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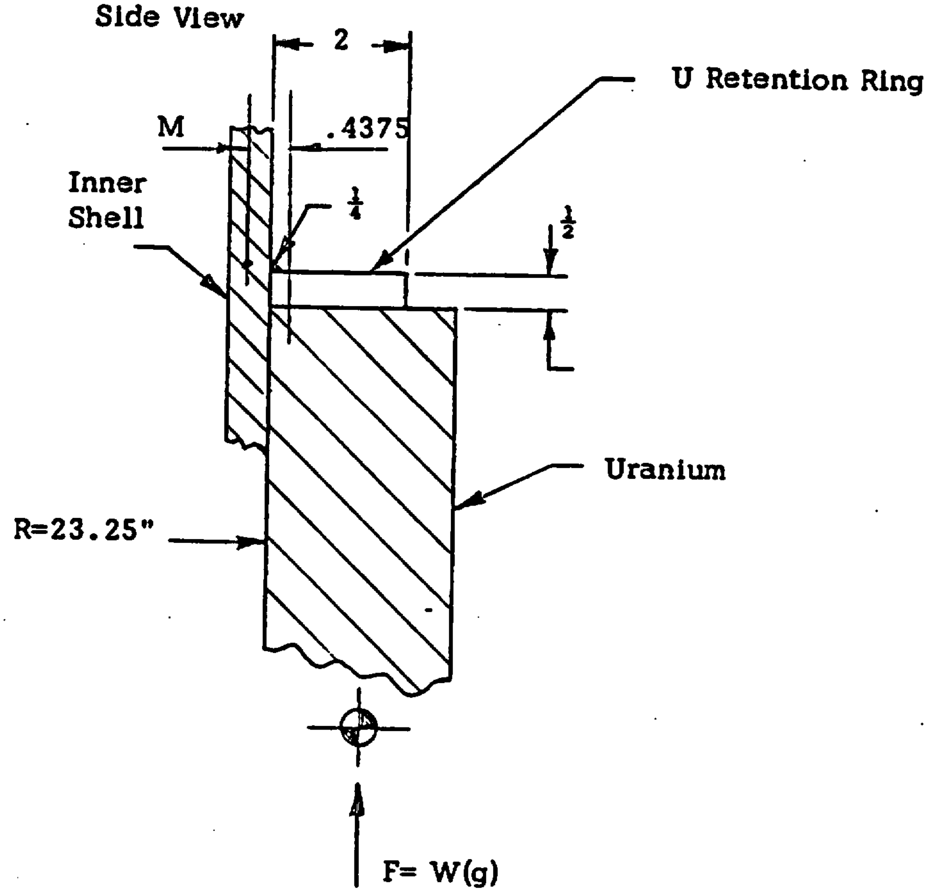
4.4.5 Lower Uranium Support Ring

Under top end impact the lower uranium shell is held in place by a .5 inch thick retaining ring which is welded around the containment vessel shell.

Side View



W = 5,170.55 lbs.

F = 30 (5170.55) = 155,116 lbs.

Bearing of uranium on retention ring:

$$A_{Br} = \pi (25.25^2 - 23.25^2) = 304.74 \text{ in.}^2$$

$$\sigma_{Br} = P/A = \frac{155,116}{304.74} = 509.0 \text{ psi}$$

Bearing strength allowable for uranium at

382° F = 0.9 x 58500 x 1.5 = 78975 psi (Sect. 1.1, 1.2)

$$M.S. = \frac{78975}{509} - 1 = 154$$

Ring deflection:

During impact, the ring will deflect slightly to allow the rigid uranium ring to place the full load on the weld (¼) at the inner shell region.

Area of weld = $2\pi R (T)$ = 2 (23.25) (π) (.25) (.707)

Area of weld = 25.82 in.^{2}

$$\sigma_s = \frac{P}{A} = \frac{155,116}{25.82} = 6,007 \text{ psi}$$

Allowable shear stress ($.6\ S_{aa} = .54\ S_u$) for 304 S/S at 382°F from Sect. 1.1 under noncontainment structure and Sect. 1.2 equals to .60(54450) = 32670 psi

$$M.S. = \frac{32670}{6,007} - 1 = 4.44$$

Stresses in the inner shell applied by the lower uranium support ring under top end impact is calculated in Sect. 3.8.

4.4.6 PWR Spacer Plug

Basic Configuration:

The Spacer Assembly consists essentially of ten (10) square blocks maintained in spaced array by a circular ring and a diametral plate. Each block extends downward into sleeve to pick up the corners or "Hard Points" of the Fuel Elements. Fuel Element loads will be transmitted to the square aluminum blocks and thence to the circular ring or plate in compression.

The spacer has the ability to sustain Impact Fuel loads if the basket contained less than ten (10) assemblies.

The Spacer System will be made from 6061-T6 aluminum with material properties taken at an operating temperature under normal conditions of transport without auxiliary cooling.

Weights:		WT	Force at 30g	
Basket	=	7054	211,620	442,650
Sleeve	=	7701	231,030	
Fuel	=	17000	510,000	(51,000 each element)

Total Impact Force = 30 (7054 + 7701 + 17,000) = 952,650 pounds)

Allowable Stresses for 6061-T6

Temperature of the Spacer is determined by an average of the Inner plate of inner closure and the top end of the basket. (Ref. Sect. VIII - Appendix D) $\tau = \frac{355 + 425}{2} = 390\ ^{o}F$

From Ref: 27, Figure 3.6.1.2.1 (c)

Table 3.6.1.0 (f)

Percent of Ult at temp. considered = 72%

F_{TU} = 42,000 psi

Allowable tensile stress ($S_{aa} = 0.9S_u$) for Al.6061 T6 at 390 ^{o}F from Sect. 1.1 under cask internal structure and Sect. 1.2 equals .72 (42,000) (.9) =

S_{TA} = 27,216 psi

Allowable Shear stress ($.6S_{aa} = 0.54S_u$) for Al. 6061 T6 at 390 ^{o}F from Sect. 1.1 under cask internal structure and Sect. 1.2 equals

S_{sa} = .72 (.54) (42,000) = 16,330 psi

Allowable Bearing Stress = S_{br} = .72 (.90) (67,000) = 43,416 psi

Allowable Weld Stress In a welding operation, dealing with either a strain hardened or heat tempered aluminum alloy, it is impossible to reduce T6 temper to a value less than 0 condition temper. Therefore, the computation of weld allowables may use 0 condition stress allowables as a conservative minimum in the applicalbe equations.

For the 0 Condition

F_{TU} = 18,000 psi (Ref. 14)

Allowable Shear in Weld = 90% (.72) (.6) (18,000) = 6,998 psi

Allowable Tension in Weld = 90% (.72) (18,000) = 11,664 psi

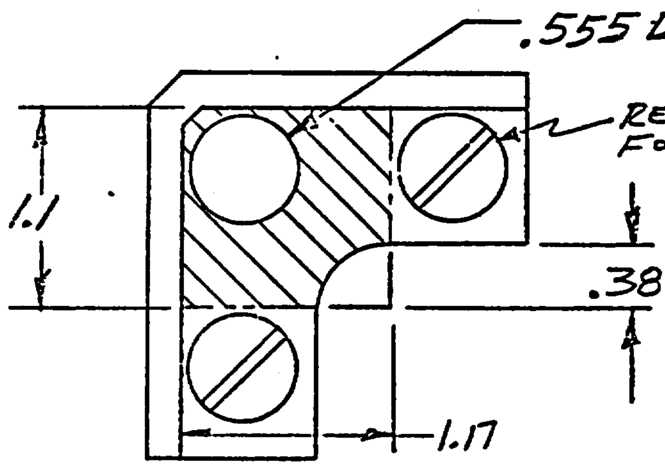
PWR Fuel Element

The PWR Fuel Element maintains structural integrity in the top impact condition due to load transmission in the pure compressive mode. The basic fuel bundle is tied together by seven spring clip grids that transmit compression to the guide thimbles which in turn resolve the load vectors into the adaptor plate or the top nozzle assembly. The adaptor plate is a structural component which allows the upper spacer to pick up axial loads on the spacer plug compression legs located at two of the four local hard points. If structural failure were to take place in the Zircaloy-4 Guide Thimbles, the relative movement of the fuel bundle to the absorber sleeve would be limited to less than one inch by the adaptor plate assembly. In all probability, the most severe failure mode that would occur in the top end impact would be a local crippling phenomenon in the guide tubes which would not pose any relative motion problems.

Adjacent hard points on the Fuel have two holes in each bearing surface, one being .875 in diameter and the other having a diameter of .555.

Top View of Fuel Pick-Up Point

Full Scale



TOP VIEW OF FUEL PICKUP POINT- FULL SCALE

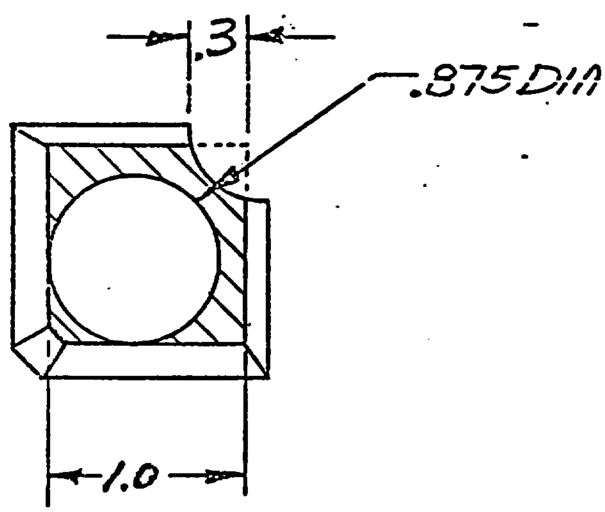
SHADED AREA INDICATES BEARING AREA UNDER COMPRESSION LEG

$$\text{Area} = (1.1)(1.17) - \pi\left(\frac{.555^2}{2}\right) - \frac{\pi}{4}(.38)^2$$

$$\text{Area} = 1.287 - .242 - .113 = .932 \text{ in}^2$$

$$S_{br} = \frac{51{,}000}{4 \times .932} = 13{,}680 \text{ psi}$$

$$\text{M.S} = \frac{43{,}416}{13{,}680} - 1 = 2.17$$

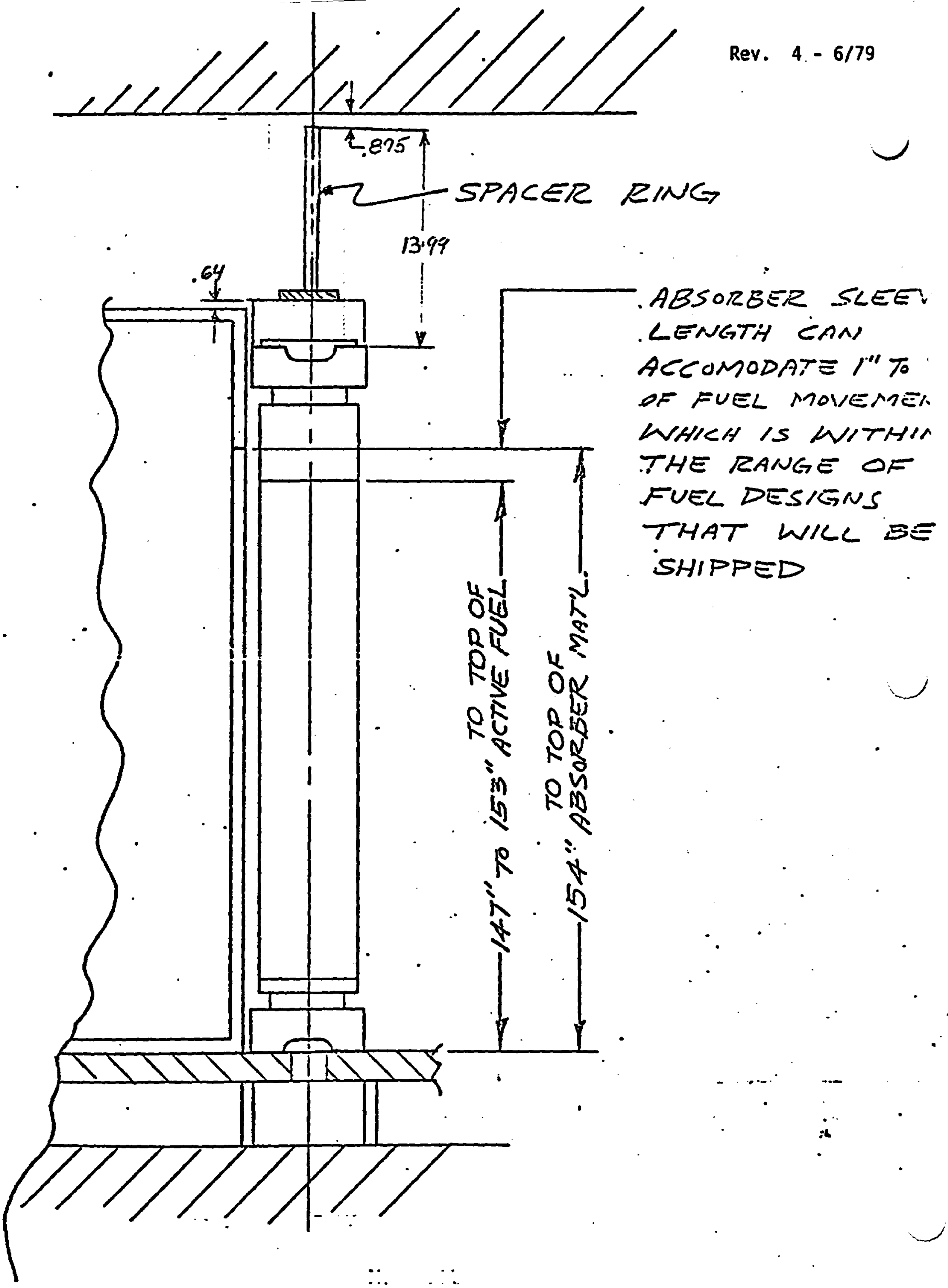


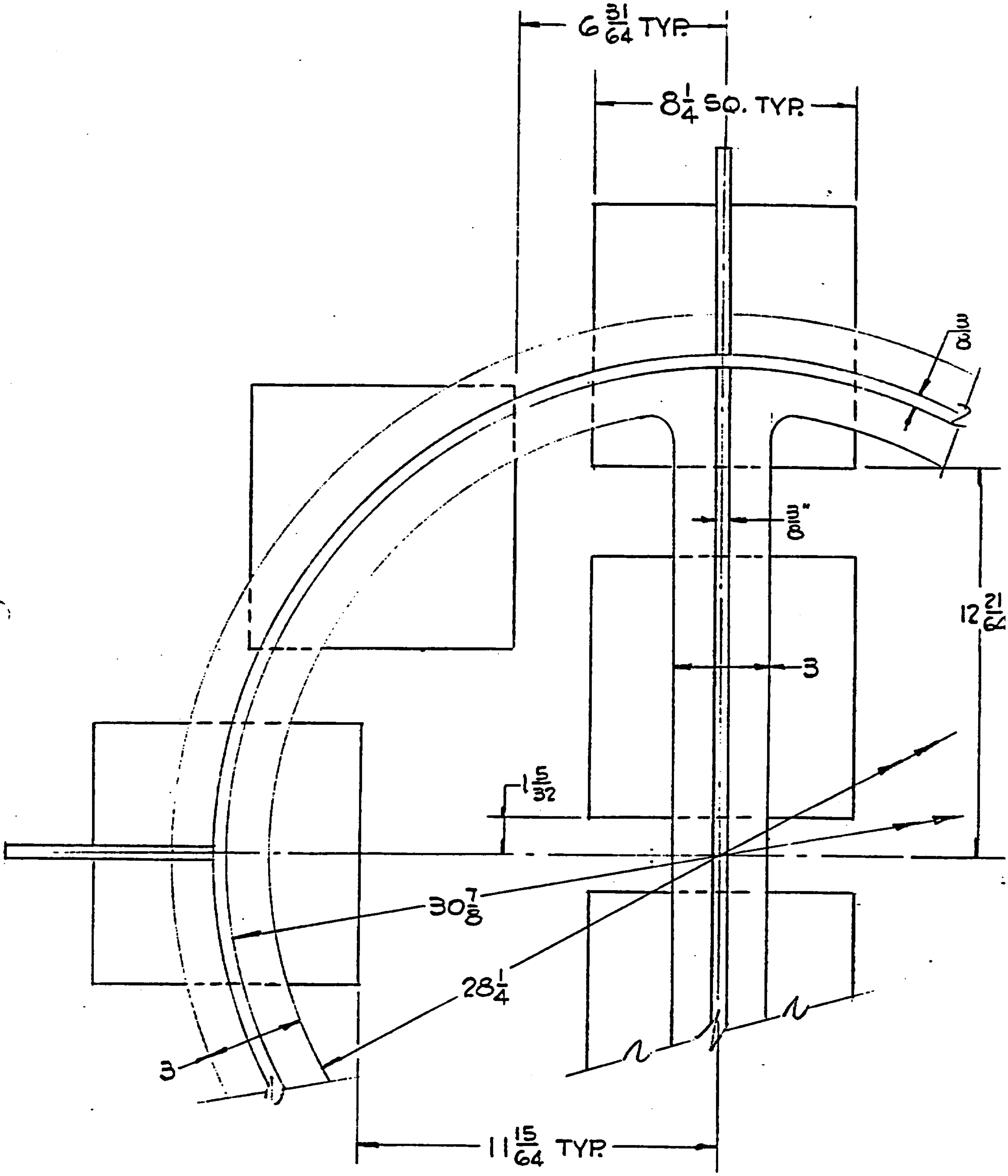
$$\text{Area} = (1.0)^2 - \pi\left(\frac{.875^2}{2}\right) - \frac{\pi}{4}(.3)^2$$

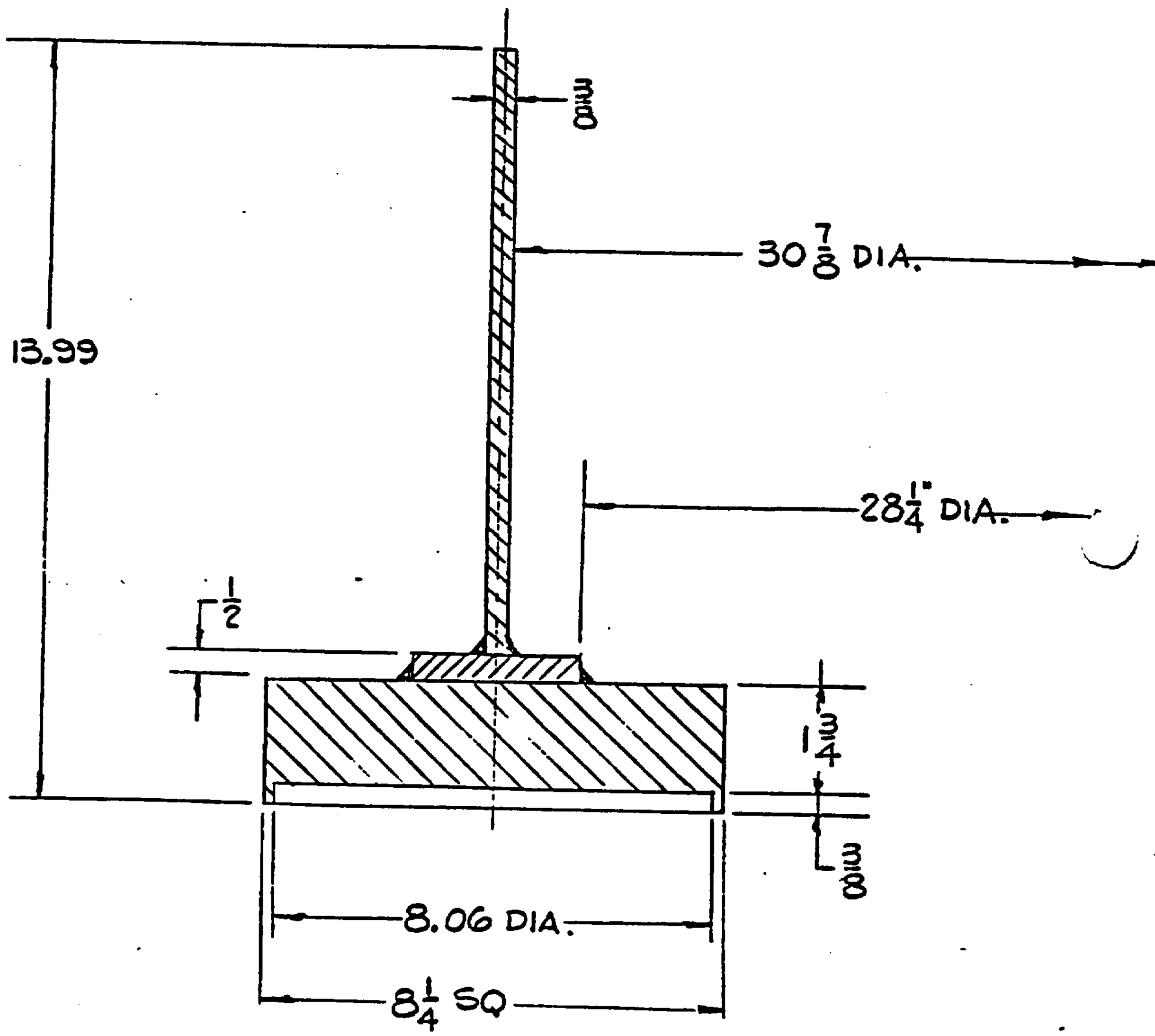
$$\text{Area} = 1.0 - .601 - .0706 = .328 \text{ in}^2$$

$$S_{br} = \frac{51{,}000}{4 \times .328} = 38{,}871 \text{ psi}$$

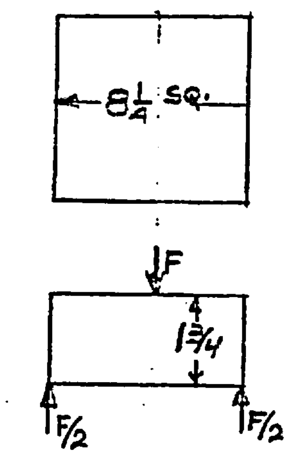
$$\text{M.S.} = \frac{43{,}416}{38871} - 1 = .117$$


.875
SPACER RING
13.99
.64
ABSORBER SLEEV
LENGTH CAN
ACCOMODATE 1" TO
OF FUEL MOVEMEN
WHICH IS WITHIN
THE RANGE OF
FUEL DESIGNS
THAT WILL BE
SHIPPED
147" TO 153" TO TOP OF ACTIVE FUEL
154" TO TOP OF ABSORBER MAT'L


6 31/64 TYP.
8 1/4 SQ. TYP.
3/8
3/8"
12 21/64
3
1 5/32
30 7/8
28 1/4
3
11 15/64 TYP.


3/8
30 7/8 DIA.
13.99
28 1/4" DIA.
1/2
1 3/4
3/8
8.06 DIA.
8 1/4 SQ

Blocks as simple beams:



$F/2 = \frac{1700 \times 30}{2} = 25{,}500$ lbs

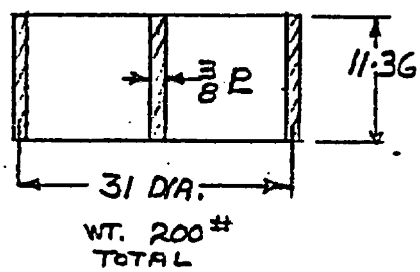
$M = 25{,}500 \times 4\text{-}1/8 = 105{,}188$ in lbs.

$Z = \frac{8\frac{1}{4} \times 1.75^2}{6} = 4.21 \text{ in}^3$

$S_b = \frac{105.188}{4.21} = 24{,}980$ psi

$M.S. = \frac{27{,}216}{24{,}980} - 1 = .089$ (Conservative)

Plate and Ring in Compression



Total load against closure head = 952,650 + 30 (200) = 958,650 lbs.

area plate and ring = $(3/8\pi 31) + (3/8 \times 30.25) = 47.86 \text{ in}^2$

$S_c = \frac{958{,}650}{47.86} = 20{,}030$ psi

$M.S. = \frac{27{,}216}{20{,}030} - 1 = .358$

Stability of cylinder

Roark - Table XVI -Case M-ends not constrained (conservative)

$S^1 = .3\ E\ t/r = .3(.9 \times 10{,}100{,}000)\ \frac{.375}{15.5} = 65{,}976$ psi (critical)

Actual $S_c = 20{,}030$ psi OK

All welds are in compression, if considered loaded at all, since stack-up of members allows direct contact for transmission of loads.

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80	4
81	3
81a	1
81b	1

4.4.6.1 Spacer Plug - BWR

The BWR spacer consists of an assembly made up of three plates approximately 9.75 in deep. The main plate which runs along the diameter and center line of the basket is 1.5 inches thick. Two plates, 3/4 in. thick, intersect the main plate at 90° and are joined to the main plate by a slot and joint arrangement. The main plates are held in position by a 1/2" thick circular top plate that will assume a position directly under the inner closure head upon complete assembly. The top circular plate also acts as a platform to which is attached the fuel retention rods . The fuel elements are restrained from axial movement toward the top by the fuel retention rods. These rods carry a pure compressive load and engage the upper grid assembly of the fuel.

The upper spacer assembly is a single unit structure that is joined by welding and contains openings in non-structural regions to facilitate placement of the spacer on the basket and fuel bundles. The welds are employed to position and retain load-carrying structural members but do not constitute a primary load path in themselves. The plate material is 6061-T6 Refer to the material curves presented in the PWR spacer analysis for impact allowables.

The fuel retention rods are 6061-T6. Material values have been presented here by use of Ref. 27, pg. 3-176, Sect. 3.6. 6061 is a very readily weldable AL-MG-SI alloy available in a wide range of product forms. It has high resistance to corrosion and has high toughness properties. The maximum temperatures: Normal conditions of transport without auxiliary cooling

For BWR

Temperature of spacer is determined by an average of the inner plate of inner closure and top end of the basket (Ref. Sect. VIII, Appendix D)

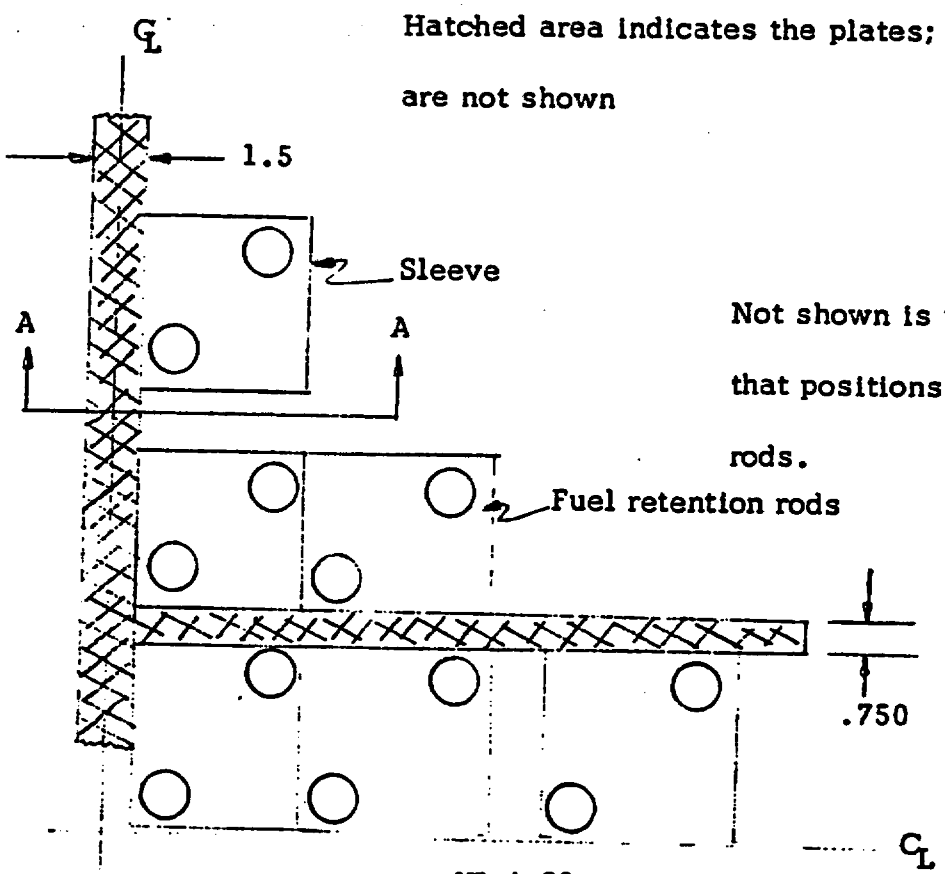
$$t = (355 + 425)/2 = 390^{o}F$$

The purpose of this analysis is to determine the structural integrity of the spacer used in the region between and the cover plate.

Top view of basket

Symetrical about both centeral lines

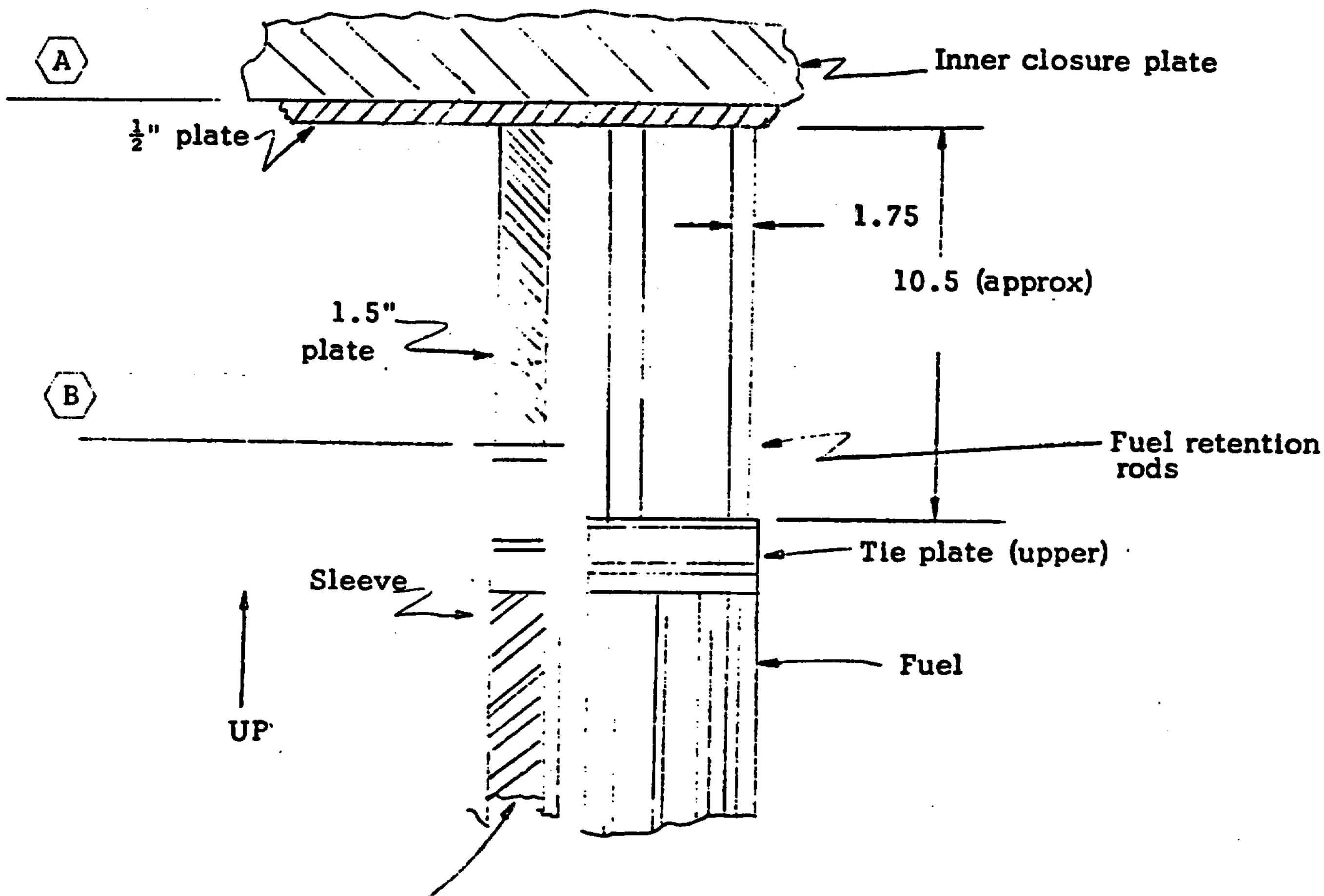
Hatched area indicates the plates; fuel elements are not shown



Not shown is the 1/2 inch plate that positions the fuel retention rods.

The 1.75 diameter aluminum rods are used to transmit the fuel loads (on top end impact) directly from the fuel to the inner closure plate. The BWR fuel assembly handle is not involved in the structural retention of the fuel bundle. The plates carry the aluminum basket and sleeve loads to the inner closure plate independently of the retention characteristics of the fuel bundles.

View A-A (Fuel assembly Handle is omitted here)



(not to scale)

Elevation from bottom head of containment vessel.

A = 179.50

B = 169.750

Height of spacer plate =179.50 - 169.750 = 9.75 in.

This height is used for the analytical model. The actual height may be somewhat less than 9.75 in.

W = 30 (750)* = 22,500 lbs.

* Design weight of BWR fuel assembly is 750 lbs.

Weight of aluminum basket = 9557.23 lbs.
Weight of sleeves = 5851.90
Design weight of 24 fuel elements = 18,000 lbs.

Total weight = 9557.23 + 5851.90 = 15,409

Shock loads are based on an impact force of 30 g.

Force = 30 (15,409) = 462,270 lbs.

Area of spacer carrying compression:

3/4 in. plate is 44.250 long - 2 are used

1.5 plate is 44.750 long - 1 is used.

$Area_1$ = (2)(.75) (44.250) = 66.375

$Area_2$ = 1.5 (44.750) = 67.125

Total = 66.375 + 67.125 = 133.50 $in.^2$

$$\sigma_c = P/A = \frac{(462,270)}{133.50} = 3,462.7 \text{ psi.}$$

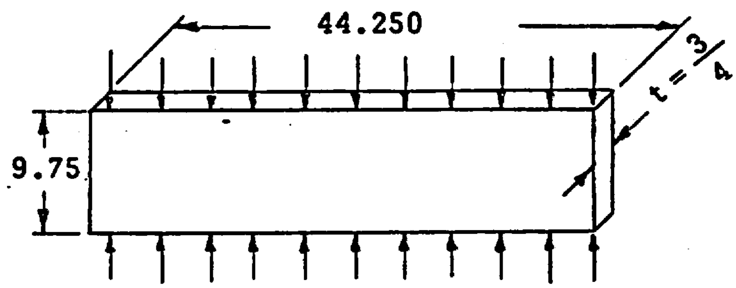
Allowable stress ($S_{aa} = 0.9\ S_u$) for Al. 6061-T6 at 390°F from Sect. 1.1 under cask internals structure and Sect. 1.2 equals to .68(42000)(.9) = 25704 psi.

Where Ref. 27, Table 3.6.1.2.1(a) indicates the use of .68 temp. effect.

$$M.S. = \frac{25704}{3462.7} - 1 = 6.42$$

Examination of stability is performed in the same manner as the PWR spacer. As a conservative method of analysis, the plate thickness is considered to be 3/4 of an inch which is a minimum value for all the plates used in the BWR spacer.

Stability is cited in Ref. 3, Table XVI, Case A



Impact Loads

$$a/b = 9.75 / 44.25 = .220$$

K = 16

Critical buckling stress = $K \left(\frac{E}{1 - v^2} \right) \left(\frac{t}{b} \right)^2$

Reference (27) , page 3-179, Figure 3.6.1.2.4

Modulus of elasticity @ 390°F = $10.1 \times 10^6 \times .90 = 9.09 \times 10^6$ psi

$S = (K) \frac{E}{1 - v^2} \left(\frac{t}{b} \right)^2$ Critical compressive stress:

$$S = \frac{16 \times 9.09 \times 10^6}{.89} (.000287) = 46900 \text{ psi}$$

Since $S > \sigma_c$ stability is not a problem.

Stresses and structural stability of fuel retention rods demonstrates there is no significant movement of the fuel relative to the absorber sleeve and basket. The fuel retention rod is a 1.75 inch diameter rod of 6061-T6 aluminum. It is secured to the top plate by small fillet welds which will not tend to materially effect the overall temper characteristics of the aluminum

Impact allowables of 6061-T6

Refer to Ref. 27, table 3.6.1.0 (f) and figure 3.6.1.2.1 (a).

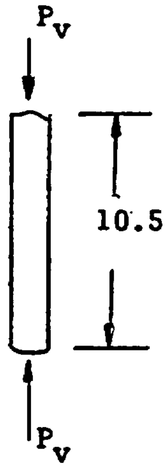
Bearing allow = 67,000 (.9) (.68 = 41004 psi

Compression allow = 42,000 (.9) (.68)* = 25704 psi

E_c (allow) = 10,100,000 (.90)* = 9,090,000 psi

* Temperature correction factor from Ref. 27.

Conservatively assume the fuel retention rod to be a round ended column with no end constraint or fixity.



$$P_v = \frac{30(750)}{2} = 11{,}250 \text{ pounds}$$

Properties:

$$\text{Area} = \pi(R)^2 = \pi\left(\frac{1.75}{2}\right)^2 = 2.406 \text{ in.}^2$$

$$I = \frac{\pi}{4}(R)^4 = .4603$$

$$\text{Radius of Gyration} = r = \sqrt{I/A} + \left[\frac{.4603}{2.406}\right]^{\frac{1}{2}} = .4374$$

Column behavior of the fuel retention rods:

Looking at the L/ r ratio;

$$L/r = \frac{10.5}{.4374} = 24.0$$

which is a very short column. If the length of a column is reduced below a certain critical value ($L/p < 120$), failure in lateral bending will occur at loads below those predicted by the Euler Formula. This is due in a great part to a reduction in the effective value of E caused primarily by changes in the slope of the stress-strain diagram. It can be noted that a good many short-column formula are given with the most satisfactory for compact shapes being the parabola-straight-line formula for a line tangent to Euler curve.

For a pin ended column having zero end restraint

$$c = 1.0$$

in

$$L' = L\sqrt{c}$$

Stresses in fuel retention rod;

r = Radius of Gyration = .4374

$L' = 10.5"$

$E = 9{,}090{,}000$

F_c = Allowable stability stress under impact.

F_{co} = allowable stability stress under impact

$$F_c = F_{co}\left[1 - 0.385\,\frac{(L'/r)}{\pi\sqrt{E/F_{co}}}\right]$$

Ref. 27, Page 1-6 equation 1.3.8.5

Substitution yields;

$$F_c = 25704\left[1 - .385\,(.363)\right] = 22112 \text{ psi}$$

As suspected, this very short column exhibits little instability.

Compressive stress

$$\sigma_c = \frac{P}{A} = \frac{11{,}250}{2.406} = 4{,}675.8 \text{ psi}$$

In column stability

$$M.S. = \frac{22112.}{4,675.8} - 1 = 3.72$$

In compression

$$M.S. = \frac{25704}{4,675.8} - 1 = 4.49$$

In bearing*

$$M.S. = \frac{41004}{4,675.8} - 1 = 7.77$$

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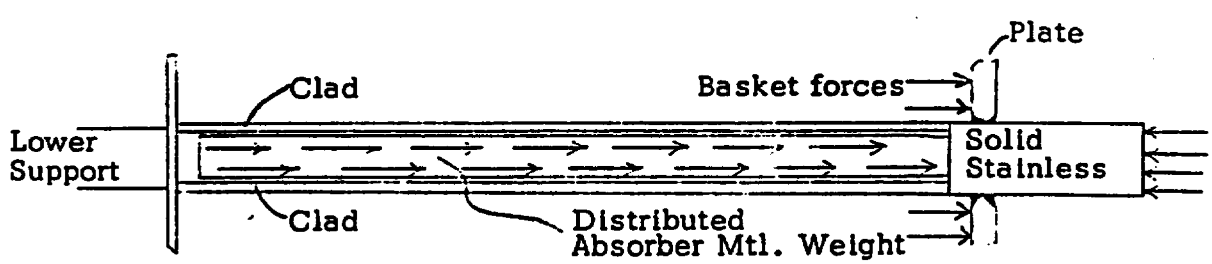
4.4.7 Analysis of Absorber Shield Structure for Impact

The absorber sleeves perform a structural contribution to the integrity of the inner basket region as well as serving to eliminate the hazards of radiation. The absorber sleeves carry the weight of the lower support structure and the aluminum basket to a reaction point at the upper spacer structure. The absorber sleeves (both PWR and BWR) are clad with stainless steel and have a heavy core material that is essentially free standing since it is not structurally attached to the stainless steel clad. The absorber sleeve then becomes a rather unique structure in its ability to transmit top end impact loads. First, the stainless steel cladding must carry the forces from the lower support structure to the solid stainless steel region of the sleeve. Secondly, the neutron absorber plate, which acts independently of the lower support structure and basket forces, must not buckle under its own distributed weight and transmits its own impact force to the solid stainless steel structure that is integral to the upper end of the absorber sleeve. Thirdly, the solid stainless steel portion must also carry the full impact of the aluminum basket.

Forces on a sleeve: (Not to scale)



Weights: (PWR)

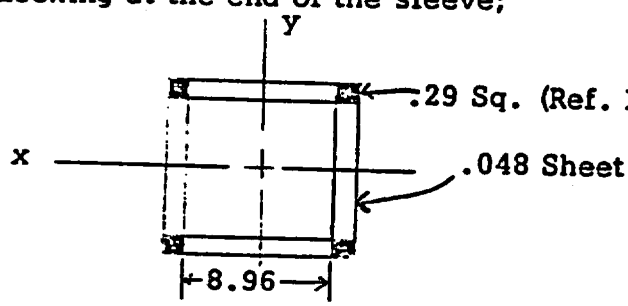
Aluminum Fuel Basket = 7059 lbs.
One Absorber Sleeve = 770.1 lbs.
Bottom Support Assy. = 182 lbs.

} Impact = 30g

Forces on Sleeve = 182(30) = 5460 lbs.

Load per Sleeve = $\frac{5460}{10}$ = 546.0 lbs.

Looking at the end of the sleeve;



$$I_x = I_y = \frac{4(.048)(8.96)^3}{12} + 4(.048)(8.96)(4.48)^2 + 4(.29)^2(4.625)^2$$

$$I_x = I_y = 11.51 + 34.52 + 7.19 = 53.22 \text{ in}^4$$

$$\text{Area} = (.048)(8.96)(4)(2) + 4(.29)^2 = 3.44 + .34 = 3.78 \text{ in.}^2$$

$$\text{Pure Compressive Stress} = P/A = \frac{546.0}{3.78} = 144.4 \text{ psi}$$

The margin is high.

Investigate the column buckling characteristics:

Length of unsupported clad = 151.00 inches

$\rho = \sqrt{I/A}$ (Radii of gyration)

$\rho = 3.75$

$L/\rho = \frac{151.00}{3.75} = 40.3$ (This indicates a short-column)

It is necessary to consider the buckling characteristics of short columns.

Ref. 27, Eq. 1.3.8.5

$$F_c = F_{co} = \left[1.0 - .385 \frac{(L'/\rho)}{\pi\sqrt{E/F_{co}}} \right]$$

Where:

F_{co} = .9(59000)=53,100 psi. For 304 S/S under cask internal structure (at 499°F) Sect. 1.1,1.2

L' = Effective length = $L/\sqrt{c}$

c = 2, where both ends are fixed against lateral movement

E = 26,100,000 psi (at 499°F)

ρ = Radii of Gyration

$$L' = \frac{151.00}{\sqrt{2}} = 106.77"$$

$$F_c = 53,100\left[1.0 - .385\left(\frac{28.5}{69.65}\right)\right]$$

F_c = 53,100(.842) = 44,734 psi

$$M.S. = \frac{44,734}{144.4} - 1 = 309$$

Consider the individual buckling of the .048 sheets in the transmitting of the lower support structure load.

Ref. 57, Case 158, Page 701.

Consider the welds attaching the .048 sheet to the stainless steel solid sections provide a degree of fixity to utilize case "A"

$$\text{Per cent of load on plates} = \frac{3.44 - .34}{3.44} = 90\%$$

$$\text{Load per plate section} = \frac{.90(546.0)}{8} = 61.4 \text{ pounds}$$

$$\text{Critical Load} = \overline{n} = \lambda_{CR} \frac{\pi^2 D}{b^2}$$

Where $b = 8.96$

$\lambda = 7.0$

$$D = \frac{E\,t^3}{12(1-v^2)} = \frac{26,100,000(.048)^3}{10.92} = 264.32$$

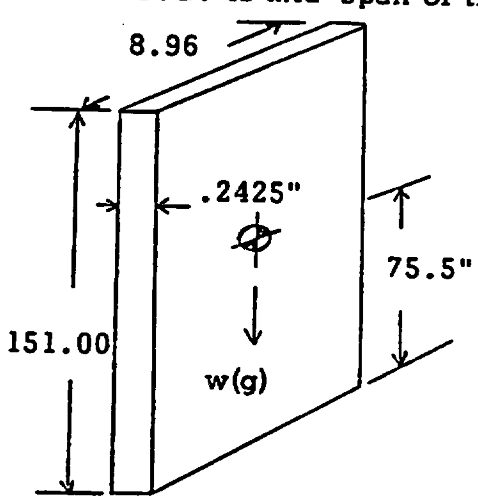
$$\overline{n} = \frac{7.0\,(\pi^2)(269.32)}{(8.96)^2} = 231.7 \text{ lbs.}$$

$$M.S. = \frac{231.7}{61.4} - 1 = 2.77$$

The absorber Material: This plate of absorber material is secured by two sheets of stainless steel clad material which would naturally tend to inhibit any classical wave form buckling. The primary load on the free standing sheet of neutron absorber material is its own weight under 30g impact loading. Since the absorber material accounts for the majority of the weight of the sleeve, this analysis conservatively assumes the total absorber weight at 770 pounds.

$$\text{Wt. of each sleeve plate} = \frac{770}{4} = 192.5 \text{ pounds}$$

Assume C.G. is mid-span of the sleeve plate



Impact load = 192.5(30) = 5775

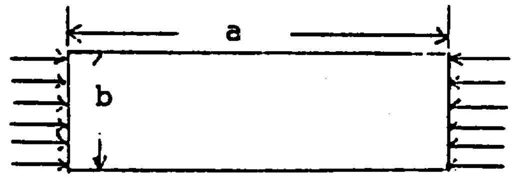
$E = 10.2 \times 10^6$ @ 499°F for absorber mat. (Sect.1
Allow. tension/Compr. = 38,610 (Ref. 19)

Area at Base = 8.96(.2425) = 2.173 in.^{2}

σ_c = P/A = 5775/2,173 = 2,657 psi

M.S. = (38,610/2,657) - 1 = 13.5

Assume the effective column length is 75.5 inches with the full load of 5775 applied at that plane. (Ref. 3, Table XVI, page 312, Case 1-A)



Lower Support Structure Forces

$$S' = K \frac{E}{1-v^2}\left(\frac{t}{b}\right)^2$$

As a/b approaches infinity

$K = 3.29$

$$S' = \frac{3.29(10{,}200{,}000)}{.91}\left(\frac{.2425}{8.96}\right)^2$$

$S' = 27{,}012$ psi (The buckling stress is well above the applied stress)

$$M.S. = \frac{27{,}012}{2{,}657} - 1 = 9.16$$

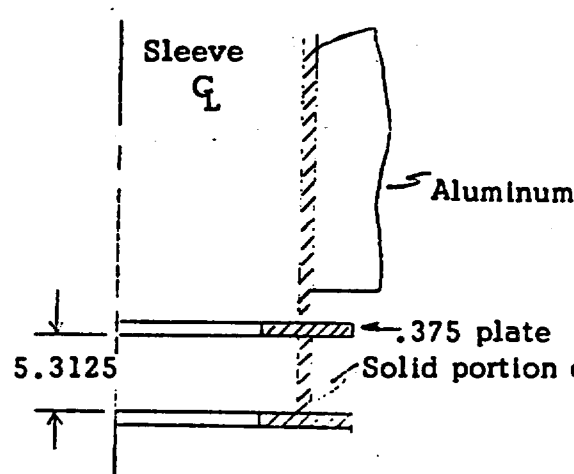
The absorber plate is able to withstand its own weight during impact.

Determine stress levels and structural integrity of the solid stainless steel portion of the absorber sleeve.

To determine length of solid section refer to page XI-4-155.

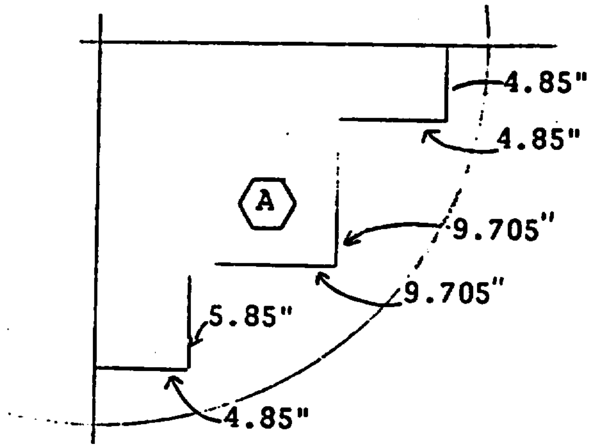
$L = 162 - (155.9375 + .75) = 5.3125"$

Weld attaching lower plate to sleeve:



Side view of sleeve & aluminum

Welded regions of .375 plate to basket.



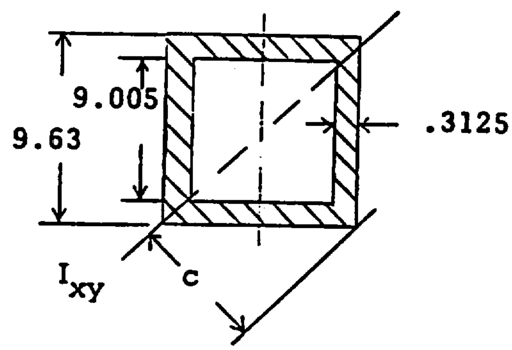
Impact Force $= F_B = 30(7059) = 211770$ pounds'

Weld Shear Stress $= \frac{F_B}{A} = \frac{211770}{42.22} = 5016$ psi

Allowable shear stress ($.6\, S_{aa} = 0.54\, S_u$) for 304 S/S at 499°F from Sect. 1.1 under cask internal structure and Sect. 1.2 equals to $.6(59000)(.9) = 31860$ psi (Full pent. weld)

$$M.S. = \frac{31,860}{5016} - 1 = 5.35$$

Section Solid Sleeve Portion:



Ref. 16, Page 5-37.

$$I_{xy} = \frac{(9.63)^4 - (9.005)^4}{12} = \frac{8600.1 - 6575.6}{12} = 168.7 \text{ in.}^4$$

$$\text{Area} = (9.63)^2 - (9.005)^2 = 92.74 - 81.09 = 11.65 \text{ in.}^2$$

$$\rho = \sqrt{I/A} = \left(\frac{168.7}{11.65}\right)^{\frac{1}{2}} = 3.805$$

$L/\rho = \frac{5.312}{3.805} = 1.4$ (The L/ρ is clearly too low to exhibit column stability characteristics)

$$c = \frac{9.63}{2(.707)} = 6.81"$$

Assume the loads are induced into the two corners of sleeve at the weld points by the aluminum basket. The sleeve must than carry pure compressive loads induced by the absorber material, bottom support, aluminum basket and local bending due to the offset loading by the aluminum basket. The load on sleeve ⟨A⟩ is a maximum as it contains the greatest amount of weld attachment.

Recall that the weld shear stress = 5016 psi.

Load per sleeve side = 5016(9.705)(.375)(.707) = 12,906 pounds

Moment about I_{xy} axis

$$M_{xy} = 12{,}906 \; \frac{6.81}{2} + 12{,}906 \; \frac{6.81}{2} = 87{,}889 \text{ in.-lbs.}$$

Max fiber stress

$$\sigma_c = \frac{\text{Lower Support}}{\text{Area}} + \frac{\text{Sleeve}}{\text{Area}} + \frac{\text{Basket}}{\text{Area}} + \frac{M_{xy}(C)}{I_{xy}}$$

$$\sigma_c = \frac{546.0}{11.65} + \frac{5775}{11.65} + \frac{211770}{10(11.65)} + \frac{87{,}889(6.81)}{168.7}$$

$$\sigma_c = 46.8 + 495.7 + 1817.7 + 3547.8 = 5908 \text{ psi}$$

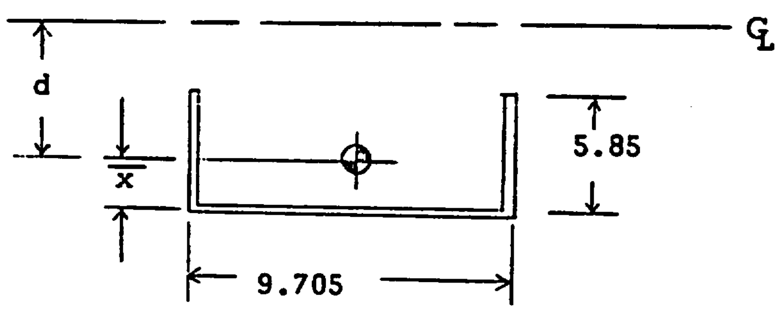
Allowable stress ($S_{aa} = 0.9\ S_u$) for 304 S/S at 499°F from Sect. 1.1 under cask internal structure and Sect. 1.2 equals to (.9)(59000)=53100 psi

$$M.S. = \frac{53{,}100}{5908} - 1 = 7.98$$

Double Sleeves

The contributing axial load on the two PWR axial sleeves (double sleeves) is in the same proportion to resisting areas as the single sleeve with the single exception of the load applied by the aluminum basket to the weld attachment. Hence, the values computed for .048 plate buckling, and absorber material behavior is the same for double sleeves as that computed for the single sleeve.

Double Sleeve Weld Attachments:



Length of weld = 2(5.85) + 9.705 = 21.405 in.

A_w = .375(.707)(21.405) = 5.675 in.^{2}

Load = 5.675(5016) = 28,465 pounds

$$\bar{x} = \frac{9.705(.375)(.375/2) + 2(5.85)(.375)(5.85/2)}{9.705(.375) + 2(5.85)(.375)}$$

$$\bar{x} = \frac{.6823 + 12.8}{30.2} = .446"$$

$$d. = \frac{9.63}{2} - .446 = 4.369"$$

I = 168.7 in.^{4} (Single sleeve resists moment)

Moment on sleeve = 28,465(4.369) = 124,363 in.-lbs.

$$\text{Bending stress} = \frac{124,363(4.815)}{168.7} = 3,549.5 \text{ psi}$$

Total compressive stress = 46.8 + 495.7 + 1817.7 + 3549.5 = 5909.7 psi

$$S_{aa} = 53,100 \text{ psi}$$

$$M.S. = \frac{53,100}{5909.7} - 1 = 7.98$$

BWR Absorber Sleeve Analysis

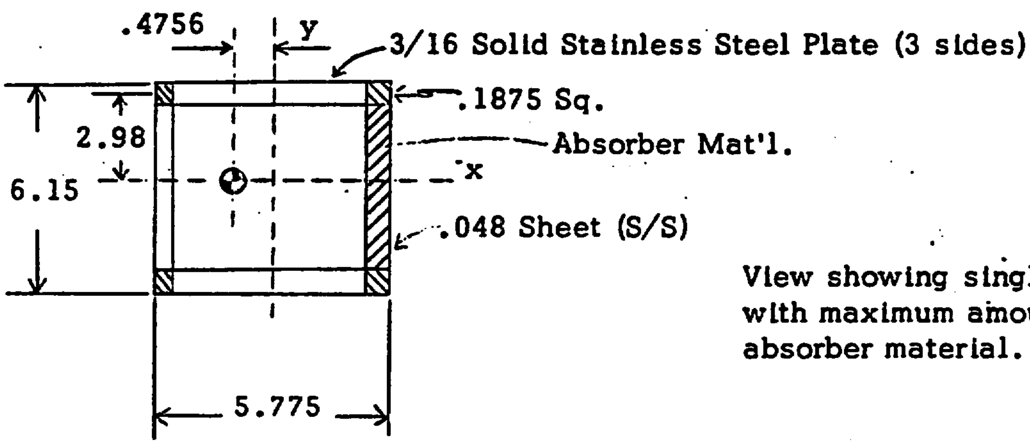
Weights: (BWR) See Weights section of SAR

Aluminum basket = 9910 lbs.
One absorber sleeve = 244 pounds
Bottom support assy. = 283 lbs.

Forces on sleeves = 283(30) = 8490 lbs.

$$\text{Load per sleeve} = \frac{8490}{24} = 353.75$$

Looking at the end of sleeve in the absorber material region,



View showing single sleeve with maximum amount of absorber material.

Determine section properties; general use is made of $(I = I + Ad^2)$

$$\text{Area} = 3(6.15)(.1875) + (2)(5.775)(.048) = 3.46 + .554 = 4.04 \text{ in.}^2$$

$$I_x = \frac{(.1875)(6.15)^3}{12} + \frac{(5.77)^3(.048)(2)}{12} + 2(6.15)(.1875)(2.98)^2$$

$$I_x = 3.63 + 1.86 + 20.48 = 25.97 \text{ in.}^4$$

Determine $\overline{x}$

$$\overline{x} = \frac{-.554(2.98) + 2.98(1.153)}{3.755} = .4756$$

$$I_y = \frac{2(.1875)(6.15)^3}{12} + 2.30(.4756)^2 + .554(2.98 + .4756)^2 + 1.15(2.98 - .4756)^2$$

$$I_y = 12.26 + .520 + 6.615 + 7.21 = 26.60 \text{ in.}^4$$

Pure Compressive Stress $= P/A = \frac{353.75}{4.04} = 87.56$ psi

Margin is very high

Investigate the column buckling characteristics:

Length of unsupported clad = 146"

$$\rho = \sqrt{I/A} = 2.536'$$

$$L/\rho = \frac{146}{2.536} = 57.66$$ (This indicates a short column)

It is now necessary to consider the buckling characteristics of short columns;

Ref: 27, Eq. 1.3.8.5

$$F_c = F_{co}\left[1.0 - .385\ \frac{(L'/\rho)}{\pi\sqrt{E/F_{co}}}\right]$$

Where:

$F_{co} = S_{aa} = 53,100$ psi

L' = Effective length $= L/\sqrt{c}$

$c = 2$, where both ends are fixed against lateral movement.

$E = 26,100,000$ psi

ρ = Radii of gyration

$$L' = \frac{146}{1.414} = 103.23"$$

$$F_c = 53,100\left[1.0 - .385\left(\frac{40.7}{69.6}\right)\right] = 41,143 \text{ psi}$$

$$M.S. = \frac{41,143}{87.56} - 1 = 469.2$$

Consider the individual buckling of the .048 stainless steel sheets in the transmitting of the lower support structure load.

Ref: 57, Case 158, Page 701 *

Load on plate = (Area)(Stress)

A = (.048)(5.775) = .277

Load on plate = 87.56(.277) = 24.27 pounds

$$\text{Critical load} = \bar{n} = \lambda_{CR} \frac{\pi^2 D}{b^2}$$

Where:

b = 5.775

$\lambda = 7.0$

$$D = \frac{E t^3}{12(1-v^2)} = \frac{26,100,000(.048)^3}{10.92} = 264.32$$

$$\bar{n} = \frac{7.0(\pi)^2 (264.32)}{(5.775)^2} = 547.5 \text{ lbs.}$$

$$\text{M.S.} = \frac{547.5}{24.27} - 1 = 21.5$$

The absorber material: This plate of absorber material is secured by two sheets of stainless steel clad material which would naturally tend to inhibit any classical wave form buckling. The primary load on the free standing sheet of neutron absorber material is its own weight under 30g impact loading.

The weight of one sleeve plate is determined by taking appropriate ratios of the PWR absorber plate.

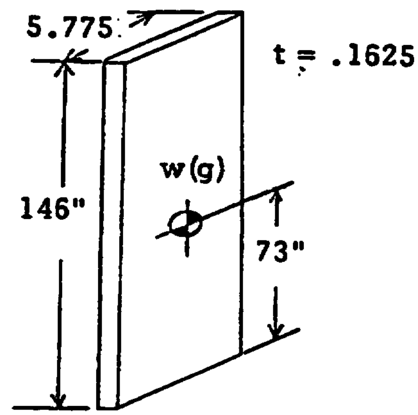
$$W_t = \left(\frac{5.77}{8.98}\right)\left(\frac{146}{151}\right)(192.5)\left(\frac{.1625}{.2425}\right) = 80.3 \text{ lbs.}$$

Impact load = 80.3(30) = 2409 pounds

$E = 10.2 \times 10^6$ at 499°F

*See appendix A

Allow. tension/compression =38,610 (Ref. 19)



Area at base = .1625(5.775) = .94 in.^{2}

$$\sigma_c = P/A = \frac{2409}{.94} = 2,562 \text{ psi}$$

$$M.S. = \frac{38,610}{2,657} - 1 = 13.5$$

Assume the effective column length is 75.5 inches with the full load of 2409 applied at that plane.

Ref. 3, Table XVI, Page 312, Case 1-A

a = 73"
b = 5.775"

$$S' = K\frac{E}{1 - v^2} \; (\Gamma / b)^2$$

Recall that, as a/b approaches infinity

K = 3.29

$$S' = \frac{(3.29)(10,200,000)}{.91} \left(\frac{.1625}{5.775}\right)^2$$

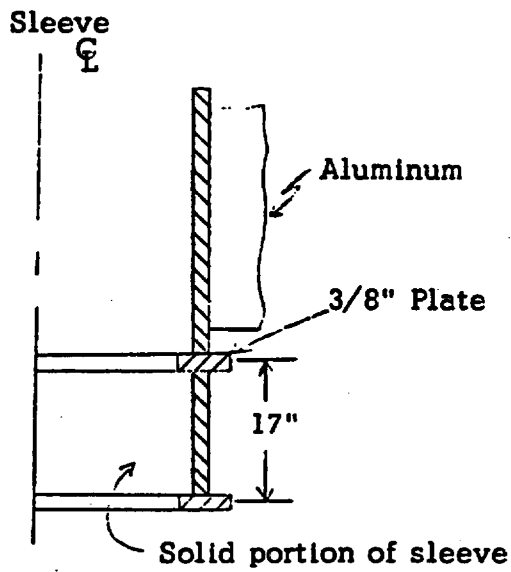
S' =29,198 psi (The buckling stress is well above the applied stress)

$$M.S. = \frac{29,198}{2,657} - 1 = 9.9$$

The absorber plate is able to withstand its own weight during impact.

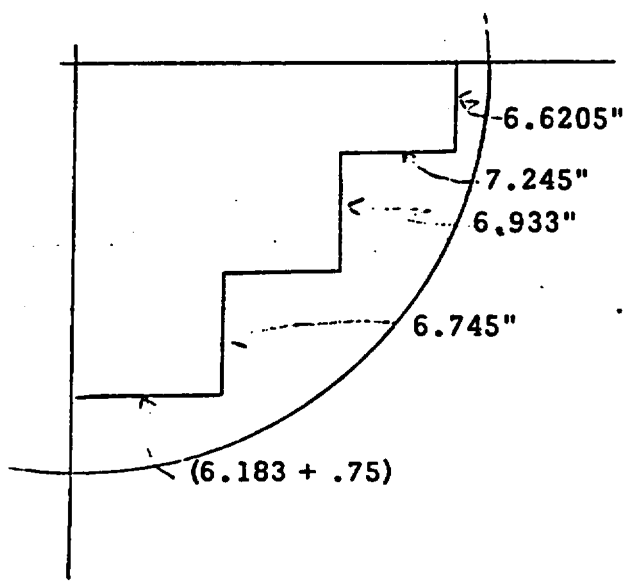
To determine length of solid section refer to page XI-3-39b

Length = 167 - 150 = 17"



Side view of sleeve and aluminum

Welded regions of .375 plate to basket;



Quadrant view looking axially at welds.

Lengths are indicated.

Total length = 6.933 + 6.745 + 6.933 + 7.245 + 6.6205 = 34.5
(Quadrant)

Total weld area = .707(.375)(4)(34.5) = 36.6 in.^{2}

BWR weights of aluminum basket = 9910

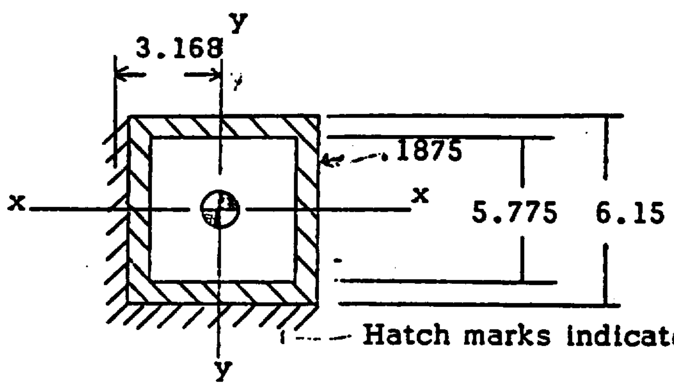
Impact Force = 30(9910) = 297,300 lbs = F_B

Weld shear stress = $F_B/A = \frac{297,300}{36.6} = 8,122$ psi

Shear allow. = .6(53,100) = 31,860 psi (Full pent weld)

$$M.S. = \frac{31,860}{8,122} - 1 = 2.9$$

By inspection of the BWR sleeve array, it is evident that the ratio of imposed load to resisting section properties is the greatest on the outermost sleeve. This is the configuration shown on page XI-4-90L , with the weld loads acting on two sides of the solid sleeve section. Hence, the loading situation and resisting section is shown below.



Load per side of stainless steel box section

Stress = 8,122 psi

Area of weld = 6.15(.707)(.375) = 1.63 in.^{2}

P = 8,122(1.63) = 13,239 pounds

Section properties;

Area = $(6.15)^2 - (5.775)^2 = 37.8 - 33.35 = 4.45$ in.^{2}

$$I_x = I_y = \frac{(6.15)^4}{12} - \frac{(5.775)^4}{12} = 119.2 - 92.7 = 26.5 \text{ in.}^4$$

$$\rho = \sqrt{I/A} = \left(\frac{26.5}{4.45}\right)^{\frac{1}{2}} = 2.44$$

$$L/\rho = \frac{17}{2.44} = 6.9$$ (The L/ρ is clearly too low to exhibit column instability characteristics)

Loads on upper solid section;

P = 30(244) + 353.75 + 2(13,239) = 7320 + 353.75 + 26,478 = 34,151.7 lbs.

Maximum Moment = 13,239(3.168)(2) = 83,882 in.lbs.

$$\text{Maximum Fiber Stress} = P/A + M_c/I = \frac{34,151.7}{4.45} + \frac{83,882(3.075)}{26.5}$$

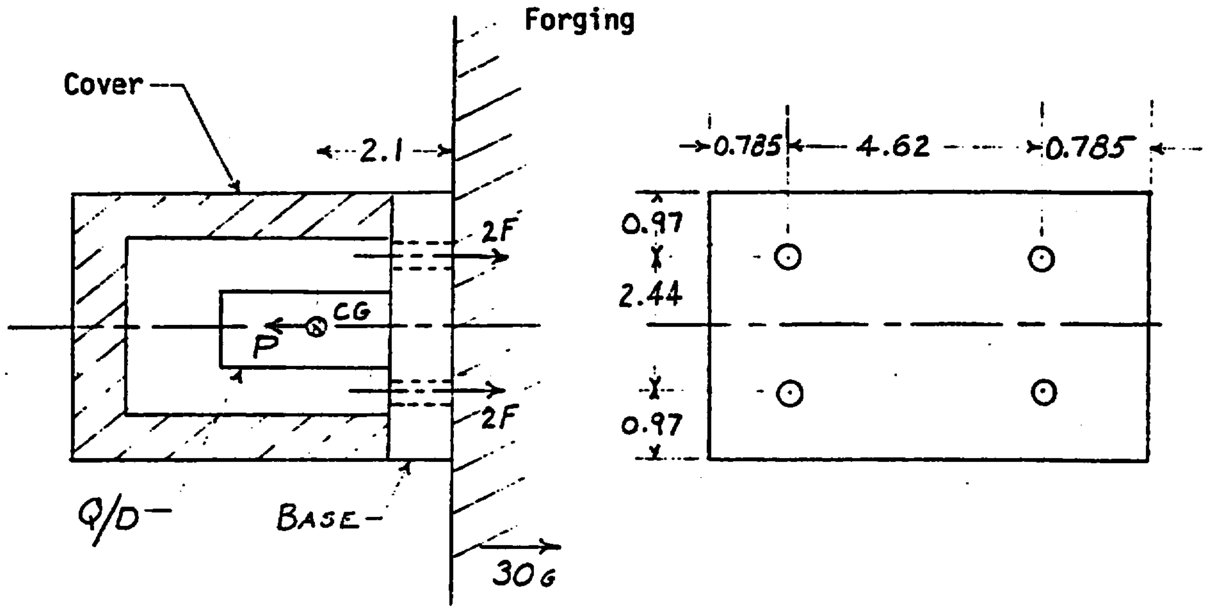
Stress = 7,674 + 9,733 = 17,407 psi

S_{aa} = 53,100 psi

$$M.S. = \frac{53,100}{17,407} - 1 = 2.05$$

4.4.8 Containment Vessel Valves

Each containment vessel valve assembly consists of a quick-disconnect valved nipple, a base plate and a cover. Details of the assembly, including seals and mounting bolts, are given in drawing 70651F. Four bolts, 3/8 - 16 UNC, hold the assembly in place on the top forging and provide ample clamping force to maintain the seal under all normal and accident conditions of transport. Cask acceleration in the top end impact is 30g. General configuration of valve assembly:



Weight and C.G. location of valve assembly are as follows:

<u>Q/D</u> W_1 = 1.9 lb. ; Y_1 = 2.5 in.

<u>Base</u> W_2 = (.285)(4.38)(6.19)(1) = 7.7 lb. ; Y_2 = 0.5 in.

<u>Cover</u> $W_3 = (.285)\left[(4.375^2)(3.63) - (\pi/4)(2.96^2)(3)\right] = 13.9$ lb.

$Y_3 = \left[19.8(2.8) - 5.9(2.5)\right](1/13.9) = 2.9$ in.

Total weight, $W = 1.9 + 7.7 + 13.9 = 23.5$ lb.

C.G. location, $Y_0 = (1/23.5)\left[.5(7.7) + 2.5\ (1.9) + 2.9(13.9)\right] = 2.1$ in.

Impact load @30g, $P_1 = 30\ (23.5) = 705$ lb.

Pressure load, $P_2 = (\pi/4)\ (3.094^2)\ (72.9) = 548$ lb. (Sect. 3.3.2)

Seal load, $P_3 = \pi(3.094)\ (700) = 6804$ lb.

(see section VI for seal seating load of 700 lb/in)

Total load, $P = 705 + 548 + 6804 = 8057$ lb.

load/bolt, $F = 8057/4 = 2014$ lb.

Since the bolts are each preloaded to 2533 lb (Sect. 4.6.4), the valve assembly remains clamped against the forging in the top end impact and the seal is maintained.

M.S. = (2533/2014) - 1 = <u>0.26</u>

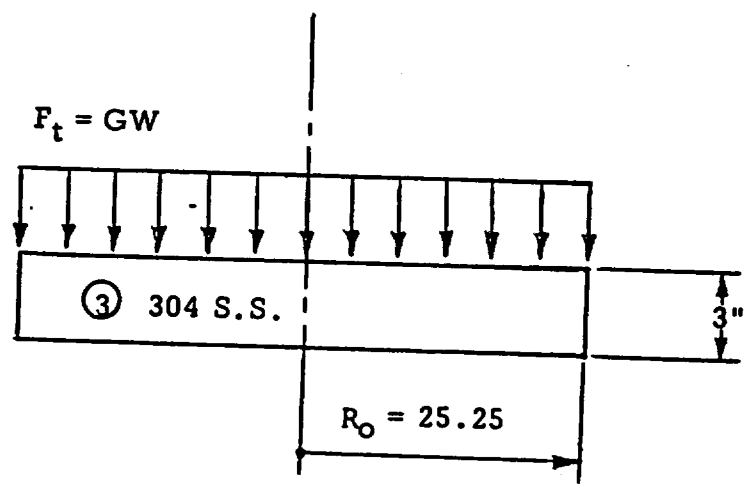
4.5 Bottom End Impact

4.5.1 Closures

Inner Closure Head

During bottom end impact the inner closure will be subjected to inward inertial forces due to its own weight and to an outward force due to pressure in the containment vessel. As will be shown, the two (2) inner plates remain in contact and act as a unit but separate from the outer plate.

Outer Plate of Inner Closure Head



Design Conditions:

Temperature	410°F
Elastic Modulus	26.6×10^6 psi
Poisson's Ratio	0.3
Weight	1905 lbs.

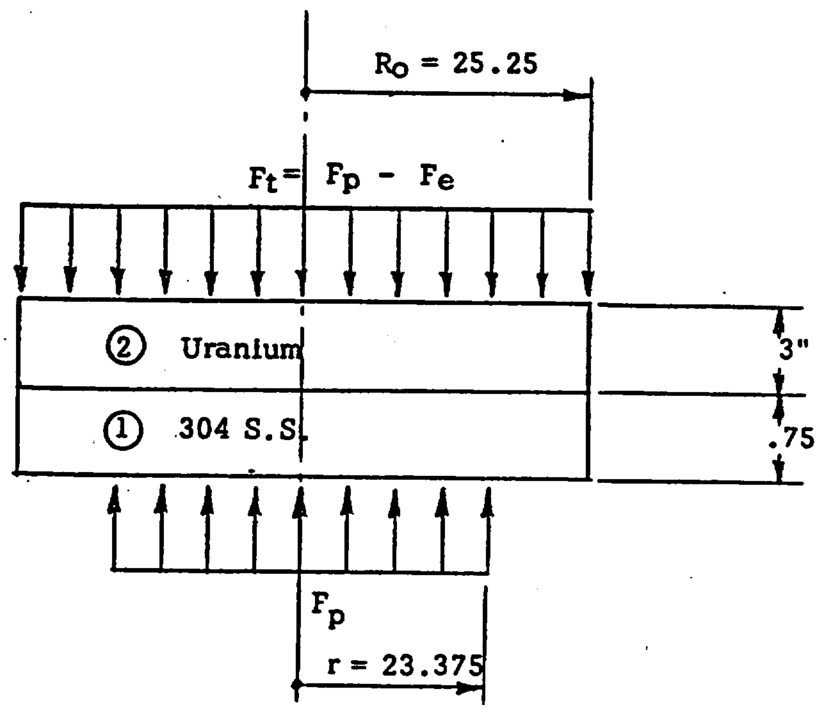
Maximum bending stress at center of plate Ref. 3 , Table X, Case 1.

$$\sigma_c = \frac{3\,(3 + v)}{8\pi t^2} F_t \qquad F_t = GW$$

$$F_t = 30 \times 1905 = 57,150 \text{ lbs.}$$

$$\sigma_c = \frac{3\,(3 + 0.3)}{8\pi\, 3^2} \times 57,150 = 2,501 \text{ psi.}$$

Two Inner Plates



Term	① 304 S.S.	② Uranium
Temp. °F	410	410
E, psi	26.6×10^6	24.8×10^6
v	0.3	0.22
Weight lbs.	575	4104

The two inner plates are constrained to have the same elastic deflection curves under lateral bending load. Hence the deflection of each plate must be the same and the total lateral load on the assembly can be divided between the individual plates in accord with each one's proportionate part of the total bending resistance.

The appropriate formulas for deflection and maximum stress are taken from Ref. 3 , Table X, Case 1

Center deflection, $y_c = \frac{3\,(1-v)\,(5+v)\,R_o^2}{16\pi E\,t^3} F = KF$

Max. stress at center, $\delta_c = \frac{3\,(3+v)}{8\pi\,t^2} F$

Equating the center deflections of the two plates gives:

$$K_1\,F_1 = K_2\,F_2$$

Also, the total load imposed on the closures must equal the sum of the individual plate loads, so that $F_t = F_1 + F_2$

Combining these equations in terms of plate ① load, F_1 gives:

$$F_t = K_1\,F_1 \left(\frac{1}{K_1} + \frac{1}{K_2} \right)$$

These equations may be evaluated to obtain the force on each plate as follows:

$F_t = F_p - F_e$

$F_e = G\,(W_1 + W_2)$

$G = 30$

$W_1 = 575$ lbs.

$W_2 = 4104$ lbs.

Containment vessel pressure is 16.45 psig (Sect. 3.3.1) and this pressure is chosen for the analysis because it will result in a higher total force (F_t).

$F_p = pA$ $\qquad$ $A = \pi\ 23.375^2$

$F_p = 16.45\ (1716.5) = 28326$ lbs. $\qquad$ $A = 1716.5\ in^2$

$F_t = 28236 - 30\ (575 + 4104)$

$F_t = -112134$ lbs. (inward force)

Evaluating the compliance constant K for each plate:

$$K_1 = \frac{3\ (0.7)\ (5.3)\ (25.25^2)}{16\ \pi\ 26.6 \times 10^6 \times .75^3} = 1.258 \times 10^{-5}\ \text{in/lb.}$$

$$K_2 = \frac{3\ (0.78)\ (5.22)\ (25.25^2)}{16\ \pi\ 24.8 \times 10^6 \times 3^3} = 2.31379\ \ 10^{-7}\ \text{in/lbs.}$$

Now F_1 can be found from the previous equation as follows:

$$112134 = 1.258 \times 10^{-5}\ F_1\ \ (7.949 \times 10^4 + 4.3219 \times 10^6)$$

$$F_1 = \frac{112134}{55.369} = 2025\ \text{lbs.}$$

$$F_2 = K_1\ F_1\ /\ K_2 = \frac{1.258 \times 10^{-5}\ (2025)}{2.31379 \times 10^{-7}} = 110110\ \text{lbs.}$$

Plate (1) Max. stress $= \dfrac{3(3.3)\ \ (2025)}{8\ \pi\ .75^2} = 1418$ psi.

Plate (2) Max. stress $= \dfrac{3\ (3.22)\ \ (110110)}{8\ \pi\ 3^2} = 4702$ psi.

Center deflection for plates (1) (2) (3) :

Y_c Plate (1) $= K_1\ F_1$

$= 1.258 \times 10^{-5}\ (2025) = 0.02548$ in.

Y_c Plate (2) $= K_2 F_2$

$= 2.31379 \times 10^{-7}$ (110110) = 0.02548 in.

Y_c Plate (3) $= K_3 F_3$

$= 1.9656 \times 10^{-7}$ (57150) = .01123 in.

where $K_3 = \frac{3 \ (0.7) \ (5.3) \ (25.25)^2}{16 \ \pi \ 26.6 \times 10^6 \ (3^3)}$

$= 1.9656 \times 10^{-7}$ lb./in.

The above deflections are all inward and show that plates (1) and (2) do separate from plate (3). Plates (1), (2) have an inward inertial force due to their own weight greater than the outward force due to internal pressure of 16.45 psig (vessel helium pressure).

If the containment vessel has an internal pressure of 80.5 psig (BWR fuel rupture) plates (1), (2) have an outward force greater than the internal inertial force due to their own weight. In this case plates (1), (2) will deflect outward in contact with plate (3), so all three plates will act together. Thus the stresses for this case will be lower than those calculated above.

Calculating effective stresses on plate (1), (2), (3)

proper formula for effective stress equals

$$S_e = \sqrt{\frac{1}{2}} \sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(T_{xy}^2 + T_{yz}^2 + T_{zx}^2)}$$

where σ_x, σ_y, σ_z are normal stresses, T_{xy}, T_{yz}, T_{zx} are shear stresses (Sect. 1.1)

Plate (1) highest stress area is at the center portion of the inner surface.

σ_x = 1418 psi (Radial Stress)

σ_y = 1418 psi (Tangential Stress)

σ_z = 16.45 psi (Axial Stress)

$T_{xy} = T_{yz} = T_{zx} = 0$

$$S_{e3} = \sqrt{\frac{1}{2}} \sqrt{(1418 - 1418)^2 + (1418 - (-16.45))^2 + (-16.45 - 1418)^2}$$

= 1434.45 psi

Allowable stress ($.8\,S_{aa} = 0.7\,S_u$) at 410°F for 304 S/S from Sect. 1.1 under containment vesset and Sect. 1.2 equals to 0.7 x 59500 = 41650 psi

$$M.S. = \frac{41650}{1434.45} - 1 = 28$$

Plate (2) highest stress area is at the center portion of the inner surface.

σ_x = 4702 psi (Radial Stress)

σ_y = 4702 psi (Tangential Stress)

σ_z = 6.88 psi (Axial Stress)

$T_{xy} = T_{yz} = T_{zx} = 0$

Solving for pressure stress σ_z between plate (1) and plate (2)

$$2025 = F + 17250 - 27044$$

$$F = 11819 \text{ lbs.}$$

$$\sigma_z = \frac{11819}{1716.5} = 6.88 \text{ psi}$$

$$S_{e3} = \sqrt{\frac{1}{2}}\sqrt{(4702 - 4702)^2 + (4702 - (-6.88))^2 + (-6.88 - 4702)^2}$$

$$= 4708.88 \text{ psi}$$

Allowable stress ($.8\, S_{aa} = 0.7\, S_u$) at 410°F for uranium from Sect. 1.1 under containment vessel and Sect. 1.2 ecuals to 0.7 x 56000 =39200 psi

$$M.S. = \frac{39200}{4708.88} - 1 = 7.32$$

Plate (3) highest stress area is at the center portion of the inner surface.

σ_x = 2501 psi (Radial Stress)

σ_y = 2501 psi (Tangential Stress)

σ_z = 0 (Axial Stress)

$T_{xy} = T_{yz} = T_{zx} = 0$

$$S_{e3} = \sqrt{\frac{1}{2}}\sqrt{(2501 - 2501)^2 + (2501 - 0)^2 + (0 - 2501)^2}$$

$$= 2501 \text{ psi}$$

$$M.S. = \frac{41650}{2501} - 1 = 15.6$$

Outer Closure Head

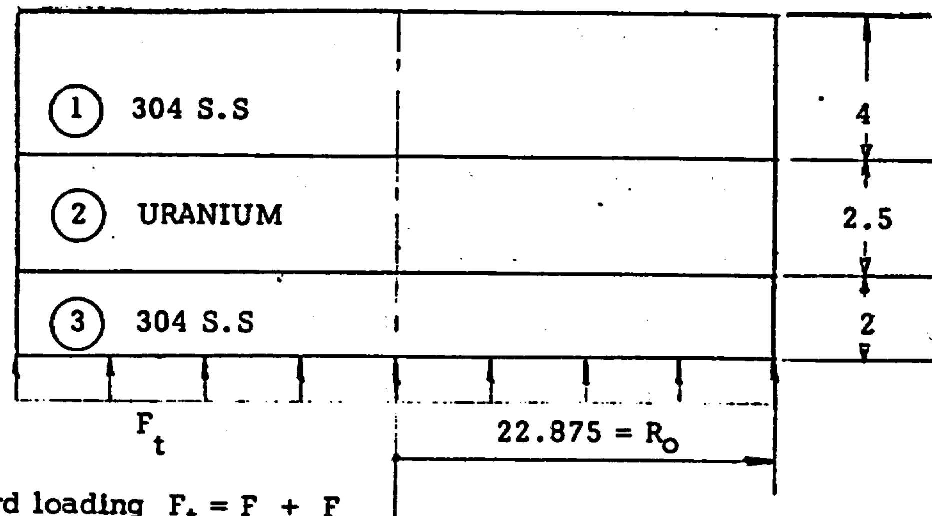
During bottom end impact the outer closure head is subjected to an inward loading due to its own weight time 30G acceleration. This results in a force equal to 69,840 lbs. less an outward force of 23,609 lbs. due to internal pressure, giving a net total inward force of 46,231 lbs.

During the top end impact, the outer closure head is subjected to an outward inertial force of 69,840 lbs. plus 23,609 lbs from internal pressure for a total of 93,449 lbs. The analysis of the outer closure head under this loading is presented in Section 4.4.1 and shows the resulting stress to be within the established limit. Comparison of forces on the outer closure head for bottom end and top end impacts shows that the resulting stress is much less than the stress calculated in Section 4.4.1 for top end impact.

4.5.2 Bottom Head

The bottom head arrangement supports the internal pressure (static load) plus the weight of the head and contents at an acceleration of 30g (dynamic load).

The impact force on the impact limiter because of its configuration, is balanced by the deceleration load of the cask bottom forging, which is outboard of the bottom head arrangment shown below.



Outward loading $F_t = F + F_p$

$F = 30(W)$ — 30g = Acceleration during bottom end impact.

$F = 30\ (40{,}900)$ — $W = 34{,}500$ (contents) + 6,400 (bottom head)

$= 1{,}227{,}000$ lbs. — $= 40{,}900$ lbs

$F_p = 21(A)$ — Internal pressure assuming total fission gas release is 72.9 psig. (BWR Sect. 3.3.2)

$A = \pi (22.875^2) = 1{,}644\ in^2$

$F_p = 72.9\ (1{,}644) = 119{,}848$ — Higher internal pressure is chosen, this results in higher outward force on the bottom head.

$F_t = 1,227,000 + 119,848$

$= 1,346,848$

Following the same method of analysis and analytical model as used in Section 3.9 the compliance constant K for each plate are as follows:

$K_1 = 5.44472 \times 10^{-7}$ in./lb.

$K_2 = 3.28146 \times 10^{-7}$ in./lb.

$K_3 = 1.66937 \times 10^{-8}$ in./lb.

From Section 3.9

$$F_t = K_3 F_3 \left(\frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3}\right)$$

$$1,346,848 = 1.66937 \times 10^{-8} F_3 (1.8366 \times 10^6 + 3.0472 \times 10^6 + 5.99028 \times 10^7)$$

$$F_3 = \frac{1,346,848}{1.08153} = 1,245,317 \text{ lbs.}$$

$$F_1 = \frac{K_3 F_3}{K_1} = \frac{1.66937 \times 10^{-8} \times 1,245,317}{5.4472 \times 10^{-7}} = 38,164 \text{ lbs.}$$

$$F_2 = \frac{K_3 F_3}{K_2} = \frac{1.66937 \times 10^{-8} \times 1,245,317}{3.28146 \times 10^{-7}} = 63,353$$

Bending stress on each plate is

Plate ① $\sigma_1 = \frac{3(3 + v)}{8 \pi t^2} F_1 = \frac{3 \times 3.3}{8 \pi (2^2)} (38164) = 3758$ psi

Plate ② $\sigma_2 = \frac{3(3 + v)}{8 \pi t^2} F_2 = \frac{3 \times 3.3 (63353)}{8 \pi (2.5^2)} = 3993$ psi

Plate ③ $\sigma_3 = \frac{3F_3}{4 \pi t^2} = \frac{3 (1245317)}{4 \pi 4^2} = 18581$ psi

Calculating effective stresses on plate (1), (2), (3).

The formula for effective stress equals

$$S_e = \sqrt{\frac{1}{2}}\ \sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(T_{xy}^2 + T_{yz}^2 + T_{zx}^2)}$$

where σ_x, σ_y, σ_z, are normal stresses, T_{xy}, T_{yz}, T_{zx} are shear stresses (Sect. 1.1)

Plate (1) The highest stress area is at the center portion of the outer surfa ce.

σ_x = 3758 psi (radial stress)

σ_y = 3758 psi (tangential stress)

σ_z = -696 psi (axial stress)

$T_{xy} = T_{yz} = T_{zx} = 0$

Calculating the pressure stress σ_z between plate (1) and plate (2)

38164 = 119848 + 1063110 - F

F = 1144794 lbs.

$$\sigma_z = \frac{1144794}{1644} = 696 \text{ psi}$$

$$S_{e3} = \sqrt{\tfrac{1}{2}}\ \sqrt{(3758 - 3758)^2 + (3758 - (-696))^2 + (-696 - 3758)^2}$$

= 4454 psi

Allowable stress ($0.8\,S_{aa} = 0.7\,S_u$) at 410°F for 304 S/S from Sect. 1.1 under containment vessel and Sect. 1.2 equals 0.7 x 59500 = 41650 psi

$$\text{M.S.} = \frac{41650}{4454} - 1 = 8.35$$

Plate (2) The highest stress area is at the center portion of the outer surface.

σ_x = 3993 psi (radial stress)

σ_y = 3993 psi (tangential stress)

σ_z = -688 psi (axial stress)

$T_{xy} = T_{yz} = T_{zx} = 0$

Calculating the pressure σ_z between plate (2) and plate (3)

63,353 = 1,144,794 + 50,550 - F

F = 1,131,991 lbs.

$$\sigma_z = \frac{1,131,991}{1644} = 688 \text{ psi}$$

$$S_{e4} = \sqrt{\tfrac{1}{2}}\ \sqrt{(3993 - 3993)^2 + (3993 - (-688))^2 + (-688 - 3993)^2}$$

= 4681 psi

Allowable Stress ($S_{aa} = 0.9\, S_u$) at 410°F for uranium from Sect. 1.1 under nonconta inment structure and Sect. 1.2 equals to .9 x 56000 =50400 psi

$$M.S. = \frac{50400}{4681} - 1 = 9.76$$

Plate (3) The highest stress is at the edge portion of the inner surface.

σ_x = 18,581 psi (radial stress)

From Ref. 3, Table X Case 6

$$\sigma_y = \frac{3\ v\ F}{4\pi t^2} = \frac{3 \times .3 \times 1,245,317}{4\ \pi\ 4^2} = 5574 \text{ psi (tangential stress)}$$

$$\sigma_z = \frac{1,131,991}{1644} = -688 \text{ psi (axial stress)}$$

$$T_{zx} = \frac{1,245,317}{45.75\ \pi\ 4} = 2166 \text{ psi}$$

$T_{yz} = T_{xy} = 0$

$$S_{e4} = \sqrt{\tfrac{1}{2}\sqrt{(18581 - 5574)^2 + (5574 - (-688))^2 + (-688 - 18581)^2 + 6(2166)^2}}$$

$= 17{,}433$ psi

Allowable stress ($S_{aa} = 0.9\ S_u$) at 410°F for 304 S/S from Sect. 1.1 under noncontainment structure and Sect. 1.2 equals to:

$0.9 \times 59500 = 53550$ psi

$$\text{M.S.} = \frac{53550}{17433} - 1 = 2.07$$

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4.5.3 SHELL ANALYSIS FOR 30G BOTTOM END IMPACT

INTRODUCTION

The bottom end impact analysis of the cask shells was performed for an equivalent 30g static force.

The shells were analyzed using the finite element idealization of Section 3.8 of this SAR. In addition, the evaluation procedure described in Section 3.8 was utilized.

SUMMARY AND CONCLUSIONS

This phase of the analysis indicated the following:

a. The primary stresses developed in the equivalent 30g loading are not large. The maximum effective primary stress of 6297 psi is developed in the outer shell at the bottom of the cask. This value is well within the allowable stress of $0.7\ S_u = 45386$ psi at that location.

b. When the 30g loading is coupled by superposition with the -40^{o}F isothermal solution and the stress solutions for the normal transport conditions, the resulting primary plus secondary stresses are all within the allowable stress values.

ANALYSIS

To analyze the cask for the 30g bottom end loading, the same finite element model as used in the analysis of the normal cycle (Section 3.8) was employed. For the solution, the cask was assumed to be at an isothermal 360^{o}F. Thus, there were no thermal stresses computed directly in the equivalent dynamic loading solution. A node in the bottom end forging of the model at a radius that approximated the line of action of the impact absorber was fixed axially in space to provide a reaction point for the solution. No internal pressures were applied to the cask in this solution. Moreover, the weight of the water was not included in this solution. The effect of the water pressure would be to place an additional hoop compressive force on the inner and outer shells, thus reducing the effective stress resulting from the compressive axial gravity loading. It was thus concluded that omission of the water in the bottom drop solution was a conservative assumption.

RESULTS

Plots of the lead pressures and shear stresses together with the membrane stresses of the inner and outer shells at the bottom of the cask are presented in Figs. 4.5.3-1 through 4. As indicated in Fig. 4.5.3-2, the unsupported region of the inner shell at the bottom of the cask is not heavily loaded in the bottom end drop. This result is due to the axial support supplied by the stiff uranium ring. Membrane stresses for the cask evaluation points discussed in Section 3.8 are presented in Table 3.8.4-1, base case no. 8. Primary plus secondary stresses for the evaluation locations are presented in Table 3.8.4-2, also listed as base case no. 8.

EVALUATION

Table 4.5.3-1 presents the primary stress evaluation for the 30g bottom end drop. The effective stresses indicated in this table result from a superposition of the indicated base case membrane stresses. The allowable stress for the evaluation of 0.7 S_u is also listed in the table. As seen in Table 4.5.3-1, the stress values are well within the allowable.

Table 4.5.3-2 presents the primary plus secondary stress evaluation. In this table, the effective stresses result from the superposition of the 30g bottom end drop stresses with the other stress results indicated at the top of the table. As shown in Table 4.5.3-2, the stresses for the load combinations examined are well within the allowable stress of 0.9 S_u.

TABLE 4.5.3-1

ACCIDENT PRIMARY STATIC AND DYNAMIC STRESS EVALUATION

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
8	1.00000	30 G BOTTOM END DROP
10	.72900	100 PSI CAVITY

LOCATION	EFFECTIVE STRESS Se3	TEMP.	STRESS ALLOWABLES 0.7SU
1	701	268	45465
3	595	248	45990
5	1525	218	46777
7	1033	316	44205
9	1122	303	44546
11	825	420	41895
13	2792	359	43076
15	2576	323	44021
17	2532	302	44572
19	1758	290	44887
21	2446	292	44835
23	6297	271	45386
25	5363	241	46174
27	2270	218	46777
29	747	260	45675
31	66	256	45780
33	826	240	46200

TABLE 4.5.3-2

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS EVA

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
2	1.00000	-40.0 F ISOTHERMAL
8	1.00000	30 G BOTTOM END DROP

LOCATION	EFFECTIVE STRESS S_{e4}	TEMP.	STRESS	ALLOWABLES 0.9SU
1	32870	-40		103500
2	19590	-40		103500
3	19192	-40		103500
4	12030	-40		103500
5	4097	-40		103500
6	753	-40		103500
7	40665	-40		103500
8	37559	-40		103500
9	20766	-40		103500
10	18644	-40		103500
11	37270	-40		103500
12	35264	-40		103500
13	21842	-40		103500
14	18430	-40		103500
15	3995	-40		103500
16	4005	-40		103500
17	18361	-40		1[illegible]0
18	16493	-40		103500
19	13940	-40		103500
20	16393	-40		103500
21	14861	-40		103500
22	12377	-40		103500
23	12405	-40		103500
24	12608	-40		103500
25	8031	-40		103500
26	7411	-40		103500
27	6671	-40		103500
28	1048	-40		103500
29	7237	-40		103500
30	7148	-40		103500
31	4773	-40		103500
32	15032	-40		103500
33	7533	-40		103500
34	12127	-40		103500

TABLE 4.5.3-2 (CONT'D)

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS EVAL

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
4	1.00000	NORMAL 70KW 130F AMBIENT
8	1.00000	30 G BOTTOM END DROP

LOCATION	EFFECTIVE STRESS Se_4	TEMP.	STRESS ALLOWABLES 0.9Su
1	24238	268	58455
2	4[illegible]080	267	58489
3	8078	248	59130
4	4566	237	59501
5	29631	218	60142
6	19576	218	60142
7	13471	316	56335
8	11813	314	56302
9	4668	303	57274
10	7346	291	57579
11	26032	420	53355
12	22950	420	53355
13	5055	359	55384
14	11284	344	55890
15	11217	323	56599
16	11044	323	56599
17	4743	302	57307
18	27153	301	57341
19	2[illegible]803	290	57712
20	14337	290	57712
21	9707	292	57645
22	10921	290	57712
23	10845	271	58354
24	1310	259	58759
25	25734	241	59366
26	29072	230	59737
27	35128	218	60142
28	25789	218	60142
29	4090	260	58725
30	8802	260	58725
31	25101	256	58860
32	16153	256	58860
33	19954	240	59400
34	28899	240	59400

TABLE 4.5.3-2 (CONT'D)

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS E/A

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
8	1.00000	30 G BOTTOM END DROP
13	1.00000	NORMAL 70KW -40F AMBIENT

LOCATION	EFFECTIVE STRESS Se_4	TEMP.	STRESS ALLOWABLES 0.9S
1	33920	103	67463
2	39084	102	67533
3	6333	81	68988
4	1230	69	70057
5	9465	50	75888
6	4939	50	75886
7	21851	148	64350
8	18789	147	64419
9	8865	133	65388
10	5190	119	66358
11	19194	242	59332
12	18144	239	59434
13	4927	185	61719
14	5010	171	62758
15	2819	149	6[illegible]81
16	2891	149	6[illegible]31
17	9535	135	65250
18	39164	134	65319
19	21962	123	66081
20	18625	123	66081
21	11748	125	65342
22	8903	124	65012
23	14512	103	67465
24	1247	90	68365
25	23072	74	69473
26	33015	63	71393
27	14859	50	75885
28	10818	50	75885
29	4911	102	67535
30	4805	102	67535
31	20944	98	67312
32	18920	98	67312
33	18425	83	68850
34	20340	83	68850

TABLE 4.5.3-2 (CONT'D)

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS EVAL

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
8	1.00000	30 G BOTTOM END DROP
12	1.00000	NORMAL 40KW -40F AMBIENT

LOCATION	EFFECTIVE STRESS Se4	TEMP.	STRESS ALLOWABLES 0.9SU
1	35575	54	74559
2	31982	53	74966
3	11491	42	78341
4	3625	33	81102
5	9279	20	85091
6	4728	20	85091
7	27130	81	68988
8	25078	80	69058
9	13230	72	69512
10	8337	65	71284
11	22024	137	65112
12	21659	135	65250
13	10085	104	67396
14	9653	95	68019
15	750	81	68988
16	777	81	68988
17	14612	73	69542
18	38510	72	69512
19	25937	68	70364
20	24199	68	70364
21	18607	67	70570
22	14145	65	70977
23	13569	54	74559
24	6939	45	77114
25	12055	38	79568
26	24134	29	82330
27	7465	20	85091
28	7207	20	85091
29	5932	52	75273
30	2640	52	75273
31	17298	50	75885
32	17122	50	75885
33	14621	38	79568
34	9831	38	79568

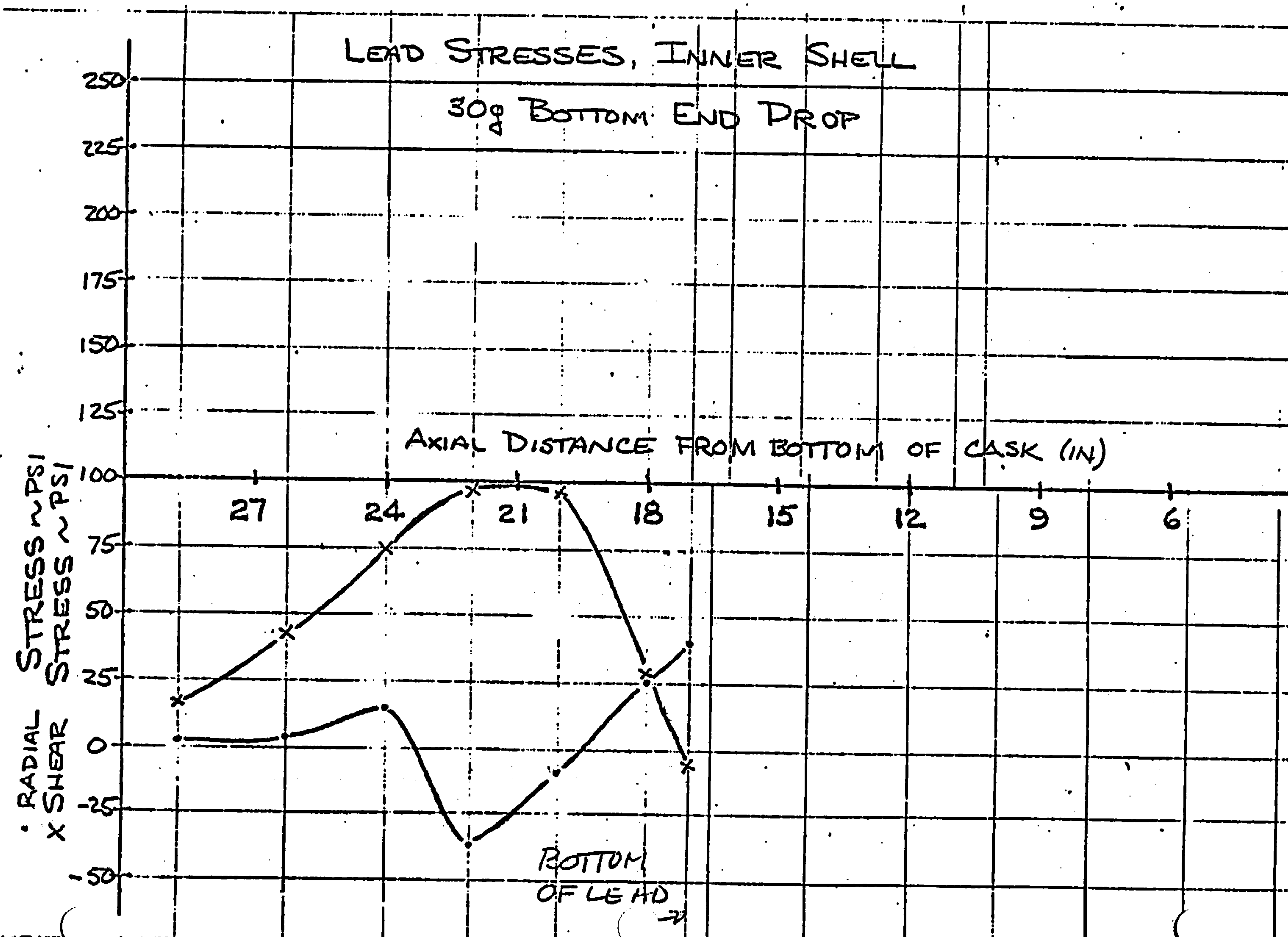

FIG. 4.5.3-1

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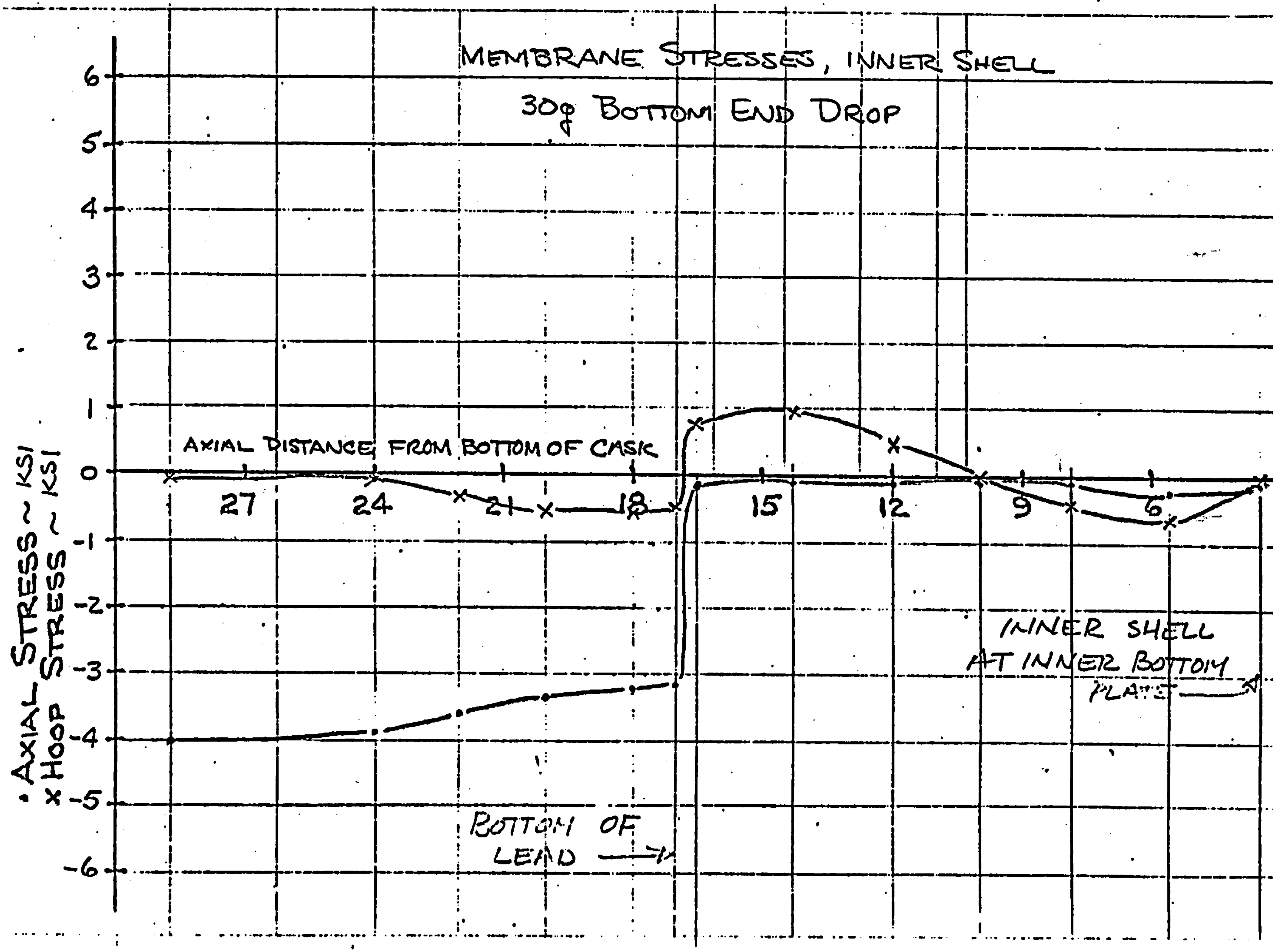

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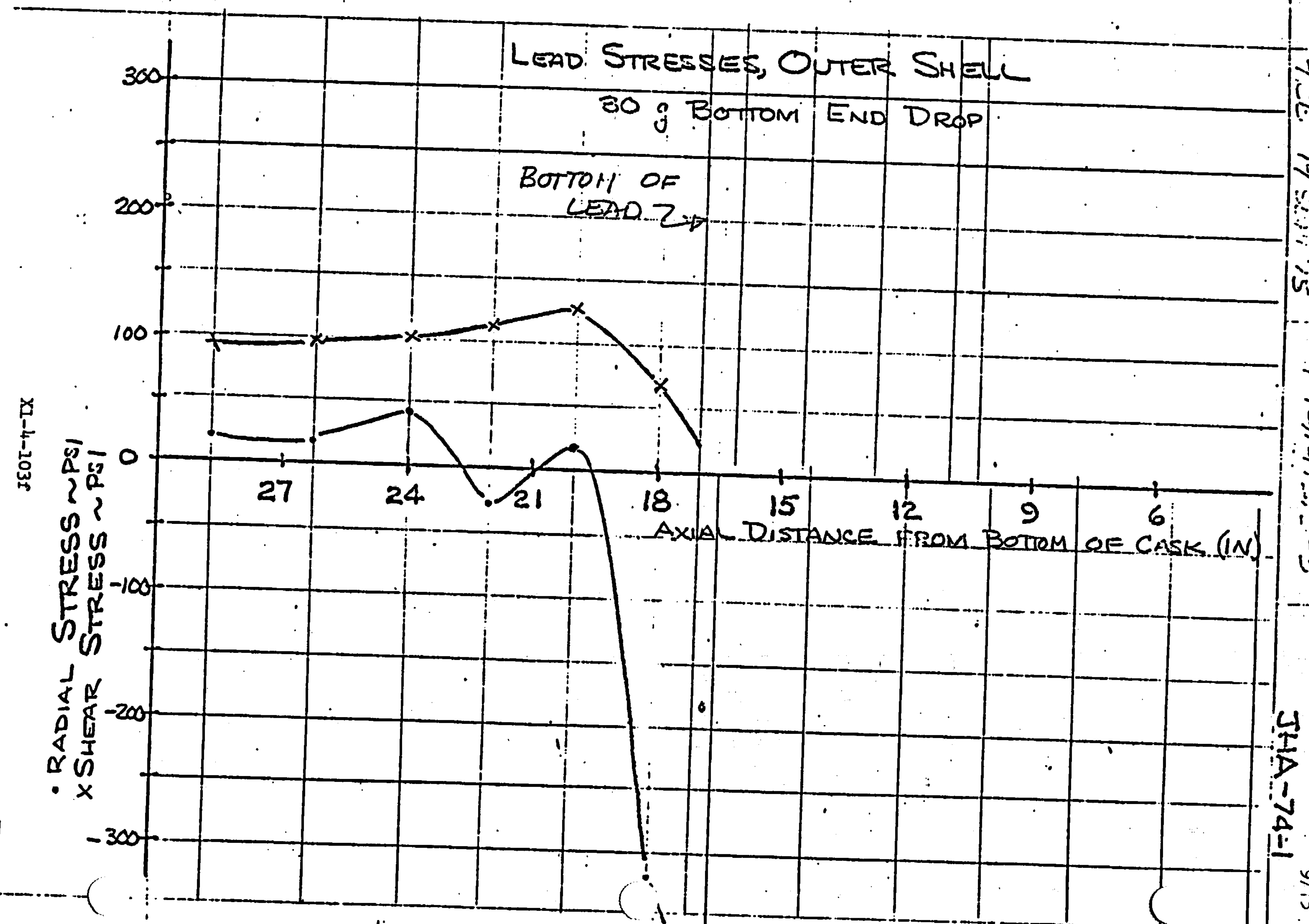

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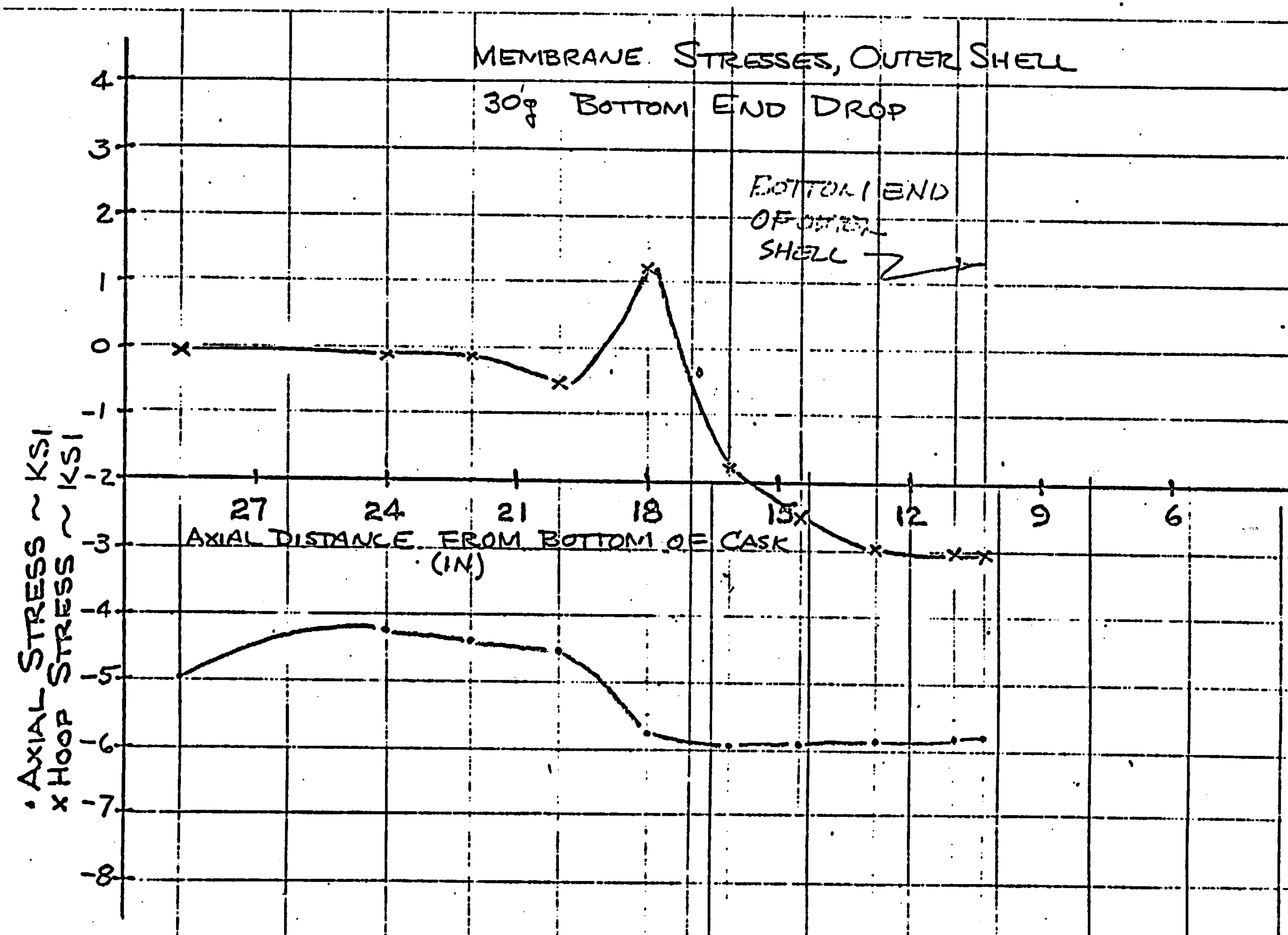

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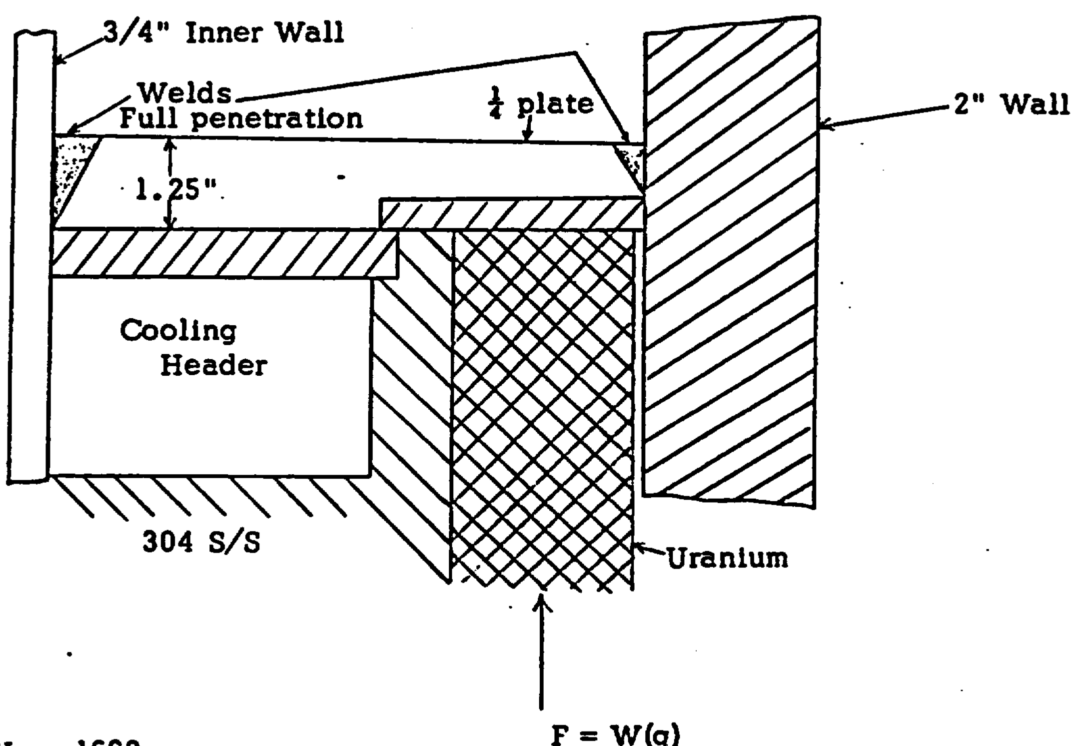
4.5.4 Lead Shielding Retaining Ring

The purpose of this ring is to limit possible axial movement of the lead shielding during a bottom end impact. Since a model test has demonstrated that the lead will not displace axially in an end impact, the ring is no longer needed and no analysis of its strength is required (see Section 4.4.4 and Appendix D).

4.5.5 Upper Uranium Support Ring

Under bottom end impact the upper uranium ring is held in place by a $\frac{1}{4}$ inch plate plus a 1" ring.



W = 1690

g = 30

F = 30 x 1690

= 50700 lbs.

During impact, the ring will deflect slightly to allow the rigid Uranium ring to place the full load on the weld at the outer shell region.

$$\text{Area of weld} = 2\pi\, 29.25\ (1.25) = 230\ \text{in}^2$$

$$\sigma_s = \frac{P}{A} = \frac{50700}{230} = 220\ \text{psi}$$

$$\text{Shear allowable} = .6\ (60500)\ (.9) = 32670\ \text{psi}$$

$$\text{M.S.} = \frac{32670}{230} - 1 = \underline{141}$$

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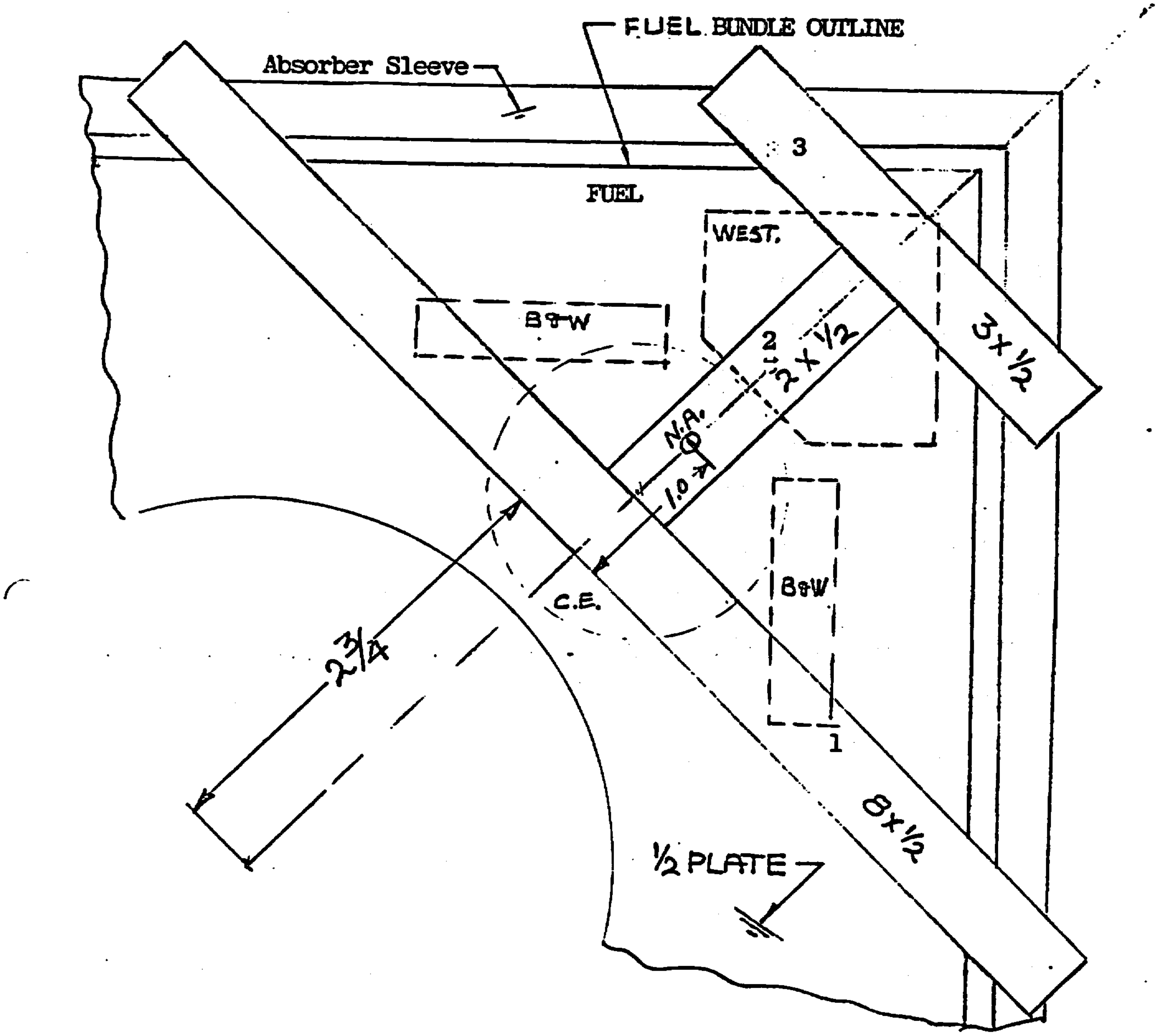
4.5.6 Bottom Support for Basket - PWR

The support for the Aluminum Basket, Absorber Sleeves, and Spent Fuel Elements consists of a lower 1/2 inch plate, an outer ring of 1/4 inch thickness and 1/2 inch thick Aluminum (6061-T6) plate welded together into a H configuration (NLI drawing 70652F). See sketch on page XI-4-109 which shows the C & E, Westinghouse and Babcock and Wilcox Fuel foot-prints with a superimposed outline of support H. The support H is in direct compression from loads due to the bottom end shock forces, the 1/2 inch thick lower plate merely acts as a device to position and maintain the orientation of the supports relative to the fuel foot print locations. The support H's are welded to the 1/2 inch circular plate. The geometry of the support H also allows the sleeve loads to be carried in direct compression.

The absorber sleeves and fuel elements are reacted at two diametrically opposed corners. Both the sleeves and the fuel elements are sufficiently supported and due to the structural integrity of both units, any differential shear within either the fuel or sleeves is negligible.


FUEL BUNDLE OUTLINE
Absorber Sleeve
3
FUEL
WEST.
B&W
2
2 x 1/2
3 x 1/2
N.A.
1.0
2 3/4
C.E.
B&W
1
8 x 1/2
1/2 PLATE

Determination of Properties

Assume an end impact load of 30_g with a fuel element design weight at 1700 pounds.

Assume an equal distribution of shock loads to each two H supports, and uniform loading over H section because of ½ inch plate.

R_1 = 30(1700)(1/2) = 25,500 pounds for fuel
R_2 = 30(770)(1/2) = 11,511 pounds for sleeve
37,011

Item	Area	y	Ay	d	Ad^2		I_o
1	8x1/2 = 4	.25	1.0	.75	2.25		.083
2	2x1/2 = 1	1.5	1.5	.5	.25		.333
3	3x1/2 = 1.5	2.75	4.125	1.75	4.59		.031
	6.5		6.625		7.09	+	.447

I_y = 7.09 + .447 = 7.537

$Z = \frac{7.537}{2} = 3.77$ at edge of item 3

Allowable stress (S_{aa} = 0.9 S_u for Al. 6061-T6 at 438°F from Sect. 1.1 under cask internals structure and Sect. 1.2 equals to (.9)(.43*)(38000) = 14706psi Refer to Ref. 27, Table 3.6.1.0(e) and Fig. 3.6.1.2.1(a)

*Temperature correction factor (Ref. 27 — t = 438°F

Analysis of various fuel stress conditions

For Westinghouse fuel - eccentricity is + 1.125 in.

$$S_f = \frac{M}{Z} = \frac{25{,}500 \times 1.125}{3.77} = 7609 \text{ psi}$$

$$S_c = \frac{P}{A} = \frac{37{,}011}{6.5} = \underline{5694} \text{ psi}$$

$$\text{Total} = 13{,}303$$

$$M.S. = \frac{14{,}706}{13{,}303} - 1 = .105$$

For Combustion Engineering fuel - eccentricity is -.5 in.

$$S_f = \frac{25{,}500 \times .5}{3.77} = 3{,}382 \text{ psi}$$

$$S_c = \underline{5{,}694} \text{ psi}$$

$$\text{Total} = 9{,}076 \text{ psi}$$

$$M.S. = \frac{14{,}706}{9{,}076} - 1 = .62$$

For Babcock and Wilcox fuel - eccentricity is - .25 in.

$$S_f = \frac{25{,}500 \times .25}{3.77} = 1{,}691 \text{ psi}$$

$$S_c = \underline{5{,}694} \text{ psi}$$

$$\text{Total} = 7{,}385 \text{ psi}$$

$$M.S. = \frac{14{,}706}{7{,}385} - 1 = .99$$

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Xi-4-112b
XI-4-112c
XI-4-112d

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The buckling stress on the 1/4 inch outer ring.

The critical buckling stress for an unpressurized thin-walled circular cylinder subjected to axial compression is given by the equation:

Ref. 4 Sect. 10-3, Page 229.

$$\frac{\sigma_{cr}}{n} = K_c \frac{\pi^2 E}{12 (1 - v)^2} \left(\frac{t}{L}\right)^2$$

For short cylinders: The buckling coefficient is expressed by,

$$K_C = K_o + \frac{12r^2 Z^2}{\pi^4 K_{co}}$$

<u>Stability of thin-walled cylinders</u>

For simply supported edges, $K_{CO} = 1$

Also, $Z = \frac{L^2}{Rt} \sqrt{1 - v^2}$

First, investigate the critical buckling of the 1/4-inch thick support plate for shock loads due to an accident condition. The temperature at which the accident takes place is assumed to be the maximum temperature at steady-state loss of coolant operation.

let,

L = 4.00 in.

R = 20.5 in.

t = .25

v = .3

$$Z = \frac{(4.00)^2}{(20.5)\ (.25)} \left[.91\right]^{\frac{1}{2}} = .953 \ \frac{14.0625}{5.125} = 2.462$$

Temp. = 438°F. (average for basket)

The equation for computation of the buckling coefficient yields;

$$K_c = K_{co} + \frac{12r^2Z^2}{\pi^4 K_{co}} \qquad R/_t = {}^{20.5}/_{.25} = 82$$

$$K_{co} = 1.00$$

r = .65 (from correlation factors curve in Ref. 4)
L = Length of Cylinder

<u>Stability of thin walled cylinders (support)</u>

Buckling coefficient evaluation:

$$K_c = 1 + \frac{12\,(.65)^2\ (2.462)^2}{\pi^4\ (1.0)}$$

$$K_c = 1 + \frac{30.73}{97.40} = 1.3155$$

$$E = 8{,}282{,}000 \text{ psi } 438^\circ \text{ F}$$

$$\sigma_{cr} = 1.315 \left[\frac{\pi^2\ E}{12\ (1-v^2)}\right] \left(\frac{t}{L}\right)^2$$

$$\sigma_{cr} = 1.315 \left[\frac{\pi^2\,(8.282)\ (10^6)}{10.92}\right] .00444$$

$$\sigma_{cr} = 43{,}703 \text{ psi} \quad \text{(no stability problem exists)}$$

The sections of the aluminum basket are very thick (16 inches) and provide a high degree of rigidity in the bending mode. The basket can conservatively be assumed to equally distribute it's load on the 1/4" thick outer ring.

Compressive Area of Outer 1/4" Thick Ring

$$\text{Area} = \pi(R_o^2 - R_1^2) = \pi(420.25 - 410.06) = 32.000 \text{ in}^2$$

Total Compressive Area = 32.00 in^2

Ref: 20122B

Basket Weight = 7054 pounds

At Impact P = 7054 (30) = 211620 pounds

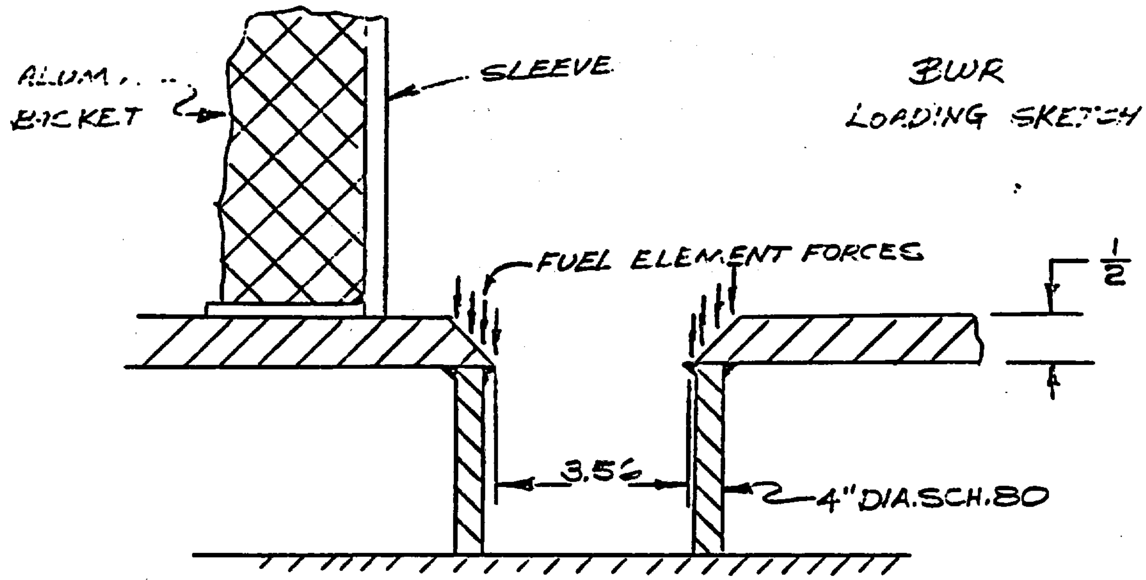
$$\sigma_c = \frac{P}{A} = \frac{211620}{32} = 6613 \text{ psi}$$

S_{aa} = 14,706 psi

$$\text{M.S.} = \frac{14,706}{6613} - 1 = 1.22$$

4.5.6.1 Bottom Support - BWR

The configuration employed in the BWR Support allows the 45° Bevel on the General Electric Fuel to rest on the 45° angle cut in the 1/2" thick Bottom Support Plate.



Refs: NLI Drawing Number 70653F, Sheets 1 and 2

Total Fuel Weight = 18,000 pounds (Ref. Section VII, Pg. 1)

Individual Fuel Weight $= \frac{18,000}{24} =$ 750 pounds

Force at Impact = 30(750) = 22,500 pounds

The placement of the Schedule 80 Pipe allows the vertical load component to be carried by the pipe directly in the Compression Mode.

Area of Pipe = 4.407 in^2

Note: PWR and BWR temps are considered the same for the bottom support. $t = 438^{o}F$ (Sect. 3.1)

Compression in Pipe (Fuel Loads only)

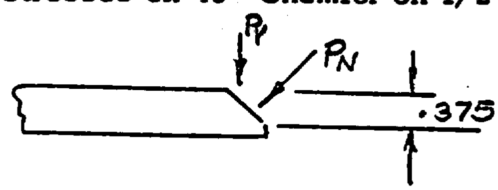
$$\sigma_c = P/A = \frac{22,500}{4.407} = 5105.5 \text{ psi}$$

Allowable stress ($S_{aa} = 0.9\ S_u$) for 304 S/S at 438°F from Sect. 1.1 under cask internals structure and Sect. 1.2 equals to 0.9 x 59500 = 53550 psi

$$M.S. = \frac{53550}{5105.5} - 1 = 9.5$$

The stresses are also below the dynamic yield strength of 39667 psi, hence the elastic stability analysis is then valid.

Compressive Stresses on 45° Chamfer on 1/2 Inch Plate



$$P_N = \frac{P_v}{\sin 45^\circ} = \frac{22,500}{.707} = 31,824 \text{ lbs.}$$

Area in Bearing $= 2\pi R\ (t)$

Where $t = \frac{.375}{.707} = .5304"$

$$A_{Br} = 2\pi \left(\frac{3.56}{2}\right) (.5304) = 5.93 \text{ in}^2$$

$$\sigma_{BR} = \frac{31,824}{5.93} = 5,364 \text{ psi}$$

$S_{aa} = 53550$ psi $\qquad$ $M.S. = \frac{53550}{5364} - 1 = 8.98$

Conservatively, the bearing allowable is used as being equal to the compressive allowable.

Determine ability of 1/2" plate to carry sleeve loads to support points. Each Quadrant of the Lower Support has 9 pipes to support the sleeve loads and the outer 1/4" thick Peripheral Ring.

Total Sleeve Weight = 5852 for all Sleeves

$$\text{Individual Sleeve} = \frac{5852}{36} = 162.5 \text{ pounds}$$

The Sleeve loading pattern is dispersed as to allow the assumption of a uniform load. Ref. 8, page 158-159.

And the distance between supports is taken as an average of all distances encountered on the placement of the four inch pipes in the Lower Spacer.

Average distance between supports

C_1 = 4"

C_2 = 3"

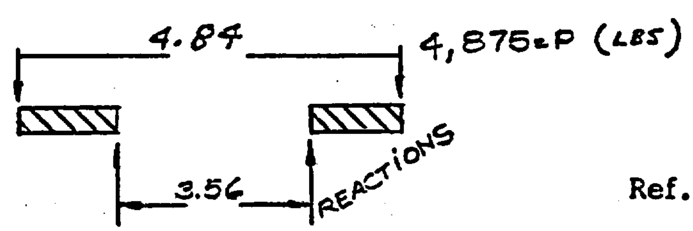
C_3 = 1.75

Ave. C = 2.92"

Make the conservative assumption that the loads (Sleeve) are placed on the outer edge of the Ring.

Sleeve Load at Impact = 162.5 (30) = 4,875 pounds

a = .22 (2.92) = .642"   a = 3.56 + 2 (.642) = 4.84"



Ref. 34, pp. 67-68

$$a/b = \frac{2.42}{1.78} = 1.36 \qquad K = 1.156 \quad \text{(Case 1)}$$

Maximum Stress in 1/2 inch Plate

$$\sigma_{max} = \frac{K P}{t^2} \qquad K = 1.156$$

$$t^2 = (.5)^2 = .25$$

$$\sigma_{(max)} = \frac{1.156\ (4875)}{.25} = 22,540 \text{ psi}$$

S_{aa} = 53550 psi

$$M.S. = \frac{53550}{22,540} - 1 = 1.37$$

Determine Ability of 1/4" Outer Ring to Carry Basket Loads

The aluminum Basket, due to the overall rigidity of the rather thick structure distributes its load equally to the 1/4" outer Ring.

Weight of Basket = 9557.3 (Ref. Weights Section of SAR)

Impact Weight = 30 (9557.3) = 286,719 pounds

Compressive Area of 1/4" thick Ring

$$\text{Area} = \pi (R_o^2 - R_i^2) = \pi (420.25 - 410.06) = 32.00 \text{ in}^2$$

$$\sigma_c = P/A = \frac{286,719}{32.00} = 8.960 \text{ psi}$$

S_{aa} = 53550 psi

$$M.S. = \frac{53550}{8,959} - 1 =. 4.97$$

The elastic stability of the BWR Bottom Support Ring (1/4" thick) is greater than that computed for the PWR Case due to the lower value of L in the BWR case. In either case, the stress required to produce instability far exceeds that of Dynamic Yield Point.

Length for PWR 1/4" Ring = 4.00" (Approx.)

Length for BWR 1/4" Ring = 3.00" (Approx.)

Compressive Pipe stresses due Sleeve Impact

Individual Sleeve Weight = 162.5 pounds

Impact Force = (162.5) (30) = 4,875

Compressive Loading in Pipe

$$\sigma_c = \frac{P}{A} = \frac{4875}{4.407} = 1105.4 \text{ psi}$$

Total Compressive Pipe Loads; (Sleeve plus Fuel)

$$\sigma_c = 1105.4 + 5105.5 = 6{,}210.9 \text{ psi}$$

S_{aa} = 53550 psi

$$\text{M.S.} = \frac{53550}{6210.9} - 1 = 7.62$$

The design allowable buckling stress for an unpressurized thin-walled circular cylinder subjected to axial compression is given by the equation:

Ref. 4 Sect. 10-3, Pg. 229

$$\frac{\sigma_{cr}}{n} = k_c \frac{\pi^2 E}{12\,(1-v)^2} \left(\frac{t}{L}\right)^2$$

For Short Cylinders; The Buckling Coefficient is expressed by

$$K_c = K_o + \frac{12\,r^2\,Z^2}{\pi^4\,K_{co}}$$

Stability of thin walled cylinders,

For simply supported edges, $K_{co} = 1$

Also,

$$Z = \frac{L^2}{R(t)} \sqrt{1 - \nu^2}$$

First, Investigate and Determine the critical buckling stress of the 4" Schedule 80 Pipes and compare this with stresses imposed by Sleeve and Fuel Loads. The temperature at which the Accident takes place is assumed to be the maximum termperature at steady-state Loss of Coolant Operation.

Let the following Parameters be shown

$$L = 3''$$

$$R = \frac{3.3125 + 2.880}{2} = 3.09$$

$$R = \frac{2.25 + 1.913}{2} = 2.0815 \quad \text{(A Mean Value)}$$

$$t = .337$$

$$v = .3$$

$$Z = \frac{(3)^2}{(2.08)(.337)} \left[.91\right]^{\frac{1}{2}} = 12.248$$

Returning to the equations for computation of the Buckling Coefficient,

$$K_c = K_{co} + \frac{12 r^2 Z^2}{\pi^4 K_{co}} \qquad R/t = \frac{2.0815}{.337} = 6.176$$

$$K_{co} = 1.00$$

$$r = .65 \text{ (from Correlation Factors Curve in Ref. 4)}$$

Stability of the Pipe Elements:

$$K_c = 1 + \frac{12\ (.65)^2\ (12.248)^2}{\pi^4\ (1.0)}$$

$$K_c = 1 + \frac{12\ (.4225)\ (150.01)}{(97.409)}$$

$$K_c = 1 + 7.8 = 8.8$$

$$E = 27{,}000{,}000 \text{ psi} \quad \text{at } 438^\circ \text{ F}$$

$$\sigma_{cr} = 12.067 \quad \frac{\pi^2\ (27 \times 10^6)}{12\ (1-v^2)} \quad \left(\frac{t}{L}\right)^2$$

$$\sigma_{cr} = 8.80 \quad \frac{\pi^2\ (27 \times 10^6)}{10.92} \quad .0126$$

$$\sigma_{cr} = 2{,}705{,}790 \text{ psi}$$

The Critical Stress for Elastic Stability greatly exceeds the Dynamic Allowable .

4.5.7 Absorber Sleeves (PWR/BWR)

Dynamic compression loads in the absorber sleeves are calculated for bottom end impact. The loading summary table is shown below.

	PWR	BWR
Top Spacer Weight (lbs.)	142	303
Top Plates Weight (lbs.)	71	69
Individual Sleeve Weight (lbs.)	770	243
Impact Load @ 30 g (lbs.)	29,490	18,450

Area of PWR sleeve $= (9.633)^2 - (8.956)^2$ Ref. Dwg. No. 70652 F

$$A = 12.58 \text{ in.}^2$$

Area of BWR sleeve $= (6.150)^2 - (5.775)^2$ Ref. Dwg. No. 70653 F

$$A = 4.45 \text{ in.}^2$$

$$\text{(BWR)} \quad \sigma_c = \frac{18450}{4.45} = 4146 \text{ psi}$$

$$\text{(PWR)} \quad \sigma_c = \frac{29490}{12.58} = 2344 \text{ psi}$$

Assume the stresses calculated are carried primarily by the neutron absorber material. The ultimate tensile strength of the absorber material is 42800 psi (Ref. 19).

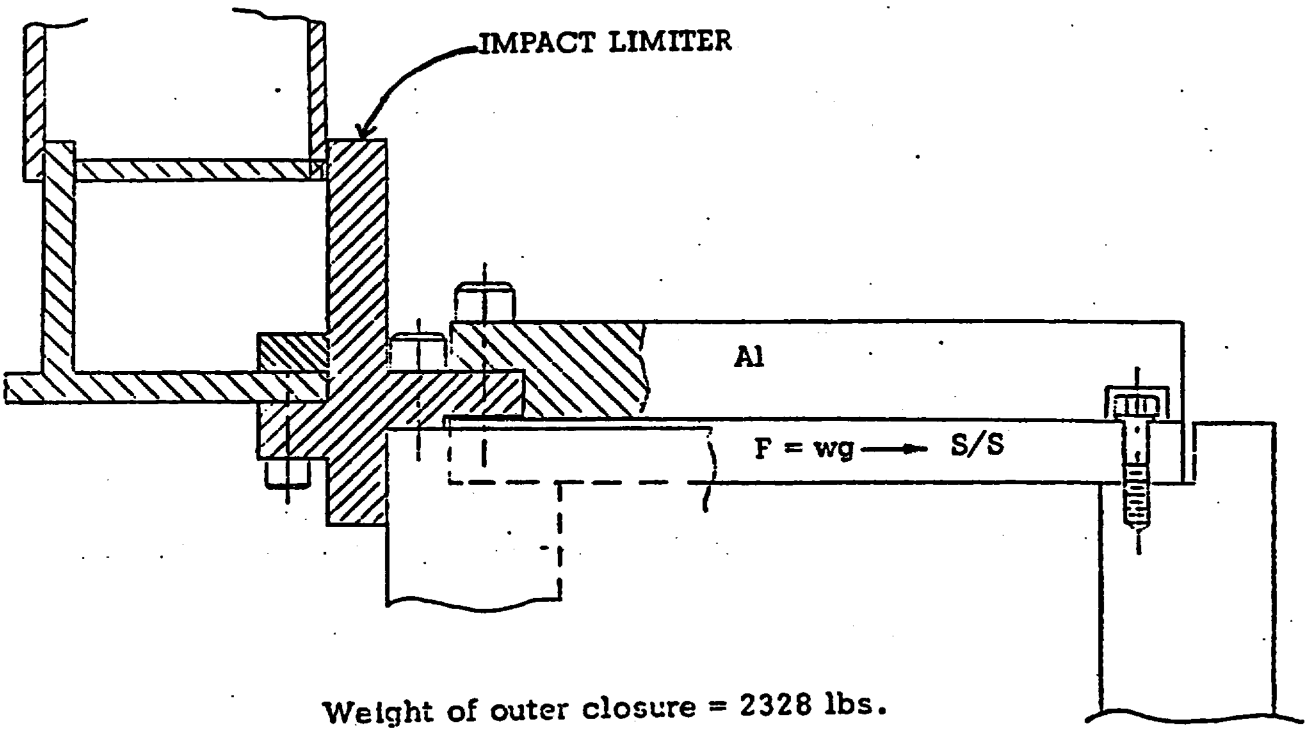
Allowable stress ($S_{aa} = 0.9\, S_u$) from Sect. 1.1 under cask internal structure and Sect. 1.2 equals to $0.9 \times 42800 = 38520$ psi.

$$\text{Minimum M.S.} = \frac{38520}{4146} - 1 = 8.29$$

4.6 Side Impact

4.6.1. Outer Closure Head

During the side impact the outer closure head is subjected to an acceleration of 81 g in the plane of the closure. Integrity of outer closure must be maintained to protect containment vessel valves.



Weight of outer closure = 2328 lbs.

Lateral force on closure F = 81 x 2328 = 188568 lbs.

Analysis of bolts in shear:

Outer closure is bolted down with 28 1¼ in. dia. bolts.

Yield strength of bolts at 250 °F is 85,000 psi (Sect. 1.2)

Ultimate tensile strength at 250 °F is 130,000 psi (Sect. 1.2)

Shear area of 1¼-8 bolts at the shear plane is 0.9406 $in.^2$

Shear Stress $S_s = \frac{F}{28A} = \frac{188,568}{28 \times 0.9408} = 7,158$ psi

Tensile stress in the bolts from preload is 12158 psi. (Sect. 3.11)

Effective Stress

$$S_{e2} = \sqrt{\tfrac{1}{2}} \sqrt{(0-0)^2 + (0-12,158)^2 + (12,158-0)^2 + 6(7,158)^2}$$

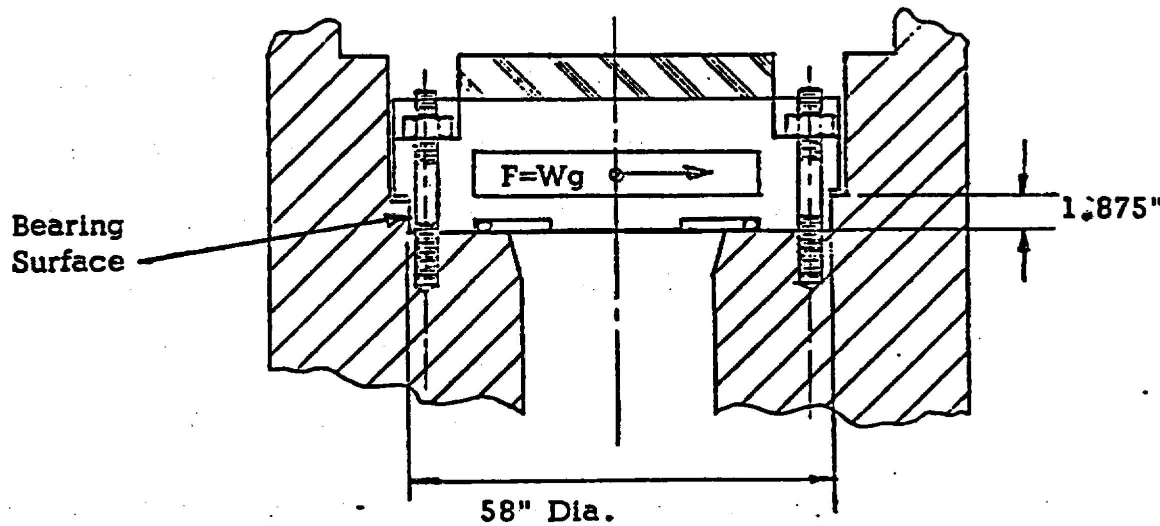
S_{e2} = 17,364 psi.

During the impact accident condition, the Allowable Stress for the bolts is conservatively set at the static yield strength of 85,000 psi. This is well below the accident allowable stress of 0.9 S_u (Sect. 1.1) and assures an adequate margin of strength in the bolts during the subsequent puncture accident condition (Sect. 4.8.1).

$$M.S. = \frac{85000}{17364} - 1 = 3.90$$

4.6.1.1 Inner Closure

During the side impact the inner closure is subjected to an acceleration of 81 g in the plane of the closure. Integrity of inner closure must be maintained to provide the desired containment of the cask contents.



Weight of Inner Closure = 7400 lbs.

Lateral force on closure, F = (81) (7400)

F = 599,400

This force will not be taken by the bolts in shear since the radial clearance between the closure head and the cask forging is less than the radial clearance between the stud and the bolt hole in the closure.

Calculating bearing stress on the 304 S.S. forging

$$S_b = \frac{P}{A} = \frac{599,400}{108.75} = 5,512 \text{ psi} \quad A = 1.875 \times 58 = 108.75 \text{ in}^2$$

Bearing stress allowable ($S_{brd} = 1.35\,S_{yd}$) for 304 S/S at 325° from Sect. 1.1 and Sect. 1.2 equals 1.35 (41200)= 55620 psi

$$M.S. = \frac{55620}{5512} - 1 = 9.09$$

Lateral movement of the inner closure before contact with the cask top forging is limited to the diametral clearance of 0.010 in. (See Dwg. 70651F, Sheet 4). The sealing surfaces of closure, forging and silver-plated O-ring seal are wide enough, radially, so that lateral displacements of 0.010 inch will not cause loss of seal integrity according to the seal manufacturer (Ref. 84). Hence, the pressure seal will be maintained for the small lateral movements of the closure that are possible.

4.6.2 Bending of Containment Shells

During the side impact the cask is subjected to a 81 G acceleration which produces large bending moments in the cask containment structure. These moments are resisted by the inner and outer containment shells and by the water jacket shell. Conservatively, additional support from the lead and the end forgings for this analysis is ignored. Stresses on the shells are within the allowables based on ultimate strengths. Hence, the containment feature of the cask design is maintained and retention of the contents is assured.

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The cask containment structure is treated as a simply supported beam loaded by a uniform transverse load due to its own weight. The moments are shared by the three shells in proportion to their stiffness because they are all constrained to have equal end slopes by the very stiff end forgings.

The analytical model for this analysis is shown in detail in Section 3.8.6. The $-40^{o}F$ isothermal condition with 81g side load results in the primary effective stress Se_3 at various locations in the shells as follows:

Loc.	Se_3 (psi) (1g)	(81g)	$S_{AA} = .7S_u$	M.S.
1	204	16524	80500 psi	3.87
3	278	22518	80500 psi	2.57
7	237	19197	80500 psi	3.19
9	372	30132	80500 psi	1.67
11	377	30537	80500 psi	1.64
13	512	41472	80500 psi	.94
17	174	14094	80500 psi	4.71
23	237	19197	80500 psi	3.19
25	200	16200	80500 psi	3.97
29	106	8586	80500 psi	8.37

ACCIDENT PRIMARY STATIC AND DYNAMIC STRESS EVALUATION

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
9	2.35000	100 PSI WATER CHAMBER
10	.72900	100 PSI CAVITY
11	-81.00000	1.0 G SIDE DROP (compression side)

LOCATION	EFFECTIVE STRESS Se_3	TEMP.	STRESS ALLOWABLES 0.7SU
1	16213	268	45465
3	22100	248	45990
5	19893	218	46777
7	20862	315	44205
9	29313	303	44546
11	29448	420	41895
13	40435	359	43076
15	45180	323	44021
17	12522	302	44572
19	13420	290	44887
21	1083	292	44835
23	21481	271	45386
25	15786	241	46174
27	13456	218	46777
29	8225	260	45675
31	157	256	45780
33	307	240	46200

Normal Transport Conditions
70 kw decay heat load
130°F ambient temperature
~~Internal pressure (all clad failed)~~

ACCIDENT PRIMARY STATIC AND DYNAMIC STRESS EVALUATION

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
9	2.35000	100 PSI WATER CHAMBER
10	.72900	100 PSI CAVITY
11	81.00000	1.0 G SIDE DROP (Tension side)

LOCATION	EFFECTIVE STRESS Se_3	TEMP.	STRESS ALLOWABLES 0.7SU
1	16874	268	45465
3	22944	248	45990
5	24514	218	46777
7	23425	315	44205
9	31154	303	44546
11	31709	420	41895
13	42536	359	43076
15	39772	323	44021
17	15726	302	44572
19	14804	290	44887
21	1083	292	44835
23	22862	271	45386
25	16665	241	46174
27	18463	218	46777
29	8952	260	45675
31	157	256	45780
33	307	240	46200

Normal Transport Condition
70 kw decay heat load
130^{O}F ambient temperature
Internal pressure (all clad failed)

ACCIDENT PRIMARY STATIC AND DYNAMIC STRESS EVALUATION

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
9	2.35000	100 PSI WATER CHAMBER
10	.16500	100 PSI CAVITY
11	-81.00000	1.0 G SIDE DROP (compression side)

LOCATION	EFFECTIVE STRESS Se_3	TEMP.	STRESS ALLOWABLES 0.7SU
1	16521	268	45465
3	22113	248	45990
5	19905	218	46777
7	20929	316	44205
9	29323	303	44546
11	29427	420	41895
13	40433	359	43076
15	45228	323	44021
17	12972	302	44572
19	13672	290	44887
21	1766	292	44835
23	21429	271	45386
25	15837	241	46174
27	13517	218	46777
29	8795	260	45675
31	412	256	45780
33	502	240	46200

Normal Transport Conditions
70 kw decay heat load
130 F ambient temperature
Internal pressure (no clad failure)

ACCIDENT PRIMARY STATIC AND DYNAMIC STRESS EVALUATION

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
9	2.35000	100 PSI WATER CHAMBER
10	.16500	100 PSI CAVITY
11	81.00000	1.0 G SIDE DROP (Tension side)

LOCATION	EFFECTIVE STRESS Se_3	TEMP.	STRESS ALLOWABLES 0.7SU
1	16613	268	45465
3	22936	248	45990
5	24500	218	46777
7	23422	316	44205
9	31169	303	44546
11	31799	420	41895
13	42558	359	43076
15	39725	323	44021
17	15414	302	44572
19	14687	290	44887
21	1766	292	44835
23	22954	271	45386
25	16613	241	46174
27	18403	218	46777
29	8377	260	45675
31	412	256	45780
33	502	240	45200

Normal Transport Conditions
70 kw decay heat load
130 °F ambient temperature
Internal pressure (no clad failure)

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS EVAL

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
2	1.00000	-40.0 F ISOTHERMAL
11	81.00000	1.0 G SIDE DROP (Tension side)

LOCATION	EFFECTIVE STRESS Se_4	TEMP.	ALLOWABLES 0.9SL
1	49689	-40	103500
2	8780	-40	103500
3	12859	-40	103500
4	35379	-40	103500
5	21269	-40	103500
6	21459	-40	103500
7	34485	-40	103500
8	33090	-40	103500
9	41048	-40	103500
10	38092	-40	103500
11	39824	-40	103500
12	37978	-40	103500
13	49568	-40	103500
14	47652	-40	103500
15	39630	-40	103500
16	39621	-40	103500
17	15343	-40	103500
18	24200	-40	103500
19	20490	-40	103500
20	21403	-40	103500
21	14566	-40	103500
22	12083	-40	103500
23	28461	-40	103500
24	21666	-40	103500
25	9775	-40	103500
26	37733	-40	103500
27	12980	-40	103500
28	16274	-40	103500
29	12585	-40	103500
30	18793	-40	103500
31	9275	-40	103500
32	11911	-40	103500
33	4699	-40	103500
34	16504	-40	103500

Normal Transport Conditions
No decay heat load
-40°F ambient
No Internal pressure

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS EVAL.

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
2	1.00000	-40.0 F ISOTHERMAL
11	-81.00000	1.0 G SIDE DROP (compression side)

LOCATION	EFFECTIVE STRESS Se_4	TEMP.	ALLOWABLES 0.9SU
1	19300	-40	103500
2	34620	-40	103500
3	39589	-40	103500
4	9803	-40	103500
5	23126	-40	103500
6	22934	-40	103500
7	55012	-40	103500
8	51427	-40	103500
9	32048	-40	103500
10	32772	-40	103500
11	54109	-40	1035[illegible]
12	52729	-40	103[illegible]
13	44301	-40	103500
14	43199	-40	103500
15	42342	-40	103500
16	42351	-40	103500
17	27976	-40	103500
18	20829	-40	103500
19	19220	-40	103500
20	18793	-40	103500
21	14566	-40	103500
22	12083	-40	103500
23	16527	-40	103500
24	22897	-40	103500
25	26688	-40	103500
26	5991	-40	103500
27	18934	-40	103500
28	15707	-40	103500
29	4765	-40	103500
30	3256	-40	103500
31	9275	-40	103500
32	11911	-40	103500
33	4699	-40	103500
34	16504	-40	103500

Normal Transport Conditions
No decay heat load
-40°F ambient
No Internal Pressure

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS EVAL.

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
4	1.00000	NORMAL 70KW 130F AMBIENT
11	81.00000	1.0 G SIDE DROP (Tension side)

LOCATION	EFFECTIVE STRESS Se_4	TEMP.	ALLOWABLES 0.9SU
1	41611	268	58455
2	29253	267	58489
3	29603	248	59130
4	19350	237	59501
5	54370	218	60142
6	7187	218	60142
7	15363	316	56835
8	15377	314	56902
9	27241	303	57274
10	35469	291	57679
11	30102	420	53865
12	28662	420	53865
13	39984	359	55384
14	46938	344	55890
15	45474	323	56599
16	45415	323	56599
17	10935	302	57307
18	10129	301	57341
19	3955	290	57712
20	24696	290	57712
21	8665	292	57645
22	9650	290	57712
23	20714	271	58354
24	24198	259	58759
25	36046	241	59366
26	8016	230	59737
27	54535	218	60142
28	12009	218	60142
29	9393	260	58725
30	3424	260	58725
31	20610	256	58860
32	13052	256	58860
33	17074	240	59400
34	24486	240	59400

Normal Transport Condition
70 kw decay heat load
130°F ambient
Internal Pressure (no clad failure)

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS EVAL.

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
4	1.00000	NORMAL 70KW 130F AMBIENT
11	-81.00000	1.0 G SIDE DROP (compression side)

LOCATION	EFFECTIVE STRESS Se_4	TEMP.	ALLOWABLES 0.9SU
1	9591	268	58455
2	53314	267	58489
3	16645	248	59130
4	25840	237	59501
5	13907	218	60142
6	41383	218	60142
7	32722	316	56835
8	31447	314	56902
9	33358	303	57274
10	26120	291	57679
11	46738	420	53865
12	44478	420	53865
13	43118	359	55384
14	38961	344	55[illegible]
15	39526	323	56599
16	39493	323	56599
17	17574	302	57307
18	35907	301	57341
19	31269	290	57712
20	6995	290	57712
21	8665	292	57645
22	9650	290	57712
23	20514	271	58354
24	14203	259	58759
25	5516	241	59366
26	29017	230	59737
27	23652	218	60142
28	40126	218	60142
29	7826	260	58725
30	14050	260	58725
31	20610	256	58860
32	13052	256	58860
33	17074	240	59400
34	24486	240	59400

Normal Transport Conditions
70 kw decay heat load
130°F ambient
Internal pressure (no clad failure)

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS EVAL

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
11	81.00000	1.0 G SIDE DROP (Tension side)
12	1.00000	NORMAL 40KW -40F AMBIENT

LOCATION	EFFECTIVE STRESS Se_4	TEMP.	ALLOWABLES 0.9SU
1	53049	54	74659
2	21225	53	74966
3	19024	42	78341
4	27025	33	81102
5	34265	20	85091
6	17680	20	85091
7	19956	81	68988
8	20476	80	69058
9	35418	72	69612
10	35733	65	71264
11	22851	137	65112
12	23515	135	65250
13	43736	104	67396
14	43874	95	68019
15	43992	81	68988
16	44038	81	68988
17	19215	73	69542
18	19634	72	69612
19	10544	68	70364
20	31624	68	70364
21	19155	67	70670
22	14501	66	70977
23	23313	54	74659
24	21373	46	77114
25	22240	38	79568
26	8288	29	82330
27	26931	20	85091
28	10163	20	85091
29	11146	52	75273
30	9095	52	75273
31	13319	50	75886
32	14009	50	75886
33	12318	38	79568
34	5396	38	79568

Normal Transport Conditions
40 kw decay heat load
-40°F ambient
Internal pressure (no clad failure)

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS F[illegible]L

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
11	-81.00000	1.0 G SIDE DROP (compression side)
12	1.00000	NORMAL 40KW -40F AMBIENT

LOCATION	EFFECTIVE STRESS Se_4	TEMP.	ALLOWABLES 0.9SU
1	20397	54	74659
2	45717	53	74966
3	30155	42	78341
4	18011	33	81102
5	10403	20	85091
6	26837	20	85091
7	44614	81	68988
8	41997	80	69058
9	30539	72	69612
10	26400	65	71284
11	46430	137	65112
12	45833	135	65250
13	41607	104	67[illegible]5
14	41280	95	68[illegible]9
15	37991	81	68988
16	37945	81	68988
17	20888	73	69542
18	47815	72	69612
19	36242	68	70364
20	17971	68	70364
21	19155	67	70670
22	14501	66	70977
23	21188	54	74659
24	18868	46	77114
25	10996	38	79568
26	24705	29	82330
27	6004	20	85091
28	22063	20	85091
29	6262	52	75273
30	8142	52	75273
31	13319	50	75886
32	14009	50	75886
33	12318	38	79568
34	5396	38	79568

Normal Transport Conditions
40 kw decay heat load
-40°F ambient
Internal pressure (no clad failure)

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS EVAL.

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
11	81.00000	1.0 G SIDE DROP (Tension side)
13	1.00000	NORMAL 70KW -40F AMBIENT

LOCATION	EFFECTIVE STRESS . Se_4	TEMP.	ALLOWABLES 0.9SU
1	51498	103	67465
2	27707	102	67535
3	22891	81	68988
4	22417	69	70057
5	34040	50	75886
6	17256	50	75886
7	11198	148	64350
8	14525	147	64419
9	31525	133	65388
10	36294	119	66358
11	14043	242	59332
12	15361	239	59434
13	41062	186	61719
14	41029	171	62758
15	46128	149	64281
16	46219	149	64281
17	19141	135	65250
18	20563	134	65319
19	6128	123	66081
20	26639	123	66081
21	12327	125	65942
22	9240	124	66012
23	19298	103	67465
24	23478	90	68365
25	33443	74	69473
26	5848	63	71898
27	34301	50	75886
28	7649	50	75886
29	10175	102	67535
30	6992	102	67535
31	16458	98	67812
32	15812	98	67812
33	15558	83	68850
34	15929	83	68850

Normal Transport Conditions
70 kw decay heat load
$-40^{\circ}F$ ambient
Internal pressure (no clad failure)

ACCIDENT PRIMARY PLUS SECONDARY STATIC AND DYNAMIC STRESS EVAL

BASE CASE	MULTIPLIER	BASE CASE DESCRIPTION
11	-81.00000	1.0 G SIDE DROP (compression side)
13	1.00000	NORMAL 70KW -40F AMBIENT

LOCATION	EFFECTIVE STRESS Se_4	TEMP.	ALLOWABLES 0.9SU
1	18487	103	67465
2	→ 52640	102	67535
3	23880	81	68988
4	22648	69	70057
5	11689	50	75886
6	27137	50	75886
7	41767	148	64350
8	37746	147	64419
9	31445	133	65388
10	23991	119	66358
11	47034	242	59332
12	45911	239	59434
13	42219	185	61719
14	42263	171	62758
15	35888	149	6[illegible]1
16	35795	149	64281
17	14623	135	65250
18	48228	134	65319
19	32412	123	66081
20	13932	123	66081
21	12327	125	65942
22	9240	124	66012
23	24051	103	67465
24	14936	90	68365
25	2821	74	69473
26	33381	63	71895
27	5102	50	75886
28	25490	50	75886
29	7105	102	67535
30	10181	102	67535
31	16458	98	67812
32	15812	98	67812
33	15558	83	68859
34	15929	83	68859

Normal Transport Conditions

70 kw decay heat load

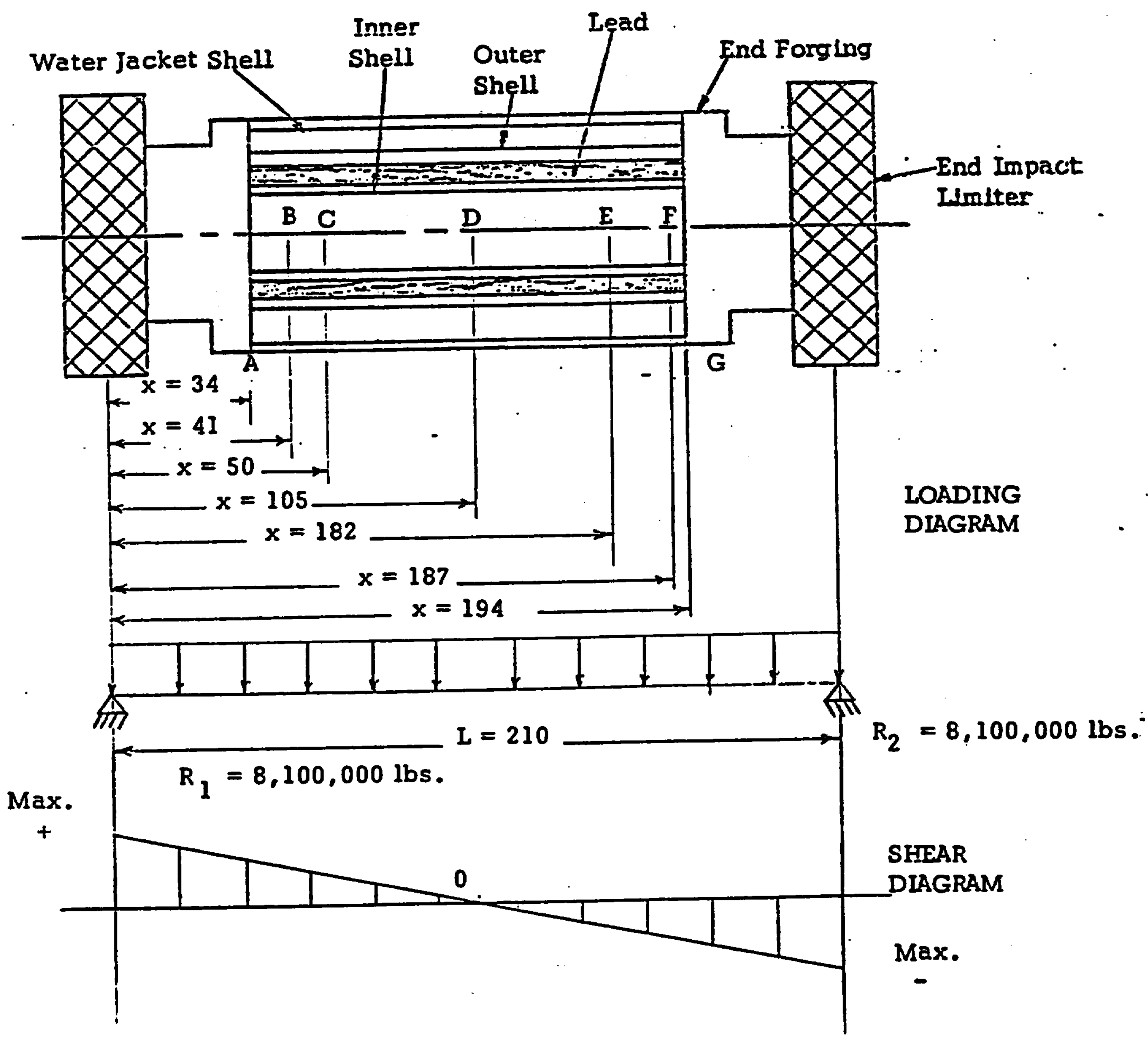
-40°F ambient

Internal pressure (no clad failure)

Shear Stress Calculations

Calculate maximum shear stress in the shells at points "A", "F", "G".

The maximum shear stress occurs 90° circumferentially from the maximum bending stress.



Calculate shear stresses in the inner shell, outer shell and water jacket shell at point "A" on a horizontal section through cask axis.

From Ref. 3 $S_s = \frac{VQ}{I_t b}$ where Q=A'z'

$V = 8,100,000 - 8,100,000 \left(\frac{34}{105}\right) = 5,477,143$ lbs.

$A_i' = \frac{107.8}{2} = 53.9$ in.^{2} (A' is shell area above centerline)

$A_o' = \frac{380.1}{2} = 190.05$ in.^{2}

$A_j' = \frac{191.4}{2} = 95.7$ in.^{2}

$z_i' = R - (R)\ (1 - \sin\alpha/\alpha) = R - R\ (1 - 2/\pi)$ (Ref. 3, Table I Case 12) (z' is distance from neutral axis to centroid of A')

$= 23.25 - 23.25\ (.363)$

$= 14.81$ in.

$z_o' = 31.25 - 31.25\ (.363)$

$= 19.9$ in.

$z_j' = 41 - 41\ (.363)$

$= 26.117$ in.

$I_i' = (0.049)\ (46.5^4 - 45^4)$

$= 28160$ in.^{4}

$I_o = (0.049)(62.5^4 - 58.5^4)$

$= 173803 \text{ in}^4$

$I_j = (0.049)(82^4 - 80.5^4)$

$= 157708 \text{ in}^4$

$I_t = 359671 \text{ in}^4$

$b = 2(.75 + 2 + .75)$

$= 7 \text{ in.}$

$Q = (14.81)(53.9) + (19.9)(190.05) + (26.117)(95.7)$

$= 7079.6 \text{ in}^3$

$$S_s = \frac{5477143\ (7079.6)}{359671\ (7)} = 15401 \text{ psi}$$

Calculate effective stress Se_3 in the shells at points "A", "F", "G" at Normal Transport Condition 70 kw, 130°F Abmient, 16.45 psig cavity pressure and 235 psig in the water jacket. This case will give the highest effective stress Se_3.

Effective stress in the inner shell Se_3 at point "A"

σ_x = 12.1 psi (Radial Stress) From Sect. 3.8

σ_y = -1315 psi (Tangential Stress) From Sect. 3.8

σ_z = -606 psi (Axial Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xz} = 176 psi (Shear stress) From Sect. 3.8

T_{xy} = 0

$$Se_3 = \sqrt{\tfrac{1}{2}}\sqrt{(12.1-(-1315))^2 + (-1315-(-606))^2 + (-606-12.1)^2 + 6\,(15401^2 + 176^2)}$$

= 26702 psi

Allowable stress ($0.8\ S_{aa} = 0.7\ S_u$) at 268 F for 304 s.s From Sect. 1.1 under containment vessel and Sect. 1.2 equals 45465 psi.

$$MS = \frac{45465}{26702} - 1 = .703$$

Effective stress in the outer shell Se_3 at point "A"

σx = -254 psi (Radial Stress) From Sect. 3.8

σy = -676 psi (Tangential Stress) From Sect. 3.8

σz = -52 psi (Axial Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xz} = 223 psi (Shear Stress) From Sect. 3.8

T_{xy} = 0

$$Se_3 = \sqrt{\tfrac{1}{2}}\sqrt{(-254-(-676))^2 + (-676-(-52))^2 + (-52-(-254))^2 + 6\,(15401^2 + 223^2)}$$

= 26684 psi

Allowable stress ($0.8\ S_{aa} = 0.7\ S_u$) at 248 $^{\circ}F$ for 304 s.s From Sect. 1.1 under containment vessel and Sect. 1.2 equals 45990 psi.

$$MS = \frac{45990}{26684} - 1 = .72$$

Calculate effective stress Se_3 in the outer shell at point "F"

$$S_s = \frac{VQ}{I_t\ B}$$

$$V = 8,100,000 - 8,100,000 \left(\frac{23}{105}\right)$$

$$= 6325714 \text{ lbs.}$$

$$S_s = \frac{6325712\ (7079.6)}{359671\ (7)} = 17787 \text{ psi.}$$

σ_x = -115 psi (Radial Stress) From Sect. 3.8

σ_y = -1012 psi (Tangential Stress) From Sect. 3.8

σ_z = -175 psi (Axial Stress) From Sect. 3.8

T_{yz} = 17787 psi (Shear Stress)

T_{xz} = -259 psi (Shear Stress) From Sect. 3.8

$$Se_3 = \sqrt{\tfrac{1}{2}}\sqrt{(-115-(-1012))^2 + (-1012-(-175))^2 + (-175-(-115))^2 + 6\,(17787^2 + (-259)^2)}$$

Se_3 = 30823 psi

Allowable stress ($0.8\ S_{aa} = 0.7\ S_u$) at 241°F for 304 s.s From Sect. 1.1 under containment vessel and Sect. 1.2 equals 46174 psi.

$$MS = \frac{46174}{30823} - 1 = .498$$

Calculate effective stress Se_3 in the inner shell at point "G"

$$S_s = \frac{VQ}{I_t\ B}$$

$$V = 8100000 - 8100000 \left(\frac{16}{105}\right)$$

$$= 6865714 \text{ lbs.}$$

$$S_s = \frac{6865714\ (7079.6)}{359671\ (7)} = 19306 \text{ psi}$$

σ_x = -8 psi (Radial Stress) From Sect. 3.8

σ_y = -46 psi (Tangential Stress) From Sect. 3.8

σ_z = -237 psi (Axial Stress) From Sect. 3.8

T_{yz} = 19306 psi (Shear Stress)

T_{xz} = T_{xy} = 0

$$Se_3 = \sqrt{\tfrac{1}{2}}\sqrt{(-8-(-46))^2 + (-46-(0237))^2 + (-237-(-8))^2 + 6\ (19306^2)}$$

$$= 33440 \text{ psi}$$

Allowable stress $0.7\ S_{aa} = 0.9\ S_u$ at 260°F for 304 s.s From Sect. 1.1 under containment vessel and Sect. 1.2 equals 45675 psi.

$$MS = \frac{45675}{33440} - 1 = .366$$

Calculate effective stresses Se_4 in the shells at points "A", "F", "G" at -40°F temperature condition. No pressure in cavity or in water jacket.

Calculate effective stress Se_4 in the inner shell, outer shell and water jacket shell at point "A".

Calculate effective stress Se_4 in the inner surface of the inner shell at point "A".

σ_x = 0 psi (Radial Stress) From Sect. 3.8

σ_y = 13740 psi (Tangential Stress) From Sect. 3.8

σ_z = 38589 psi (Axial Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xy} = T_{zx} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(0-13740)^2 + (13740 - 38589)^2 + (38589-0)^2 + 6\,(15401)^2}$$

= 43119 psi

Allowable stress ($S_{aa} = 0.9\ S_u$) at -40°F for 304 s.s. From Sect. 1.1 under all cask structures and Sect. 1.2 equals 0.9 x 115000 = 103500 psi.

$$MS = \frac{103500}{43119} - 1 = 1.40$$

Calculate effective stress Se_4 in the inner surface of the outer shell at point "A".

σ_x = 2505 psi (Radial Stress) From Sect. 3.8

σ_y = -9884 psi (Tangential Stress) From Sect. 3.8

σ_z = -19246 psi (Axial Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress) From Sect. 3.8

T_{xz} = 769 psi (Shear Stress) From Sect. 3.8

T_{yz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(2502 - (-9884))^2 + (-9884-(-19246))^2 + (-19246-2505)^2 + 6\,(15401^2 + 769^2)}$$

= 32717 psi

$$MS = \frac{103500}{32717} - 1 = 2.16$$

Calculate effective stress Se_4 in the inner surface of the water jacket shell at point "A"

σ_x = 0 psi (Radial Stress) From Sect. 3.8

σ_y = -443 psi (Tangential Stress) From Sect. 3.8

σ_z = -1150 psi (Axial Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xy} = T_{zx} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(0-(-443))^2 + (-443-(-1150))^2 + (-1150-0)^2 + 6(15401^2)}$$

$$= 26694 \text{ psi}$$

$$MS = \frac{103500}{26694} - 1 = 2.87$$

Calculate effective stress Se_4 in the outer surface of the outer shell at point "F".

σ_x = 0 psi (Radial Stress) From Sect. 3.8

σ_y = 3398 psi (Tangential Stress) From Sect. 3.8

σ_z = 23117 psi (Axial Stress) From Sect. 3.8

T_{yz} = 17787 psi (Shear Stress)

T_{xy} = T_{zx} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(0-3398)^2 + (3398 - 23117)^2 + (23117-0)^2 + 6(1778)^2}$$

$$= 37637 \text{ psi}$$

$$MS = \frac{103500}{37637} - 1 = 1.75$$

Calculate effective stress Se_4 in the inner surface of the water jacket shell at point "F".

σ_x = 0 psi (Radial Stress) From Sect. 3.8

σ_y = 48 psi (Tangential Stress) From Sect. 3.8

σ_z = -2953 psi (Axial Stress) From Sect. 3.8

T_{yz} = 17787 psi (Shear Stress)

$T_{xy} = T_{zx} = 0$

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(0-48)^2 + (48-(-2953))^2 + (-2953-0)^2 + 6(17787^2)}$$

$= 30947$ psi

$$MS = \frac{103500}{30947} - 1 = 2.34$$

Calculate effective stress Se_4 in the outer surface of the inner shell at point "G"

σ_x = 0 psi (Radial Stress) From Sect. 3.8

σ_y = 3400 psi (Tangential Stress) From Sect. 3.8

σ_z = 11675 psi (Axial Stress) From Sect. 3.8

T_{yz} = 19306 psi (Shear Stress)

$T_{xy} = T_{zx} = 0$

$$Se = \sqrt{\tfrac{1}{2}}\sqrt{(0-3400)^2 + (3400-11675)^2 + (11675-0)^2 + 6(19306)^2}$$

$= 35019$ psi

$$MS = \frac{103500}{35019} - 1 = 1.95$$

Calculate effective stress Se_4 in the shells at points "A", "F", "G" at normal temperature condition, with a cavity pressure of 16.45 psig. and a water jacket pressure of 235 psig.

Calculate effective stress Se_4 in the outer surface of the inner shell at point "A"

σ_x = 322 psi (Radial Stress) From Sect. 3.8

σ_z = -44706 psi (Axial Stress) From Sect. 3.8

σ_y = -29619 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xz} = -262 psi (Shear Stress) From Sect. 3.8

T_{xy} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(322-(-44706))^2 + (-44706-(-29619))^2 + (-29619-322^2) + 6(15401^2 + 262^2)}$$

= 47829 psi

Allowable stress ($S_{aa} = 0.9\ S_u$) at 267°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 58489 psi.

$$MS = \frac{58489}{47829} - 1 = .223$$

Calculate effective stress Se_4 in the inner surface of the outer shell at point "A"

σ_x = 127 psi (Radial Stress) From Sect. 3.8

σ_z = 3881 psi (Axial Stress) From Sect. 3.8

σ_y = -5672 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xz} = 234 psi (Shear Stress) From Sect. 3.8

T_{yx} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(127-3881)^2 + (3881-(-5672))^2 + (-5672-127)^2 + 6\,(15401^2 + 234^2)}$$

= 27950 psi

Allowable stress (S_{aa} = 0.9 S_u) at 248°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 59130 psi.

$$MS = \frac{59130}{27950} - 1 = 1.11$$

Calculate effective stress Se_4 in the inner surface of the water jacket at point "A"

σ_x = -235 psi (Radial Stress) From Sect. 3.8

σ_z = 36467 psi (Axial Stress) From Sect. 3.8

σ_y = 10384 psi (Tangential Stress) From Sect. 3.9

T_{yz} = 15401 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-235-36467)^2 + (36467-10384)^2 + (10384-(-235))^2 + 6(15401^2)}$$

Se_4 = 42209 psi

Allowable stress ($S_{aa} = 0.9\ S_u$) at 218°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 60142 psi.

$$MS = \frac{60142}{42209} - 1 = .424$$

Calculate effective stress Se_4 in the inner surface of the outer shell at point "F".

σ_x = 0.0 psi (Radial Stress) From Sect. 3.8

σ_z = 17066 psi (Axial Stress) From Sect. 3.8

σ_y = -5032 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 17787 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(0-17066)^2 + (17066-(-5032))^2 + (-5032-0)^2 + 6(17787^2)}$$

= 36763 psi

Allowable stress ($S_{aa} = 0.9\ S_u$) at 241°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 59366 psi.

$$MS = \frac{59366}{36763} - 1 = .614$$

Calculate effective stress Se_4 in the inner surface of the water jacket shell at point "F"

σ_x = -235 psi (Radial Stress) From Sect. 3.8

σ_z = 42790 psi (Axial Stress) From Sect. 3.8

σ_y = 10154 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 17787 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-235-42790)^2 + (42790-10154)^2 + (10154-(-235))^2 + 6(17787^2)}$$

$$= 49611 \text{ psi}$$

Allowable stress ($S_{aa} = 0.9\ S_u$) at 218°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 60142 psi.

$$MS = \frac{60142}{49611} - 1 = .212$$

Calculate effective stress Se_4 in the outer surface of the inner shell at point "G"

σ_x = 0.0 psi (Radial Stress) From Sect. 3.8

σ_z = -6140 psi (Axial Stress) From Sect. 3.8

σ_y = -1467 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 19306 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(0-(-6140))^2 + (-6140-(-1467))^2 + (-1467-0)^2 + 6(19306^2)}$$

= 33897 psi

Allowable stress ($S_{aa} = 0.9\ S_u$) at 260°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 58725 psi.

$$MS = \frac{58725}{33897} - 1 = .732$$

Calculate effective stress Se_4 in the shells at points "A", "F", "G" at 70 kw, -40 F ambient, with a cavity pressure of 16.45 psig and a water jacket pressure of 33 psig.

Calculate effective stress Se_4 in the outer surface of the inner shell at point "A"

σ_x = 1200 psi (Radial Stress) From Sect. 3.8

σ_z = -42980 psi (Axial Stress) From Sect. 3.8

σ_y = -26541 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xz} = -382 psi (Shear Stress) From Sect. 3.8

T_{xy} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(1200-(-42980))^2 + (-42980-(-26541))^2 + (15401-1200)^2 + 6(15401^2 + 382^2)}$$

Se_4 = 43862 psi

Allowable stress ($S_{aa} = 0.9\ S_u$) at 103°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 67465 psi.

$$MS = \frac{67465}{43868} - 1 = .540$$

Calculate effective stress Se_4 in the inner surface of the outer shell at point "A"

σ_x = -189 psi (Radial Stress) From Sect. 3.8

σ_z = -4345 psi (Axial Stress) From Sect. 3.8

σ_y = -7474 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xz} = 20 psi (Shear Stress) From Sect. 3.8

T_{yz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-189-(-4345))^2 + (-4345-(-7474))^2 + (-7474-(-189))^2 + 6\,(15401^2 + 20^2)}$$

$$= 27416 \text{ psi}$$

Allowable stress ($S_{aa} = 0.9\ S_u$) at 81°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 68988 psi.

$$MS = \frac{68988}{27416} - 1 = 1.52$$

Calculate effective stress Se_4 in the inner surface of the water jacket shell at point "A"

σ_x = -33 psi (Radial Stress) From Sect. 3.8

σ_z = 14221 psi (Axial Stress) From Sect. 3.8

σ_y = 5449 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-33-14221)^2 + (14221-5449)^2 + (5449-(-33))^2 + 6(15401^2)}$$

$$= 29439 \text{ psi}$$

Allowable stress ($S_{aa} = 0.9\ S_u$) at 50°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 75886 psi.

$$MS = \frac{75886}{29439} - 1 = 1.57$$

Calculate effective stress Se_4 in the outer surface of the outer shell at point "F"

σ_x = -33 psi (Radial Stress) From Sect. 3.8

σ_z = -20066 psi (Axial Stress) From Sect. 3.8

σ_y = -6764 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 17787 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-33-(-20066))^2 + (-20066-(-6764))^2 + (-6764-(-33))^2 + 6(17787^2)}$$

$$= 35510 \text{ psi}$$

Allowable stress (S_{aa} = 0.9 S_u) at 63°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 71898 psi.

$$MS = \frac{71898}{35510} - 1 = 1.02$$

Calculate effective stress Se_4 in the inner surface of the water jacket shell at point "F"

σ_x = -33 psi (Radial Stress) From Sect. 3.8

σ_z = 20685 psi (Axial Stress) From Sect. 3.8

σ_y = 5353 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 17787 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-33-20685)^2 + (20685-5353)^2 + (5353-(-33))^2 + 6(17787^2)}$$

= 36000 psi

Allowable stress (S_{aa} = 0.9 S_u) at 50°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 75886 psi.

$$MS = \frac{75886}{36000} - 1 = 1.1$$

Calculate effective stress Se_4 in the inner surface of the inner shell at point "G"

σ_x = -16 psi (Radial Stress) From Sect. 3.8

σ_z = 2075 psi (Axial Stress) From Sect. 3.8

σ_y = 1078 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 19306 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-16-2075)^2 + (2075-1078)^2 + (1078-(-16))^2 + 6(19306^2)}$$

= 33488 psi

Allowable stress (S_{aa} = 0.9 S_u) at 102°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 52527 psi.

$$MS = \frac{52527}{33488} - 1 = .57$$

Calculate effective stress Se_4 in the shells at points "A", "F", "G" at 40 kw, -40°F ambient, with a cavity pressure of 16.45 psig and a water jacket pressure of 14 psig.

Calculate effective stress Se_4 in the inner surface of the inner shell at point "A"

σ_x = -16 psi (Radial Stress) From Sect. 3.8

σ_z = 39188 psi (Axial Stress) From Sect. 3.8

σ_y = 1688 psi (Tangential Stress)

T_{yz} = 15401 psi (Shear Stress)

T_{xz} = T_{xy} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-16-39188)^2 + (39188-1688)^2 + (1688-(-16))^2 + 6(15401^2)}$$

$$= 46740 \text{ psi}$$

Allowable stress ($S_{aa} = 0.9\ S_u$) at 54°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 74659 psi.

$$MS = \frac{74659}{46740} - 1 = .597$$

Calculate effective stress Se_4 in the inner surface of the outer shell at point "A"

σ_x = 1158 psi (Radial Stress) From Sect. 3.8

σ_z = -10397 psi (Axial Stress) From Sect. 3.8

σ_y = -9797 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xz} = 730 psi (Shear Stress) From Sect. 3.8

T_{yz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(1158-(-10397))^2 + (-10397-(-9797))^2 + (-9797-1158)^2 + 6(15401^2 + 730^2)}$$

$$= 28985 \text{ psi}$$

Allowable stress ($S_{aa} = 0.9\ S_u$) at 42°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 78341 psi

$$MS = \frac{78341}{28985} - 1 = 1.7$$

Calculate effective stress Se_4 in the inner surface of the water jacket at point "A"

σ_x = -14 psi (Radial Stress) From Sect. 3.8

σ_z = 10775 psi (Axial Stress) From Sect. 3.8

σ_y = -2449 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 15401 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-14-10775)^2 + (10775-(-2449))^2 + (-2449-(-14))^2 + 6(15401^2)}$$

Se_4 = 30479 psi

Allowable stress ($S_{aa} = 0.9\ S_u$) at 29°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 85091 psi.

$$MS = \frac{85091}{30479} - 1 = 1.79$$

Calculate effective stress Se_4 in the outer surface of the outer shell at point "F"

σ_x = -14 psi (Radial Stress) From Sect. 3.8

σ_z = -9923 psi (Axial Stress) From Sect. 3.8

σ_y = -3114 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 17787 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-14-(-9923))^2 + (-9923-(-3114))^2 + (-3114-(-14))^2 + 6(17787^2)}$$

= 32034 psi

Allowable stress ($S_{aa} = 0.9\ S_u$) at 29°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 82330 psi.

$$MS = \frac{82330}{32034} - 1 = 1.57$$

Calculate effective stress Se_4 in the inner surface of the water jacket shell at point "F"

σ_x = -14 psi (Radial Stress) From Sect. 3.8

σ_z = 12558 psi (Axial Stress) From Sect. 3.8

σ_y = 3533 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 17787 psi (Shear Stress

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-14-12558)^2 + (12558-3533)^2 + (3533-(-14))^2 + 6(17787^2)}$$

$$= 32790 \text{ psi}$$

Allowable stress ($S_{aa} = 0.9\ S_u$) at 20°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 85091 psi.

$$MS = \frac{85091}{32790} - 1 = 1.59$$

Calculate effective stress Se_4 in the inner surface of the inner shell at point "G"

σ_x = -16 psi (Radial Stress) From Sect. 3.8

σ_z = 3249 psi (Axial Stress) From Sect. 3.8

σ_y = 1563 psi (Tangential Stress) From Sect. 3.8

T_{yz} = 19306 psi (Shear Stress)

T_{xy} = T_{xz} = 0

$$Se_4 = \sqrt{\tfrac{1}{2}}\sqrt{(-16-3249)^2 + (3249-1563)^2 + (1563-(-16))^2 + 6(19306^2)}$$

$$= 33558 \text{ psi}$$

Allowable stress ($S_{aa} = 0.9\ S_u$) at 52°F for 304 s.s From Sect. 1.1 under all structures and Sect. 1.2 equals 75273 psi.

$$MS = \frac{75273}{33558} - 1 = 1.24$$

4.6.2.1 BUCKLING OF CONTAINMENT SHELLS DURING SIDE IMPACT

This section addresses the cask shell stability under the 81g side impact. As discussed below, this analysis also demonstrates the stability of the cask shells for all the 30 ft. drops and the side pin-puncture loading.

Briefly, the procedure used to evaluate the stability of the containment shells under the 81g side loading was as follows:

a. The peak stress-strain condition at the cask mid-plane for the side drop was determined by computing the radius of curvature of the cask center line at the cask mid-plane in order to produce in the inner shell, lead, outer shell, and water jacket shell the static 81g equilibrium moment computed for this condition in Section 3.8.6. In this calculation the shells and lead deform plastically (static properties were used for both the stainless steel and the lead), and axial buckling of the water shell was considered.

b. The maximum computed compressive stress in the shells was then assumed to act uniformly (axisymmetrically) around the shells as a constant axial load or stress. (Baker, Ref. 4, in Figs. 10-9 and 10-13 shows this assumption to be conservative.) Using the axisymmetric buckling analysis for the cask composite structure given in Appendix C, the axial buckling stress was determined for the system with the 70 kw, 130° F ambient stress condition imposed as a pre-buckling stress state. The result was then compared to the maximum computed stress for the 30 ft. side drop.

Since the method of analysis assumed the maximum load to act axisymmetrically, this analysis also applies to the 30 ft. end drops. The axisymmetric axial load produced by the end drops are much less severe than the resulting axial load from the 81g side drop. (In the 30g end drops, the maximum axial stresses developed in the shells were elastically calculated to be approximately 8 ksi, while for the 30 ft. side drop, the maximum elastic stress was about 40 ksi.) Additionally, the pin-puncture drop on the cask side produced only 7.07g lateral acceleration (Section 4.8.2) and was shown to produce no concentrated strains in the outer shell. Therefore, for the method of analysis applied, the 81g side drop condition represents a worst-case for shell stability among the cask drop conditions.

Material Properties

Static material properties are used in this analysis. For the stainless steel, stress-strain properties are needed to strain levels of approximately 3% and at temperatures between 326° F and 413° F. The stress-strain curve selected is presented in Fig. 4.6.2.1-1. The curve represents design data for 304 stainless steel at 400° F taken from the NERVA Program Materials Handbook, Ref. 7 (The selected curve has the same 0.2% yield and ultimate stress as given in Section 1.2 for stainless steel.) The 400° F curve was used for all three shells of the cask. This is a conservative assumption since most of the bending resistance is developed in the 350° F outer shell. Additionally shown in Fig. 4.6.2.1-1 are empirical representations of the curve required in the analysis.

The stress-strain properties selected for the lead are shown in Fig. 4.6.2.1-2. The curve represents the 450° F static properties developed from the NL Research Laboratory experimental lead study, Ref. 81. (The temperature 450° F represents a convenient and highly conservative average lead temperature for the 70 kw, 130° F ambient normal condition.) The required empirical representation of the lead curve is also given in Fig. 4.6.2.1-2.

Buckling of Water Shell Under Axial Load

To insure that a conservative estimate for the bending resistance of the water shell was obtained, the buckling of this shell alone under axial load was examined.

The stress-strain representation assumed for the shell is the second empirical relation given in Fig. 4.6.2.1-1.

The stiffening affects of the combined internal and external cooling fins were assumed to influence strongly the stability of the water shell. Thus, one possible mode of shell buckling is for axial waves to form between the cooling fins. To examine this mode, the fins were assumed to restrict the axial half wave length to 2.0 in. or less. The 2.0 in. is a conservative assumption since the free length of shell between either an external or internal fin is less than this assumed half wave length. Another mode of buckling is for the shell to buckle in longer wave lengths than the spacing between the fins. To determine the buckling of the water shell for the longer wave lengths, the following expression was used[1]:

$$\frac{P_{cr}\,(\text{supported})}{P_{cr}\,(\text{unsupported})} = \sqrt{1 + \frac{A_F}{\delta t_s}}$$

where

P_{cr} is the axial buckling stress,

A_F is the fin area,

δ is the axial fin spacing,

t_s is the shell thickness.

For the 10/24 cask fin configuration, the resulting buckling stress ratio is 1.40.

To compute the buckling loads, the shell buckling development of Appendix C was used. The axial stress was used as the load parameter and the water chamber was conservatively assumed to be non-pressurized. In the search for the minimum buckling load, the axial half wave length and the number of circumferential waves were varied.

---

[1]Eq. 6, Hutchinson, J. W., and Amazigo, J. C., AIAA Journal, March 1976, p. 392.

The buckling solution for the water shell indicates that the axial buckling stress is in excess of 45.0 ksi for the 2.0 in. axial half wave length. For the unsupported shell, the computed axial buckling stress was 28.0 ksi. The resulting buckling load is therefore 39.2 ksi and is the smaller of the two predictions. This stress is above the levels reached in the 81g side drop loading. Therefore, the water shell can be assumed stable in the calculations which follow.

Shell Loading for the 81g Side Drop

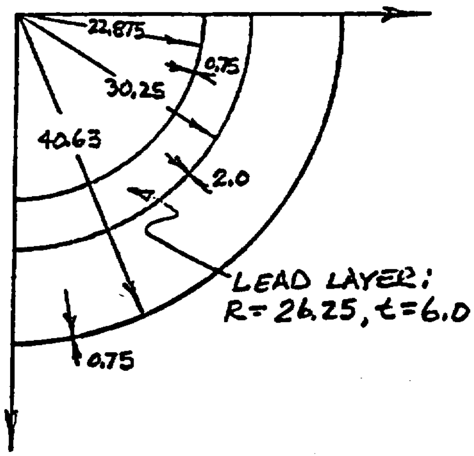
In the SAR, Section 3.8.6, the equilibrium moment at the cask mid-plane is computed for 1g side loading. The results of this calculation are $\overline{m} = 5.25(10)^6$ in. lbs. Therefore, the required equilibrium moment for the 81g side loading is $81.0\,\overline{m} = \overline{M} = 4.253(10)^8$ in. lbs.

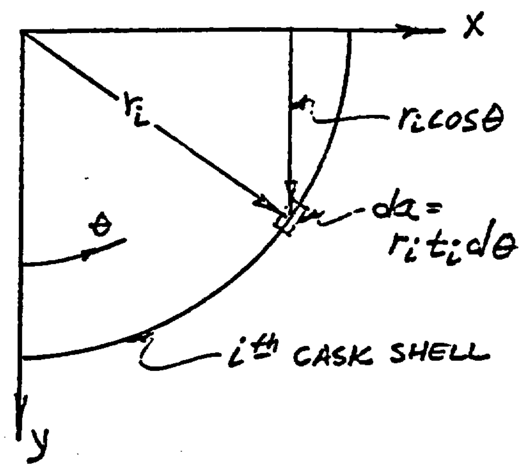
Plastically Computed Bending Moment

To determine the state of stress and strain in the shells which correspond to the 81g equilibrium moment, the radius of curvature at the cask center was increased incrementally. At each step in this loading the internal resisting moment was computed using the inelastic stress-strain curves of Figs. 4.6.2.1-1 and 2, and the assumption that plane sections remain plane. When the internal resisting moment in the shells and lead equaled the equilibrium moment, the loading was stopped and the desired stresses and strains in the shells computed.

The stresses which exist in the cask before the drop are predominantly strain-controlled and are relatively small, approximately 0.2%. In the 81g side drop, the strains are on the order of 2%; therefore, the final stress results were not significantly affected by the initial normal condition stress state. But, to insure the conservativism of the final result, the initial strain from the normal condition was included.

In the analysis presented below, the integrations in the cask shells are performed along each shell's mid-surface, and the axial stress (the only stress considered) is assumed not to vary through the shell thickness. The sketches below define the basic dimensions used in the analysis.





From the plane-sections-remain-plane assumption, the strain at a distance y from the cask neutral surface is given by:

$$\epsilon = \frac{y}{\bar{R}}$$

where $\bar{R}$ is the radius of curvature of the deformed center line at the cask mid-plane. Substituting this strain expression into an empirical stress-strain expression of the form (see Figs. 4.6.2.1-1 and 2):

$$\sigma = \sigma_0 \epsilon^m$$
$$= \sigma_0 \left(y/\bar{R}\right)^m = \sigma_0 \left[\frac{r\cos\theta}{\bar{R}}\right]^m$$

Computing the increment of moment about the x-axis for one of the shells yields:

$$dM = y\sigma\, da = (r_i \cos\theta)(\sigma)\, r_i t_i\, d\theta$$

$$= (r_i \cos\theta)\left(\sigma_o \left[\frac{r_i \cos\theta}{\bar{R}}\right]^m\right) r_i t_i\, d\theta$$

or:

$$dM = t_i\, r_i^{2+m}\, \sigma_o \frac{1}{\bar{R}^m} (\cos\theta)^{1+m}\, d\theta$$

Once an estimate for $\bar{R}$ was made, the moment expression was integrated for each of the three shells and the lead. The moment contribution for each shell was then summed to yield the internal resisting moment for the particular cask radius used. The results of these integrations vs. $\bar{R}$ are given in Fig. 4.6.2.1-3.

As shown in Fig. 4.6.2.1-3, the predicted radius of curvature for the cask center line is 2060 in. This result corresponds to a maximum strain in each of the shells as listed below:

inner shell:

$$\epsilon_i = 0.0111 \text{ in./in.}$$

outer shell:

$$\epsilon_o = 0.0147 \text{ in./in.}$$

water shell:

$$\epsilon_w = 0.0197 \text{ in./in.}$$

Referring to Fig. 4.6.2.1-1, it is evident that the predicted axial stress in the water shell is adequately within the computed axial instability loading for this shell.

To include the influence of the initial state of the shells, the maximum shell strains from the 70 kw, 130° F ambient normal condition were added directly to the computed drop condition strains. From the axisymmetric finite element solution (with elastic-plastic stainless steel), the maximum computed effective strains are 0.0009 in./in. for the inner shell and 0.0003 in./in. for the outer shell. Adding the normal condition strains to the drop condition strains results in 0.012 in./in. strain for the inner shell and 0.0150 in./in. strain for the outer shell. Referring to Fig. 4.6.2.1-1, the maximum estimated axial stress in the inner shell-lead-outer shell composite was computed to be 32.0 ksi.

Buckling of the Inner Shell-Lead-Outer Shell Composite

Using the buckling development of Appendix C, the shell composite system was examined for axial buckling. Starting from the initial stress conditions of the 70 kw, 130° F ambient normal solution, the axial load in the inner and outer shells was increased incrementally. The second empirical relation of Fig. 4.6.2.1-1 was used to describe the stainless steel curves in the buckling solution. Additional loading parameters for the computation are listed below.

	Inner Shell	Lead	Outer Shell
Hoop Stress	-19000	-650	6250
Axial Stress	-P(1)	-650	-P
Radial Stress	-325	-850	-325
Temperature	400° F	450° F	400° F
Tangent Slope	empirical curve(2)	2000 psi(3)	empirical curve
Secant Slope	empirical curve	10000 psi(3)	empirical curve

NOTES:

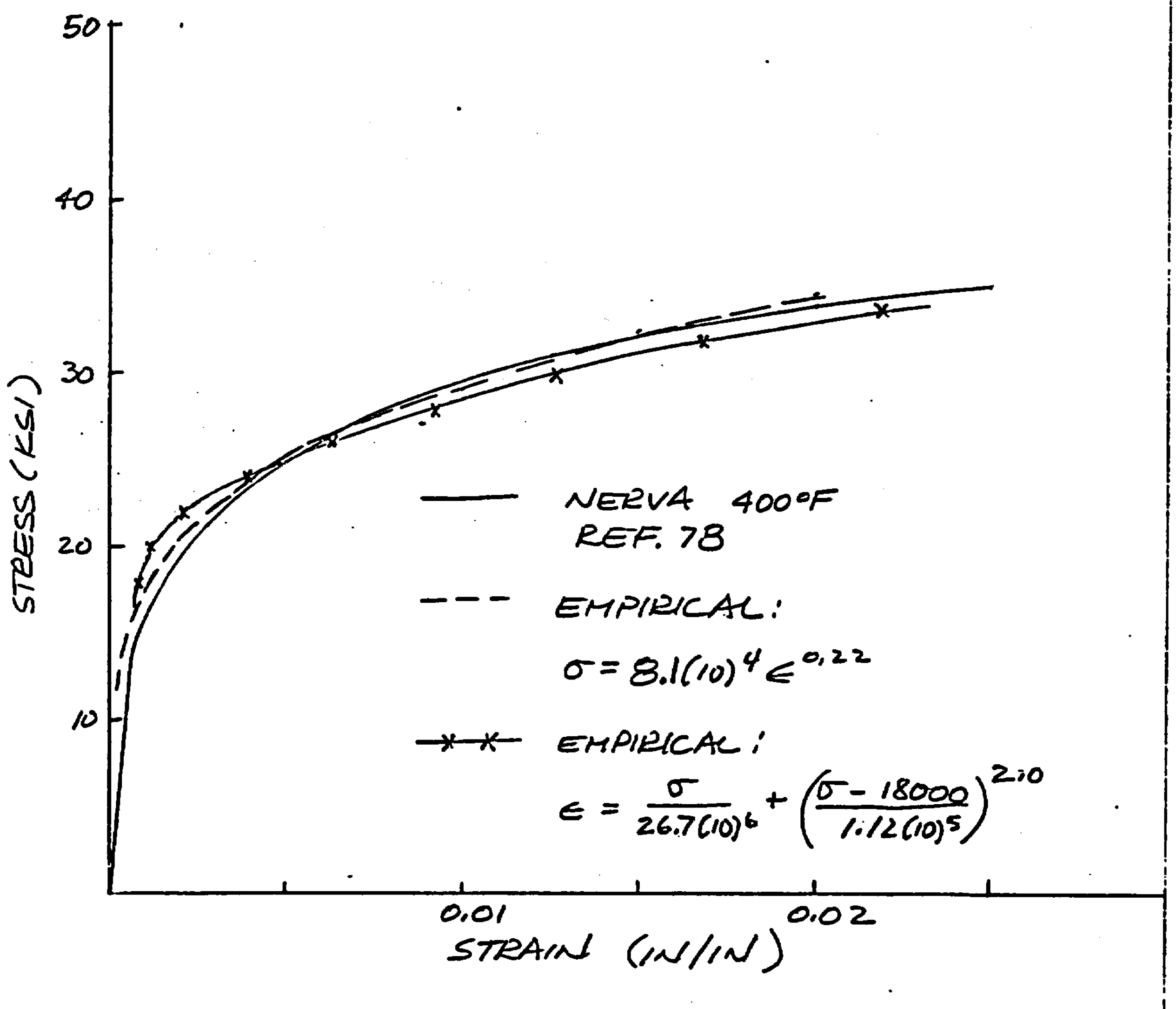
1. The axial load in the shells was taken as the loading parameter.
2. From the stress components, an equivalent stress was computed

which was used with the empirical representation of the stress-strain curve to determine the secant and tangent moduli.

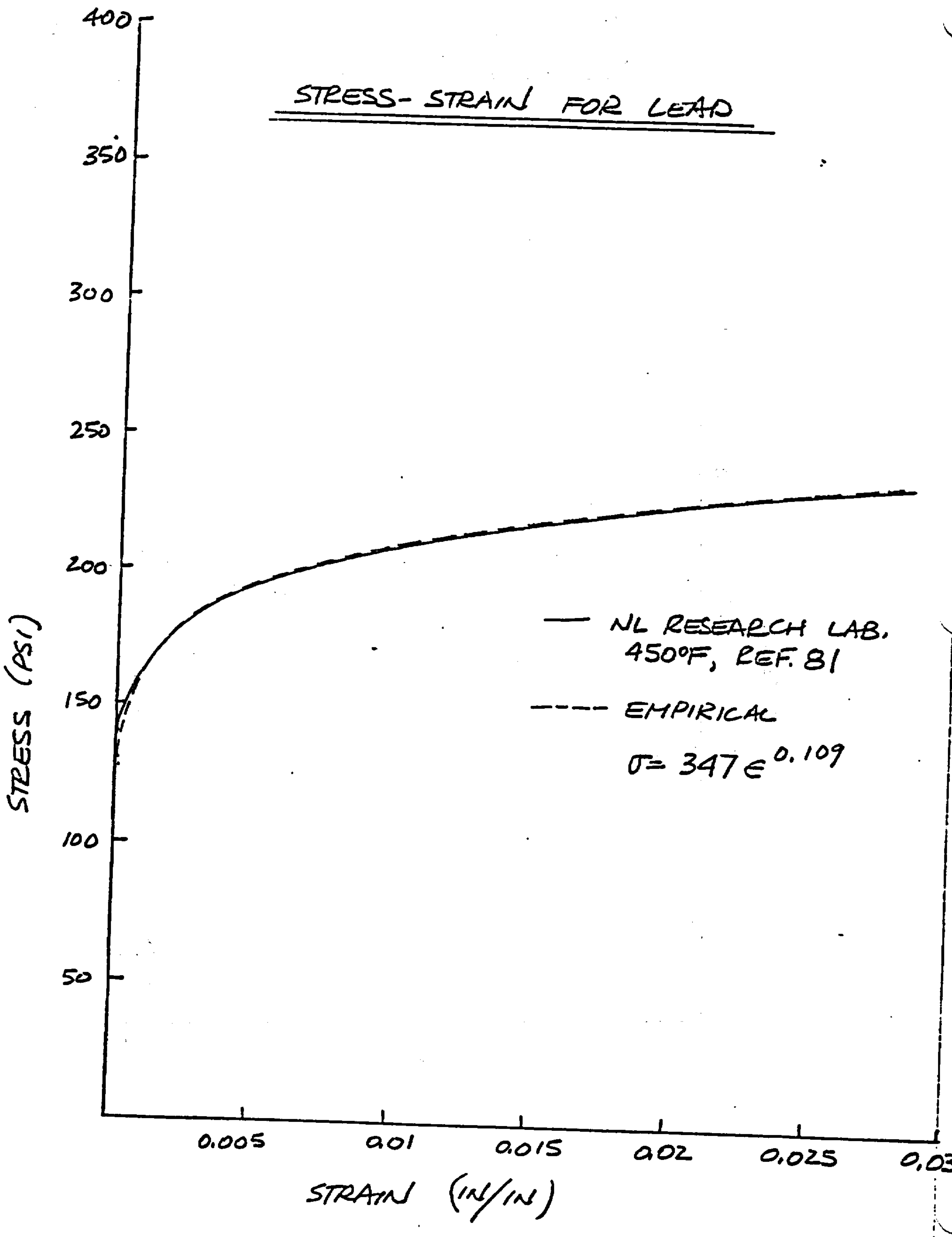
3. Conservative estimates from Fig. 4.6.2.1-2 at an average lead strain of approximately 2.0%. In this analysis, the deformation theory of Appendix C was used to determine the lead moduli.

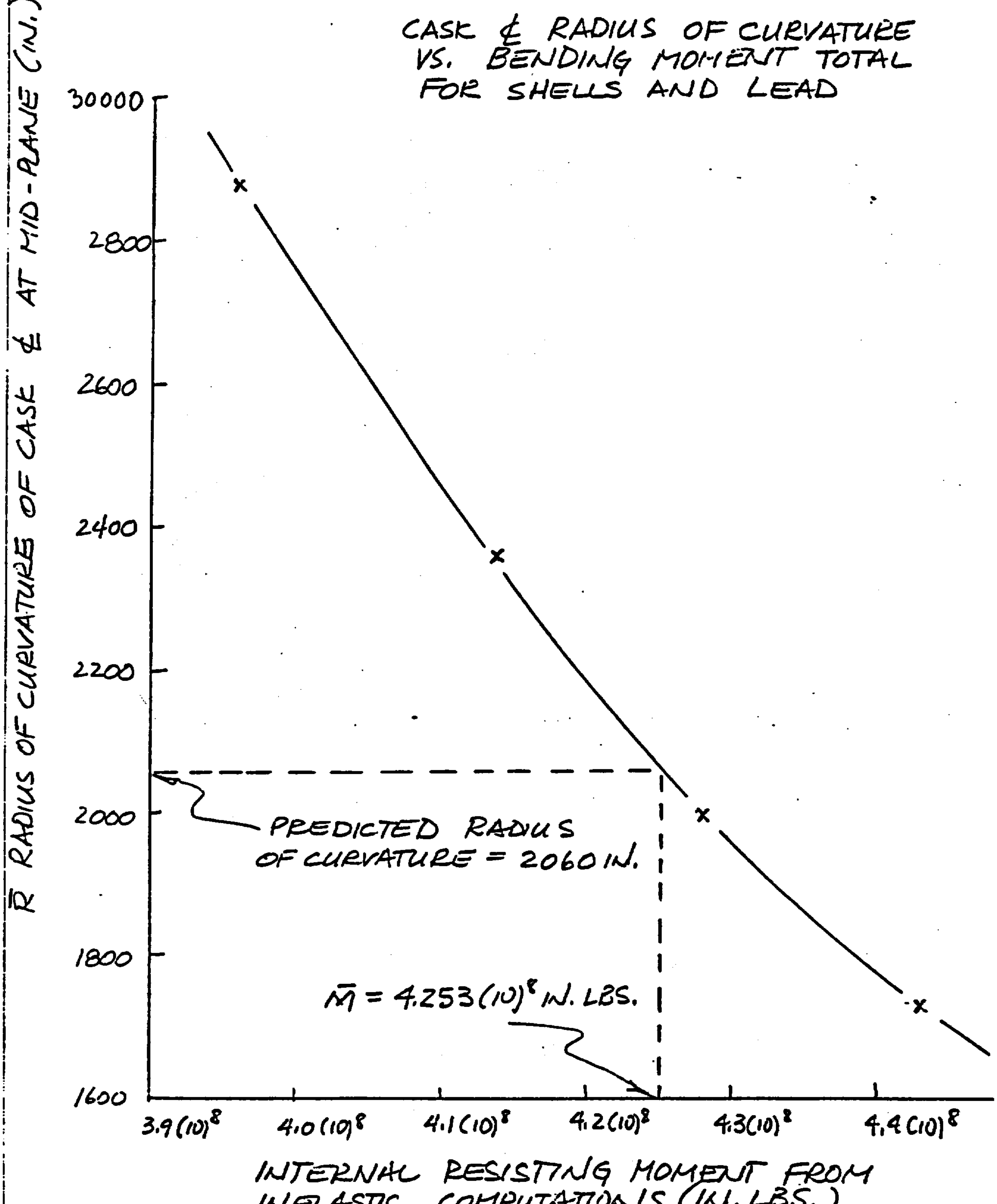
The results of the buckling calculation indicate that the inner shell buckles in an axisymmetric mode with an axial half wave length of approximately 8.0 in. at a stress of 42.0 ksi. This buckling stress is well above the calculated 32.0 ksi maximum 81g side drop axial stress in either of the two shells.

STATIC STRESS STRAIN CURVE FOR 304 S.S. @ 400°F



SHELL BUCKLING UNDER 81G SIDE DROP (CONT.)



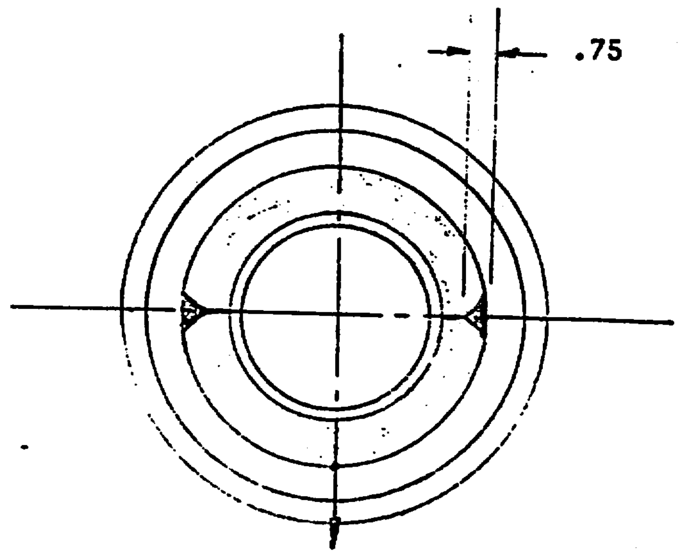

CASK ℄ RADIUS OF CURVATURE VS. BENDING MOMENT TOTAL FOR SHELLS AND LEAD
R̄ RADIUS OF CURVATURE OF CASK ℄ AT MID-PLANE (IN.)
30000
2800
2600
2400
2200
2000
1800
1600
PREDICTED RADIUS OF CURVATURE = 2060 IN.
M̄ = 4.253(10)^8 IN. LBS.
3.9(10)^8
4.0(10)^8
4.1(10)^8
4.2(10)^8
4.3(10)^8
4.4(10)^8
INTERNAL RESISTING MOMENT FROM INELASTIC COMPUTATIONS (IN. LBS.)

4.6.3 Lower Uranium Ring

The lower uranium ring is subjected to a tension load through the longitudinal weld joint which joins the cylindrical shield halves together as a result of side impact loading of 81 G.

F= 0.5 WG W= Uranium Weight

F= 0.5 x 5171 x 81 = 209425 lbs.



$$S_t = \frac{F}{A}$$

A= Area of Weld

A= (.750) (14) (2) = 21 in.^{2}

$$S_t = \frac{209425}{21} = 9973 \text{ psi}$$

Allowable stress (S_{aa} =0.9 S_u) at 382°F for uranium from Sect. 1.1 under noncontainment structure and Sect. 1.2 equals 0.9 x 58500 = 52650 psi.

$$M.S. = \frac{52650}{9973} - 1 = 4.23$$

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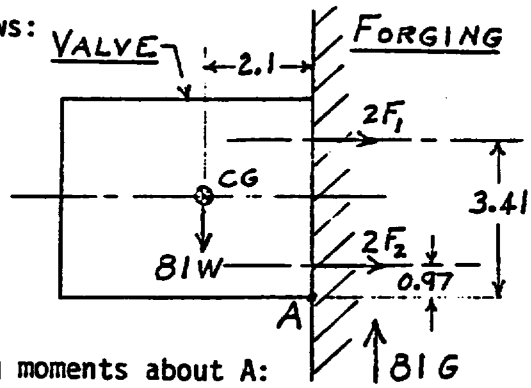
4.6.4 Containment Vessel Valves

In the side impact the cask acceleration of 81g acts in a transverse direction to the valve assembly, which causes a moment tending to open the joint between the valve assembly and the forging. Loading is as follows:



See Sect. 4.4.8 for weight, dimensions, pressure and seal loads.
A - rotation axis of joint.
$F_2 = (.97/3.41)F_1 = 0.28446F_1$

Taking moments about A:

$(2.1)\ (81)\ (23.5) = 3.41(2F_1) + 0.97(2\ F_2) = 6.82F_1 + 1.94\ (0.28446)F_1$

$F_1 = 3997/7.37 = 542$ lb.

Adding pressure and seal loads gives $F\ max = (1/4)(6804 + 548) + 542 = 238$

Bolt preload torque is set at 200± 10 in-lb to provide a <u>minimum</u> bolt load of 190/(.2)(.375) = 2533 lb. and thus maintain the joint seal in the side impact. Allowable bolt stress @325°F = (2/3)(88000) = 58667 psi. (Sect. 1.2.24) At maximum preload the tensile stress is 210/(.075)(.9775) = 36129 psi.

M.S. = (58667/36129) - 1 = <u>0.62</u>

4.6.5 PWR Absorber Sleeve Supports

This analysis considers two cases of the supports for the sleeves under side drop loadings of 81 g's. The first case is that of the separator between each pair of sleeves in a double PWR sleeve assembly. Such an assembly occurs in two places on the 0°-180° centerline of the basket. The second case is that of a built-up support for a single PWR sleeve, and occurs twice on the 90°-270° centerline. Both cases are analyzed by similar methods.

Fuel loadings. During the side drop shock condition of 81 g's, the fuel elements transmit individual loads from the pin sections to the spacer grid structures, which in turn transmit the fuel loads to the underlying absorber sleeve structure.

This analysis does not take into account either the inherent stiffness of the spacer grids or the fact that they impose less than a purely uniform load pattern on the absorber sleeves and so on their supports. The load at each spacer grid is, however, assumed uniformly distributed over its width of 1.5" and its length of 9.633", which is the width of the absorber sleeve. This area constitutes the "foot" of the spacer grid, but for calculating bending of the supports it is conservatively considered a load line 9.633" long. The actual 1.5 in. wide foot would somewhat reduce the sharp peak moments developed by line loadings.

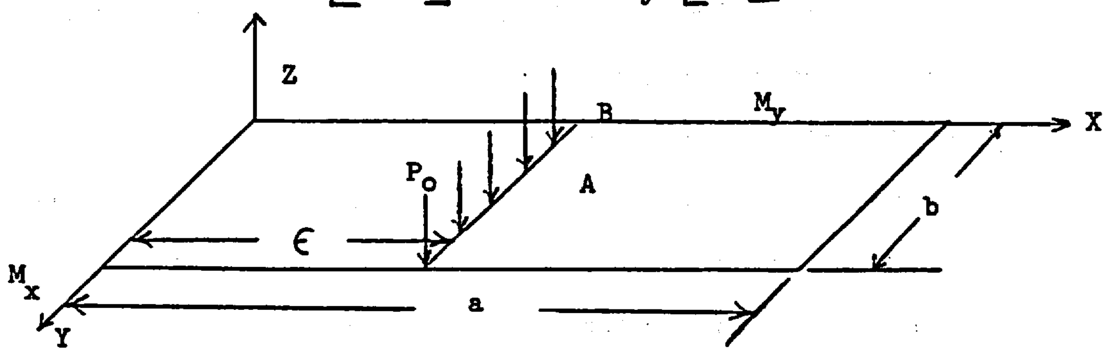
Each fuel assembly is assumed to have 2 end nozzles and a minimum of 6 intermediate spacer grids, and therefore 6 load lines. Assuming relative load values of 1/2 at each end and 1 at each spacer grid, each grid and load line will be supporting 1/7 of the total weight of the fuel assembly.

For a nominal overall length of 161", the spacing distance between load lines is $\frac{161}{7} = 23$ in. The fuel sleeve is assumed not to contribute to the bending resistance of the assembly of sleeves and support spacer.

The spacer in case 1 and the support in case 2 are each treated for analysis purposes as a flat plate with simply supported edges and multiple load lines across its short span spaced at 23 in. intervals.

The solution to this plate problem is given in reference 57 (Szilard-pp 57-61). This is an expansion of two variables in a double Fourier sine series. For strip loadings of a simply supported plate, reference is made to No. 6 diagram and to formulae for: *

P_{mn}, W_m [1.6.4a], M_x and M_y [1.6.6]



$$P_{mn} = \frac{8P_o}{\pi an} \sin\left(\frac{m\pi\epsilon}{a}\right) \qquad \begin{pmatrix} m = 1, 2, 3, \ldots \\ n = 1, 3, -5, \ldots \end{pmatrix}$$

P_o = lbs/in along load line

ϵ = distance to load line

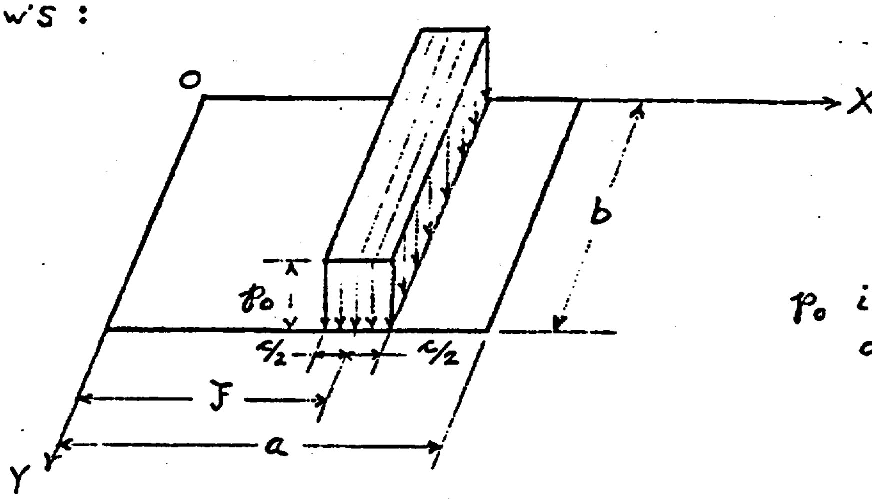
$$W_{mn} = \frac{P_{mn}}{D\pi^4 \left[\frac{m^2}{a^2} + \frac{n^2}{b^2}\right]^2}$$

$$= \frac{8P_o \sin\left(\frac{m\pi\epsilon}{a}\right)}{\pi an \,(D\pi^4)\left[\frac{m^2}{a^2} + \frac{n^2}{b^2}\right]^2}$$

$$M_x = \pi^2 D \sum_m \sum_n \left[\frac{m^2}{a^2} + v\,\frac{n^2}{b^2}\right] W_{mn} \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right)$$

* See Page XI-4-123b-1 for derivation of corrected expression for P_{mn}.

The equation for the load coeffient, P_{mn}, of a line load as given in Reference 57 was found to be incorrect. Hence, the expression for P_{mn} was rederived as follows:



p_o is uniform pressure over loaded region.

Let $p_z(x,y) = p_o = P/bc$ where P is total load.

From Equation 1.5.34 (Ref. 57)

$$P_{mn} = \frac{4}{ab}\int_{\xi - c/2}^{\xi + c/2}\int_0^b (P/bc)\sin\left(\frac{m\pi x}{a}\right)\sin\left(\frac{n\pi y}{b}\right)dx\,dy$$

$$P_{mn} = \frac{4P(1-\cos n\pi)}{abcn\pi}\int_{\xi - c/2}^{\xi + c/2}\sin\left(\frac{m\pi x}{a}\right)dx$$

$$P_{mn} = \frac{4P(1-\cos n\pi)}{bcmn\pi^2}\left[\cos\frac{m\pi}{a}(\xi - c/2) - \cos\frac{m\pi}{a}(\xi + c/2)\right]$$

$$P_{mn} = \frac{8P(1-\cos n\pi)}{bc\,mn\pi^2}\sin\left(\frac{m\pi\xi}{a}\right)\sin\left(\frac{m\pi c}{2a}\right)$$

To evaluate this expression for a line load where $c=0$, multiply by $(2a/2a)$, giving

$$P_{mn} = \frac{4P(1-\cos n\pi)}{abn\pi} \sin\left(\frac{m\pi\xi}{a}\right)\left[\frac{\sin\left(\frac{m\pi c}{2a}\right)}{m\pi c/2a}\right]$$

Let $(m\pi c/2a) = \alpha$; then $\lim_{\alpha \to 0} \left(\frac{\sin\alpha}{\alpha}\right) = 1$

$$\therefore P_{mn} = \left(\frac{4P}{abn\pi}\right)(1-\cos n\pi)\sin\left(\frac{m\pi\xi}{a}\right) \qquad (m,n = 1,2,3,\ldots)$$

Since P/b equals the line load, P_o (see p. XI-4-123b), and the term $(1-\cos n\pi)$ equals two for odd n and zero for even n, the equation for P_{mn} can alternatively be written as shown on page XI-4-123b as

$$P_{mn} = \left(\frac{8P_o}{\pi a n}\right)\sin\left(\frac{m\pi\epsilon}{a}\right) \qquad \text{where } \epsilon = \xi,\ m = 1,2,3,\ldots,\ n = 1,3,5,\ldots$$

$$M_x = \frac{8P_o}{a\pi^3} \sum_m \sum_n \frac{\left[\frac{m^2}{a^2} + v \frac{n^2}{b^2}\right]}{n\left[\frac{m^2}{a^2} + \frac{n^2}{b^2}\right]^2} \sin\left(\frac{m\pi\xi}{a}\right) \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right)$$

$$(m = 1,2,3,\ldots.)$$
$$(n = 1,3,5,\ldots.)$$

Similarly,

$$M_y = \pi^2 D \sum_m \sum_n \left[\frac{n^2}{b^2} + v \frac{m^2}{a^2}\right] W_{mn} \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right)$$

$$= \frac{8P_o}{a\pi^3} \sum_m \sum_n \frac{\left[\frac{n^2}{b^2} + v \frac{m^2}{a^2}\right]}{n\left[\frac{m^2}{a^2} + \frac{n^2}{b^2}\right]^2} \sin\left(\frac{m\pi\xi}{a}\right) \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right)$$

The following computer programs were used to evaluate M_x and M_y above. Further analyses were made by hand.

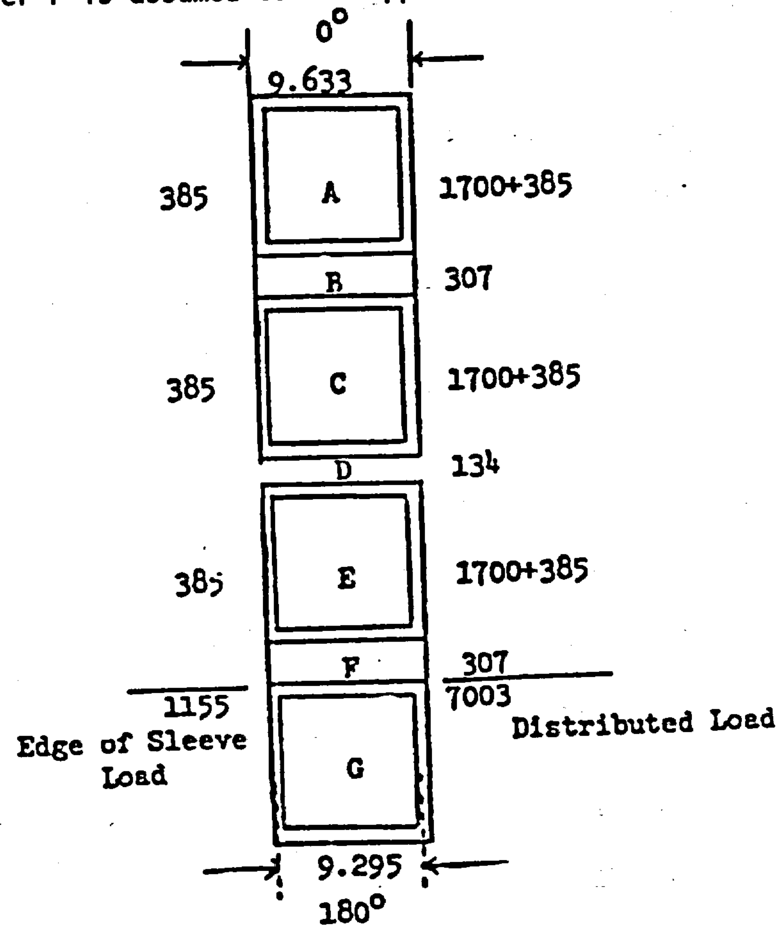
```
THIS PROGRAM CALCULATES BENDING MOMENTS IN A PLATE (MX)
      READ ,PO,A,B,PSI,X,Y
      TOT=0.
      DO 1 M=1,300
      C1=(M*3.14159*PSI)/A
      S1=SIN(C1)
      C2=(M*3.14159*X)/A
      S2=SIN(C2)
      SUM1=0.
      DO 2 N=1,300,2
      C3=(N*3.14159*Y)/B
      S3=SIN(C3)
      R1=(M/A)**2+.300*(N/B)**2
      R2=((M/A)**2+(N/B)**2)**2
      SUM1=SUM1+(R1*S1*S2*S3)/(N*R2)
    2 CONTINUE
      TOT=TOT+SUM1
    1 CONTINUE
      ANS=(8.*PO*TOT)/(A*31.0063)
      PRINT 3,ANS
    3 FORMAT (1X,E12.5)
      STOP
```

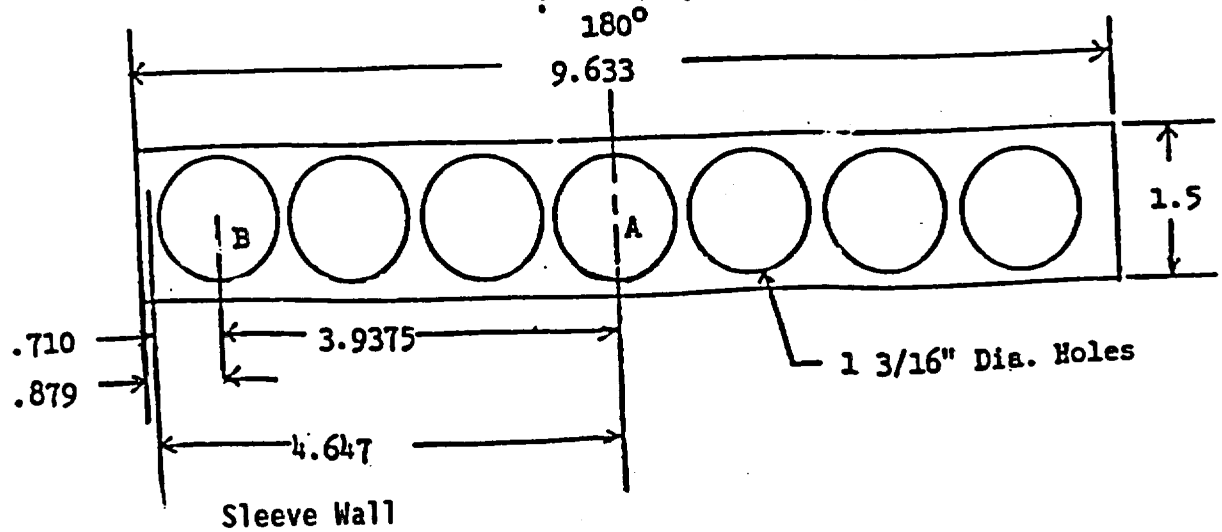
```
THIS PROGRAM CALCULATES BENDING MOMENTS IN A PLATE (MY)
SZILARD PG.60
      READ ,PO,A,B,PSI,X,Y
      TOT=0.
      DO 1 M=1,300
      C1=(M*3.14159*PSI)/A
      S1=SIN(C1)
      C2=(M*3.14159*X)/A
      S2=SIN(C2)
      SUM1=0.
      DO 2 N=1,300,2
      C3=(N*3.14159*Y)/B
      S3=SIN(C3)
      R1=(N/B)**2+.300*(M/A)**2
      R2=((M/A)**2+(N/B)**2)**2
      SUM1=SUM1+(R1*S1*S2*S3)/(N*R2)
    2 CONTINUE
      TOT=TOT+SUM1
    1 CONTINUE
      ANS=(8.*PO*TOT)/(A*31.0063)
      PRINT 3,ANS
```

4.6.5.1 Spacer Plate between Double PWR Sleeves. (Case 1)

In the following diagram, analysis is made for spacer F, which is assumed to bear the loads from fuel sleeves A, C, and E, and spacers B and D. Aluminum ligament D is assumed to have failed in shear.

Spacer F is assumed to be supported at its edges by sleeve G.





"X" Section of Spacer B and F

Spacers B and F are composed of stainless steel (type 304) blocks $1\frac{1}{2}$" thick, 9.633 in. wide and about 16 in. long, welded together for a total desi[gn] length of 161 in. The welds are of sufficient size to allow the assembly to be considered as a continuous beam.

The holes are water passages which extend the entire 161" length.

The minimum section area of the plate is thru each hole. The I and S values are very conservatively taken on this section.

Overall thickness 1.5 in.

Max. hole dia. = 1.187 ± .015 = 1.202 max.

$$I_y \text{ min.} = \frac{1.5^3 - 1.202^3}{12} = .136529 \text{ in.}^4/\text{in. length.}$$

The equivalent thickness of a solid plate would be

$$.136529 = \frac{t^3}{12} \text{ where } t = 1.1788 \text{ in.}$$

Total distributed load = 7003 lbs. = weight of 3 fuel cells and sleeves, spacers B and F, and ligament D.

Total impact load for each load line along spacer is

$$F = \frac{7003 \times (81g)}{7} = 81,035 \text{ lbs.}$$

$$P_o = \frac{81,035}{9.633} = \underline{8412} \text{ lbs/in. along each load line.}$$

also

$b = 9.633$ in.

$a = 161$ in.

ϵ = variable

x = Variable

$y = a/2 = 4.8165$ in.

These values are for the center point A of the simply supported beam shown.

The following table and graph are the moments calculated for the following values of a, b and y, with $\mathcal{E}$ and x varied as shown.

P_o = 8412 a = 161 b = 9.633 y = 4.8165

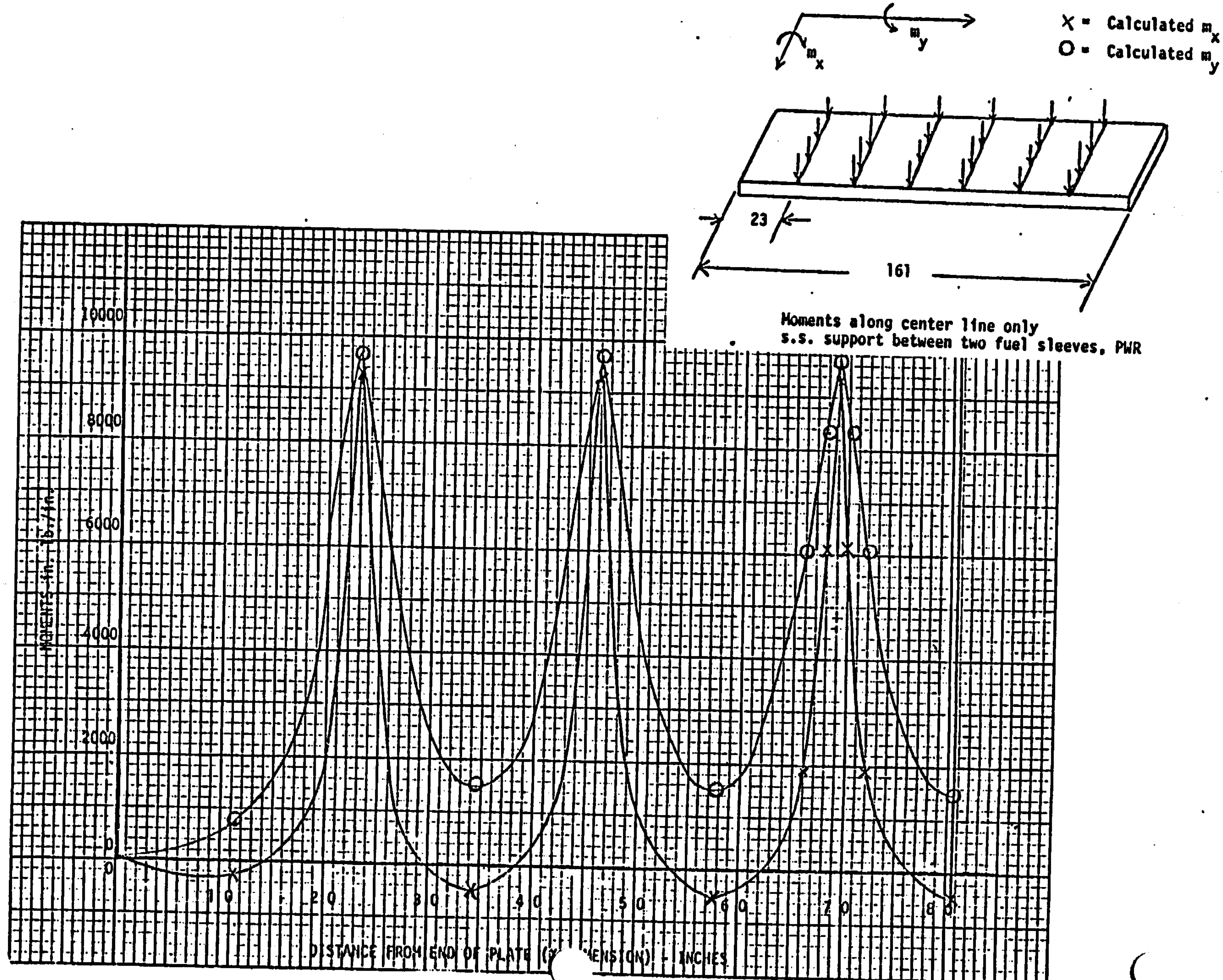
$\mathcal{E}$	x	m_x	m_y
23	11.5	-259	751
	23	9358	9674
	34.5	-257	753
	46	-17.7	30
46	23	-17.7	30
	34.5	-259	753
	46	9358	9674
	57.5	-257	753
	66	-42.2	70
	68	-23.3	39
	69	-17.7	30
69	46	-17.7	30
	57.5	-258	753
	66	1935	5968
	68	6142	8500
	69	9358	9674
	70	6142	8500
	72	1935	5967
	80.5	-258	753
92	69	-17.7	30
	70	-24.5	39
	72	-42.5	69
	80.5	-257.7	753

Suming M_x and m_y at x = 69 inches gives the maximum m_x equal to 9323 in lb/in and the maximum m_y equal to 9734 in lb/in.

Variation in m_x and m_y in y deviction

x = 69 inches

m_x	m_y	y
0	0	0
4375	4060	.879
7574	7257	2.192
9176	8859	3.504



X = Calculated m_x
O = Calculated m_y
m_x
m_y
23
161
Moments along center line only
s.s. support between two fuel sleeves, PWR
MOMENTS in. lb./in.
10000
8000
6000
4000
2000
0
0
10
20
30
40
50
60
70
80
DISTANCE FROM END OF PLATE (X DIMENSION) - INCHES

The peak values are

M_x = 9323 in. lbs/in

M_y = 9734 in. lbs/in

The moments peak and drop off sharply near each load line to negligible values so that the peak moments remain unchanged at the point of each load application.

<u>For the Y cross section</u> of the spacers (thru the 7 holes);

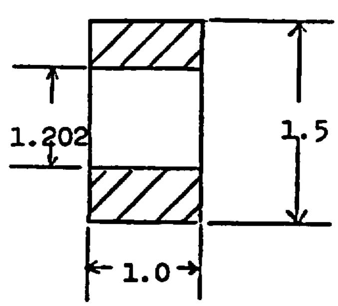
$$I_y = \frac{9.633 \times 1.5^3}{12} - 7 \times \frac{\pi}{64} 1.202^4 = .1992 \text{ in}^4 \text{ for total width}$$

$$I_y \text{ per inch} = \frac{.1992}{9.633} = .2068 \text{ in}^4/\text{in}$$

$$Z_y \text{ per inch} = \frac{.2068}{.75} = .2757 \text{ in}^3/\text{in}$$

$$A \text{ per inch} = \frac{(9.633 \times 1.5) - 7 \times \pi/4\ 1.202^2}{9.633} = .675 \text{ in}^2/\text{in}$$

<u>For the X cross section</u> per inch along the center water hole;



$$I_x \text{ per inch} = \frac{1.5^3 - 1.202^3}{12} = .136529 \text{ in}^4/\text{in}$$

$$Z_x \text{ per inch} = \frac{.136529}{.75} = .18204 \text{ in}^3/\text{in}$$

For point A at the middle of the load line;

$$S_{bx} = \frac{Mx}{Zy} = \frac{9323}{.2757} = \underline{33816\ psi}$$

$$S_{by} = \frac{My}{Zx} = \frac{9734}{.18204} = \underline{53472\ psi}$$

$$S_{sy} = 0 \text{ at midpoint of transverse beam}$$

$$S_{sx} = \frac{P_o}{2 \times .675} = \frac{8412}{1.35} = \underline{6231\ psi}$$

$$S_t = \sqrt{1/2}\ \sqrt{33816^2 + 53472^2 + (54372-33816)^2 + 6\ (6231)^2}$$

$$= .7071\ (67986) = \underline{48073\ psi}$$

Allowable stress ($S_{aa} = 0.9\ S_u$) for 304 S.S. at 466°F from Sect. 1.1 accident conditions and Sect. 1.2 equals (0.9) (59000) = 53100 psi

$$M.S. = \frac{53100}{48073} - 1 = \underline{.105}$$

For point B, thru the outermost hole.

$$S_{bx} = \frac{4060}{.2757} = 14726 \text{ psi}$$ (see p. XI-4-123g)

$$S_{by} = \frac{4375}{.18204} = 24033 \text{ psi}$$

$$S_{sx} = 6231 \text{ psi}$$

S_{sy} is now to be calculated, again conservatively.

$$S_{sy} = \frac{P_o}{2 \times \text{Area}}$$

$$P_o = 8412 \text{ lbs/in}$$

$$\text{Area} = (1.5-1.202)(1) = .298 \text{ in}^2$$

$$S_{sy} = \frac{8412}{.596} = 14{,}114 \text{ psi}$$

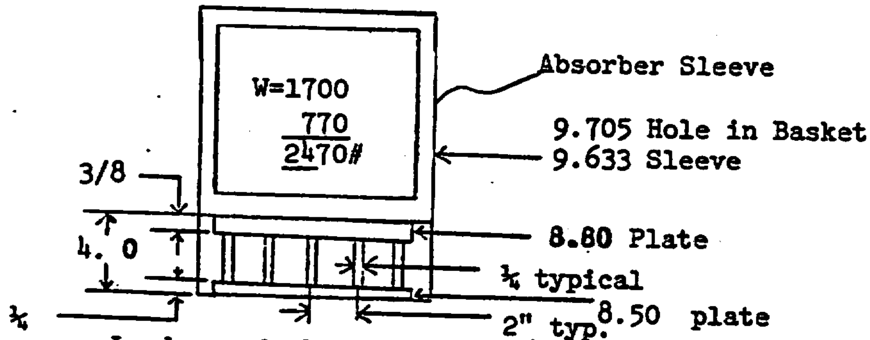
$$S_t = \sqrt{1/2}\sqrt{14726^2 + 24033^2 + (24033-14726)^2 + 6(6231)^2 + 6(14{,}114)^2}$$

$$= .7071\ (48055) = 33979 \text{ psi}$$

$$\text{M.S.} = \frac{53100}{33979} - 1 = .563$$

4.6.5.2 Support under single PWR Sleeve (Case 2)

This support consists of a 3/8 in. S.S. top plate 9¼ in. wide and 161 in. long beneath the sleeve, combined with 5 welded 1/4 in. thick S.S. plates on edge to form a water passage, and a bottom S.S. plate 1/4 thick.

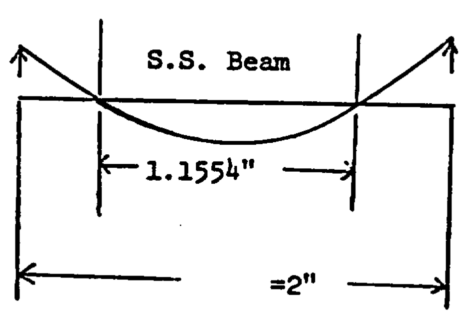


Load per fuel spacer grid (7 divisions of 23 in length)

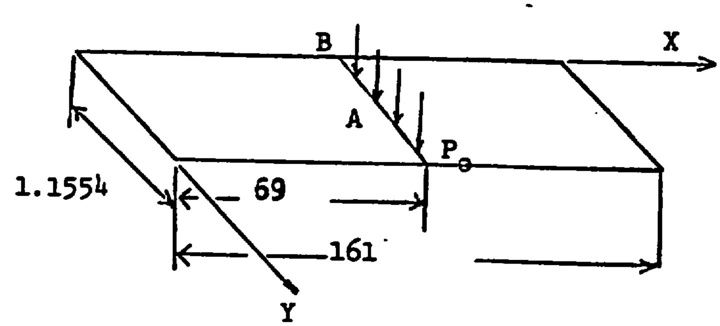
$$F = \frac{2470 \times 81g}{7} = 28{,}581 \text{ lbs.}$$

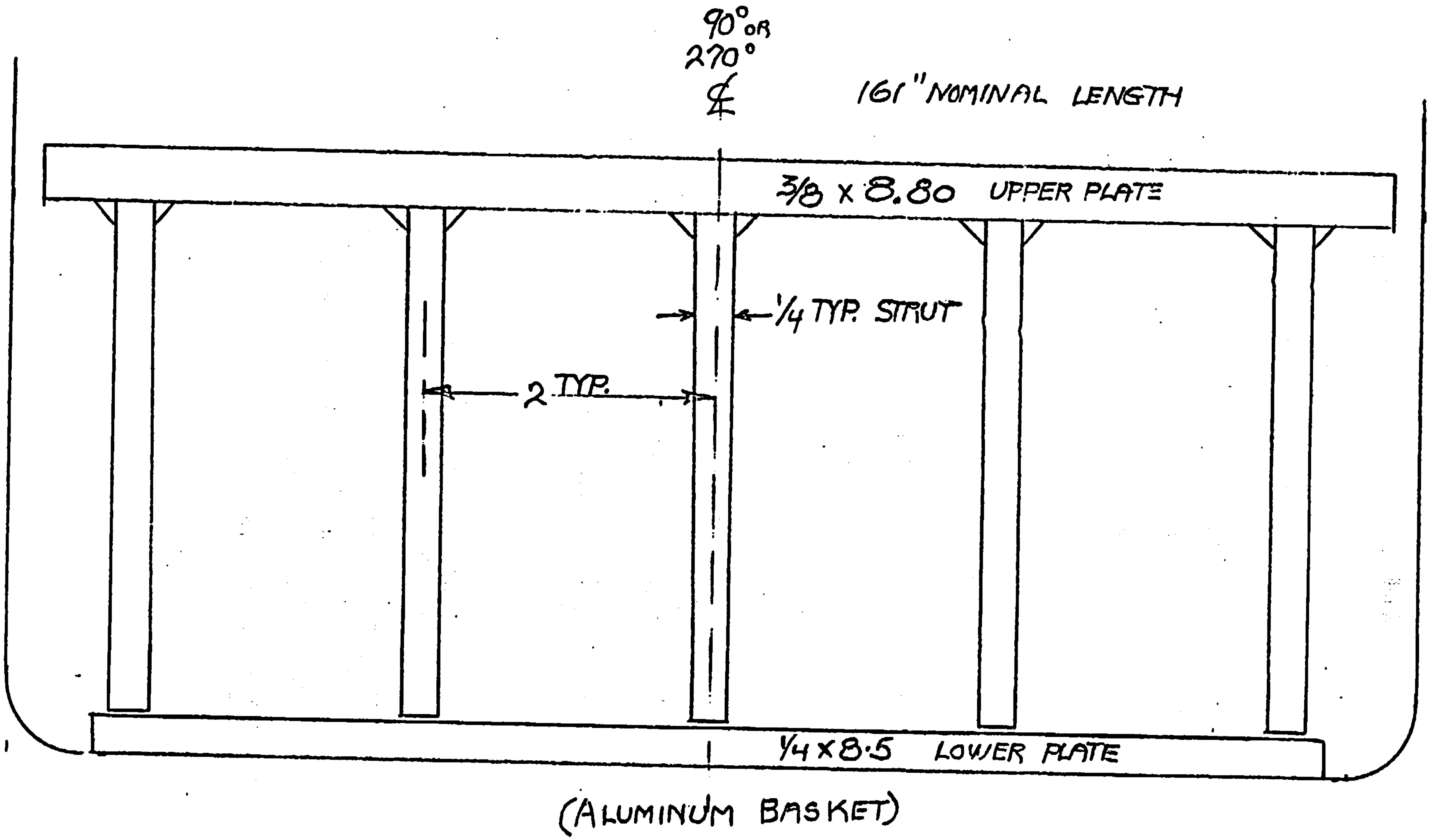
$$P_o = \frac{F}{9.25} = 3090 \text{ lbs/in}$$

Since the slope over each longitudinal strut is zero, the effect is that of fixed ends for the 2" wide section of the support plate. Such a beam can be considered as 2 end cantilever sections with a central length which is simply supported between the points of inflection.



Moment Diagram





SUPPORT FOR SINGLE SLEEVE

304 S.S. WELDED

The method of analysis is now similar to that used in 4.6.5.1. Only the third load line is analyzed because 4.6.5.1 has shown identical peak values for M_x and M_y at the various load line positions.

$$a = 161 \text{ in.}$$

$$b = 1.1554 \text{ in.}$$

$$\epsilon = 69 \text{ in.}$$

$$x = 3a/7 = 69 \text{ in.}$$

$$y = b/2 = .5777 \text{ in.}$$

By computer summation of expressions for M_x and M_y

$$M_x = 270 \text{ in lbs/in}$$

$$M_y = 379 \text{ in lbs/in}$$

<u>For 3/8 plate</u>

$$Z \text{ per inch} = \frac{1 \text{ x} .375^2}{6} = .0234375 \text{ in}^3/\text{in}$$

$$S_{bx} = \frac{270}{.0234375} = \underline{11520} \text{ psi}$$

$$S_{by} = \frac{379}{.0234375} = \underline{16171} \text{ psi}$$

$$S_{sy} = 0$$

$$S_{sx} = \frac{P_{o/2}}{(1x3/8)} = \frac{3090}{.75} = \underline{4120} \text{ psi}$$

$$S_t = \sqrt{1/2}\sqrt{11520^2 + 16171^2 + (16171-11520)^2 + 6 \times 4120^2}$$

$$= .7071\,(22753) = 16089 \text{ psi}$$

$$\text{M.S.} = \frac{53100}{16089} - 1 = 2.30$$

Bearing on aluminum ligament of S.S. strut plates

While the load from the fuel elements is applied to the strut plates at intervals of 23 inches, the stiffness of these plates as deep beams is very effective in distributing this load into the supporting aluminum ligament. This is checked by calculating the deflection of a 23 inch segment of a continuous beam under uniform loading with fixed ends. In this case the actual load lines become reaction supports in the application of formula from Ref. 3, Table III Case 33

Load on each strut plate over 23 in. length and 2 in. width

$$W = 3090 \text{ lbs/in} \times 2'' = 6180 \text{ lbs per strut plate}$$

$$y = \frac{1}{384}\frac{W\ell^3}{EI}$$, in this case represents a waviness of contacting edge.

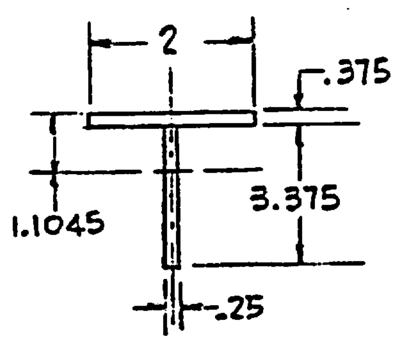
$$\ell^3 = 23^3 = 12167 \text{ in}^3$$

$$E = 26.3 \times 10^6 \text{ psi}$$

I is calculated for the T shape shown in the sketch.

To find Neutral Axis

$$\text{Area Moment} = (3/8 \times 2)(3/16) + (1/4 \times 3.375 (2.1325) = 1.799$$



$$Y_c = \frac{1.79}{.75 + .87875} = 1.1045 \text{ in.}$$

$$I = \frac{2 \times .375^3}{12} + \frac{.25 \times 3.375^3}{12} + .75\,(1.1045 - .1875)^2$$

$$+ \; .87875\,(2.1325 - 1.1045)^2$$

$$I = 2.3688 \text{ in}^4$$

Max. deflection of strut

$$y = \frac{6180 \times 12167}{384 \times 26.3 \times 10^6 \times 2.3688} = .00314 \text{ in}$$

This small deformation supports the assumption that the load distribution is substantially uniform on the aluminum ligament.

Bearing stress on aluminum, allowing only 1/4" load width, very conservatively,

$$S_{br} = \frac{6180}{26.83 \times 1/4} = 921 \text{ psi}$$

Allowable stress ($S_{bra} = 1.35\ S_u$) for 1180 aluminum at 466°F From Sections 1.1 and 1.2 equals (1.35) (2700) = 3645 psi.

$$M.S. = \frac{1.35 \times 2700}{921} - 1 = 2.96$$

Top Plate - stress at supports (at strut plates)

Since the slope of the top 3/8 plate is zero as it passes over each of the strut plates, the section between two strut plates is the equivalent of a fixed end beam. The maximum moment is at these ends and is double the moment at the center, which has already been calculated (M_y)

$$M_x = \underline{270} \text{ in lbs/in}$$

$$2M_y = 2(379) = 758 \text{ in lbs/in}$$

$$Z = \frac{1 \times .375^2}{6} = .0234375 \text{ in}^3/\text{in}$$

$$S_{bx} = \frac{270}{.0234375} = \underline{11520 \text{ psi}}$$

$$S_{by} = \frac{758}{.0234375} = \underline{32341} \text{ psi}$$

$$S_{sy} = \frac{2" \times P_o}{2 \times (1 \times .375)} = \frac{3090}{.375} = \underline{8240 \text{ psi}}$$

$$S_{sx} = \frac{P_o}{2 \times (1 \times .375)} = \frac{3090}{.75} = \underline{4120 \text{ psi}}$$

$$S_t = \sqrt{\tfrac{1}{2}}\sqrt{11520^2 + 32341^2 + (32341-11520)^2 + 6(8240)^2 + 6(4120)^2}$$

$$= .7071\ (46059) = \underline{32568\ \text{psi}}$$

$$\text{M.S.} = \frac{53100}{32568} - 1 = \underline{0.63}$$

Buckling Analysis of strut plates

A series of 5 strut plates extend the full 161 in. length of the support. The section of each is 3.515 in. high by .250 in. thick. Each plate passes under the transverse foot of the fuel spacers, giving a minimum projected loaded area of 1.5 in x .25 in = .375 in^2 in the strut plate. This part of the strut plate is very conservatively treated as a column standing alone under the fuel spacer load, without benefit from the adjacent material in the continuous plate.

The dynamic elastic limit is taken as 60% of the dynamic yield strength (Ref. 5, Sect. III App. I Table I-2.4)

At 466°F the ultimate tensile strength of 304 s/s = 59000 psi and the dynamic yield strength = 2/3 x 59000 = 39,333 psi (Sect. 1.1 and 1.2).

Thus the dynamic elastic limit = .60 x 39,333 = 23,600 psi

Load in each plate from foot = 2 x P_o = 2 x 3090 = 6180 lbs

The critical load for a strut with one end fixed

(Ref. 3, Table XV-Case 1) is

$$P' = \frac{\pi^2 EI}{4\ell^2} = \frac{\pi^2 (26.3)(10^6)\left(\frac{1.5 \times .25^3}{12}\right)}{4 \times 3.375^2} = \underline{11{,}109\ \text{lbs}}$$

$$\text{M.S.} = \frac{11{,}109}{6180} - 1 = .798$$

Compressive stress in column

$$S_c = \frac{6180}{1.5 \times .25} = \underline{16480\ \text{psi}}$$

$$\text{M.S.} = \frac{23600}{16480} - 1 = .432$$

Since the applied load of 6180is less than the critical elastic buckling load of 11,109 lbs and the compressive stress of 16480 psi is less than the elastic limit of 23,600 psi, buckling of the strut plate will not occur.

Conclusions

The principal function of the above spacer and support is to maintain the given volume of water between the fuel sleeves or between a fuel sleeve and the basket material. The above analyses show that the construction and stresses are satisfactory under the 81g side impact and any deformation of the structures will not occur to reduce the water content in the holes.

Relative Flow Rates

All water flow into the water channels will be through holes of the same size and geometry as the water passage itself. The flow into the fuel sleeves will be somewhat retarded by the separators of the fuel bundles. Therefore, the water level in the water channels will be equivalent to or higher than the water level in the fuel sleeves at all times.

BWR Absorber Sleeves

The criticality analysis has shown that the BWR fuel assemblies are undermoderated, therefore, if an absorber sleeve is crushed it will tend to lower the cask reactivity. For this reason the cask will remain in a subcritical condition for all postulated accidents.

Side Drop Effect on BWR/PWR Fuels

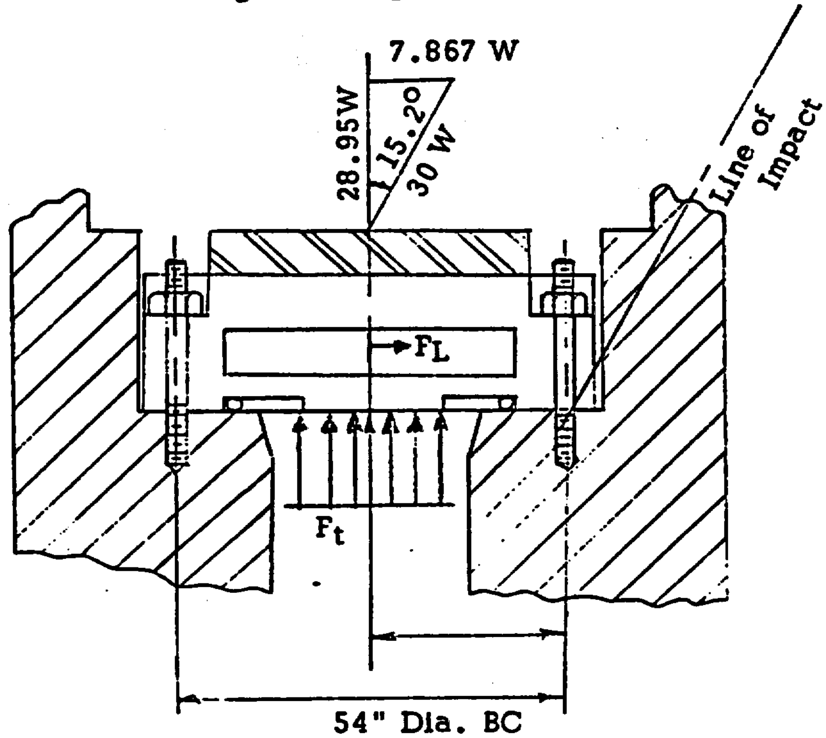
During the side drop impact the fuel elements bear directly against the sleeve wall at the numerous spring clip grids. The grids preserve the structural integrity of the fuel bundle and allow the direct transmission of side loads. The sleeves fit closely in the spaces provided in the aluminum basket and the walls of the basket provides a uniform bearing support for imposed fuel loads. The basket will move to a position of full radial contact in the peripheral regions with the inner shell structure. The inner shell structure then transmits the side impact loads to the upper and lower end forgings.

The means of side impact load transmission allows the load paths in the fuel/sleeve/basket interface regions to be strictly in the pure compression mode and therfore maintaining the overall structural integrity of the system. This method of load transmittal allows the sleeves to maintain the relative position to the fuel element that is indicated in the overall design.

4.7 Corner Impact

4.7.1 Inner Closure Rotation

During the corner impact the inner closure head is subjected to an acceleration of 30g at an angle of 15.2° (Section 4.3.4)



$W = W_1 + W_2$

W_1 = 7400 lbs. Closure weight

W_2 = 34100 lbs. Contents weight

$W = 7400 + 34100 = 41500$ lbs.

$F_t = W(28.95) = 41500 \times 28.95 = 1201425$ lbs.

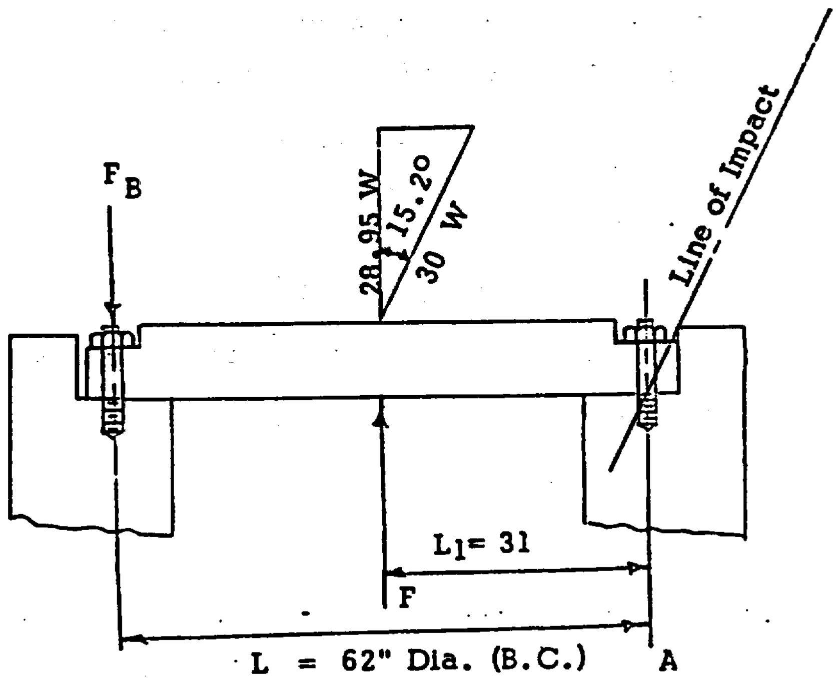
$F_L = W_1(7.867) = 7400 \times 7.867 = 58216$ lbs.

During the corner impact the lateral load F_L = 58216 lbs will not be taken by the bolts in shear. It will be taken by the forging in bearing as shown in section 4.6.1.1 (81 G side impact analysis). Bearing stresses in this case are much lower than stresses that will occur during the side impact.

The axial force F_t will be taken by the bolts in tension; the joint analysis and bending stresses of the inner closure head are calculated in section 4.4.1 for the 30g top end impact, which is a more severe loading than the corner impact.

4.7.1.1 Outer Closure Rotation.

During the corner impact the outer closure is subjected to an acceleration of 30g at an angle of 15.2° (Section 4.3.4)



Weight of outer closure = 2328 lbs.

Loading from pressure between inner and outer closures is.

18265 lbs. (Sect. 3.11)

To be highly conservative, assume that only 5 bolts resist the combined inertial and pressure loads

$$F = 28.95\ (2328) + 18{,}265 = 85{,}661 \text{ lbs.}$$

Moment on joint, $M_o = 31(85661) = (2.6555)10^6$ in-lbs.

Joint resisting moment from only 5 bolts is

$$M_B = 31\ F_B \left[2 + (\cos\phi + 1)^2 + (\cos 2\phi + 1)^2\right]$$

where ϕ = 6.428 degrees (angle between bolts).

F_B = maximum bolt force.

Equating M_O to M_B and solving for F_B gives

$$(2.6555)10^6 = 31\ F_B \left[2 + (1.9937)^2 + (1.9749)^2\right]$$

$$F_B = (2.6555)10^6 / (31)(9.875) = 8675 \text{ lbs.}$$

Minimum bolt preload = 10,450 lbs. (Sect. 3.11)

M.S. = (10450/8675)-1 = 0.205

Since the outer closure can withstand the larger bending loads of the top end impact (Sect. 4.4.1.1), its integrity and that of the bolts and seal are maintained in the corner impact.

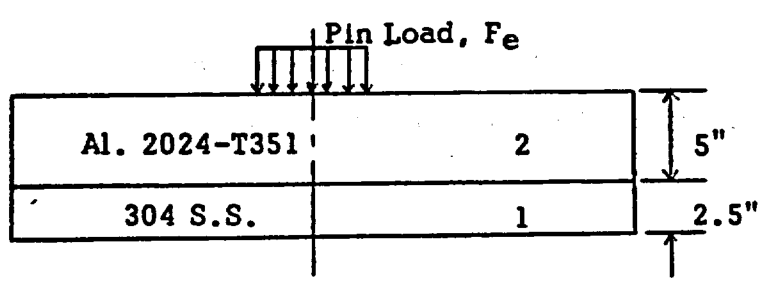
4.8 Puncture

In the hypothetical puncture accident specified in 10 CFR Part 71 (Par. 71.36 and Appendix B) the cask must withstand a drop from a height of 40 inches onto a 6 inch diameter steel pin without unacceptable reduction of shielding or loss of contents. In this cask design the only potentially vulnerable parts of the containment structure are the outer closure, the bottom head, and the outer shell. That these parts of the containment structure will maintain their integrity in the puncture accident is demonstrated by the analyses which follow.

4.8.1 Outer Closure

Integrity of outer closure must be maintained in the hypothetical puncture accident to protect containment vessel valves and protect inner closure head. The maximum possible load that could be imposed on the outer closure during the puncture impact would be the failure load of the puncture pin itself. See Sect. 4.8.2 where F_c = $(1.413716)10^6$ lbs. With this force the maximum possible cask acceleration during the puncture impact is

$$= (1.414)(10^6) / 200,000 = 7.07 \text{ g}$$



The aluminum plate shown in the analytical model is part of the end impact limiter and rests directly on the surfact of the outer closure head.

Inner plate (1) (Outer Closure)

Temperature (Sect. 3.1) $= \frac{410 + 326}{2} = 368^\circ F$

Material - 304 Stainless Steel

Radius to edge support 29.5"

Poisson's ratio - 0.3 (Sect. 1.2)

Modulus of elasticity - $(26.8) \times 10^6$ psi (Sect. 1.2)

Outer plate (2)

Temperature 250° F (Sect. 3.1)

Material - 2024-T 351 Aluminum

Radius to edge support - 29.5"

Poisson's ratio - 0.33 (Sect. 1.2)

Modulus of elasticity - 9.6×10^6 psi (Sect. 1.2)

For plate, edges supported, uniform load over entire surface

From Reference 3, Table X, Case I

Max. Stress at center, $\sigma = \frac{3\ (3 + v)}{8\ \pi\ t^2}\ F$

Center deflection, $y_c = \frac{3\ (1 - v)\ (5 + v)\ R_o^2}{16\ \pi\ E\ t^3}\ F = KF$

The two plates are constrained to have the same elastic deflection curves under lateral bending loads. Hence, the deflection of each plate must be the same and the total lateral load applied to the assembly can be divided among the individual plates in accord with each one's proportionate part of the total bending resistance.

Equating the center deflection of the two plates gives - $K_1F_1 = K_2F_2$

Also, the total load imposed on the closures must equal the sum of the individual plate loads, so that

$$F_t = F_1 + F_2$$

Combining these equations in terms of outer plate load F_2, gives -

$$F_t = K_2F_2\left(\frac{1}{K_1} + \frac{1}{K_2}\right)$$

These equations may be evaluated to obtain the force on each plate as follows:

$$F_t = F_c - F_1$$

$$F_1 = \alpha(W_1 + W_2) + F_p$$

$$= 7.07\ (1948 + 1340) + 23609$$

$$= 46855 \text{ lbs.}$$

α = 7.07 g
W_1 = 1948 lbs.
W_2 = 1340 lbs.
F_p = 23609 lbs.
(See Sect. 4.4.1.1) pressure loading between inner and outer closure head

$$F_t = 1.414 \times 10^6 - 46855$$

$$= 1.367 \times 10^6 \text{ lbs.}$$

Determine the compliance constant K for each plate

$$K_1 = \frac{3 \times 0.7 \times 5.3 \times 29.5^2}{16\pi \times 26.8 \times 10^6 \times 2.5^3} = 4.6017 \times 10^{-7} \text{ in/lb.}$$

$$K_2 = \frac{3 \times 0.67 \times 5.33 \times 29.5^2}{16 \pi \times 9.6 \times 10^6 \times 5^3} = 1.54566 \times 10^{-7} \text{ in/lb}$$

Now F_2 can be found from the previous equations as follows:

$$1.367 \times 10^6 = 1.54566 \times 10^{-7} F_2 \left[2.17313 \times 10^6 + 6.469728 \times 10^6\right]$$

$$F_2 = \frac{1.367 \times 10^6}{1.33589} = 1.023 \times 10^6 \text{ lbs.}$$

$$F_1 = \frac{K_2 F_2}{K_1} = \frac{1.54566 \times 10^{-7} \times 1.023 \times 10^6}{4.6017 \times 10^{-7}} = 343{,}614 \text{ lbs}$$

Bending stress in each plate is determined as follows:

Plate (1) $\sigma_1 = \frac{3\,(3 + v)}{8 \pi t^2} F = \frac{3\,(3 + .3)\,343{,}614}{8 \pi\, 2.5^2} = 21{,}656$ psi

Plate (2) $\sigma_2 = \frac{3\,(3 + v)}{8 \pi t^2} F = \frac{3\,(3 + .33)\,1023000}{8 \pi\, 5^2} = 16265$ psi

Calculate effective stress S_{e4} on Plate (1) and Plate (2).

Plate (1) highest stress area is at the center portion of the inner surface. From Sect. 1.1

$\sigma_x = 21656$ psi (radial stress)

$\sigma_y = 21656$ psi (tangential stress)

$\sigma_z = 8.35$ psi (axial stress)

$T_{xy} = T_{yz} = T_{zx} = 0$

$$S_{e4} = \sqrt{\tfrac{1}{2}}\sqrt{(21656 - 21656)^2 + (21656 - (-8.35))^2 + (-8.35 - 21656)^2}$$

$$S_{e4} = 21664.35 \text{ psi}$$

Allowable stress ($S_{aa} = 0.9\ S_u$) at 368°F for 304 S.S. From Sect. 1.1 under noncontainment structure and Sect. 1.2 equals 0.9 x 60500 = 54450 psi.

$$M.S. = \frac{54450}{21664.35} - 1 = 1.513$$

Plate (2) calculate effective stress at the center portion of the outer surface.

σ_x = - 16265 psi (radial stress)

σ_y = - 16265 psi (tangential stress)

σ_z = - 50000 psi (axial stress) compression stress under the pin.

$T_{xy} = T_{yz} = T_{zx} = 0$

$$S_{e4} = \sqrt{\tfrac{1}{2}}\sqrt{(-16265 - (-16265))^2 + (-16265 - (-50000))^2 + (-50000 - (-16265))^2}$$

$= 33735$ psi

Allowable stress ($S_{aa} = 0.9S_u$) at 250°F for 2024-T351. From Sect. 1.1 under noncontainment structure and Sect. 1.2 equals 0.9 x 49280 = 44352 psi.

$$M.S. = \frac{44352}{33735} - 1 = .315$$

Plate (2) calculate effective stress, S_{e4}, at the center portion of the inner surface:

σ_x = 16265 psi (radial stress)

σ_y = 16265 psi (tangential stress)

σ_z = -135 psi (axial stress)

$T_{xy} = T_{yz} = T_{zx} = 0$

Calculate the pressure stress of σ_z between Plate (1) and Plate (2).

1023000 = 1413716 - F - 9474

F = 381242 lbs.

$$\sigma_z = \frac{381232}{\pi\, 30^2} = 135 \text{ psi}$$

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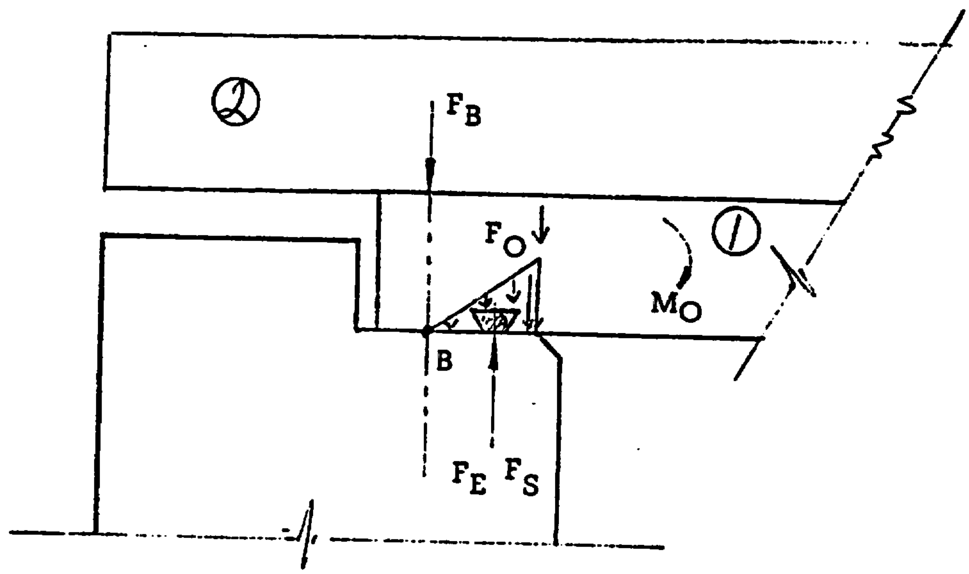
$$S_{e4} = \sqrt{\tfrac{1}{2}}\sqrt{(16265 - 16265)^2 + (16265 - (-135))^2 + (-135 - 16265)^2}$$

$$= 16400 \text{ psi}$$

$$M.S. = \frac{44352}{16400} - 1 = 1.7$$

The stresses calculated for the two plates are conservative because the edge moment provided by the bolts will reduce them further.

Closure Bolt Analysis



F_B = Bolt load @ 62" dia.

F_O = Inward load @ 59 " dia.

F_E = Force on clamp face @ 60" dia.

F_S = Seal Force @ 60" dia.

$\sum F$: $F_B + F_O - F_E - F_S = 0$

$\sum M_B$: $L_E F_E - L_O F_O + L_S F_S = M_O$

Calculating the edge slope of plate (1) = edge slope of plate (2).

The appropriate formula for edge slope θ for a simply supported plate, uniformly loaded is taken from Ref. 3, Table X, Case 1.

$$\theta = \frac{3(1-v)aF}{2\pi E t^3} = \frac{3(1-.3)\,29.5 \times 343614}{2\pi\, 26.8 \times 10^6 \times 2.5^3} = 0.0081 \text{ Rad.}$$

Due to the inward loading on the outer closure of 343614 lbs., assume the bolts will slightly yield.

Assume a bolt stress of S_{e4} = 87800 psi which is below 90% of the ultimate strength of 130000 psi (Sect. 1.2)

$$\epsilon_t = \frac{\sigma}{E} + \frac{\sigma - \sigma_E}{E_p}$$

Where ϵ_t = Total strain

σ_E = Elastic limit

E_p = Plastic Modulus (straight line basis)

E = Elastic modulus

σ = Actual Stress

From Ref. 25

$E = 27.3\ (10^6)$ psi

Bolt stress = 80,000 psi @ .0002 plastic strain

$.0002 = (80{,}000 - \sigma_E)/E_p$ and $.002 = (85{,}000 - \sigma_E)/E_p$

Solving for E_p and σ_E a conservative estimate can be obtained by approximating E_p with a straight line through the .02% and .2% strain points.

$\sigma_E = 79{,}445$ psi , $E_p = 2.78 \times 10^6$ psi

ϵ_T = 87800 /(27.3) 10^6 + (87800 - 79,445)/(2.78) 10^6

ϵ_T =0.00622 in/in

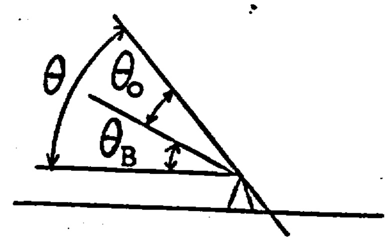
ϵ_{clamp} = clamp stress/(27.3) (10^6)

= 61978 /(27.3)(10^6)

= 0.00227 in/in

ϵ_{net} = 0.00622 - 0.00227.

= 0.00395 in/in



$$\theta - \theta_B = \theta_O$$

$$\theta_B = \frac{\Delta_B}{1.5}$$

θ = edge slope of a simply supported plate

θ_B = the angle at the maximum bolt elongation

θ_O = the angle due to uniform edge moment.

Δ_B = Bolt length x ϵ_{net}

= 2.625 x 0.00395

= 0.01037 in

$$\theta_B = \frac{0.01037}{1.5} = 0.00691 \text{ Rad.}$$

θ_o = 0.0081 - 0.00691 = .00119 Rad.

For the two plates in the outer closure

$M = M_1 + M_2$ and $\theta = \theta_1 = \theta_2$ since edge slopes must be equal.

$$M_1 = \frac{E_1 t_1^3 \theta_1}{12 (1-v_1) R_1} = \frac{(26.8 \times 10^6)(2.5)^3 \theta_1}{12(1 - 0.33)\, 29.5}$$

$M_1 = 1{,}689{,}870\ \theta_1$

$$M_2 = \frac{E_2 t_2^{\,3} \theta_2}{12(1-v_2)R_2} = \frac{(9.6 \times 10^6)\ (5^3)\ \theta_2}{12(1 - 0.33)\ 29.5}$$

$$M_2 = 5{,}062{,}906\ \theta_2$$

$$M_o = [1{,}689{,}870 + 5{,}062{,}906]\ \ 0.00119 = 8036 \text{ in-lb/in}$$

$$\Sigma F : F_B - F_E - F_S + F_o = 0$$

$$\Sigma M_B : L_E F_E - L_o F_o + L_S F_S = M_o$$

$$F_B = \frac{87800 \times 0.950 \times 28}{62\ \pi} = 11{,}990 \text{ lbs/in}$$

$$\Sigma F = 11{,}990 - F_E - 100 + 7{,}357 = 0$$

$$F_E = 19247 \text{ lbs/in}$$

$$\Sigma M_B = 1 \times 19{,}247 - 1.5 \times 7{,}357 + 1 \times 100 = 8{,}036 \text{ in-lbs/in}$$

$$8{,}311 \text{ in-lbs/in} = 8{,}036 \text{ in lbs/in}$$

This is sufficiently close agreement, so the assumed bolt stress of 87,800 psi is correct.

Allowable stress ($S_{aa} = 0.9S_u$) for outer closure bolts from Sect. 1.1 under Noncontainment Structure Criteria and Sect. 1.2 equals:

$$0.9 \times 130{,}000 = 117{,}000 \text{ psi}$$

$$M.S. = \frac{117{,}000}{87{,}800} - 1 = .332$$

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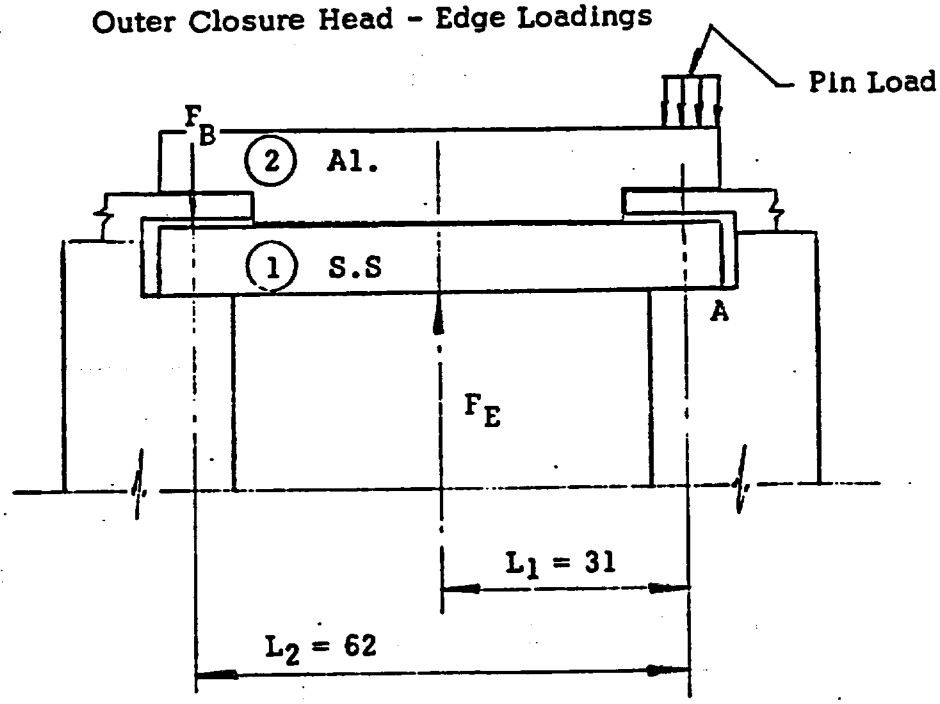
4.8.1.1 Outer Closure Head - Edge Loadings



Acceleration during the puncture impact is = 7.07 g (Sect. 4.8.1)

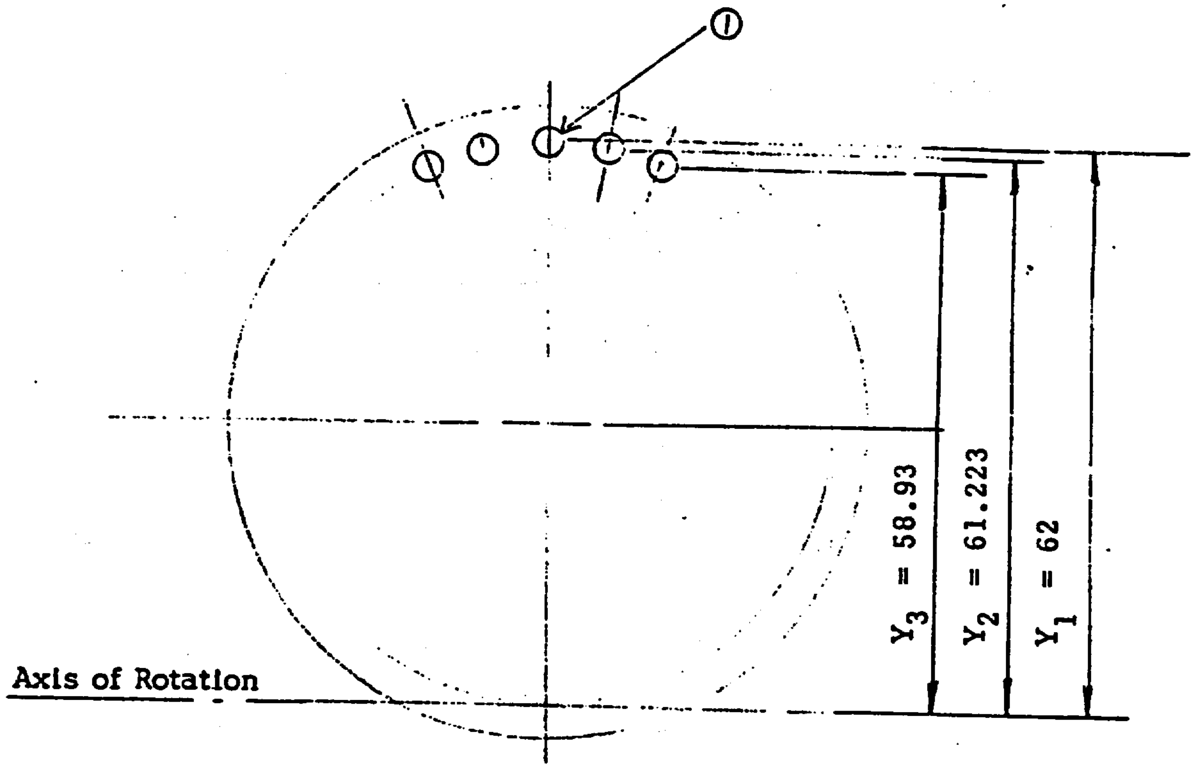
The load that the outer closure head bolts will have to support is the inertial load of the S/S plate and the pressure loading between the inner and outer closure. See sketch in section 4.4.1.1 for clarification.

Plate (1) weight = 2328 lbs.

Pressure loading = 23609 lbs (Sect. 4.4.1.1)

F_E = 7.07(2328) + 23609

= 40068 lbs.



Force on any bolt, $F_B = \frac{yF_o}{2R}$, where F_o is Force on bolt (1) and y is distance from axis of rotation to each bolt.

Applied moment, $L_I F_E = y_1 \frac{y_1 F_o}{2R} + 2y_2 \frac{y_2 F_o}{2R} + 2y_3 \frac{y_3 F_o}{2R}$

$$= \frac{F_o}{2R} \left(y_1^2 + 2y_2^2 + 2y_3^2 \right)$$

$$31\,(40068) = \frac{F_o}{2(31)} \left[62^2 + (61.223^2) + 2(58.93^2) \right]$$

$$1242108 = \frac{F_o}{62} \, (18286)$$

$$F_o = \frac{1242108}{295} = 4210 \text{ lbs.}$$

Conservatively assumes only 5 bolts resist moment out of 28 bolts

Minimum preload on the bolts from Sect 3.11 is 10,450 lbs. This clamping force of 10,450 is higher than F_o = 4210 lbs. Hence no added stress will be applied to the bolts during the puncture edge loading

$$M.S. = \frac{10450}{4210} - 1 = 1.48$$

The outer closure plate will take the bending stress due to its inertial forces (See Sect. 4.4.1.1)

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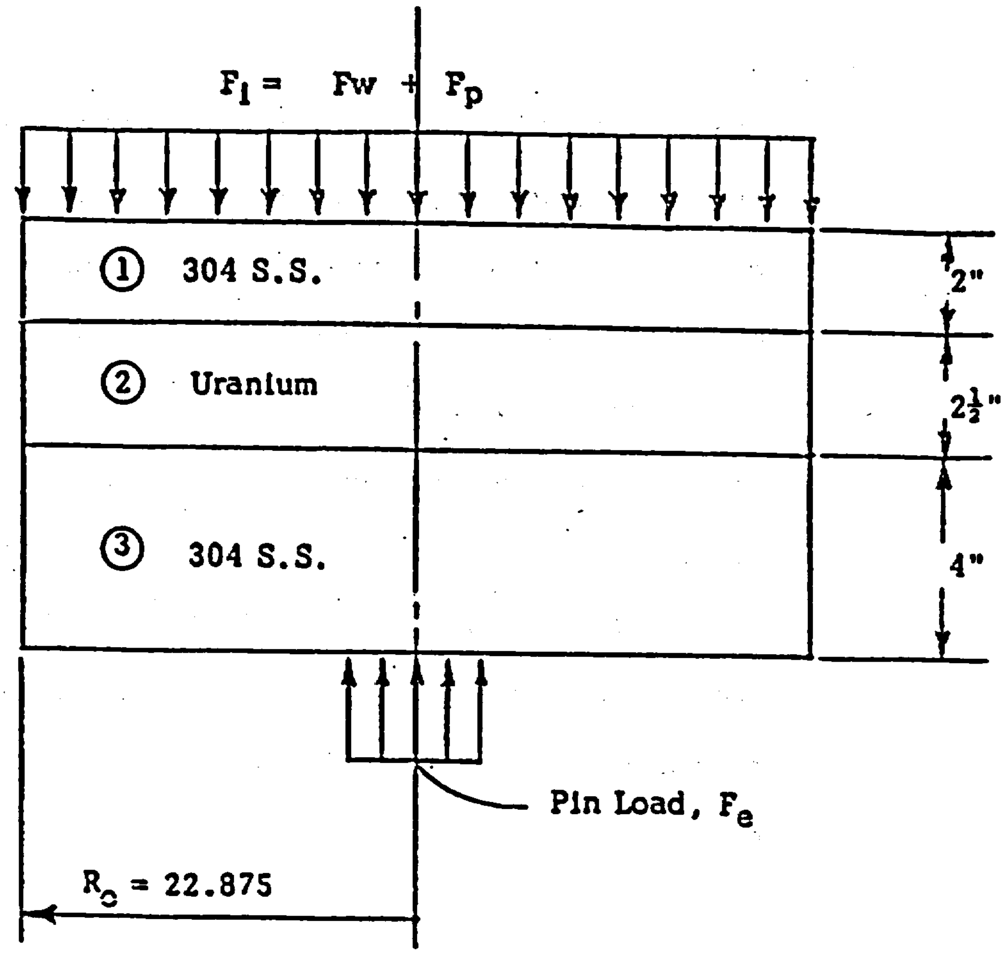
4.8.2 Bottom Head

Integrity of the bottom head must be maintained in the hypothetical puncture accident to provide the required containment of the cask contents. The maximum possible load that could be imposed on the outer closure during the puncture impact would be the failure load of the puncture pin itself.

From Ref. 66, the pin failure load has been established to be 50000 psi. From this value the largest force that can be developed by the puncture pin is $F_e = (50000)(\pi)(3^2) = 1.413716(10^6)$ lb. With this force the maximum possible cask acceleration during the puncture impact is $\delta = (1.414)10^6/200,000 = 7.07$ g. The design load of 200,000 lbs. equals the gross weight of the cask.



Term	① Inner S.S.	② Uranium	③ Outer S.S.
Temp. ^oF	410	410	410
E, psi	26.6×10^6	24.8×10^6	26.6×10^6
v	0.3	0.22	0.3
Weight lbs.	1240	2900	2260

The three plates are constrained to have the same elastic deflection curves under lateral bending loads. Hence, the deflection of each plate must be the same and the total lateral load supplied to the assembly can be divided among the individual plates in accord with each one's proportionate part of the total bending resistance. The appropriate formulas for deflection and maximum stress are taken from Reference 3, Table X Case 1 (for plate 1,2)

Center deflection, $y_c = \frac{3\,(1-v)\,(5+v)\,R_o^2}{16\,\pi\,E\,t^3}\,F = KF$

Max. stress at center, $\delta_c = \frac{3\,(3+v)}{8\,\pi\,t^2}\,F$

Ref. 3, Table X, Case 6 (For Plate 3)

Center deflection, $y_c = \frac{3\,(1-v^2)\,R_o^2}{16\,\pi\,E\,t^3}\,F = KF$

Max. stress at edge, $\delta = \frac{3}{4\pi\,t^2}\,F$

Equating the center deflection of the three plates gives

$$K_1 F_1 = K_2 F_2 = K_3 F_3$$

Also, the total load imposed on the closures must equal the sum of the individual plate loads, so that

$$F_t = F_1 + F_2 + F_3$$

Combining these equations in terms of outer plate loads F_3, gives

$$F_t = K_3 F_3 \left(\frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} \right)$$

These equations may be evaluated to obtain the force on each plate as follows:

$$F_t = F_e - F_i$$

$$F_i = \alpha F_w + F_p$$

Worst case conditions exist with minimum internal force which occurs with PWR fuel and only helium in the containment vessel.

$F_p = Ap$ $\qquad$ $A = \pi(22.875^2) = 1644 \text{ in.}^2$

$p =$ 16.45 psig (helium pressure only and PWR fuel).

$F_p = 1644 \times 16.45 = 27044$ lbs.

$F_w = 38000$ lbs. (PWR loading)

$$F_t = F_e - (\alpha F_w + F_p)$$

$$= 1.414 \times 10^6 - (7.07 \times 38{,}000 + 27{,}044)$$

$$F_t = 1.118 \times 10^6 \text{ lbs.}$$

Following the same method of analysis as used in Section 3.9, the compliance constant K for each plate is as follows:

$K_1 = 5.44472 \times 10^{-7}$ in./lb.

$K_2 = 3.28146 \times 10^{-7}$ in./lb.

$K_3 = 1.66937 \times 10^{-8}$ in./lb.

Now F_3 can be determined as follows:

$$F_t = K_3F_3 \left(\frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} \right)$$

$$1.118 \times 10^6 = 1.66937 \times 10^{-8} F_3 \; (1.8366 \times 10^6 + 3.04742 \times 10^6 + 5.99028 \times 10^7)$$

$$F_3 = \frac{1.118 \times 10^6}{1.08153} = 1{,}033{,}721 \text{ lbs.}$$

$$F_1 = \frac{K_3F_3}{K_1} = \frac{1.66937 \times 10^{-8} \times 1{,}033{,}721}{5.44472 \times 10^{-7}} = 31{,}694 \text{ lbs.}$$

$$F_2 = \frac{K_3F_3}{K_2} = \frac{1.66937 \times 10^{-8} \times 1{,}033{,}721}{3.28146 \times 10^{-7}} = 52{,}588 \text{ lbs}$$

Bending stress in each plate is determined as follows:

Plate (1) $\sigma_1 = \frac{3\,(3 + v)}{8\pi t^2} F_2 = \frac{3 \times 3.3}{8\pi\, 2^2} \times 31{,}694 = 3{,}121 \text{ psi}$

Plate (2) $\sigma_2 = \frac{3\,(3 + v)}{8\pi\, t^2} F_2 = \frac{3 \times 3.22}{8\pi\, 2.5^2} \times 52{,}588 = 3{,}234 \text{ psi}$

Plate (3) $\sigma_3 = \frac{3}{4\pi t^2} F_3 = \frac{3}{4\pi\, 4^2} \times 1{,}033{,}721 = 15{,}424 \text{psi}$

Calculating effective stresses on Plates (1), (2), (3)

The formula for effective stress equals

$$S_e = \sqrt{\frac{1}{2}} \sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6\left(T_{xy}^2 + T_{yz}^2 + T_{zx}^2\right)}$$

Where σ_x, σ_y, σ_z are normal stress, and T_{xy}, T_{yz}, T_{zx} are shear stresses (Sect. 1.1)

Plate (1)

The highest stress area is at the center portion of the inner surface

σ_x = 3121 psi (radial stress)

σ_y = 3121 psi (tangential stress)

σ_z = -158 psi (axial stress)

$T_{xy} = T_{yz} = T_{zx} = 0$

$$S_{e3} = \sqrt{\tfrac{1}{2}}\sqrt{(3121 - 3121)^2 + (3121 - (-158))^2 + (-158 - 3121)^2}$$

= 3279 psi

Allowable stress (.8 $S_{aa} = 0.7S_u$) at 410°F for 304 S/S. From Sec. 1.1 under containment vessel and Sect. 1.2 equals 0.7 x 59500 = 41650 psi

$$M.S. = \frac{41650}{3279} - 1 = 11.7$$

Plate (2)

The highest stress area is at the center portion of the inner surface

σ_x = 3234 psi (radial stress)

σ_y = 3234 psi (tangential stress)

σ_z = -177 psi (axial stress)

Calculating pressure stress σ_z between Plate (1) and Plate (2) :

31694 = F - 8768 - 250,456

F = 290,918 lbs.

$$\sigma_z = \frac{290,918}{1,644} = 177 \text{ psi}$$

$$S_{e4} = \sqrt{\tfrac{1}{2}}\sqrt{(3234 - 3234)^2 + (3234 - (-177))^2 + (-177 - 3234)^2}$$

= 3411 psi

Allowable stress ($S_{aa} = 0.9\ S_u$) at 410°F for uranium from Sect. 1.1 under non-containment structure and Sect. 1.2 equals 0.9 x 56,000 = 50,400 psi

$$M.S. = \frac{50,400}{3,411} - 1 = 13.7$$

Plate (3)

Calculate effective stress S_{e4} at the edge portion of the outer surface.

σ_x = 15424 psi (radial stress)

From Ref. 3, Table X, case 6 $\sigma_y = \frac{3 v F}{4 \pi t^2}$

$$\sigma_y = \frac{3 \times 0.3 \times 1,033,721}{4 \pi 4^2} = 4627 \text{ psi}$$

σ_z = 0 (axial stress)

$T_{xy} = T_{yz} = 0$

$$T_{zx} = \frac{1,033,721}{45.75 \pi \ 4} = 1798 \text{ psi}$$

$$S_{e4} = \sqrt{\tfrac{1}{2}} \sqrt{(15424 - 4627)^2 + (4627 - 0)^2 + (0 - 15424)^2 + 6(1798)^2}$$

= 14058 psi

Allowable stress ($S_{aa} = 0.9 \ S_u$) at 410°F for 304 S.S. from Sect.1.1 under non-containment structure and Sect. 1.2 equals 0.9 x 59,500 = 53,550 psi

$$M.S. = \frac{53,550}{14,058} - 1 = 2.81$$

Plate (3)

Calculate effective stress S_{e4} at the center portion of the outer surface.

From Ref. 3 Table X, case 6 - Bending stress at the center of the plate.

$$\sigma_3 = \frac{(1 + v)(3)(F)}{8 \pi t^2}$$

$$\sigma_3 = \frac{(1 + .3)(3)(1,033,721)}{8 \pi 4^2} = 10,026 \text{ psi}$$

σ_x = -10026 psi (radial stress)

σ_y = -10026 psi (tangential stress)

σ_z = -50,000 psi (axial stress) compression stress under the pin.

$T_{xy} = T_{yz} = T_{zx} = 0$

$$S_{e4} = \sqrt{\tfrac{1}{2}} \sqrt{(-10026 - (-10026))^2 + (-10026 - (-50000))^2 + (-50000 - (-10026))^2}$$

= 39,974 psi

$$M.S. = \frac{53,550}{39,974} - 1 = .340$$

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4.8.3 Cask Shell

The greatest damage from a drop of 40 inches upon a 6 in. 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 a result of a plastic hinge.

The maximum possible load that could be imposed on the cask during the puncture impact would be the failure load of the puncture pin itself. See Sect. 4.8.2 where

$$F_c = (1.414 \times 10^6) \text{ lbs.}$$

The analysis and the characteristics of the containment structure are as follows:

Inner Shell:

Outside diameter = 46.5 in.

Thickness = 0.75 in.

Temperature = 410 $^{\circ}$F (Sect. 3.1)

Material - Stainless Steel, Type 304

Modulus of Elasticity, $E = 26.6 \times 10^6$ psi

Outer Shell:

Outside diameter = 62.5 in.

Thickness = 2 in.

Temperature = 353 $^{\circ}$F (Sect. 3.1)

Material - Stainless Steel, Type 304

Modulus of Elasticity, $E = 26.9 \times 10^6$ psi

Water Jacket Shell:

Outside diameter = 82 in.

Thickness = 0.75 in.

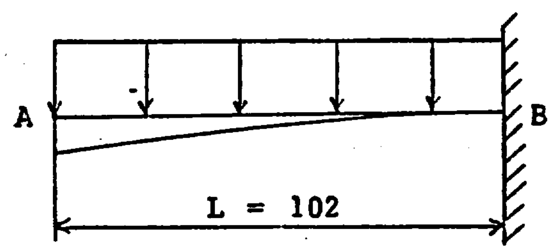
Temperature = 326^{o} F (Sect. 3.1)

Material - Stainless Steel, Type 304

Modulus of Elasticity, $E = 27 \times 10^6$ psi

Calculating the bending stresses in the cask, the cask is treated as a cantilever with uniform load. The moments are shared by the three shells in proportion to their stiffness because they are all constrained to have equal end slopes by the very stiff end forgings.

$W = 1.414 \times 10^6$ lbs.



For a cantilever beam, uniform load from Ref. 3, Table III, Case 3

$$\text{Max. } M = -\frac{1}{2} WL \text{ at } B$$

$$M = \frac{1.414\ (10^6)\ (102)}{2} = 7.2114\ (10^7) \text{ in.-lbs.}$$

Inner Shell EI = 7.4906 (10^{11}) lb.-in^2 (Sect. 4.6.2)

Inner Shell Section Modulus = 1211.2 in^3 (Sect. 4.6.2)

Outer Shell $EI = 4.6753\ (10^{12})\ lb.\text{-}in^2$ (Sect. 4.6.2)

Outer Shell Section Modulus $= 5561.7\ in^3$ (Sect. 4.6.2)

Water Jacket $EI = 3.1936\ (10^{12})\ lb.\text{-}in^2$ (Sect. 4.6.2)

Water Jacket Section Modulus $= 3846.5\ in^3$ (Sect. 4.6.2)

Dividing total moment among three shells in proportion to their values of EI gives the following:

Total Effective $EI = 8.61796\ (10^{12})\ lb.\text{-}in^2$

$$\text{Inner Shell } M = \frac{7.4906\ (10^{11})\ (7.2114\ (10^7))}{8.61796\ (10^{12})}$$

$$= 6.268\ (10^6)\ \text{in.-lbs.}$$

$$\text{Outer Shell } M = \frac{4.6753\ (10^{12})\ (7.2114\ (10^7))}{8.61796\ (10^{12})}$$

$$= 3.912\ (10^7)\ \text{in.-lbs.}$$

$$\text{Water Jacket Shell } M = \frac{3.1936\ (10^{12})\ (7.2114\ (10^7))}{8.61796\ (10^{12})}$$

$$= 2.672\ (10^7)\ \text{in.-lbs.}$$

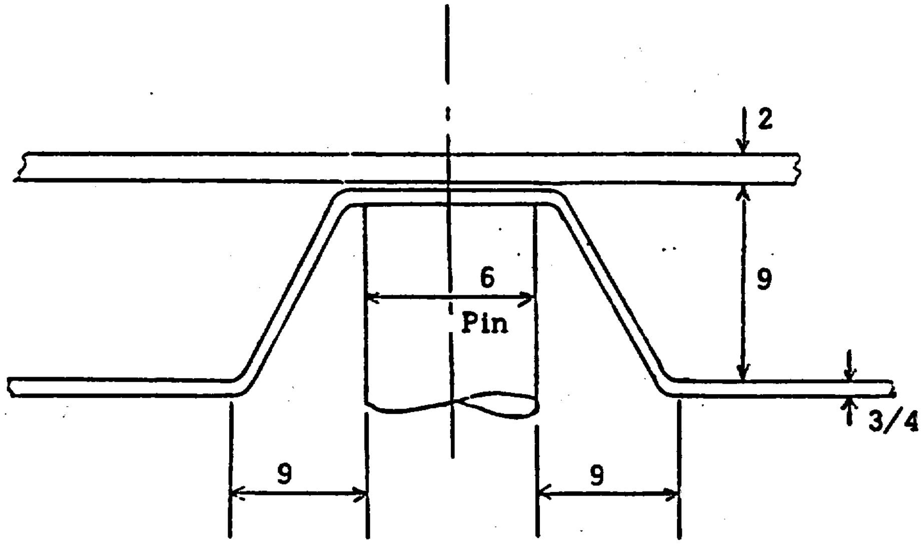
Shell bending stresses then, are as follows:

$$\text{Inner Shell, } \sigma = \frac{6.268\ (10^6)}{1211.2} = 5175 \text{ psi}$$

$$\text{Outer Shell, } \sigma = \frac{3.912\ (10^7)}{5561.7} = 7034 \text{ psi}$$

$$\text{Water Jacket Shell, } \sigma = \frac{2.672\ (10^7)}{3846.5} = 6946 \text{ psi}$$

Calculate the force required for indentation of Water Jacket and the force on the pin from hydraulic pressure.



Assume only 152 inches of 156 inches of Water Jacket is stretched radially. The wall is .75 inches thick with a 80.5 inch inside diameter. Assume the 6 inch diameter pin pushed the Jacket wall 9 inches to contact the outer cask shell. Very conservatively, a volume of water, equal to

$$v = 0.2618\,h\ (D^2 + Dd + d^2)$$

$$= 0.2618(9)\ (24^2 + 24(6) + 6^2)$$

$$= 1781.3\ in^3 \text{ is displaced as shown}$$

The volume of 1781 in^3 will distend the Jacket.

$$\Delta r = \frac{1781.3\ in^3}{\pi 80.5\ (152)} = .04634 \text{ inches}$$

The strain due to hoop stress is $2\ \pi(.04634) = .2912$ inches

The hoop stress is $\frac{.2912}{\pi\,80.5} \times E$ $\quad E = 27 \times 10^6$ (Sect. 2.1)

$$\text{Hoop Stress} = \frac{.2912}{\pi\,80.5}\ 27 \times 10^6 = 31089 \text{ psi OK} < 41333 \text{ psi}$$

Hydraulic pressure developed is

$$S = P\frac{R}{t};\ 31089 = P\ \frac{40.25}{.75};\ P = \frac{31089}{53.6} = 580\ \text{psi}$$

Bursting Pressure is $P_u = 2S_u\ \frac{b - a}{b + a}$ $\qquad a = 40.25$

$b = 41$

$$P_u = 2\ (62000)\ \frac{41 - 40.25}{41 + 40.25} = 1145\ \text{psi}$$

Force on pin required to shear .75 inch

Area of 6 inch diameter pin 28.27 in^2

Area of Jacket in shear = $.75\,\pi\,6$ = 14.14 in^2

Shearing stress = .75 (62000) = 46500 psi

Force on the pin to shear = 14.14 in^2 (46500) = 657510 lbs.

Maximum force on the pin from hydraulic pressure is:

Area of 24 inch Annulus (max.) = 424.12 in^2

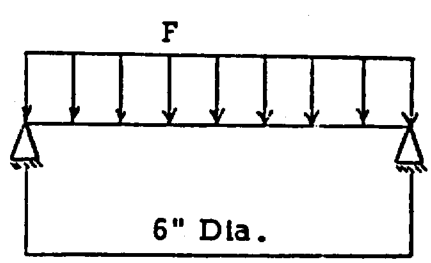
F max. = (424.12)(580) = 245990 lbs.

The Jacket does not shear, but deforms.

<u>Force on the pin required for indentation of Water Jacket.</u>

Calculate the force on the pin required to plastically deform the Water Jacket Shell over the 6" Dia. circle.

No credit is taken for the force to stretch the shell, or bending of the fins.



The limit load to plastically deform the 6" Dia plate, from Ref. 59 for a plate edges supported uniform load over entire surface equals

$$P_e = 6\pi M_p$$

$$M_p = \frac{t^2}{4} y_s$$

$$M_p = \frac{.75^2}{4} (41333)$$

$$= 5813 \text{ lbs.}$$

$$P_e = 6\pi (5813)$$

$$= 109572 \text{ lbs.}$$

Dynamic yield strength for 304 S/S at 326° F equals 41333 psi

t = .75 in.

Adding the force from the hydraulic pressure to the force required to deform the 6" Dia. plate equals

$$F_t = 245990 + 109572$$

$$= 355562 \text{ lbs.} < 657510 \text{ lbs.}$$

The Jacket still does not shear, but deforms.

Net force left on the pin just before it starts to puncture the Water Jacket equals to

$$F_{net} = 1.414 (10^6) - (355562)$$

$$= 1.0584 (10^6) \text{ lbs.}$$

Calculating the shear force required to penetrate the outer shell. In order to penetrate the outer shell, the pin will have to go through the Water Jacket Shell, three layers of fins and the Outer Shell.

Water Jacket Shell UTS = 62000 psi at 326°F (Sect. 3.1, 2.1)

$$\text{Shear A} = 6\,\pi\,.75$$

$$= 14.14 \text{ in}^2$$

$$\text{Shear Force} = .75\,(62000)(14.14)$$

$$= 657510 \text{ lbs.}$$

External Fins UTS = 62000 psi (Sect. 3.1, 2.1)

$$\text{Shear A} = .1875\,\pi\,6$$

$$= 3.54 \text{ in}^2$$

$$\text{Shear Force} = .75\,(62000)(3.54)$$

$$= 164610 \text{ lbs.}$$

Internal Fins UTS = 69120 psi at 340°F (Sect. 3.1, 2.1)

Two layers of internal Fins

$$\text{Shear A} = .1875\,(2)\,(\pi)\,(6)$$

$$= 7.08 \text{ in}^2$$

$$\text{Shear Force} = .75\,(69120)(7.08)$$

$$= 367027 \text{ lbs.}$$

Net force left on the pin just before it starts to puncture the fins on the outer shell

$$F_{net} = 1.0584\ (10^6) - \left[657510 + 164610 + 183514\right]$$

$$= 52766 \text{ lbs.}$$

The pin is now in contact with the fins on the outer shell with a force of 52766 lbs. This force is not sufficient to shear thru the fins since the force required to shear one fin thickness equals 183,514 lbs.

This force of 52,766 lbs. creates a pressure stress on the outer shell equal to $52766/3^2\pi = 1866$ psi. This pressure stress plus the bending stress resulting from the 7.07g (Sect. 4.8.2) acceleration loading during the puncture accident is much less severe than the stresses due to the 81g side impact (Sect. 4.6.2). The analysis in Sect. 4.6.2 shows that the integrity of the shell structure is maintained. Therefore, during the puncture accident, integrity of the cask is maintained.

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4.9 Fire Accident Condition

Following exposure to the hypothetical fire accident specified in 10 CFR Part 71 (par. 71.36 and Appendix B), fuel and cask temperatures reach their maximum values under any transport condition as a result of loss of coolant and water from the neutron shield. The pressure in the inner containment vessel likewise increases to its maximum value.

4.9.1 Determination of pressure in containment vessel

Fuel temperature (PWR fuel) 824° F Avg. (Sect. 4.1.)

$$P = \frac{.498\,(1545)\,(824 + 460)}{76.8\,(144)}$$ (See Sect. 3.3.1)

P = 89.3 psia or 74.6 psig

If fuel rods do not fail, pressure is based on fuel temperature at 824° F avg. The containment vessel initially filled with helium at 68°F and one atmosphere pressure.

$$P = \frac{(824 + 460)\,(14.7)}{(68 + 460)} = 35.7 \text{ psia or } 21 \text{ psig}$$

Determination of pressure in containment vessel for BWR type fuel

$$P = \frac{.467\,(1545)\,(824 + 460)}{64\,(144)} = 100.5 \text{ psia or } 85.8 \text{ psig}$$

If fuel rods do not fail, pressure is based on fuel temperature at 824^{o} F avg. The containment vessel initially filled with helium at 68^{o}F and one atmosphere pressure

$$P = \frac{(824 + 460)(14.7)}{(68 + 460)} = 35.7 \text{ psia or } 21 \text{ psig}$$

Summary of Internal Pressure Conditions

Normal Post Fire Pressure

@	PWR Temperatures	21 psig
@	BWR Temperatures	21 psig

Post Fire Pressures - Fuel Failure

PWR Fuel	74.6 psig
BWR Fuel	85.8 psig

4.9.2 Radial Thermal Expansion of Aluminum Basket (PWR/BWR)

Sect. 3.8 describes a two dimensional axial calculation to determine stresses and displacements in the ends of the cask body using the ANSYS program. From this analysis it is possible to determine the radial displacement of the inner shell at any axial position. At the center of the cask, the radial displacement of the inner shell is calculated to be .116" outward. This displacement is for the post fire condition.

In order to eliminate or reduce radial interference that could result from differential thermal expansion of the aluminum basket and the inner shell of the containment vessel, a radial clearance of 0.105 in. has been provided. An analysis of the radial expansion and resulting interference stresses follows:

<u>Aluminum Basket</u>

Average temperature, $T_a = 655^{o}F$ (Post fire area weighted average)

Coefficient of thermal expansion, $\alpha_a = (14.40(10^{-6})$ in./in./^{o}F

Outside radius, $R_a = 22.395$ in.

$\Delta R_a = R_a \alpha_a \Delta T_a = 22.395(14.40)(10^{-6})(655 - 68) = 0.1893$ in.

Radial clearance, $L_g = 0.105$ in.

For the fire accident condition, a comparison of Radial Thermal Expansions can be made.

In summary:

Aluminum Basket = .1893 in. = ΔR_a

Inner Shell = .116 in. = ΔR_5

Radial Clearance = .105 in = L_G

Since $L_G + \Delta R_5 > \Delta R_a$, no interference exists

Material Creep of Aluminum Basket

The material behavior of the aluminum basket at temperatures at the post fire environment are considered here and the consequence of a deformation/creep phenonemon. Before computing the strain growth characteristics of the basket it is necessary to ascertain the semantics involved.

Creep takes place when a strong, continuous membrane force exists within the aluminum. Usually the membrane force causing creep and the possible eventual stress rupture is of a non self limiting characteristic, such as an induced mechanical load or a pressure stress.

When a thermal stress is present in a free-standing structural form, such as the basket, a stress situation exists that is clearly self-limiting. A self-limiting stress provides that the induced stress level will be lowered as material strain takes place. This characteristic of lowered stress levels and the general subsidization of strains in the self-limiting stress application is known as relaxation.

Relaxation takes place with the aluminum basket, as there are no strong internal membrane forces in the basket, hence there are low resulting creep strains.

Peripheral elements of the aluminum basket have been tabulated in Tables I and II for both the PWR and BWR case. The element numbers, temperatures, and resulting final strain (EPGEN) has been reduced from ANSYS Computer Output. The temperatures at which strain data is tabulated is slightly less than the fire accident temperature. Since strain is a linear function dependent upon temperature, a new strain at the fire-accident temperature can be determined by

$$\epsilon_2 = \frac{\Delta_{T\,(f)}}{\Delta_{T\,(c)}} \; (\epsilon_1)$$

where,

ϵ_2 = Fire Accident Element Strain

$\Delta_T(f)$ = Delta Temperature to fire accident temperature

$\Delta_T(c)$ = Delta Temperature at Computer Run

ϵ_1 = Computer Output Strain.

Average Peripheral Strain in the BWR Basket = .000579

Since $\Delta_T (f) = 655 - 68 = 587^{\circ}F$

$\Delta_T (c) = 550 - 68 = 482^{\circ}F$

$$\epsilon_2 = \frac{587}{482} (.000579) = .00071$$

As can be seen from the curve on "Stress Relaxation on Aluminum Basket," the .00017 strain encountered in the outer peripheral regions of the basket are low. A lower limit on relaxation stresses is desirable in order to compute elemental strain in which the transient creep is treated as an instantaneous strain. To predict relaxation an instantaneous drop in stress from σ_o to σ_1 is given by

$$\frac{(\sigma_o - \sigma_1)}{E} = K \sigma_1^{\,b}$$ Ref: 54, page 123

Dealing with strain relaxation

$$\frac{\sigma_o - \sigma_1}{E} = \epsilon_R$$

By knowing the strain growth, ϵ_2, the material constant K for the aluminum basket can be computed. Say relaxation takes place as to allow the full thermally induced strain to be removed, such that

$$\epsilon_2 = \epsilon_R = .00071$$

Some material constants for aluminum at 100°C and 2,000 hours are as follows:

$m = .10$

$b = 5.2 \times 10^{-5} = .000052$

$n = .385$

} Table 2-1, Page 12 Ref. 54

for $\epsilon = .00071$

$\sigma = 1170$ See chart on Stress Relaxation on Aluminum Basket

$$K = \frac{.00071}{(1170)^{.000052}} = .00071$$

Determine the equivalent relaxation stress that will be present over the 1000 hour recovery time to induce a complete peripheral creep flow in the basket.

From 4.9.2 the remaining gap is = (.11776 + .105 - .1893) = .033

Returning to creep strain equation that indicates the stress level for a full creep flow into the gap region during the 1000 hour recovery time.

$$\epsilon_c = K\sigma^n t^m$$ Ref. 54 pg. 117

$$\sigma^n = \frac{\epsilon_c}{Kt^m}$$

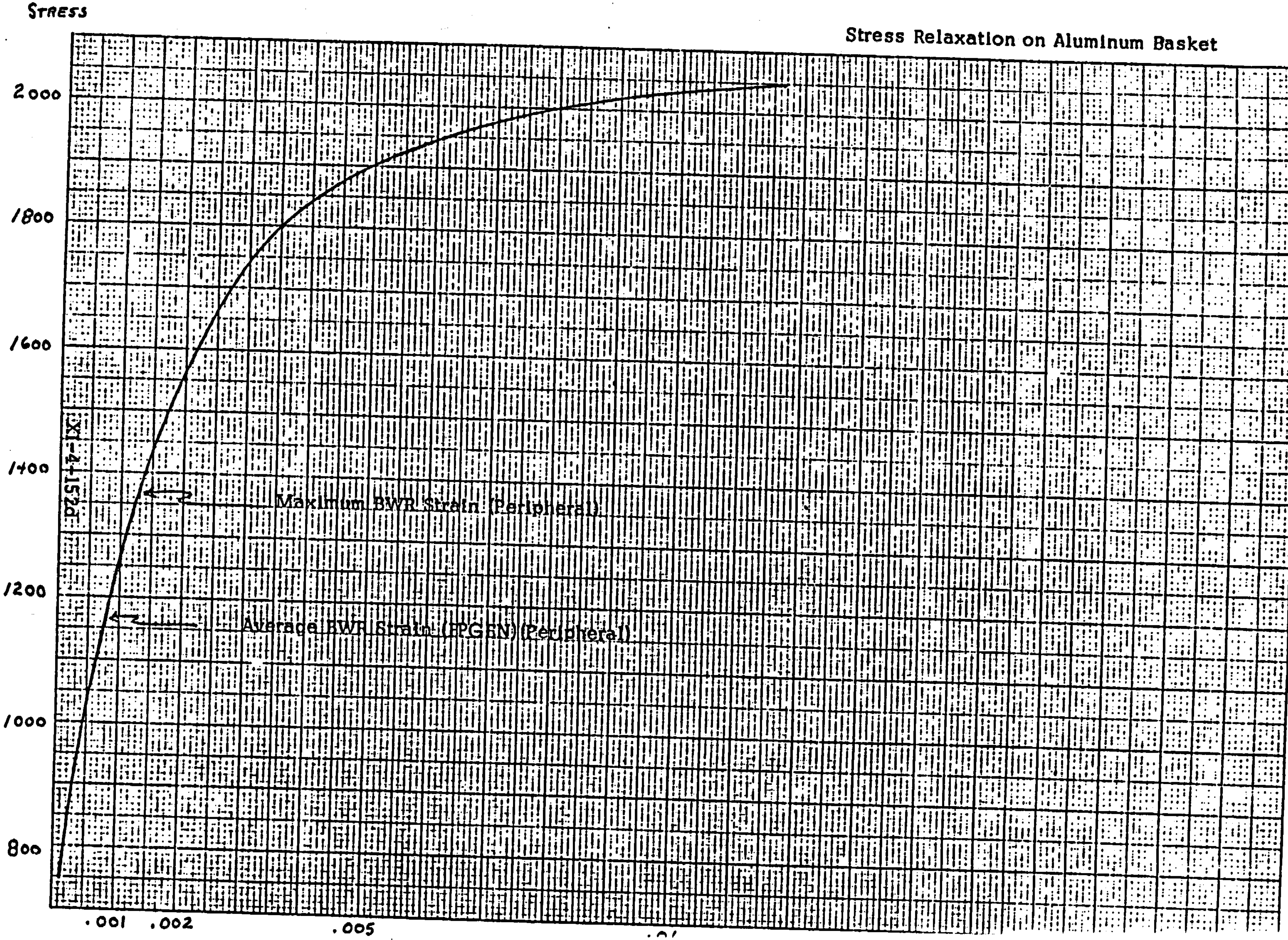
$$\sigma^n = \frac{.033}{(.00071)(1000)^{.10}} = \frac{.033}{.0014}$$

$$\sigma = \left[23.3\right]^{\frac{1}{n}} = 3,561 \text{ psi}$$

The previous strain - relaxation computations indicates that a stress of 3,561 is necessary to creep the basket structure into the .033

remaining gap that exists after thermal growth. This required high stress is greatly in excess of a peripheral stress existing on either BWR or PWR basket.

K&E 10 X 10 TO THE ½ INCH 359-11
KEUFFEL & ESSER CO. MADE IN U.S.A.



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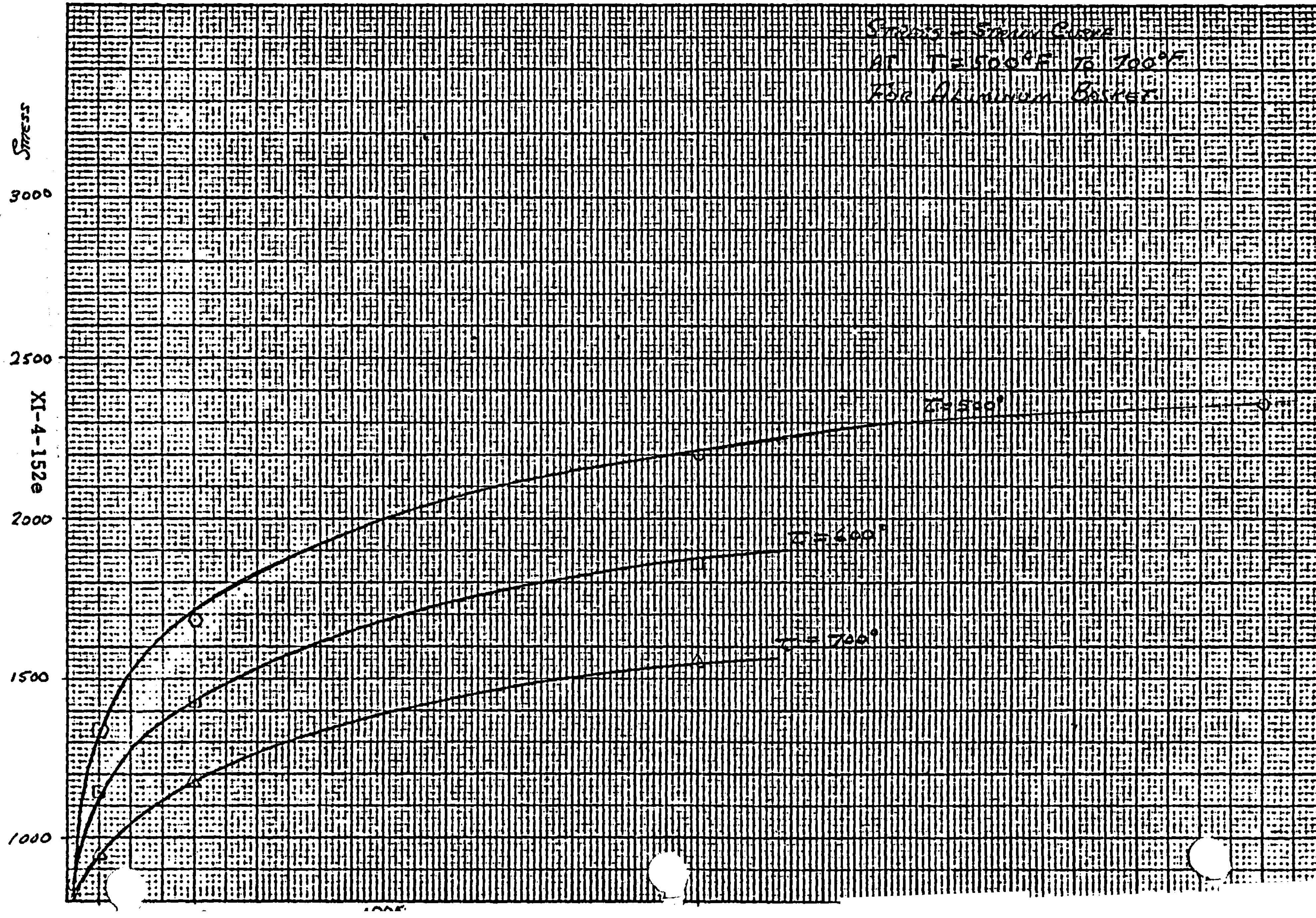
Stress - Strain Curve
At T=500°F to 700°F
For Aluminum Basket
Stress
3000
2500
2000
1500
1000
T=500°
T=600°
T=700°

XI-4-152e

PERIPHERAL PWR BASKET ELEMENTS

TABLE I

El No	Temp	EPGEN	El No	Temp	EPGEN
4	569	.000214	278	575	.00031
10	570	.000223	284	575	.000295
16	570	.000248	293	575	.000291
20	570	.000277	303	573	.000271
25	572	.000308	308	573	.000278
32	573	.000401	316	575	.0003405
38	576	.0004968	336	577	.0004978
42	579	.0006248	343	577	.000573
121	583	.0006566	366	574	.000233
129	582	.000299	371	573	.000203
147	577	.000162	376	572	.0001855
152	577	.0001501			
173	577	.000179			
202	577	.000158			
211	576	.000186			
224	575	.000484			
229	575	.000734			

PERIPHERAL BWR BASKET ELEMENTS

TABLE II

El No	Temp	EPGEN	El No	Temp	EPGEN
62	549	.00107	373	548	.000072
71	549	.000778	395	547	.00029
76	547	.00061	422	544	.00131
85	546	.00060	433	545	.00094
94	545	.00073	444	547	.000664
105	546	.0011	449	550	.00079
167	550	.0014	454	552	.00121
205	553	.00044			
218	553	.000325			
230	553	.00027			
257	552	.000196			
294	552	.0001786			
312	551	.0001240			
329	550	.0000993			
338	549	.0000746			
360	548	.0000597			

$$\text{Ave EPGEN} = \frac{.01333}{23}$$

$$= .000579$$

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4.9.3 Axial Thermal Expansion - PWR Basket

The Table on page XI-4-154 compares various relative expansion values of the cask components. From this table the dimensions of each component and the associated gap between components are determined for both conditions of the fuel containing either Zircaloy-4 or stainless steel guide tubes.

It can be seen from the figure on page XI-4-155 that under post-accident conditions minimal gaps exists between the spacer plug and the underside of the closure head and the top of the fuel basket and the spacer plug. This arrangement provides adequate fuel support yet does not impose thermal expansion forces on the internal structure or the containment vessel.

Relative Elongations:

Gap between aluminum basket and steel retention sleeve - Cold gap = 0.75

$$\Delta_G = \Delta_2 - \Delta_3 = 0.9947 - 1.318 = -.3233 \text{ in.}$$

Zircaloy-4 Rods -

Gap between aluminum spacer plug and under side of closure head -

Cold gap = .875

$$\Delta_G = (\Delta_4 + \Delta_7) - (\Delta_1 + \Delta_5 + \Delta_6)$$

$$\Delta_G = (.710 + .0602) - (.411 + .0228 + .1067) = +.2297$$

Stainless Steel Rods -

$$\Delta_G = (\Delta_4 + \Delta_7) - (\Delta_{15} + \Delta_5 + \Delta_6)$$

PWR CONFIGURATION - POST FIRE CONDITION

Item	Length	Temp. °F	ΔT °F	∝ mean in/in/°F	Material	Elongation
Δ_{15} Fuel Element (1) (4) (5)	160.515	938	870	10.2×10^{-6}	S.S. - 304	$\Delta_{15} = 1.424$
Δ_1 Fuel Element (1)(4)(5)	160.515	938	870	*2.95×10^{-6}	Zr. -4	$\Delta_1 = .411$
Δ_2 Absorber Sleeve	162.00	637	619	9.92×10^{-6}	S.S. Type 304	$\Delta_2 = .9947$
Δ_3 Aluminum Basket	155.9375	655	587	14.4×10^{-6}	Al. (Gr. 1180)	$\Delta_3 = 1.318$
Δ_4 Containment Shell(3)	156.000	535	467	9.75×10^{-6}	S.S. Type 304	$\Delta_4 = .710$
Δ_5 Lower Support Structure	4.125	628	560	9.86×10^{-6}	S.S. Type 304	$\Delta_5 = .0228$
Δ_6 Upper Spacer Plug (2)	13.895	628	560	14.25×10^{-6}	Al.(5052-H32)	$\Delta_6 = .1067$
Δ_7 Top Forging	11.500	601	533	9.82×10^{-6}	S.S. Type 304	$\Delta_7 = .0602$

Notes:

(1) Fuel element length is to the point where spacer compression less pick up hard point of the fuel element.

(2) Difference in spacer plug length shown here at 12.375 in. and SAR calculations at 12.000 in. is considered insignificant.

(3) Calculations for containment shell are based on axial expansion of 2 in. outer shell, A length = 156 in. is used here.

(4) The guide thimbles control fuel growth due to temperature.

(5) Irradiated length

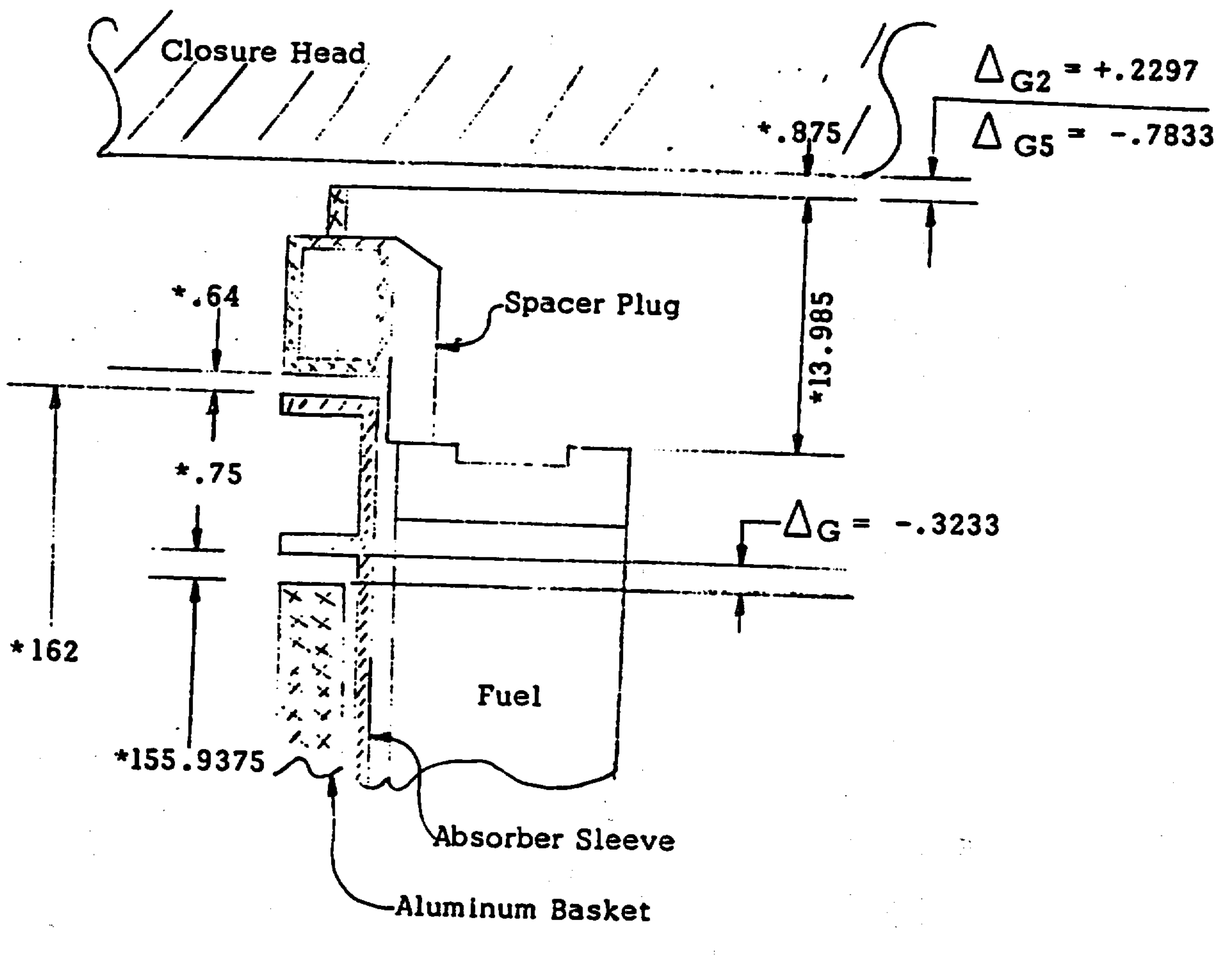
* Generally accepted data by most manufacturers . Ref. 30 indicates α values vs. temperature of a slightly lower value. The above figure of alpha represents a more conservative approach.

$$\Delta_G = (.710 + .0602) - (1.424 + .0228 + .1067) = -.7833$$

The worst case is with stainless steel guide tubes. However, a gap still exists in the post fire condition (.875 - .7833 = .0917 gap)

Thermal Expansion of PWR Basket

Gap summary for fuels containing either zircaloy - 4 or stainless steel guide tubes



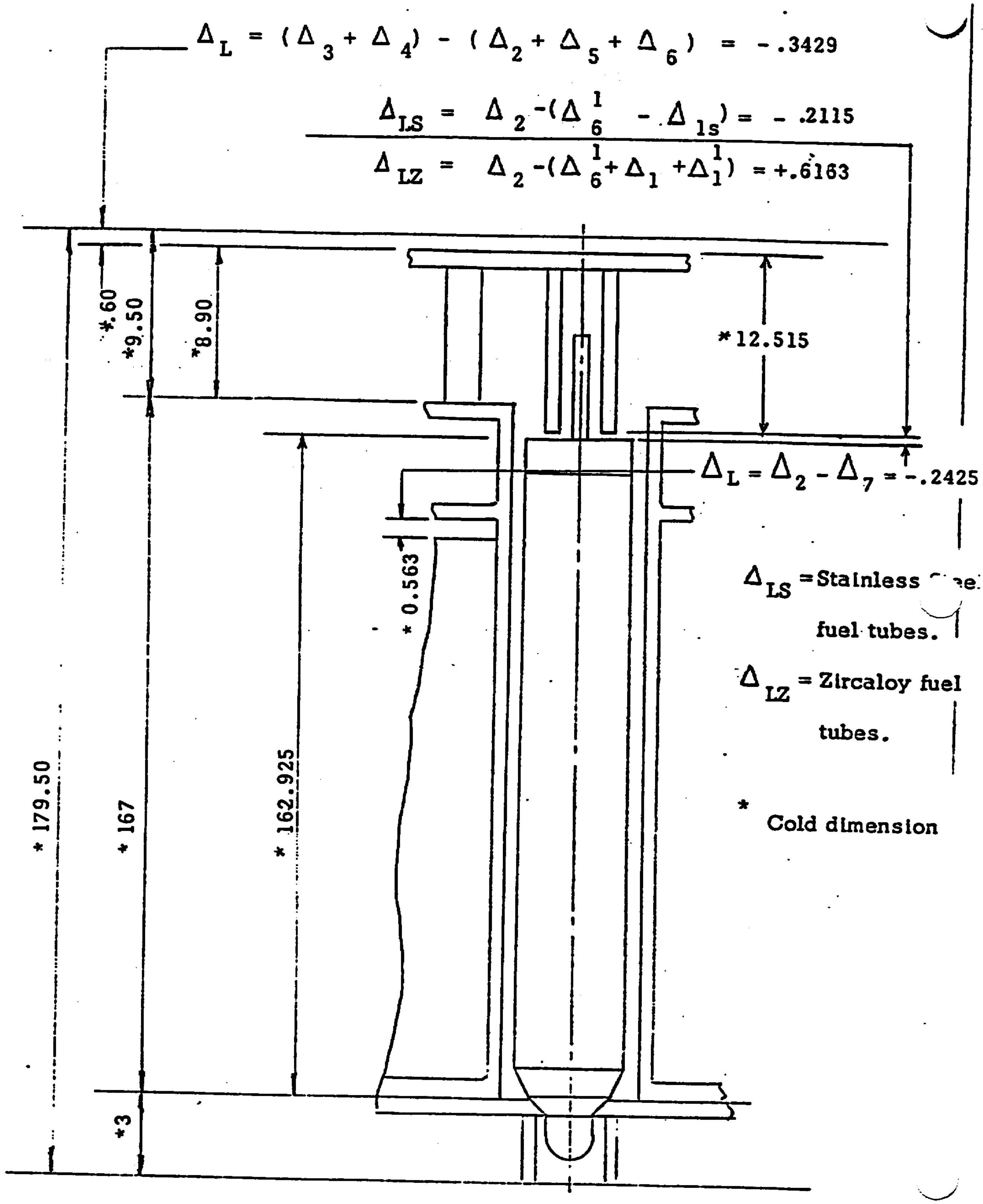
* Cold dimensions

4.9.3.1 Axial Thermal Expansion - BWR Basket

The Table on Page XI-4-158 compares various relative expansion values of the cask components. From this table, the dimensions of each component and the associated gap between components are determined for both conditions of fuel containing either Zircaloy-4 or stainless steel fuel tubes.

It can be seen from the figure on Page XI-4-157 that under post accident conditions minimal gaps exist between the spacer plug and the underside of the closure head and the fuel and the spacer plug. This arrangement provides adequate fuel support yet does not impose thermal expansion forces on the internal structure or the containment vessel.

BWR - Axial Expansion



	Length in.	Temp. °F.	Δ T °F.	(mean) α in/in °F.	Material	Elongation
Fuel Element (4)	162.925	805	737	10.06×10^{-6}	S.S. Type 304	$\Delta_{1s} = 1.208$
Fuel Element (1)(3)	157.890	805	737	$^{*}2.95 \times 10^{-6}$	Zr.-4	$\Delta_{1} = 0.343$
Fuel Element (2)	5.035	805	737	10.06×10^{-6}	S.S. Type 304	$\Delta_{1}^{1} = 0.0373$
Absorber Sleeve	167	687	619	9.92×10^{-6}	S.S. Type 304	$\Delta_{2} = 1.0255$
Containment Shell	156	535	467	9.75×10^{-6}	S.S. Type 304	$\Delta_{3} = 0.710$
Top Forging	11.5	601	533	9.82×10^{-6}	S.S. Type 304	$\Delta_{4} = 0.0602$
Lower Support Structure	3	628	560	9.86×10^{-6}	S.S. Type 304	$\Delta_{5} = 0.0166$
Spacer Plug	8.9	628	560	14.25×10^{-6}	Al.5052-H32	$\Delta_{6} = 0.071$
Difference In Spacer Arm Length	3.615	628	560	14.25×10^{-6}	Al.5052-H32	$\Delta_{6}^{1} = 0.0289$
Aluminum Basket	150	655	587	14.4×10^{-6}	Al. Gr. 1180	$\Delta_{7} = 1.268$

(1) Active Portion of Fuel

(2) Inactive Fuel Portion

(3) Fuel Element Length Reflects a 3/4 in. growth due to irradiation.

(4) Stainless Steel Fuel Tubes

* See Ref. 30. A maximum value is used here in order to obtain a conservative elongation value.

4.9.4 Thermal Expansion of Lead Shield

Appendix B derives the equations and outlines the method of calculation to determine the interference stress between the inner and outer shell and the lead shield. The results of the Appendix B calculations are then used in the ANSYS program to compute the combined stresses in the cask shells which is presented in section 4.9.6