ lbs.

$R_4 = (50.0/2)(306.6) = 7{,}664.6$ lbs.

Bearing Area

$A_{br} = D_{IC}\,t = (8.5)(0.25) = 2.125$ in.^{2}

Where, $t = 0.25$ in. (Support Segment Thickness)
$D_{IC} = 8.5$ in. (Chord of 82° Segment Arc)

Liner - 3 Element (Drwg. No. 347-291-F3)
NLI 1/2 Cask

Support Segment Bearing on Cask (Cont'd.)

Allowable Stress

Support Segment - 6061-T6 Alum Alloy - $S'_{br} = S_y = 31.2$ ksi
Cask - 304 stainless steel - $S'_{br} = S_y = 21.0$ ksi

Calculated Stress

$$S_{br} = \frac{P_2}{A_{br}} = \frac{15,329.3}{2.125} = 7,214 \text{ psi}$$

Margin of Safety

$$M.S. = \frac{S'_{br}}{S_{br}} - 1 = \underline{\underline{\text{Large}}}$$

Liner -3 Element (Drwg. No. 347-291-F3)
NLI 1/2 Cask

Support Segment - Compression and Stability Load

$P_{ss} = R_2 = 15{,}329$ lbs. (Ref. page XI-G12)

Segment Properties

The -13 segment is considered as a 0.25 inch thick plate 8.5 inches long and 3.5 inches wide.

$A_c = (8.5)(0.25) = 2.125 \text{ in.}^2$ (Compressive area)

$a = 3.5$ in. $\quad a/b = 0.41 \quad t = 0.25$ in. $\quad E = 10.0 \times 10^6$ psi

$b = 8.5$ in. $\quad {}^*K = 6.92 \quad \nu = 0.33$

Allowable Stresses (6061-T6 Alum. Alloy)

$S_c' = S_y = 34.2$ ksi

$${}^*S'_{stab} = \frac{KE}{1-\nu^2}\left(\frac{t}{b}\right)^2 = 67.2 \text{ ksi}$$

Calculated Stresses

$$S_c = \frac{P_{ss}}{A_c} = 7{,}214 \text{ psi}$$

Margins of Safety

$$\text{M.S.} = \frac{S'_c}{S_c} - 1 = \underline{\underline{\text{Large}}}$$

$$\text{M.S.} = \frac{S'_{stab}}{S_c} - 1 = \underline{\underline{\text{Large}}}$$

* Ref. No. 2, page 318, Case 1.

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Liner - 3 Element (Drwg. No. 347-291-F3)
NLI 1/2 Cask

Bottom Plate Bearing on Spacer

Load (1-Foot End Drop)

$$P_{BP} = (39.4)(2935) = 115{,}639 \text{ lbs.}$$

Bearing Area

$$A_{br} = (\pi/4)(9.0^2 - 8.5^2) = 6.87 \text{ in.}^2$$

Allowable Stress (6061-T6 Alum. Alloy)

$$S'_{br} = S_y = 34.2 \text{ ksi}$$

Calculated Stress

$$S_{br} = \frac{P_{BP}}{A_{br}} = 16{,}832 \text{ psi}$$

Margin of Safety

$$M.S. = \frac{S'_{br}}{S_{br}} - 1 = \underline{\underline{+1.03}}$$

Liner - 3 Element (Drwg. No. 347-291-F3)
NLI 1/2 Cask

Spacer - Compression and Stability Load (1-Foot End Drop)

$P_s = (39.4)(2935) = 115{,}639$ lbs.

Spacer Properties

O.D. = 9.0 in.
I.D. = 85 in.
$r = 4.375$ in.

$A = (\pi/4)(9.0^2 - 8.5^2) = 6.87 \text{ in.}^2$
$t = 0.25$ in.
$E = 10.0 \times 10^6$ psi

Allowable Stresses (6061-T6 Alum. Alloy)

$S'_c = S_y = 34200$ psi

$^{*}S'_{stab} = \frac{0.3Et}{r} = 171{,}430$ psi

Calculated Stress

$$S_c = \frac{P_s}{A} = 16{,}832 \text{ psi}$$

Margins of Safety

$$M.S. = \frac{S'_c}{S_c} - 1 = \underline{\underline{+1.03}}$$

$$M.S. = \frac{S'_{stab.}}{S_c} - 1 = \underline{\underline{\text{Large}}}$$

* Ref. No. 2, page 352, Case M.

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