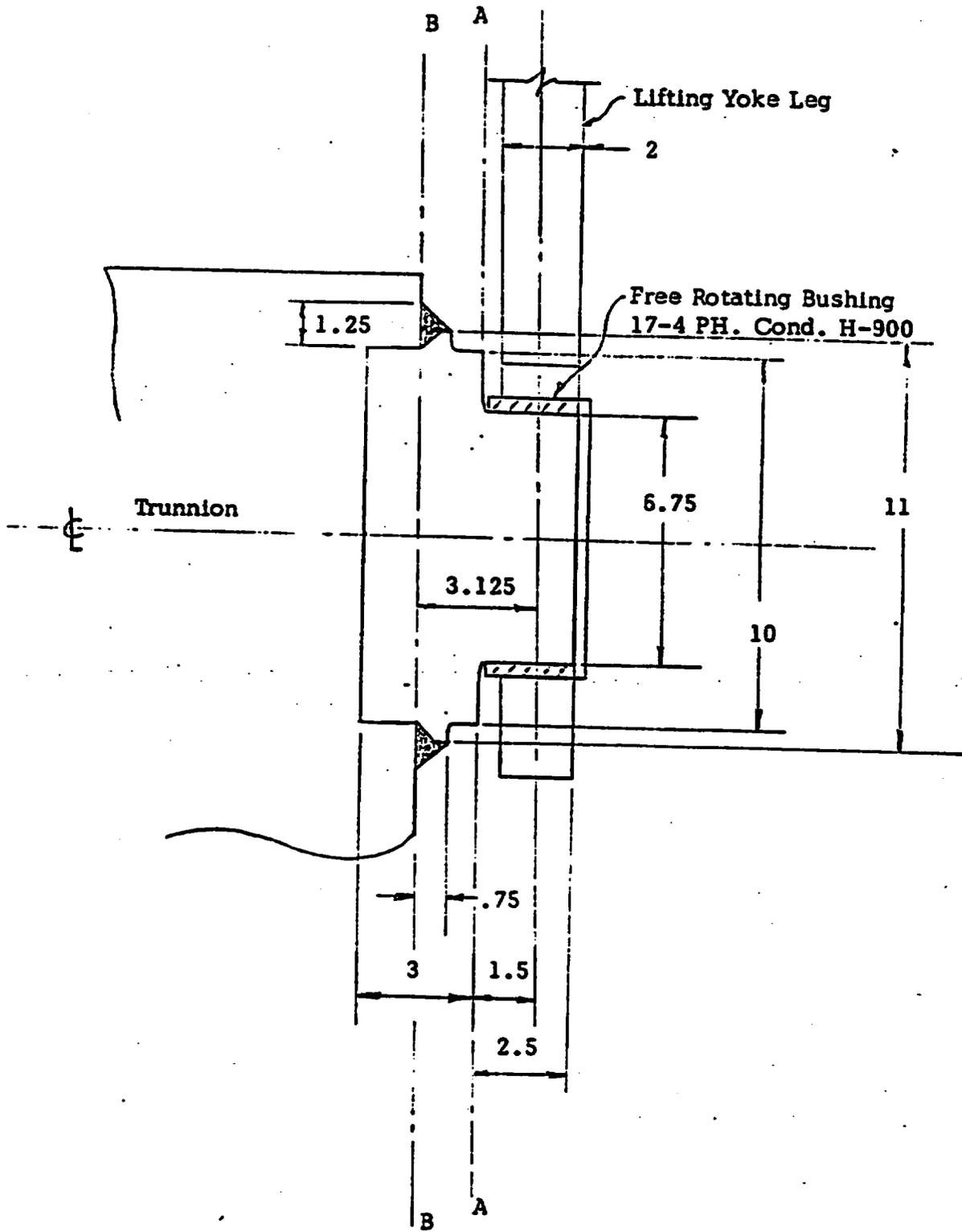


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TRUNNION DETAIL

Section B-B Weld - 45° conical throat

$$\begin{aligned} \text{Throat area of weld} &= t \pi d \\ &= (.707 \times 1.25) \pi (10 + \frac{1.25}{2}) = 29.5 \text{ in.}^2 \end{aligned}$$

Z of throat area of weld

$$\begin{aligned} Z &= \pi r^2 t \text{ of thin cylinder} \\ &= A \left(\frac{r}{2} \right) = 29.5 \left(\frac{10.625}{4} \right) = 78.36 \text{ in.}^3 \end{aligned}$$

Bending moment $M_3 = 300,000 (3.125) = 937,500 \text{ in. lbs.}$

$$S_B = \frac{M_B}{Z} = \frac{937,500}{78.36} = 11,964 \text{ psi.}$$

$$S_S = \frac{300,000}{29.5 + \frac{\pi}{4} 10^2} = \frac{300,000}{108.04} = 2777 \text{ psi}$$

Combined tension stress

$$\begin{aligned} S_t &= \frac{11964}{2} + \sqrt{5982^2 + 2777^2} \\ &= 5982 + 6595 = 12,577 \text{ psi} \end{aligned}$$

$$\text{M.S.} = \frac{24750}{12577} - 1 = .968$$

Section A-A 6.75 dia. trunnion section at root

$$\text{Area of 6.75 dia.} = 35.78 \text{ in.}^2$$

$$Z = .098 (6.75)^3 = 30.14 \text{ in.}^3$$

$$\text{Bending Moment } M_B = 300,000 (1.5) = 450,000 \text{ in. lbs.}$$

$$S_b = \frac{450,000}{30.14} = 14930 \text{ psi}$$

Shear stress

$$S_s = \frac{300,000}{35.78} = 8384 \text{ psi}$$

Combined stress

$$S_t = \frac{14930}{2} + \sqrt{74652 + 8384^2} = 18691 \text{ psi.}$$

$$\text{M.S.} = \frac{24,750}{18691} - 1 = .324$$

2.1.2 Outer Closure Lifting Lugs

Outer Closure Head weight is 2438 lbs. Design weight 3000 lbs.

Design load = 3 x 3000 = 9000 lbs.

Outer closure head is provided with three threaded eye bolts. The eye bolts are removed during shipment and the tapped holes in the closure head are fitted with threaded plugs.

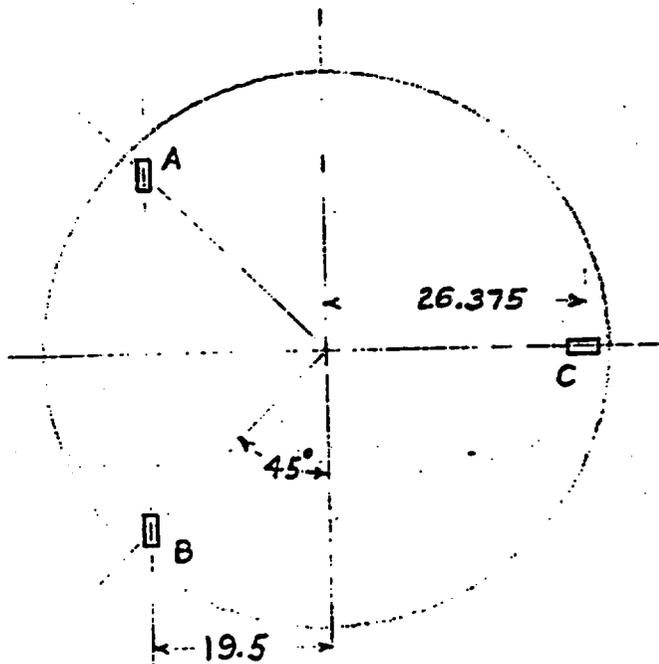
The eye bolts are 3/4" (shank) diameter, each rated at a strength of 23,400 lbs. which is approximately five times the design load carried by one eye bolt taking half the design load.

2.1.3 Inner Closure Lifting Lugs

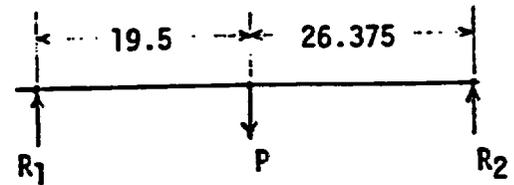
Inner closure design weight is 7400 lb., (sect. VII).

Design load at 3g, $P = 3(7400) = 22200$ lb.

The inner closure is provided with three threaded eye bolts, which are inserted into tapped holes in the closure for handling. Closure head will be handled by a lift rig having lift points located at the same dimensions from its center of lift as the closure head eye bolts are located from the center of the head.



Lifting force model:



Sum forces:

$$R_1 + R_2 = P = 22200$$

Moments about R_1 :

$$19.5 P = 45.875 R_2$$

Solving for eye bolt forces:

$$R_2 = 22200 (19.5/45.875) = 9437 \text{ lb.}$$

$$R_1 = 22200 - 9437 = 12763 \text{ lb.}$$

$$\text{eye bolts A and B, } F_A = F_B = 12763/2 = \underline{6382} \text{ lb.}$$

$$\text{Eye bolt C, } F_C = \underline{9437} \text{ lb}$$

The minimum rated tensile strength of the 1 inch diameter eye bolts is 46850 lb., which is nearly 5 times the design load of 9437 lb. on the most heavily loaded eye bolt.

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XI-2-6

2.2 Tie-Downs

The tie-down system consists of (1) cask lugs with mating plate anchorages welded to the railcar center sill to take vertical upward and longitudinal loads, and (2) V saddles to take vertical downward and lateral loads.

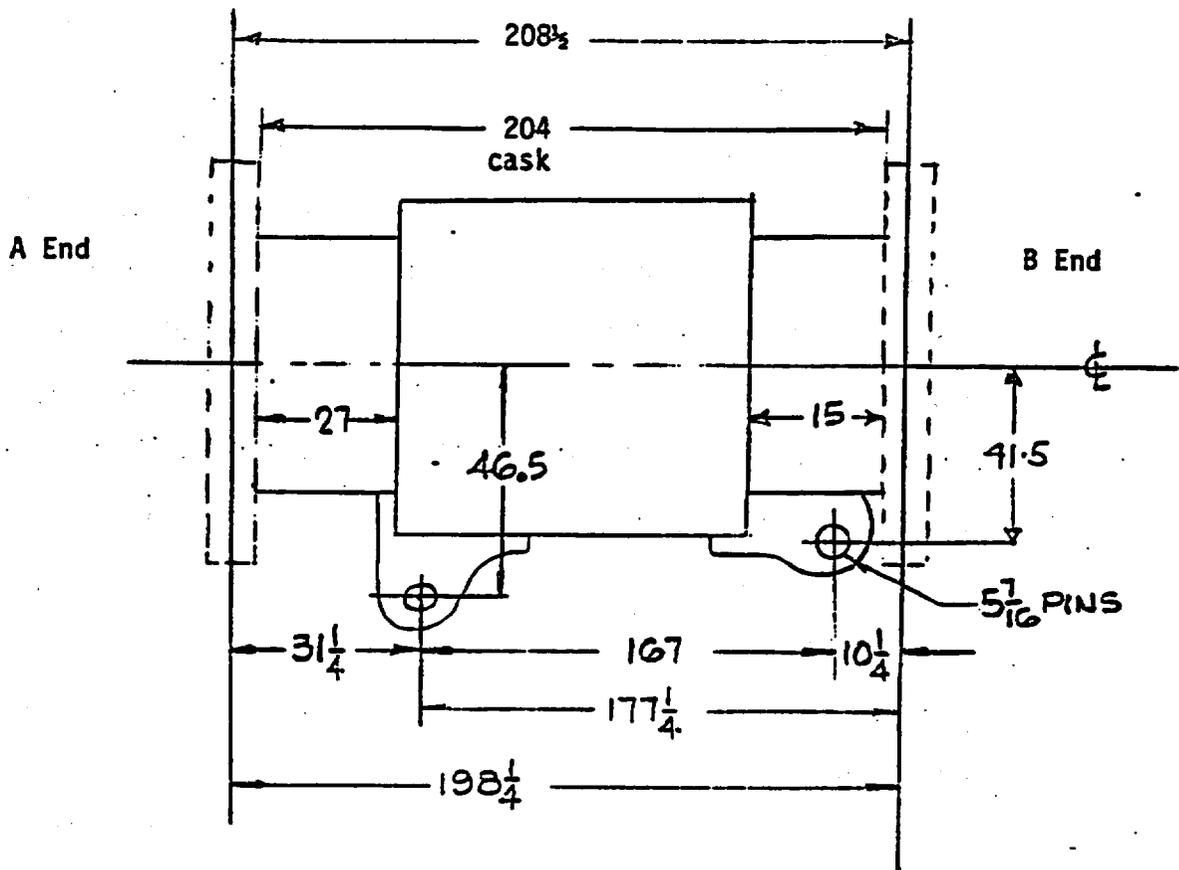
The rear cask lugs take the entire 10g longitudinal loads in both directions, thus allowing the front cask lug to be free from longitudinal loads so that clearances can be provided longitudinally for cask/railcar relative expansions.

The lugs and pins of the cask at both ends are designed for combined loads of 10g longitudinally, 5g transversely, and 2g vertically, in both directions, without exceeding 90% of the yield point stress.

The cask lugs are 17-4 ph material and the plates of the anchorage are T-1 steel, both at a nominal 100,000 psi Y.P. Welds between the bottom of the plate weldment and the mounting blocks on the center sill itself are calibrated to break at a load less than the actual strength of the tie-down unit of lugs and plates, but greater than the minimum specified design loads.

The cask weight rests on and between the 45⁰ flat saddle bearing plates (which form a 90⁰ V support). Low friction wear bearing linings are provided as part of the saddle construction, thus allowing axial motion between the saddle cover plate and the edge of the impact structure which supports the entire cask weight and vertical reactions.

The front tie-down has 1 cask lug and 2 side plates, while the rear tie down, designed for 10g longitudinal, has 2 cask lugs and 3 side plates spaced alternately. Both have 5 7/16 in. dia. pins.



Analysis of Applied Loads (Specified by 10 CFR - Part 71.31 (d))

10g Longitudinal = $10 \times 220,000 = 2,200,000$ lbs at rear tie down.

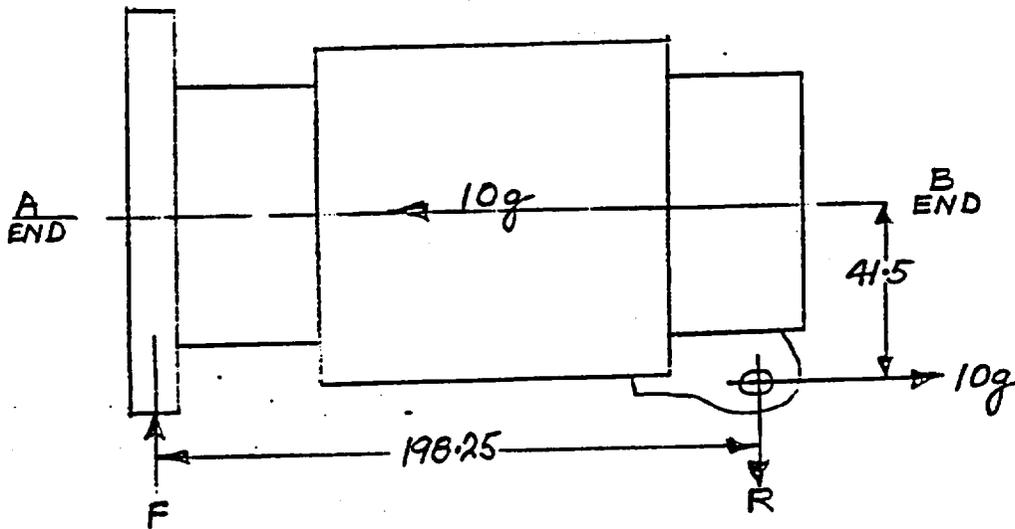
5g Transverse = $\frac{5 \times 220,000}{2} = 550,000$ lbs at front and rear saddles

2g Vertical = $\frac{2 \times 220,000}{2} = 220,000$ lbs at front & rear tie-down or saddle.

Each of the above loads can be applied in either + or - directions.

The particular combinations which produce the maximum loads on a certain structure are developed in the following cases.

- (1) A end Impact Car - 10g forward on cask



Pitching couple at F and R

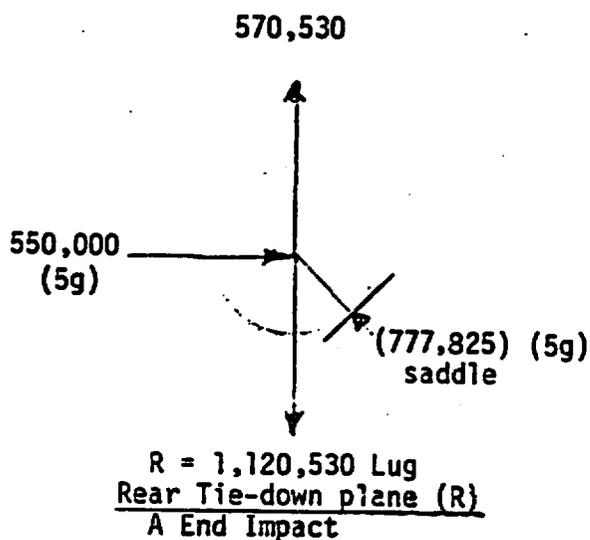
$$+F = \frac{2,200,000 \times 41.5}{198.25} = 460,530 \text{ lbs.}$$

Verticals for rear tie down at R

- ↑ 460,530 pitching 10g
- ↑ 220,000 2g V
- ↓ 110,000 1g static wt.
- ↑ 570,530 Net tension

Verticals for front support at F

- ↓ 460,530 pitching 10g
- ↓ 220,000 2g V
- ↓ 110,000 1g static wt.
- ↓ 790,530 Net Compression



Rear tie-down cask lug and pin, net loads

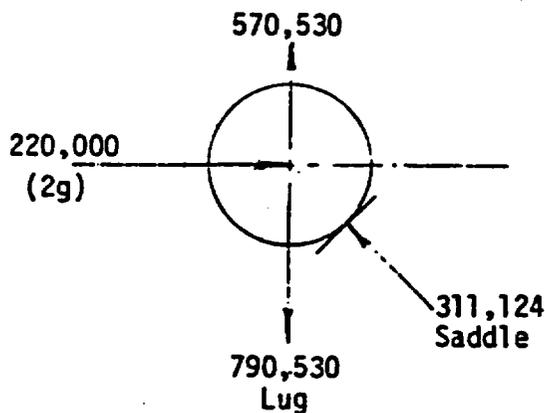
1,120,530 lbs. ↑ vertical, tension

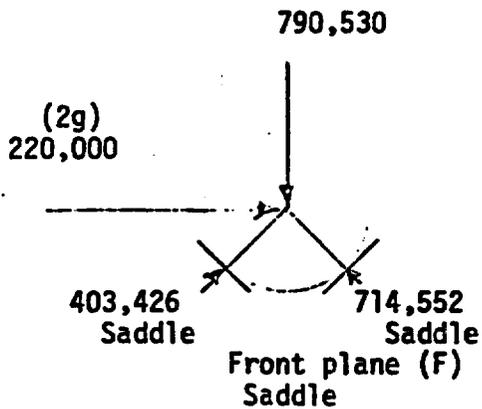
2,200,000 lbs. ← toward A end.

2,468,924 lbs. ↘ resultant

Front tie-down is not loaded with 10gL acting toward A end.

Front and Rear saddles are designed for 2g transverse instead of 5g, since the whole package of car and cask is unstable at about .5g, and 2g design loading provides adequate large margin of safety.



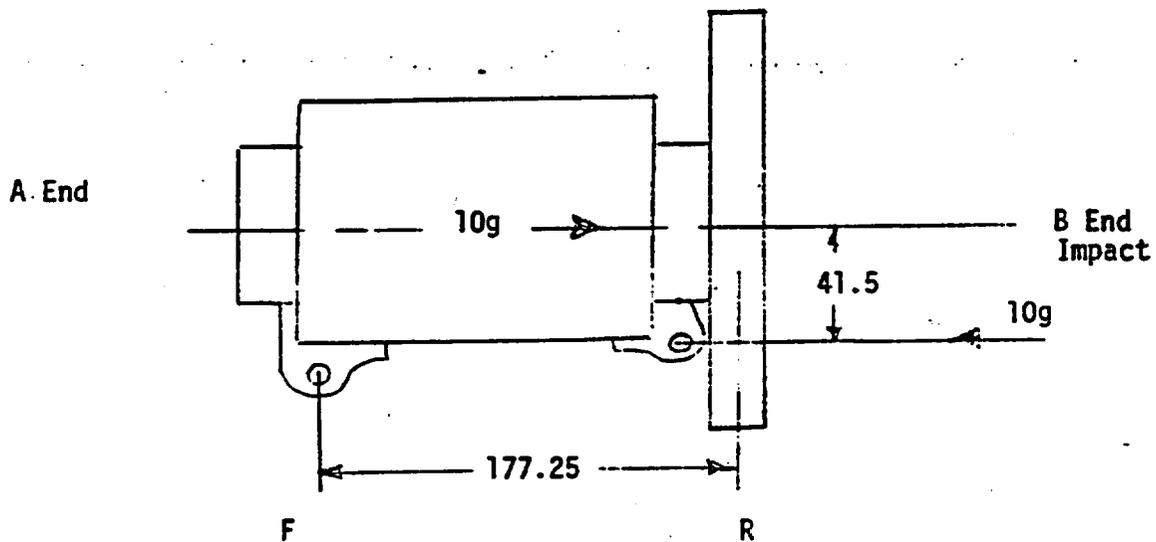


- ↓ 460,530 pitching
- ↓ 220,000 2g V
- ↓ 110,000 1g static wt.
- ↓ 790,530 net compression

A End Impact

(2) For 10g L acting toward B end of car and bottom of cask

B end Impact

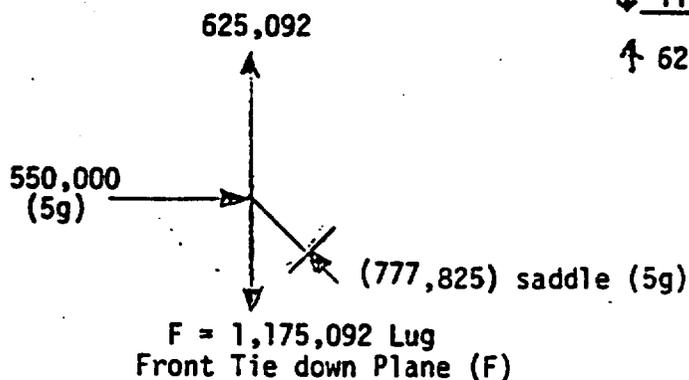


Pitching couple at F and R.

$$+F = \frac{2,200,000 \times 41.5}{177.25} = 515,092 \text{ lbs.}$$

For front tie-down at F

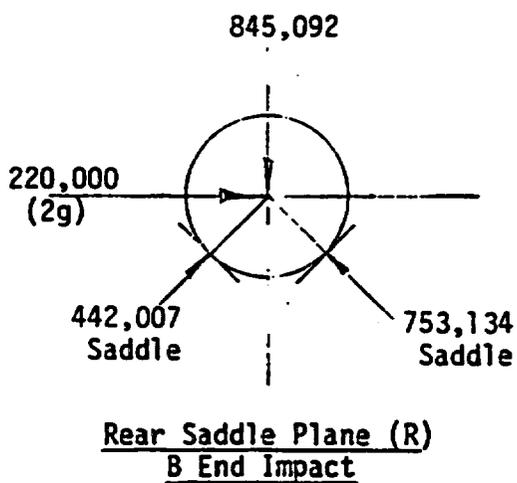
- ↑ 515,092 pitching
- ↑ 220,000 2g V
- ↓ 110,000 1g Static
- ↑ 625,092 net tension



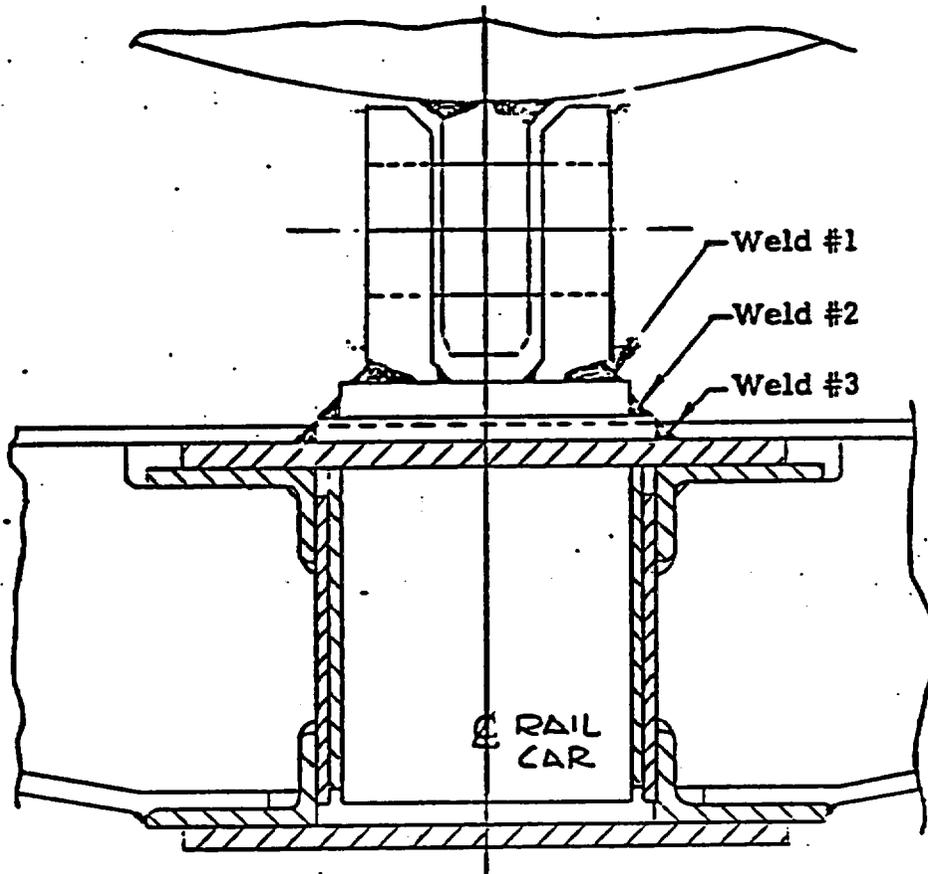
B End Impact

For Rear Saddle at R, with 2g transverse

- ↓ 515,092 pitching
- ↓ 220,000 2g V
- ↓ 110,000 1g static wt.
- ↓ 845,092 net compression



2.2.1 Tie-Down-Cask Closure Head End ("A" End)



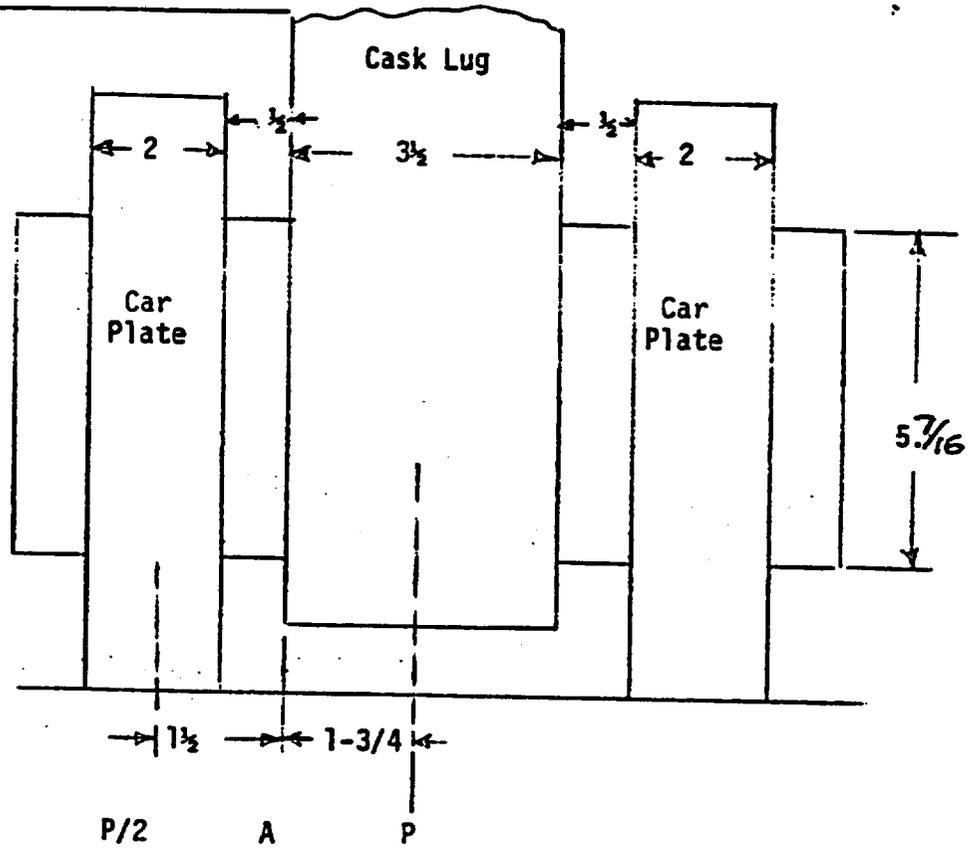
Front Tie Down

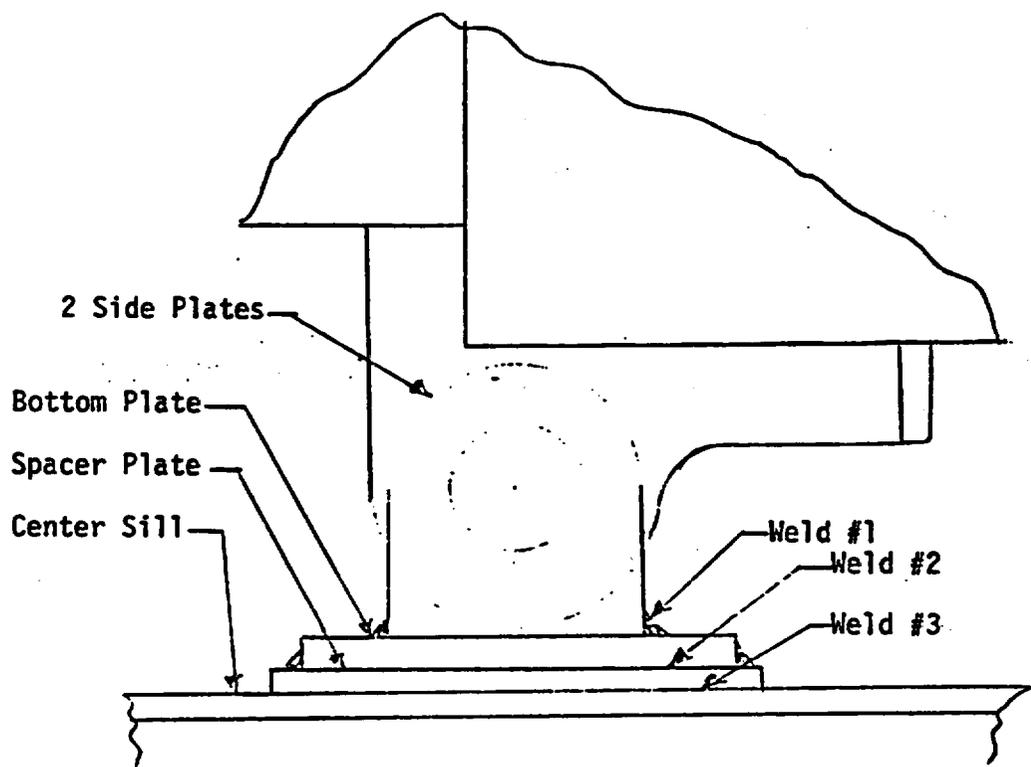
"A" End.

Also Closure Head End of Cask

2.2.1.1

Front Tie-Down Analysis





Front Tie Down

"A" End

2.2.1.2 Front Pin analysis

$$P = 1,175,092 \text{ lbs on lug (from B end impact - XI-2-12)}$$

$$P/2 = 587,546 \text{ lbs on each plate}$$

$$\text{Pin dia} = 5\text{-}7/16 \text{ in.}$$

Material 17-4 ph (100,000 psi Y.P.)

$$Z = .098 (5.4375)^3 = 15.755 \text{ in}^3$$

$$A = \pi/4 (5.4375)^2 = 23.22 \text{ in}^2$$

Pin stresses at section A

$$M_A = P/2 \times 1.5 = 881,319 \text{ in lbs.}$$

$$S_b = \frac{881,319}{15.755} = 55,939 \text{ psi}$$

$$S_s = \frac{587,546}{23.22} = 25,303 \text{ psi}$$

$$\text{Combined stresses} = \sqrt{55,939^2 + 3 \times 25,303^2} = 71063 \text{ psi}$$

$$\text{M.S.} = \frac{.9 \times 100,000}{71063} - 1 = .266$$

Pin stresses at center P

$$M_B = P/2 \times 3\frac{1}{2} - P/2 \times 7/8 = P/2 \times 2.375 = 1,395,422 \text{ in lbs.}$$

$$S_b = \frac{M}{Z_B} = 88,570 \text{ PSI} \quad S_s = 0$$

$$\text{M.S.} = \frac{.9 \times 100,000}{88,570} - 1 = .016$$

2.2.1.3 Front Cask Lug Analysis

Section through C.L. of pin

$$\text{Area} = 3\text{-}1/8 \times 3\text{-}1/2 - \frac{.866}{2} = 10.5045 \text{ in}^2$$

Tension Load - 1,175,092 lbs. (from B end impact XI-2-12)

Hoop stress across area - tension

$$S_t = \frac{1,175,092}{2 \times 10.5045} = 55,933 \text{ psi}$$

$$\text{M.S.} = \frac{9 \times 100,000}{55,933} - 1 = .609$$

Shear tearout at 40° from C.L.

$$\text{Area} = 2 \left(3 \frac{5}{8} \times 3\text{-}1/2 - \frac{.75}{2} \right) = 24.625 \text{ in}^2$$

$$S_s = \frac{1,175,092}{24.625} = 47,719 \text{ psi}$$

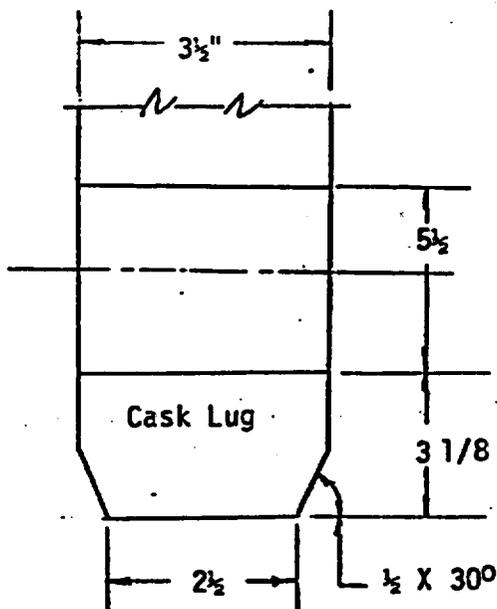
$$\text{M.S.} = \frac{.6 \times .9 \times 100,000}{47,719} - 1 = \underline{\underline{.131}}$$

Tension area thru diameter

$$\text{Area} = (13 - 5.5) 3.5 = 26.25 \text{ in}^2$$

$$S_t = \frac{1,175,092}{26.25} = 44,765 \text{ psi}$$

$$\text{M.S.} = \frac{.9 \times 100,000}{44,765} - 1 = 1.01$$



C.G. of welds - to find \bar{X} and \bar{Y}

| <u>Weld</u> | <u>Area (A)</u> | <u>X</u> | <u>AX</u> | <u>Y</u> | <u>AY</u> |
|-------------|-------------------------|-------------|--------------|----------|-----------|
| 3½C | 3½ x .707 = 2.47 | 0 | 0 | 5.5 | 13.585 |
| 4½A | 4½ x 3 = 13.5 | 2¼ | 30.38 | 5.5 | 74.25 |
| 5.5A | 5.5 x 3 = 16.5 | 4½ | 74.25 | 2.75 | 45.375 |
| 3A | 3 x 3 = 9.0 | 6 | 54.0 | 0 | 0 |
| 15B | 15 x 1.414 = 21.21 | 15 | 318.15 | 0 | 0 |
| 3½D | 3½ x 1.06 = <u>3.71</u> | 22½ | <u>83.47</u> | <u>0</u> | <u>0</u> |
| | 66.39 | x 8.44 | =560.25 | 2.00 | 133.21 |
| | | - 4.50 | | | |
| | | <u>3.94</u> | | | |

$$I_{yy} = \frac{3.5 \times (.707)^3}{12} + 2.47 (8.44)^2 + \frac{3 \times (4.5)^3}{12} + 13.5 (6.19)^2$$

$$+ \frac{2 \times 5.5 (1.5)^3}{12} + 16.5 (3.94)^2 + \frac{3 \times 3^3}{12} + 9.0 (2.44)^2$$

$$+ \frac{1.414 (15)^3}{12} + 21.21 (6.56)^2 + \frac{7 \times (.53)^3}{12} + 3.71 (14.06)^2 = 3081.8 \text{ in}^4$$

$$I_{xx} = \frac{.707 (3.5)^3}{12} + 2.47 (3.5)^2 + \frac{2 \times 4.5 \times (1.5)^3}{12} + 13.5 (3.5)^2$$

$$+ \frac{3 \times (5.5)^3}{12} + 16.5 (.75)^2 + \frac{3 \times 3^3}{12} + 9.0 (2.0)^2$$

$$+ \frac{2 \times 15 \times (.707)^3}{12} + 21.21 (2.0)^2 + 1.06 \frac{(3.5)^3}{12} + 3.71 (2.0)^2 = 398.7 \text{ in}^4$$

$$I_p = 3081.8 + 398.7 = 3480.5 \text{ in}^4$$

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Eccentricity of vertical load = 1.94"

$$P_1 = 1,175,092 \text{ lbs} \quad (\text{XI-2-16})$$

$$\uparrow = P_1 \times \text{ecc} = 1,175,092 \times 1.94 = 2,279,678 \text{ in. lbs.}$$

$$\frac{T}{I_p} = \frac{2,279,678}{3480.5} = 655.$$

$$S_s = \frac{T \rho}{I_p} = \left(\frac{T}{I_p}\right) \rho = 655 \rho \text{ for any point}$$

Stress at point (3)

$$\rho = \sqrt{2^2 + 14.06^2} = 14.2 \text{ in.}$$

$$S_s = 655 \times 14.2 = 9301 \text{ psi} \uparrow \text{ compression}$$

S_x = Uniform tension stress over all welds

$$= \frac{1,175,092}{66.39} = 17,700 \text{ psi tension} \downarrow$$

$$\text{Net stress} = 17,700 - 9301 = 8399 \text{ psi tension}$$

$$\text{M.S.} = \frac{.9 \times 30,000}{8399} - 1 = 2.21$$

Stress at point (2)

$$\rho = 3.94$$

$$S_s = 655 \times 3.94 = 2581 \text{ psi} \downarrow$$

$$S = 17,700 \text{ psi} \uparrow$$

$$\text{Net stress} = 17,700 \text{ psi} + 2581 \text{ psi} = 20,281 \text{ psi} \downarrow$$

$$\text{M.S.} = \frac{.9 \times 30,000}{20,281} - 1 = .33$$

Stress at point (1)

$$\rho = \sqrt{8.44^2 + 3.52^2} = 9.14"$$

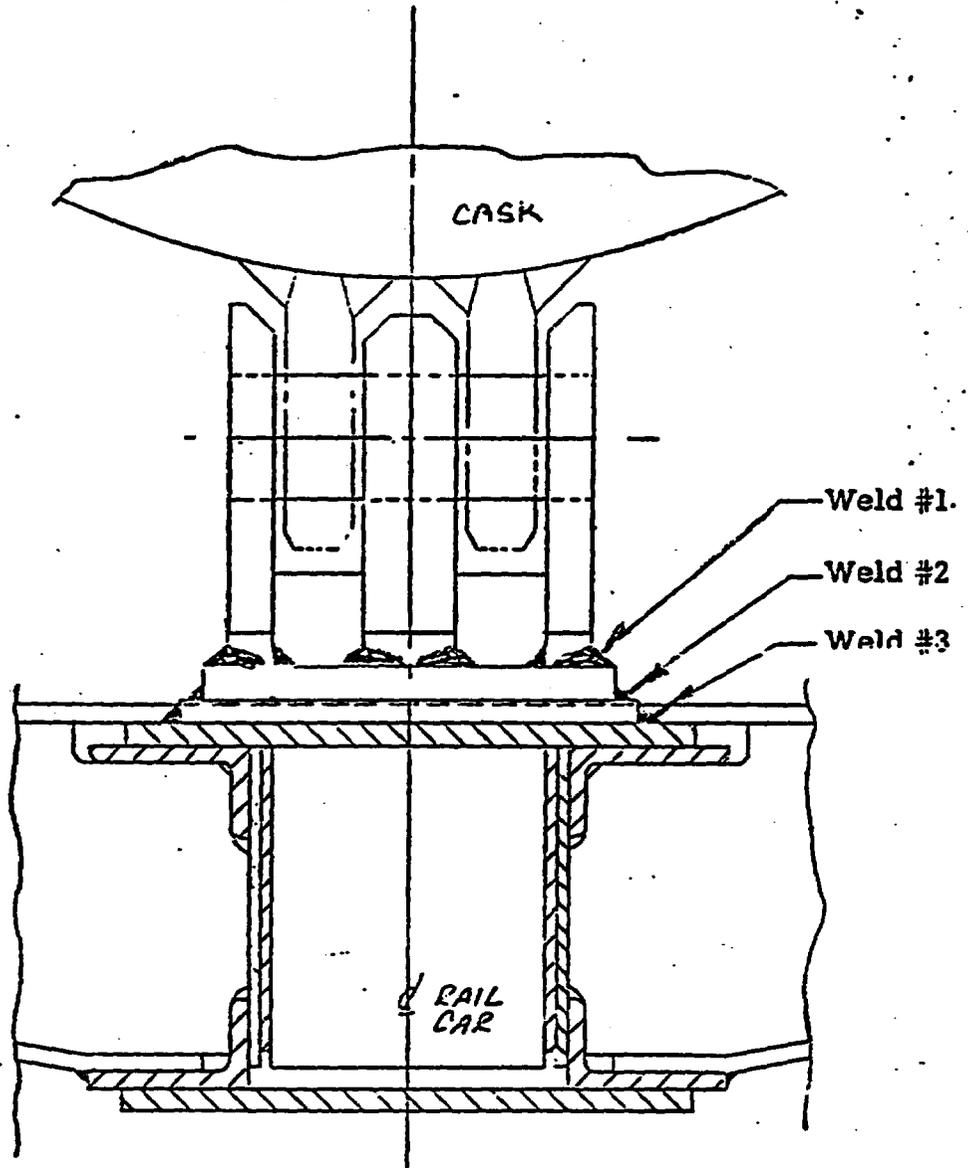
$$S_s = 655 \times 9.14 = 5989 \text{ psi} \downarrow$$

$$S = 17,700 \text{ psi} \downarrow$$

$$\text{Net stress} = 17,700 + 5985 = 23,685 \text{ psi} \downarrow$$

$$\text{M.S.} = \frac{.9 \times 30,000}{23,685} - 1 = .14$$

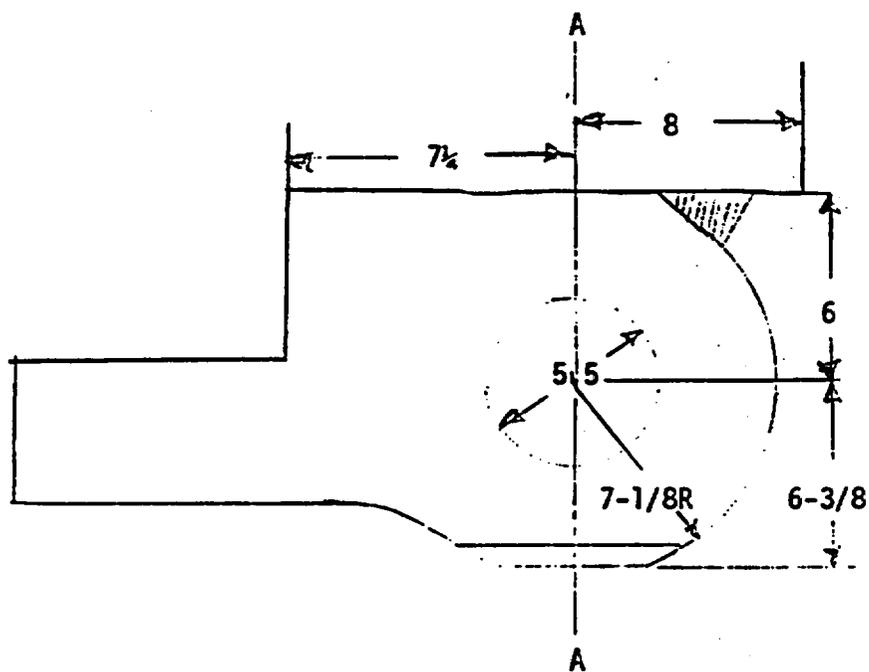
2.2.2 Tie down - bottom end of cask ("B" End)



Rear Tie Down

"B" End

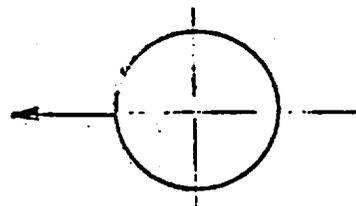
Bottom of Cask



Rear Cask Lug (B End)

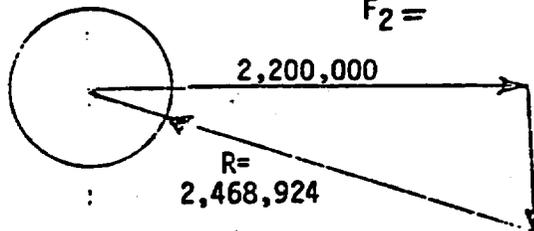
For impact at "B" End

F.
2,200,000 lbs



For impact at "A" End

F₂ =



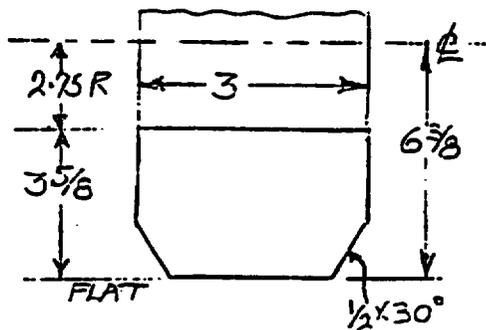
R =
2,468,924

F₃ =

1,120,530

2.2.2.1 Rear Cask Lugs - B - End

Material - 17 - 4-PH 100,000 psi y.p.



Min. area at vertical section =
 $(3 \times 3\text{-}5/8) - \frac{1}{2} \times .866 = 10.44 \text{ in.}^2$

Max tension load = 2,468,924 lbs
 on both lugs (Impact at "A" End)

$$\text{Hoop stress thru section A} = \frac{2,468,924}{(12.375 - 5.5)3 \times 2} = 59,852 \text{ psi}$$

$$\text{M.S.} = \frac{.9 \times 100,000}{59,852} - 1 = .50$$

Shear tear - out at 40° to load line

$$\text{Area} = 2 (3 \times 4.66) = 27.96 \text{ in}^2$$

$$S_s = \frac{2,468,924}{2 \times 27.96} = 44,151 \text{ psi}$$

$$\text{M.S.} = \frac{6 \times .9 \times 100,000}{44,151} - 1 = .223$$

Bearing stress at P

$$\text{Bearing area 2 lugs} = 2 \times 5.5 \times 3 = 33 \text{ in}^2$$

$$S = \frac{2,468,924}{33} = 74,816 \text{ psi}$$

$$\text{M.S.} = \frac{.9 \times 1.5 \times 100,000}{74,816} - 1 = .80$$

2.2.2.2 Cask pin - B End - 5-7/16 dia.

Material - 17-4 PH 100,000 psi Y.P.

$$Z = .098 (5.4375)^3 = 15.755 \text{ in}^3$$

$$A = \frac{\pi}{4} (5.4375)^2 = 23.22 \text{ in}^2$$

Load distribution

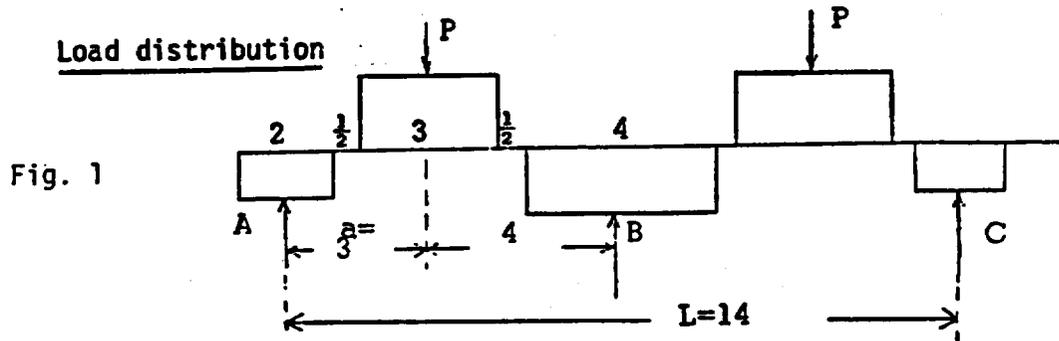


Fig. 2

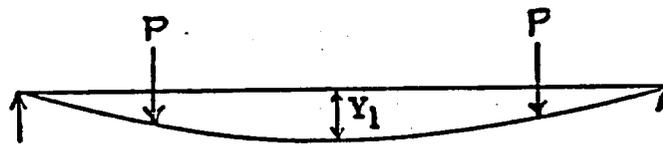


Fig. 3

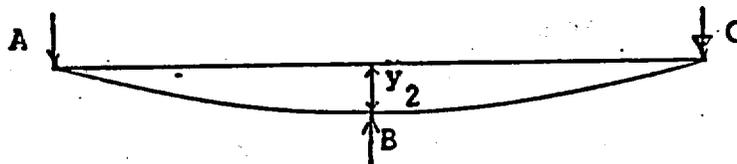


Fig. 2 Shows center deflection y_1 , caused by two loads (P) on simply supported beam (support B removed)

Fig. 3 shows support force B required to return center point to same level as ends A and C. $y_2 = y_1$

Max. load = $2P = 2,468,924$ from impact at A end .

$P = 1,234,462$ lbs.

$$\text{From Fig. 2 } Y_i = \frac{Pa}{24EI} (3L^2 - 4a^2)$$

$$a = 3 \quad L = 14$$

$$Y_i = \frac{P \times 3}{24EI} (3 \times 14^2 - 4 \times 3^2) = \frac{69P}{EI}$$

$$\text{From fig. 3 } Y_2 = Y_i = \frac{BL^3}{48EI} = \frac{B \times 14^3}{48EI} = \frac{57.1666 B}{EI}$$

$$\text{Equating } 69P = 57.1666 B$$

$$B = 1.207 P$$

$$A = C = \frac{2 - 1.207P}{2} = .3965P$$

$$\text{But max. } P = 1,234,462 \text{ lbs}$$

$$B = 1,489,000 \text{ lbs}$$

$$A = C = 489,464 \text{ lbs}$$

$$\text{Moment at P} = 489,464 \times 3 - \frac{1,234,462}{2} \times 7.5 = 1,005,469 \text{ in lbs}$$

$$\text{Moment at B} = 489,464 \times 7 - \frac{1,234,462}{2} \times 4 + \frac{1,489,000}{2} \times 1 = 767,100 \text{ in lbs}$$

$$\text{At P, Max } S_b = \frac{1,005,469}{15,755} = 63,819 \text{ psi at P}$$

$$\text{Shear at P} = 489,464 - \frac{1,234,462}{2} = -127,768 \text{ lbs}$$

$$S_s = \frac{-127,768}{23.22} = -5502 \text{ psi}$$

$$\text{Comb. } S_x = \sqrt{63819^2 + 3(5502)^2} = 64,527 \text{ psi}$$

$$\text{M.S.} = \frac{.9 \times 100,000}{64,527} - 1 = .394$$

$$\text{At B, Bearing } S_{br} = \frac{1,489,000}{4 \times 5.5} = 67,682 \text{ psi}$$

$$\text{M.S.} = \frac{.9 \times 100,000}{67,682} - 1 = .329 \text{ conservative}$$

2.2.2.3 Welds of lugs to cask

The two lugs welded to cask are of 17-4ph metal, while the cask itself is 304 S.S. The weld metal is similiar to the base metal and is calculated at 30,000 psi Y. P. The weld is critical thru its throat area.

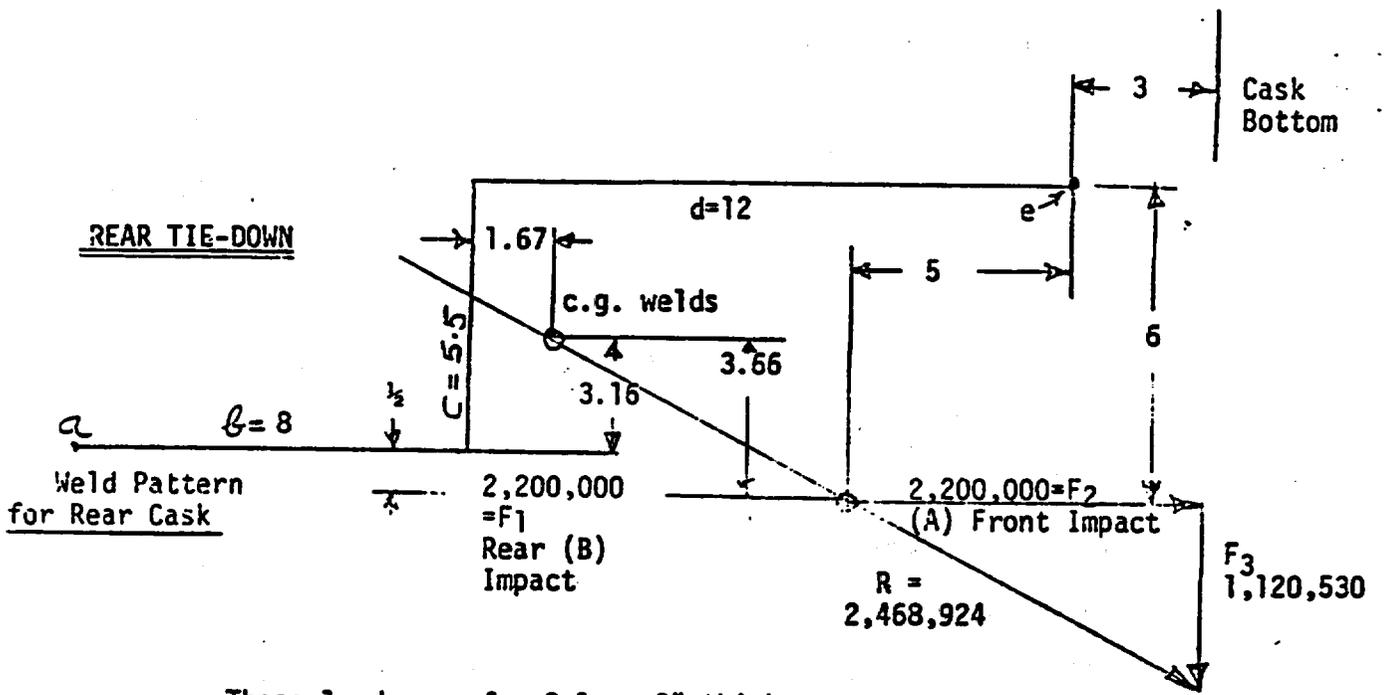
There are two conditions of applied loads acting through the center of the pin. When the force applied at the cask C.G. acts toward the B end of the railcar, the rear tie-down is loaded only by the 10g L force while the vertical forces are taken in compression on the rear saddle itself. When the force of 10g L is acting in the direction of the top (A) end, then the rear lug pin is loaded with both 10g L and the resultant tension of applied vertical forces (of pitching, etc.)

For an impact at (A) end, the total resultant load of 2,468,924 lbs. acts almost through the c.g. of the welds, and so negligible torsional stresses are developed. The main stress is the average uniformly distributed stress of:

$$S = \frac{2,468,924 \text{ lbs.}}{166.86} = 14,800 \text{ psi}$$

Due to the direction of the resultant, this is largely tension across areas a, c and e and largely shear across areas b and d. Even for a point which may be completely oriented to pure shear, this stress is safely low

$$\text{M.S.} = \frac{.6 \times .90 \times 30,000}{14,800} - 1 = .095 \text{ shear (Actually conservative on Y.P.)}$$



These loads are for 2 lugs 3" thick

Throat widths - single lug. (w)

- a 1½" fillet $w = 1.5 \times .707 = 1.06$ in.
 b, c, d each 2 (1½ fillets + 1/2 penetration)
 $w = 2 \times 1.50 = 3.0$ in.
 e 1½ fillet $w = 1.25$ in.

To determine C.G. welds

| | x w = | A | \bar{x} | $A\bar{x}$ | y | $A\bar{y}$ |
|---|-----------|-------------|-----------|------------|------------|---------------|
| a | 3 x 1.06 | 3.18 | 0 | 0 | 0 | 0 |
| b | 8 x 3.0 | 24.0 | 4 | 96 | 0 | 0 |
| c | 5.5 x 3.0 | 16.5 | 8 | 132 | 2.75 | 45.375 |
| d | 12 x 3.0 | 36.0 | 14 | 504 | 5.5 | 198. |
| e | 3 x 1.75 | <u>3.75</u> | <u>20</u> | <u>75</u> | <u>5.5</u> | <u>20.625</u> |
| | | 83.43 | (9.67) | 807 | (3.16) | 264 |

Polar moment of inertia I_p for each lug

$$\begin{aligned}
 &= \sum \left(\frac{w^3}{12} + w^2 x^2 + \frac{w^3}{12} + w^2 y^2 \right) \text{ where } w^2 = A \\
 &= \sum A \left(\frac{w^2}{12} + \frac{w^2}{12} + x^2 + y^2 \right) \\
 I_p &= 3.18 \left(\frac{1.06^2}{12} + \frac{3^2}{12} + 9.67^2 + 3.16^2 \right) = 24 \left(\frac{3^2}{12} + \frac{8^2}{12} + 5.67^2 + 3.16^2 \right) \\
 &\quad + 16.5 \left(\frac{3^2}{12} + \frac{5.5^2}{12} + 1.67^2 + .41^2 \right) + 36 \left(\frac{3^2}{12} + \frac{12^2}{12} + 4.33^2 + 2.34^2 \right) \\
 &\quad + 3.75 \left(\frac{1.25^2}{12} + \frac{3^2}{12} + 10.33^2 + 2.34^2 \right) \\
 &= 331.769 + 1157.228 + 102.76 + 1331.082 + 423.993 \\
 &= 3346.858
 \end{aligned}$$

For both lugs; $2I_p = 6693.7$ in.⁴

$2A = 166.86$ in.²

For an impact at the rear (B) end, $F_1 = 2,200,000$ lbs and the vertical loads are all taken by the impacter rather than by the lug.

$$\text{Average stress } S = \frac{2,200,000}{166.86} = 13,185 \text{ psi for all welds.}$$

Torque due to eccentricity is

$$T = 2,200,000 \times 3.66 = 8,052,000 \text{ in. lbs.}$$

$$T/I_p = \frac{8,052,000}{6693.7} = 1202.9 \text{ psi}$$

Shear stress at any point of the welds to cask.

$$S_s = \rho (T/I_p) = 120.219 \rho$$

For point (a) $\rho \sqrt{9.67^2 + 3.16^2} = 10.2 \text{ in.}$

$$\uparrow S_c = 10.2 (1202.9) = 12,270 \text{ psi (compression against jacket)}$$

$$\leftarrow S_s = 13,185 \text{ psi is here a shearing stress}$$

$$S_c = \sqrt{12,270^2 + 3 \times 13,185^2} = 25,925 \text{ psi}$$

$$\text{M.S.} = \frac{.9 \times 30,000}{25,925} - 1 = .04 \text{ conservative because compressive stress.}$$

For midpoint of (c)

$$\rho = \text{app. } 1.67 \quad \uparrow S_s = 1.67 \times 1202.9 = 2,009 \text{ psi (shear)}$$

$$S_c = \sqrt{13,185^2 + 3 \times 2,009^2} = 13,636 \text{ psi (compression)}$$

$$\text{M.S.} = \frac{.9 \times 30,000}{13,636} - 1 = .98$$

For point (e)

$$\rho = \sqrt{10.33^2 + 2.34^2} = 10.59$$

$$\downarrow S_x = 10.59 \times 1202.9 = 12,741 \text{ psi tension}$$

$$\leftarrow S = 13,185 \text{ psi shear}$$

$$S_t = \sqrt{12,741^2 + 3 \times 13,185^2} = 26,151 \text{ psi}$$

$$\text{M.S.} = \frac{.9 \times 30,000}{26,151} - 1 = .48$$

2.2.3 Railcar tie-downs and saddles Accident analysis

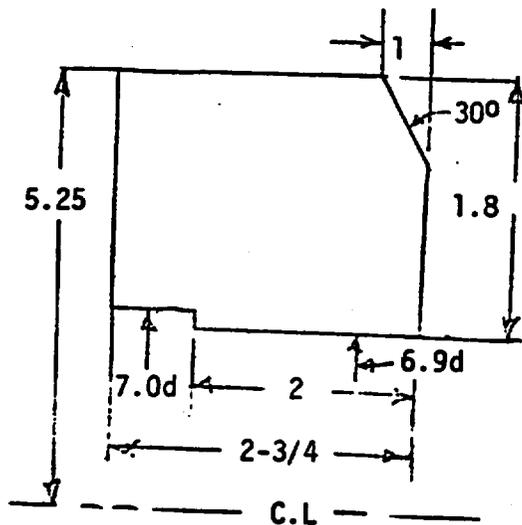
This analysis deals with those parts of the tie-down system which are attached to the railcar rather than the cask body itself. The same applied loadings are used as in the previous analyses of the cask members, namely 10g longitudinal, 5g lateral and 2g vertical. The difference in this analysis is that, since it is for an accident condition, the allowable design strengths are 100% of the ultimate tensile stress, contrasted with 90% of the Y.P in the previous cask member analysis. To prove that in an accident, the failure would be thru the railcar attachment welds, a final summary of M.S. values is made, comparing these railcar members with the cask ratings - for the accident condition only.

| <u>Material properties</u> | - U.T.S. values are taken as |
|----------------------------|--|
| 75,000 psi | - 304 stainless steel (also 308 weld rod & wire) |
| 115,000 psi | - 17-4PH Stainless steel |
| 115,000 psi | - T-1 steel |
| 95,000 psi | - 8018 weld rod & wire |
| 75,000 psi | - 7018 weld rod & wire |

Refer to sections 2.2.1 and 2.2.2 for pertinent drawings and part dimensions and weld designations.

2.2.3.1

Front tie-down - Railcar plates - T-1 steel - 2 plates
Impact at B end



Section through C.L.

$$\text{Area} = (1.8 \times 2.75) - \left(\frac{3}{4} \times 0.5\right) - \frac{(1.732)}{2} = 4.047$$

$$\text{Load per plate} = \frac{1,175,092}{2} = 587,546 \text{ lbs.}$$

Hoop stress across section

$$S_t = \frac{587,546}{2 \times 4.047} = 72,590 \text{ psi}$$

$$\text{M.S.} = \frac{115,000}{72,590} - 1 = .584$$

Shear tearout at 40° from C.L.

$$\text{Area} = 2 \left(2.6 \times 2.75 - .05 \times .75 - \frac{1.732}{2}\right) = 12.49$$

$$S_s = \frac{587,546}{12.49} = 47,041 \text{ psi}$$

$$\text{M.S.} = \frac{.6 \times 115,000}{47,041} - 1 = .466$$

Tension on diameter

$$\text{Area} = (11 - 6.9) 2.75 - 2(.05 \times .75) = 11.20 \text{ in}^2$$

$$S_t = \frac{587,546}{11.237} = 52,459 \text{ psi}$$

$$\text{M.S.} = \frac{115,000}{52,459} - 1 = \underline{\underline{1.192}}$$

$$\text{Bearing SBR} = \frac{587,546}{6.9 \times 2} = 42,576 \text{ psi}$$

$$\text{M.S.} = \frac{115,000}{42,576} - 1 = 1.7$$

Front tie-down - Railcar welds

Welds 1 and 3 are superior in strength to that of weld 2 which is sacrificed in tension (the only load imposed under transit conditions or when impacted at B end in the accident condition). When impacted at the A end, weld 2 can easily be overturned and separated but only after the rear tie-down has initially released the cask as a free body.

Weld 1 - 2 side plates to bottom plate - (T-1 to T-1) (95,000 UTS welds)

1-1/8" J weld on outer side of each plate

1" fillet welds all around both plates =

$$2 \times (11 \times 1\frac{1}{2} \text{ throat} + .707 \times 16\frac{1}{2}) = 56.3 \text{in}^2$$

P = 1,175,092 lbs. on pin and 2 plates \uparrow impact at B end

$$S_t = \frac{1,175,092}{56.3} = 20,872 \text{ psi}$$

$$\text{M.S.} = \frac{95,000}{20,872} - 1 = 3.55$$

Weld 3 - spacer plate to center sill (T-1 to 64,000 psi UTS steel)
(75,000 UTS weld)

2 welds each 1" fillet x 20" long

Area = 2 x 1 x .707 x 20 = 28.28in² - throat of weld

$$S_t = \frac{1,175,092}{28.28} = 41,552 \text{ psi}$$

$$\text{M.S.} = \frac{75,000}{41,552} - 1 = .80$$

Interface with center sill is along 1" face of weld.

Area = 2 x 1 x 20 = 40in.²

$$S_t = \frac{1,175,092}{40} = 29,377 \text{ psi}$$

$$\text{M.S.} = \frac{64,000}{29,377} - 1 = 1.178$$

Weld 2 (sacrificial) bottom plate to spacer plate (T1 - T1) - Tension only
9/16 in. fillet welds along 18 in. sides only (95,000 UTS weld)

$$\text{Area} = 2 \times 18 \times 9/16 \times .707 = 14.317 \text{ in}^2$$

$$S_t = \frac{1,175,092}{14.317} = 82,077 \text{ psi}$$

$$\text{M.S.} = \frac{95,000}{82,077} - 1 = .157 \text{ tension } \uparrow$$

Therefore, in severe fore and after impact as specified (10g)
the welds will all have a generous M.S.

Front tie down weld #2 - accident condition - A end impact
Weld 2 will be separated, not in tension, but by longitudinal
bending after the rear tie down has separated.

Bending strength of weld #2 in break away after rear tie down parts

$$Z_{xy} = \frac{2 (9/16 \times .707) 18^2}{6} = 42.95 \text{ in}^3$$

$$\text{Area} = 2(9/16 \times .707) 18 = 14.317 \text{ in}^2$$

$$\text{Moment arm} = 6 - 1/8 + 1 \frac{1}{2} = 7 - 5/8 \text{ in. height.}$$

F = hor. force required to cause failure in bending and shear.

$$M = 7.625F$$

$$S_b = \frac{7.625 F}{42.95} = .1775F$$

$$S_s = \frac{F}{14.317}$$

$$\text{Comb. stress} = \sqrt{(.1775F)^2 + 3 \left(\frac{F}{14.317} \right)^2} = .1216F$$

$$F \text{ max.} = \frac{95,000}{.1216} = 781,250 \text{ lbs to break away thru 95,000 UTS welds}$$

$$\frac{781,250}{220,000} = 3.55g$$

Longitudinal force to break away weld #2

2.2.3.3 Rear tie-down - Railcar plates

T-1 Steel -- 115,000 psi UTS - 3 plates

Impact at A end

Load on central plate = 1,489,000 lbs.

Hoop stress - - at top - flatted

$$A_1 = (4 \times 2\frac{1}{2}) - (.75 \times 1.3) = 9.025 \text{ in}^2$$

$$S_t = \frac{1,489,000}{2 \times 9.025} = 82493 \text{ psi}$$

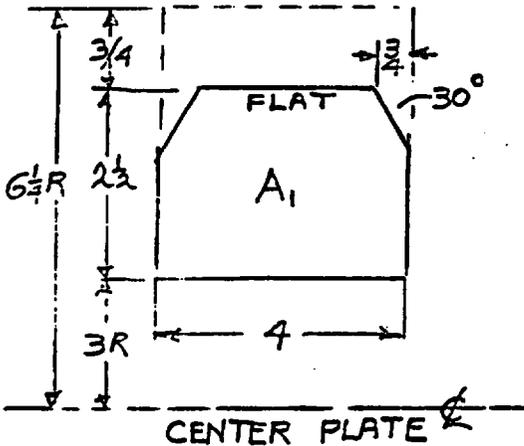
$$\text{M.S.} = \frac{115,000}{82493} - 1 = .394$$

Shear tear out at 40° to 45° load line.

$$\text{Area} = (4 \times 3\frac{1}{2}) - (3/4 \times 1.3) = 13.025 \text{ in}^2$$

$$S_s = \frac{1,489,000}{2 \times 13.025} = 57,160 \text{ psi}$$

$$\text{M.S.} = \frac{.6 \times 115,000}{57,160} - 1 = .207$$



Load on each side plate = 489,464 lbs.

Hoop stress at top - flatted

$$A_2 = (3 \times 2) - (.75 \times 1.3) = 5.5125 \text{ in}^2$$

$$S_t = \frac{489,464}{2 \times 5.5125} = 44,396 \text{ psi}$$

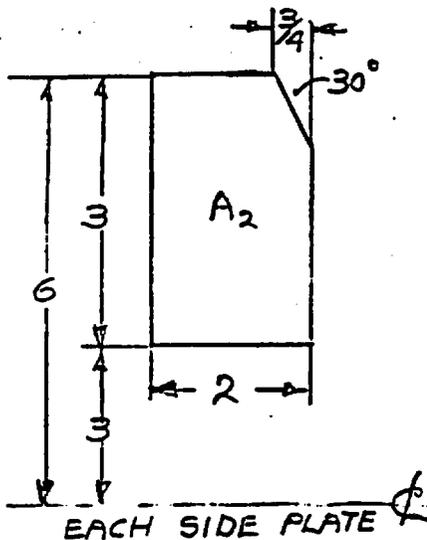
$$\text{M.S.} = \frac{115,000}{44,396} - 1 = 1.59$$

Shear tear out at 40° to 45° load line

$$\text{Area} = 2 \times 3\frac{1}{2} - \frac{.75 \times 1.3}{2} = 6.51 \text{ in}^2$$

$$S_s = \frac{489,464}{2 \times 6.51} = 37,593 \text{ psi}$$

$$\text{M.S.} = \frac{.6 \times 115,000}{37,593} - 1 = .835$$



Impact at B end

Stresses in all plates are reduced from those calculated above in the rates of $\frac{2,200,000}{2,468,924}$, so they are not critically stressed in this condition.

2.2.3.4

Rear tie-down - welds to railcarWeld 1 - 3 plates to bottom plate of weldment

(T-1 to T-1) - weld 95,000 psi UTS.

4 x 42½" of 1-1/8"J + 1" fillet = 170" x 1½" throat = 255in²

(2 x 42½ + 16) of 1" fillet = 101 x .70 throat = 71.4 in²
 Total - weld #1 = 326.4 in²

Weld 2 - (Sacrificial) Bottom of weldment to center sill spacer plates

(T-1 to T-1) - weld 75,000 psi UTS.

88" of 1-3/8" fillet - 88 x 1-3/8" x .707 = 85.55in²

Weld 3 Spacer plates to railcar center sill (2)

(T-1 to railcar sill (A441 - 67,000 psi UTS)

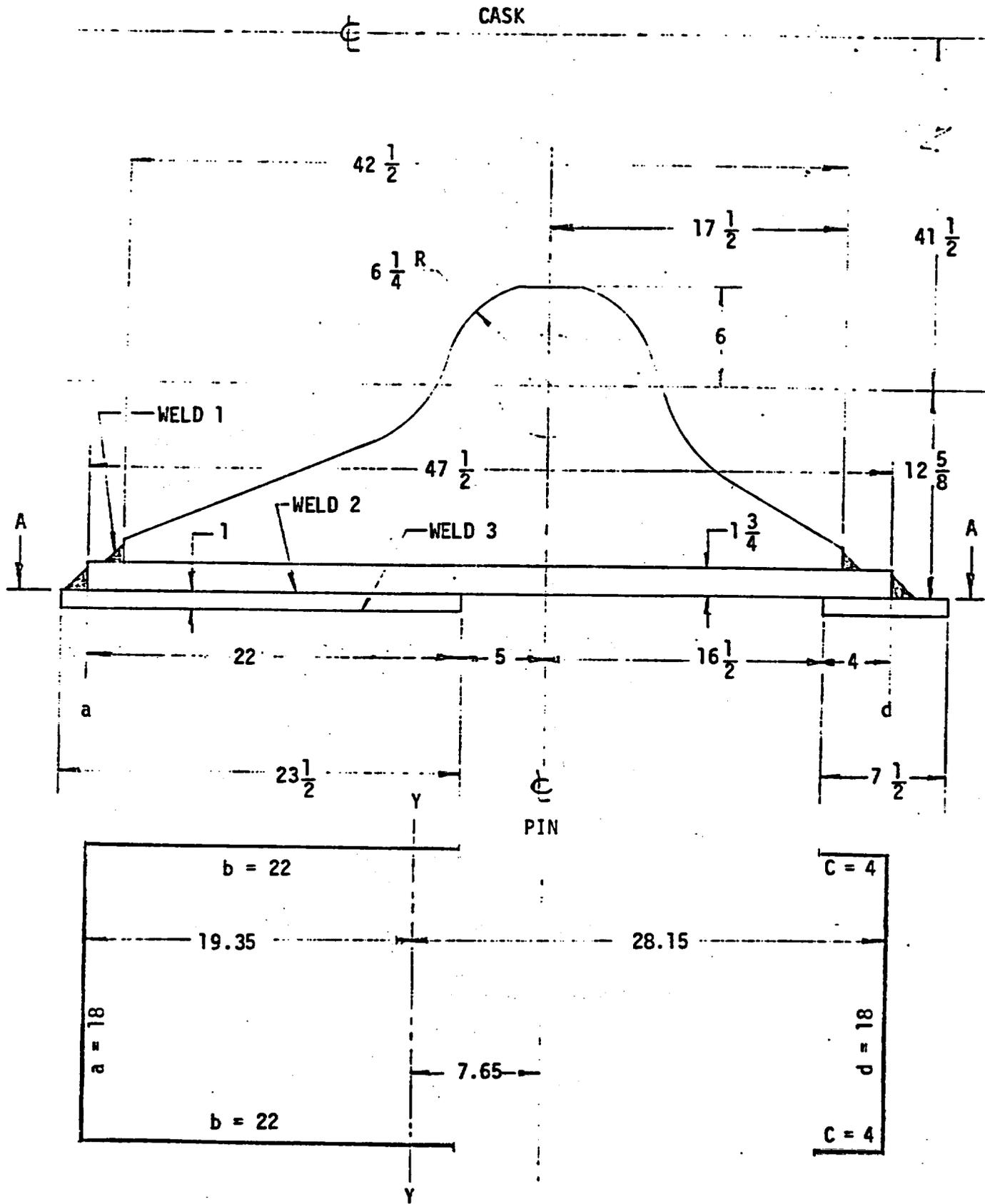
Weld 75,000 psi UTS.

142" of 3/4" penetration + 1" fillet

142" x 1½" throat = 177.5in²

Weld 1 and 3 are obviously much stronger than weld 2.

Weld 2 is sacrificial and will be analyzed in detail.



WELD PATTERN OF PLANE -A- WELD 2

X1-2-22m

To find C.G. of welds - Weld 2

$$a \quad 18 \quad x \quad 0 \quad = \quad 0$$

$$2b \quad 2x22 \quad x \quad 11 \quad = \quad 484.$$

$$2c \quad 2x4 \quad x \quad 45.5 \quad = \quad 364$$

$$\underline{d \quad 18} \quad x \quad \underline{47.5} \quad = \quad \underline{855}$$

$$88 \quad x \quad (19.35) \quad = \quad 1703 \text{ in}^2$$

$$I_{yy} = 18 \times 19.35^2 + 2 \times \frac{22^3}{12} + 2 \times 22 \times 8.35^2 + \frac{2 \times 4^3}{12} + 2 \times 4 \times 26.15^2 + 18 \times 28.15^2$$

$$= 31327. \text{in}^3 \text{ for line welds}$$

$$\text{Weld throat} = 1-3/8" \times .707 = .972 \text{in}$$

$$Z_d = \frac{31327}{28.15} = 1112.86 \text{ in}^2 \text{ for line welds } x .972" = 1082 \text{ in}^3$$

$$Z_a = \frac{31327}{19.35} = 1619 \text{ in}^2 \text{ for line welds } x .972" = 1574 \text{ in}^3$$

$$\text{Area} = 19.35 \text{ in}^2$$

$$\text{weld length} = 88", \text{ weld throat area} = 85.54 \text{ in}^2$$

For impact at rear (B) end - Loads on tie-down at rear - weld 2

$$F_1 = 2,200,000 \text{ lbs.}$$

$$M = 2,200,000 \times 12 \frac{5}{8} = 27,775,000 \text{ in lbs.}$$

This puts tension at a & compression at d.

There is no vertical load in this case.

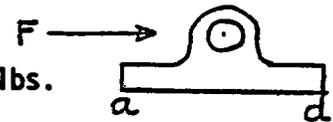
$$\text{Shear stress } S_s = \frac{2,200,000}{85.54} = 25,719 \text{ psi}$$

$$\text{at } \underline{a} \text{ max tension } S_t = \frac{M}{1574} = 17646 \text{ psi}$$

$$\text{Comb. stress } S = \sqrt{17646^2 + 3 \times 25.719^2}$$

$$= 47914 \text{ psi}$$

$$\text{M.S.} = \frac{75,000}{47,914} - 1 = .565 \text{ at } \underline{a}$$



For impact at front end (A) - Loads on tie-down at rear. - weld 2

$$F_1 = 2,200,000 \text{ lbs. } \leftarrow$$

$$R = 2,468,924 \leftarrow$$

$$V_1 = 1,120,530 \text{ lbs. } \uparrow$$

$$M = F_1 \times 12-5/8 + V_1 \times 7.65 = 36,072,055 \text{ in lbs.}$$

This puts tension at D and compression at A

Stress at D (Sac. Weld #2)

$$\frac{M}{Z_d} = \frac{36,072,055}{1,082} = 33,338 \text{ psi}$$

$$\frac{V_1}{A} = \frac{1,120,530}{85.54} = 13,099 \text{ psi}$$

$$\uparrow \text{ Total tension} = 33,338 + 13,099 = 46,437 \text{ psi}$$

$$\text{Shear stress } K = 25,719 \text{ psi}$$

$$\text{Comb. stress } S = \sqrt{46,437^2 + 3 \times 25,719^2} = 64,349 \text{ psi}$$

The max stress at the plane A of the sacrificial weld is thus 64,349 psi

The weld is between T-1 plates.

$$\text{Weld M.S.} = \frac{75000}{64,349} - 1 = .165 \text{ at } \underline{d}$$

This assures adequate strength for 10gL + 5gT + 2gV condition, yet allows controlled break-away above this.

2.2.3.5

Saddle bearing stresses - Impact limiters on support structure

The compression loads on a 450 saddle structure are maximum as following:

753,134 lbs. Rear saddle, impact at B end, 2g lateral

Minimum bearing area of each flat on radial base plate of impact structure is $28 \times 1.5 = 42 \text{ in}^2$

$$\text{SBR} = \frac{753,134}{42} = 17,932 \text{ psi}$$

The material is 6061 - T 6511 with allowable compressive stress at y.p. = 36,000 psi

$$\text{M.S.} = \frac{36,000}{17,932} - 1 = 1.00$$

This is for the net area of the edge of the end plate itself, and neglects the load distribution afforded by the actual flange width itself, thus actually giving lower stresses.

2.2.3.6

Summary of Tie-Down Stresses

The various parts which are positioned between the cask proper and the railcar proper, and which constitute the tie-down and support system, are subject to different design requirements.

Summaries of stresses are therefore presented in two groupings, each consistent within itself.

Condition A has loadings of 10g L, 5g T, and 2g V and applies to integral cask tie down lugs and associated pins, with stresses limited to .9 x y.p.

Condition B is the accident condition. The same loadings of 10gL, 5 gT, and 2 gV are applied to the railcar plate weldments which mate with the above pins and are welded to the railcar center sill. Stresses are calculated on the U.T.S of base metal and of weld material for these parts.

To make a consistent comparison of these railcar parts with the above cask related parts, it is necessary to calculate a new set of M.S. values for the latter, based on U.T.S rather than .9 x y.p (as in condition A). This will permit determining how and where separation would occur between cask and railcar in the case of maximum accident.

Stress Summary - Condition A - Transit - .9 x y.p. Allowable

| <u>Impact at</u> | <u>Member</u> | <u>Stress</u> | <u>M.S. (.9 y.p.)</u> |
|------------------|----------------|--------------------|-----------------------|
| B End | Front Pin | 88,570 psi bending | .016 |
| " | Front Lug | 47,719 psi shear | .131 |
| " | Front Lug Weld | 23,685 psi tension | .14 |
| A End | Rear Pin | 64,527 psi comb. | .394 |
| " | Rear Lug | 44,151 psi shear | .223 |
| " | Rear Lug weld | 14,800 psi shear | .095 + |
| Bend | " " " | 25,925 psi comp. | .04 + |

Stress Summary - condition B - Accident - U.T.S allowable

(Note - above cask related members herewith re-evaluated at their U.T.S)

| <u>Impact at</u> | <u>Member</u> | <u>Stress</u> | <u>M.S. (U.T.S)</u> |
|------------------|------------------------|--------------------|---------------------|
| A End | Rear Pin | 64,527 psi comb. | .782 |
| " | Rear Lugs | 44,151 psi shear | .563 |
| " | Rear lug welds | 14,800 psi shear | 2.04 |
| " | Rear center rail plate | 57,160 psi shear | .207 |
| " | Rear side rail plate | 37,593 psi shear | .835 |
| " | Rear Weld #2 | 64,349 psi comb. | .165 * |
| B End | Front pin | 88,570 psi bending | .298 |
| " | Front Lug | 47,719 psi shear | .446 |
| " | Front Lug Weld | 23,685 psi Tension | 2.166 |
| " | Front rail plates | 47,041 psi Shear | .466 |
| " | Front Weld #1 | 20,872 psi Tension | 3.55 |
| " | Front Weld #2 | 82,077 psi Tension | .157 * |
| " | Front Weld #3 | 41,552 psi Tension | .80 |
| " | Rear Lug Weld | 25,925 psi Comp. | 1.892 |
| " | Rear Weld #2 | 47,914 psi comb. | .565 |

Conclusions - Tie-down Stress Analyses.

Condition A - Transit

The rear, or B end, tie-down takes the entire 10gL in both directions, plus the resultant vertical loads from all three (3) axes of component loadings. The stresses are greater when impact occurs at the front, or A end, putting the cask lug largely in direct tension.

The front, or A end, tie-down takes only vertical loads in tension, which develop only when impact is at the B end.

Satisfactory M.S. values apply to all cask related members at a conservative .9 x Y.P. stress for 10 gL, 5 gT, and 2g V for the transit condition.

Condition B - Accident

For A end impact all members have M.S. values greater than the .165 value for the sacrificial weld #2 of the rear (B end) tie-down weldment. This initiates the break-away procedure, followed by rip off of weld #2 of the front (A end) tie down weldment. Break-away of the cask occurs there at a longitudinal force of 11.65 gL.

For B end impact also all members are superior in strength to the #2 welds.

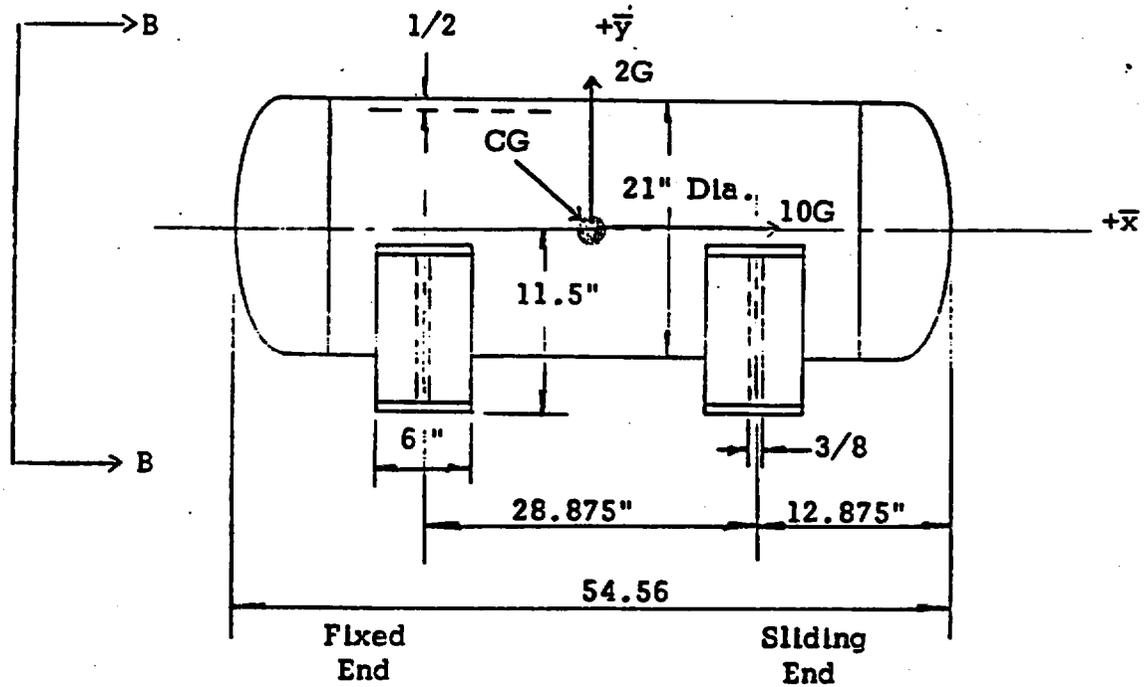
The sequence in the break-away, however is reversed. First the front weld #2 breaks in tension as the front end of the cask rises, followed by break-away of the rear weld #2.

Saddle Weldments are integral with the railcar structure and carry only compressive loads. They are not part of the accident failure situation.

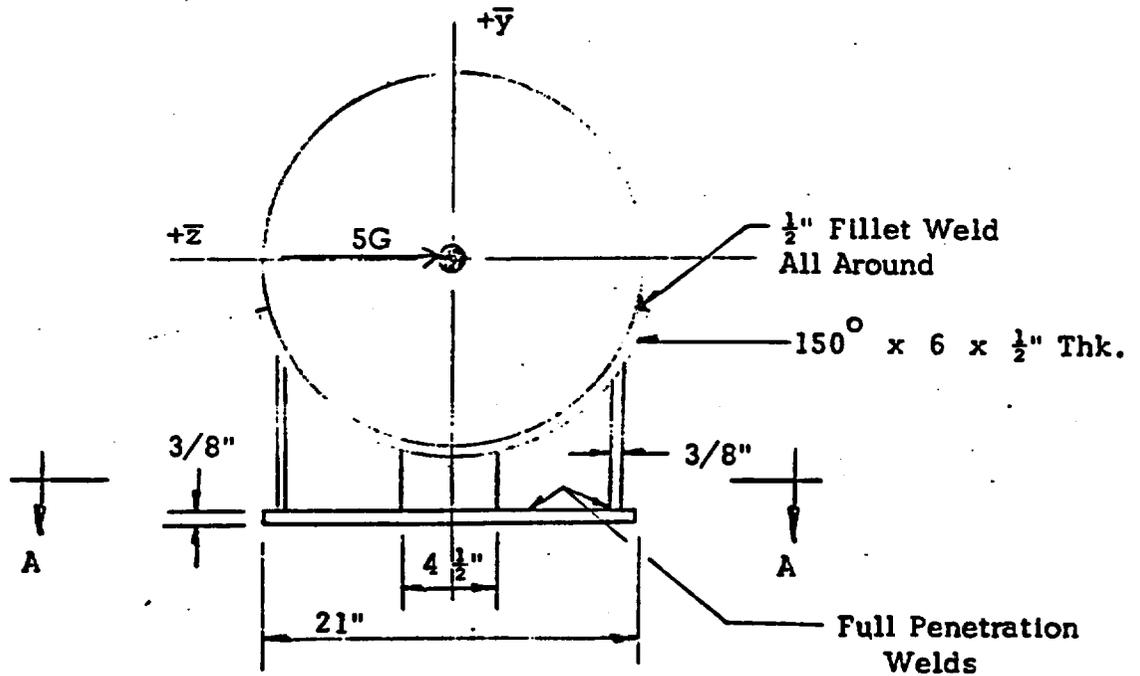
In conclusion, the design satisfies the specification requirement that the #2 welds are sacrificial in the accident condition.

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2.2.4 Expansion Tank Tie-Down



All Materials are 304 S.S.



VIEW B-B

There are four (4) expansion tanks as shown on NL Dr. 70665F.

The larger of the two tank sizes is used in the analysis.

Expansion Tank Design Weight - 1100 lbs. (for one tank)

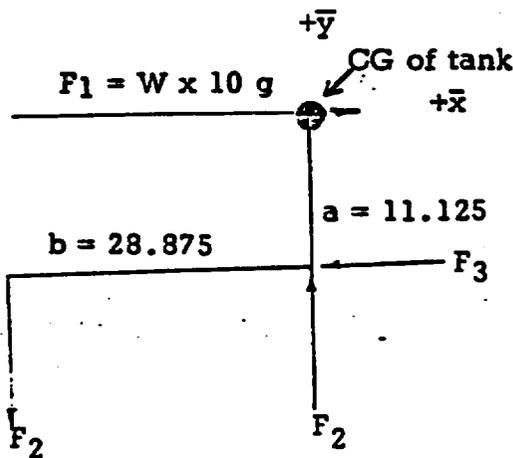
Tank tiedowns are designed to withstand simultaneously shock loads

of 10g longitudinal, 5g transverse and 2g vertical at the CG of the

tank, also they are designed with one tiedown fixed, the other will

slide to take care of thermal expansion.

Analysis of Section "A" - "A" for 10g longitudinal loadings



$$W = 1100 \text{ lbs.}$$

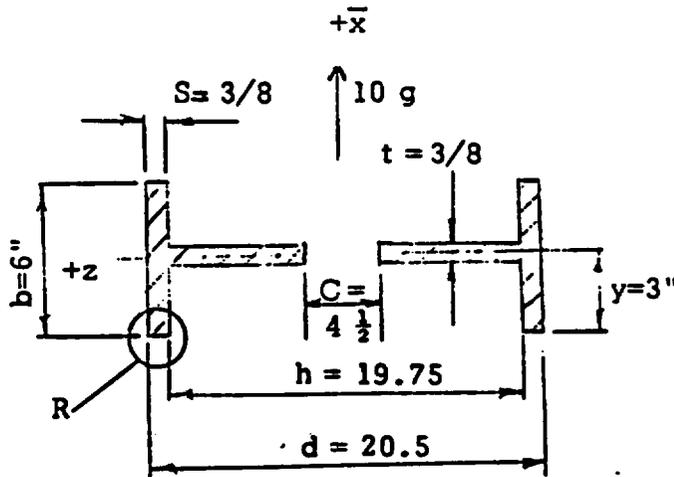
$$F_1 = 1100 \times 10 = 11000 \text{ lbs.}$$

$$F_1 (11.125) = F_2 (28.875)$$

$$11000 (11.125) = F_2 (28.875)$$

$$F_2 = 4238 \text{ lbs.}$$

Support Reactions



$$\begin{aligned} \text{Area } A &= [b(d-c)] - [h-c(b-t)] \\ &= [6(20.5-4.5)] - [19.75-4.5(6-.375)] \\ &= 10.22 \text{ in.}^2 \end{aligned}$$

Section "A-A"

$$St = \frac{F_2}{A} = \frac{4238}{10.22} = 415 \text{ psi}$$

$$S_s = \frac{F_1}{A}$$

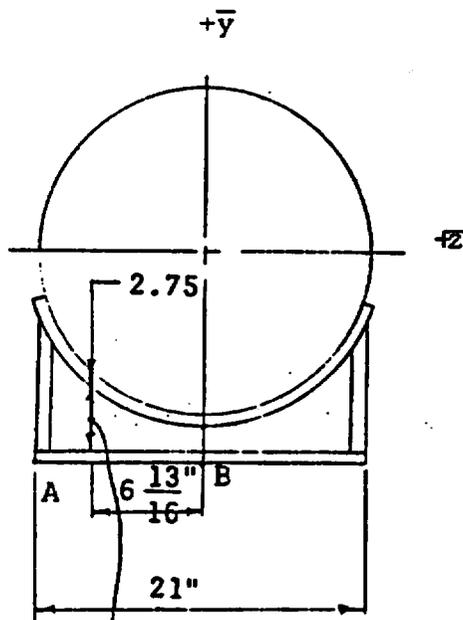
$$= \frac{11000}{10.22}$$

$$= 1076 \text{ psi}$$

$$F_1 = 11000 \text{ lbs.}$$

(The 10 g loading will be supported by the fixed support)

Bending stress thru Section A-A



Moment arm taken about
2/3 the distance between
B-A

$$M = 1100 (10 \text{ g}) (2.75)$$

$$= 30250 \text{ in. lbs.}$$

Section modulus for Section A-A

$$Z = \frac{2 S b^3 + (h-c) t^3}{6b}$$

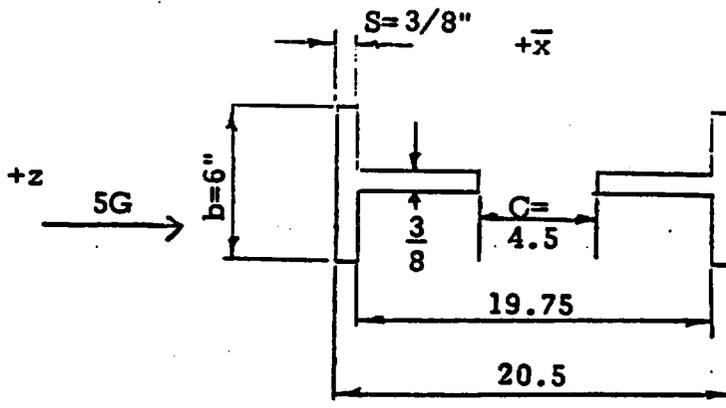
$$= \frac{2(.375)(6^3) + (19.75-4.5)(.375^3)}{6 \times 6}$$

$$= 4.52 \text{ in.}^3$$

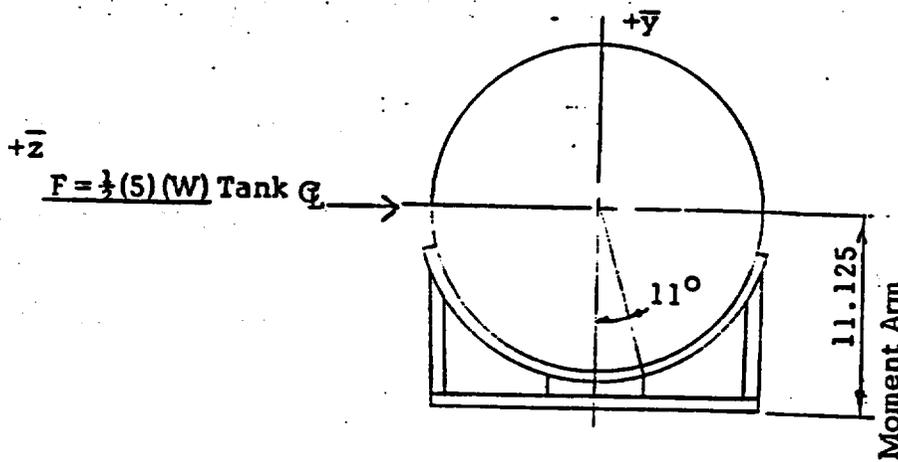
Bending stress $S_b = \frac{M}{Z}$

$= \frac{30250}{4.52} = 6692 \text{ psi}$

Analysis of Section "A-A" for 5 g Lateral Loading



Section "A-A"



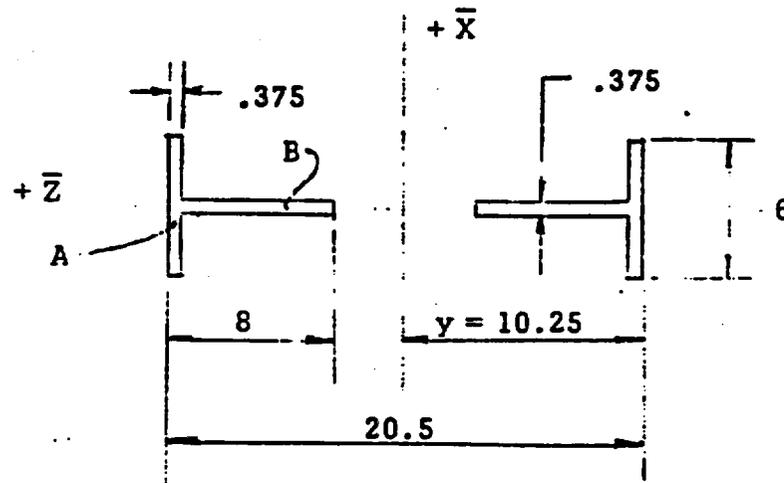
$F = \frac{1}{2} (5) (1100)$
 $= 2750 \text{ lbs.}$

(The 5 G loading will be supported by both tie-downs).

$$M = 2750 (11.125)$$

$$= 30594 \text{ in. lbs.}$$

Section Modulus for Section "A-A"



| $I_o \text{ in}^4$ | $d, \text{ in.}$ | $Ad^2, \text{ in}^4$ |
|--|------------------|----------------------|
| $A = \frac{2(6 \times .375^3)}{12} = 0.0527$ | 10.0625 | 455.6 |
| $B = \frac{2(.375 \times 7.625^3)}{12} = 27.7$ | 6.0625 | 210.18 |
| $I_o = 27.752$ | | 665.78 |

$$I = 27.752 + 665.78 = 693.5 \text{ in}^4$$

$$Z = \frac{693.5}{10.25} = 67.66 \text{ in}^3$$

Bending Stress thru Section "A-A"

$$S_b = \frac{M}{Z} \quad M = 30594 \text{ in. - lbs.}$$

$$Z = 67.66$$

$$= \frac{30594}{67.66} = 452 \text{ psi}$$

$$S_s = \frac{F}{A}$$

$$F = 4238 \text{ lbs.}$$

$$A = 10.22 \text{ in}^2$$

$$= \frac{4238}{10.22} = 415 \text{ psi}$$

Analysis of Section A-A for 2g vertical loading in-addition to 1 g vertical static weight applied front and rear supports.

2g - 1 g static load = 1 g (Section "A-A" in tension)

$$S_t = \frac{F}{A}$$

$$F = \frac{1100}{2}$$

$$= \frac{550}{10.22} = 54 \text{ psi}$$

$$= 550 \text{ lbs.}$$

$$= 10.22 \text{ in}^2$$

Calculate effective stress S_{e2} through Section "A-A" at point "R"

(See page 23, 24)

$$\sigma_x = 0 \text{ psi}$$

$$\sigma_y = 415 + 6692 + 452 + 54 = 7613 \text{ psi}$$

$$\sigma_z = 0 \text{ psi}$$

$$T_{xy} = 1076 \text{ psi}, T_{yz} = 415 \text{ psi}, T_{zx} = 0 \text{ psi}$$

$$S_{e2} = \sqrt{1/2 \sqrt{(0 - 0)^2 + (0 - 7613)^2 + (7613 - 0)^2 + 6(1076^2 + 415^2)}}$$

$$= 7871 \text{ psi}$$

Allowable stress ($S_{ad} = 0.9 S_{yd}$) at 250°F for 304 S/S from Sect. 1.1

under normal conditions and Sect. 1.2 equals to $.9 \times 41660 = 37500 \text{ psi}$

$$M.S. = \frac{37500}{7871} - 1 = 3.76$$

Calculating localized stresses for a cylinder that is supported by
a saddle - From Ref. 3

$$S \text{ max.} = K \frac{P}{t^2} \log_e \left(\frac{r}{t} \right)$$

$$K = 0.02 - 0.00012 (B-90) \quad B = 150^\circ - 22^\circ = 128^\circ$$

$$= 0.02 - 0.00012 (128-90)$$

$$= 0.01544$$

$$t = .5 \text{ in.}$$

$$K = 0.01544$$

$$P = 7380.37 \text{ lbs.}$$

$$R_{10g} = 4238 \text{ lbs.}$$

$$R_{5g} = 1492.37 \text{ lbs.}$$

$$R_{3g} = 1650 \text{ lbs.}$$

$$R_{3g} = \frac{(3)(1100)}{2}$$

$$= 1650 \text{ lbs.}$$

$$\frac{1}{2} M = L (R_{5g})$$

$$\frac{1}{2} (5500) (11.125) = 20.5 (R_{5g})$$

$$R_{5g} = 1492.37 \text{ lbs.}$$

$$P = 1650 + 1492.37 + 4238$$

$$= 7380.37 \text{ lbs.}$$

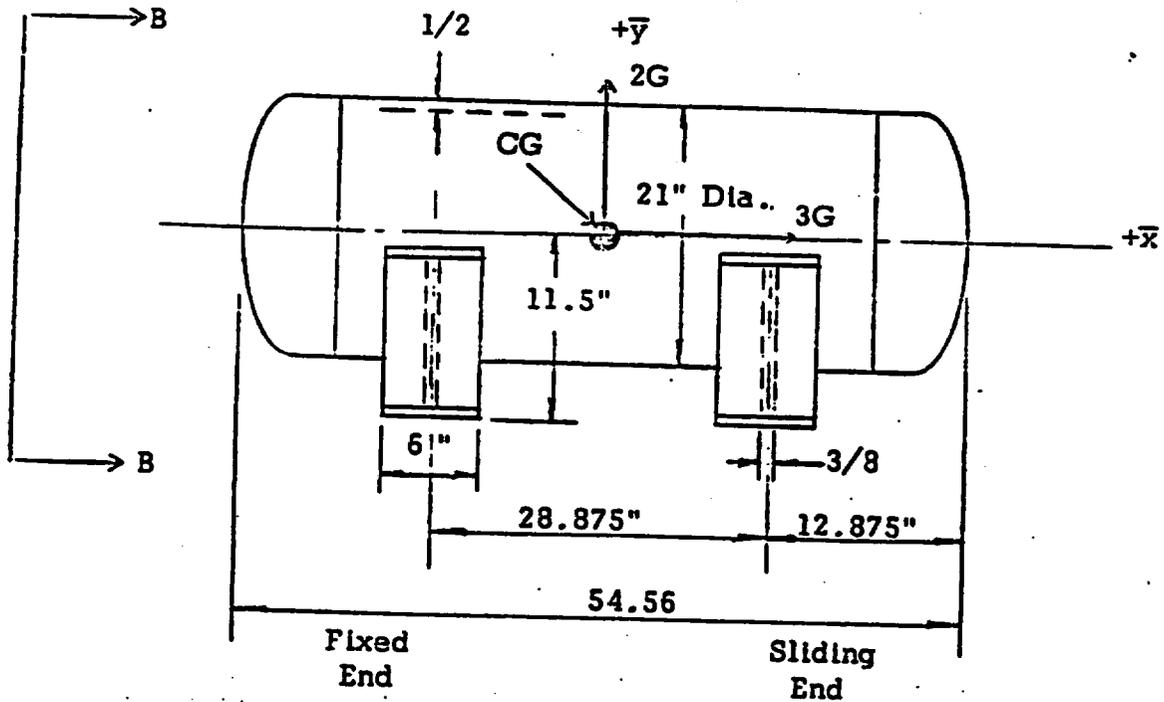
$$S \text{ max} = 0.01544 \left(\frac{7380.37}{.5^2} \right) \log_e \frac{10.5}{.5} = 603 \text{ psi.}$$

Allowable stress ($S_{as} = 0.9 S_{ys}$) at 350°F for 304 S/S from Sect. 1.1
under normal condition and Sect. 1.2 equals to $0.9 \times 21500 = 19350 \text{ psi.}$

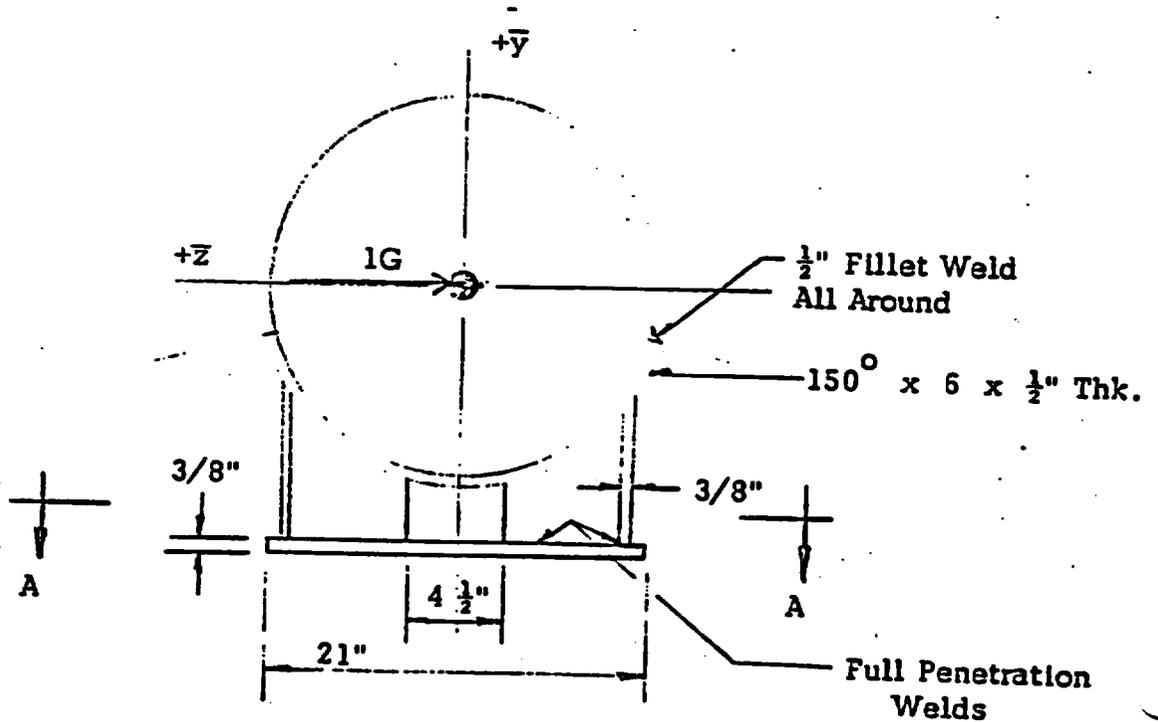
$$M.S. = \frac{19350}{603} - 1 = 31.08$$

BOLT STRESS CALCULATIONS:

The bolt loads are those anticipated during normal transport conditions as established on pg. XI-3-8 (3g longitudinal, 1g transvers, 2g vertical) simultaneously at the CG of the tank:



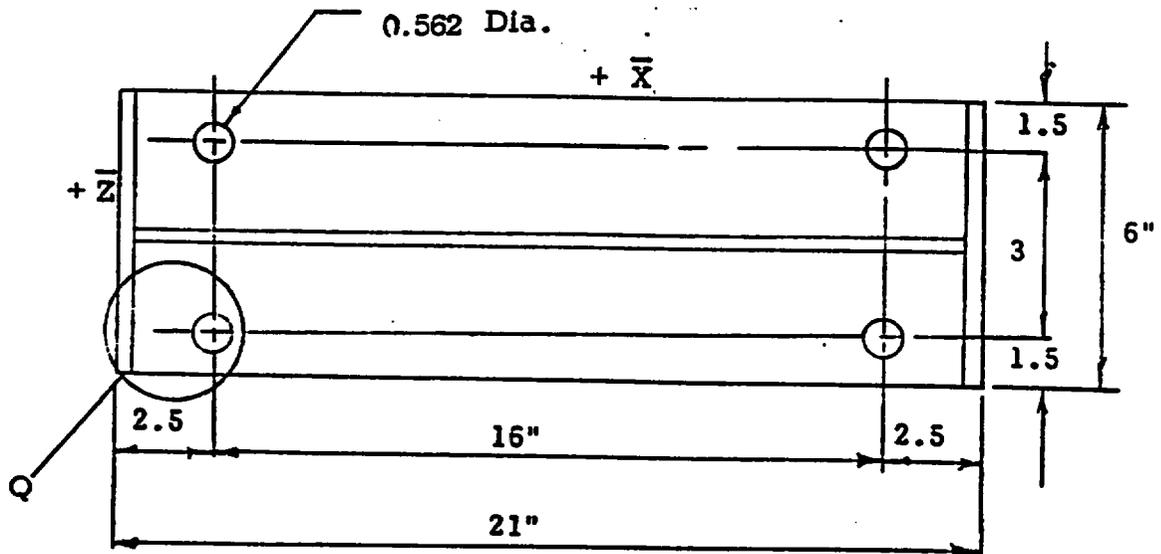
All Materials are 304 S.S.



VIEW B-B

XI-2-29a

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Fixed End Tie-down Base Plate (Sect. A-A) pg XI-2-29a

Expansion tank will be bolted down with 8 of 1/2 -13 UNC bolts,
ASTM A193 Grade B7.

Tensile strength = 125,000 psi.; Yield strength = 105,000 psi.

Tensile stress area, $A_t = 0.1419 \text{ in.}^2$

Thread shear area, $A_s = 0.1257 \text{ in.}^2$

For lg lateral loadings

$$(1100) (1) (11.5) = F_1 (4) (16)$$

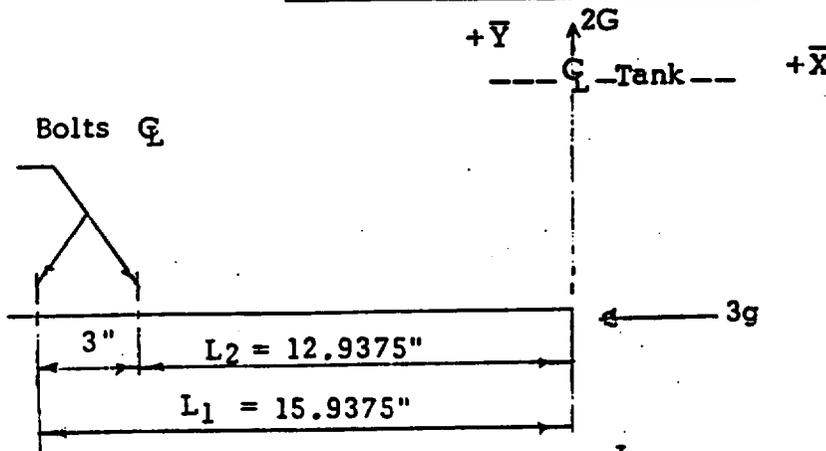
$$F_1 = 198 \text{ lbs.}$$

$$8 F_2 = 1100 (1)$$

$$F_2 = 137.5 \text{ lbs.}$$

$$S_t = \frac{198.}{0.1419} = 1395 \text{ psi.}$$

$$S_s = \frac{137.5}{0.1257} = 1094 \text{ psi}$$

For 3 g Longitudinal Loading

$$(1100) (3) (11.5) = \left(\frac{L_1}{L_2} F_3 L_1 + F_3 L_2 \right) (2)$$

$$37950 = \left(\frac{15.9375}{12.9375} \right) \left[F_3 (15.9375) + F_3 (12.9375) \right] (2)$$

$$F_3 = 533 \text{ lbs.}$$

$$4 F_4 = 1100 (3)$$

$$F_4 = 825 \text{ lbs.}$$

$$S_t = \frac{533}{0.1419} = 3756 \text{ psi.}$$

$$S_s = \frac{825}{0.1257} = 6563 \text{ psi.}$$

Analysis of bolt joint for for 2g vertical loading in addition to 1g vertical static weight applied front and rear supports 2g - 1g static = 1g tension on the bolts.

1 g Upward Loading

$$S_t = \frac{137.5}{0.1419} = 969 \text{ psi.}$$

$$8 F_5 = 1100 (1)$$

$$F_5 = 137.5 \text{ lbs.}$$

Calculate effective stress on the bolt at point Q (See page XI-2-30)

$$\sigma_x = 0 \text{ psi}$$

$$\sigma_y = 1395 + 3756 + 969 = 6120 \text{ psi}$$

$$\sigma_z = 0 \text{ psi}$$

$$T_{xy} = 6563 \text{ psi}, T_{yz} = 1094 \text{ psi}, T_{zx} = 0 \text{ psi}$$

$$S_{e1} = \sqrt{1/2} \sqrt{(0 - 0)^2 + (0 - 6120)^2 + (6120 - 0)^2 + 6(1094^2 + 6563^2)}$$

$$= 13048 \text{ psi}$$

Allowable stress ($S_{as} = .75 S_{ys}$) under normal condition from Sect. 1.1

and Sect. 1.2 equals $0.75 (105000) = 78750 \text{ psi}$; $S_{e1} = 13048 \leq S_{All} = 78750 \text{ psi}$

Total tension force on each bolt at the fixed end is 868.5 lbs.

Apply a clamping force to all bolts of 1200 lbs.

$$S_t = \frac{1200}{0.1419} = 8457 \text{ psi}$$

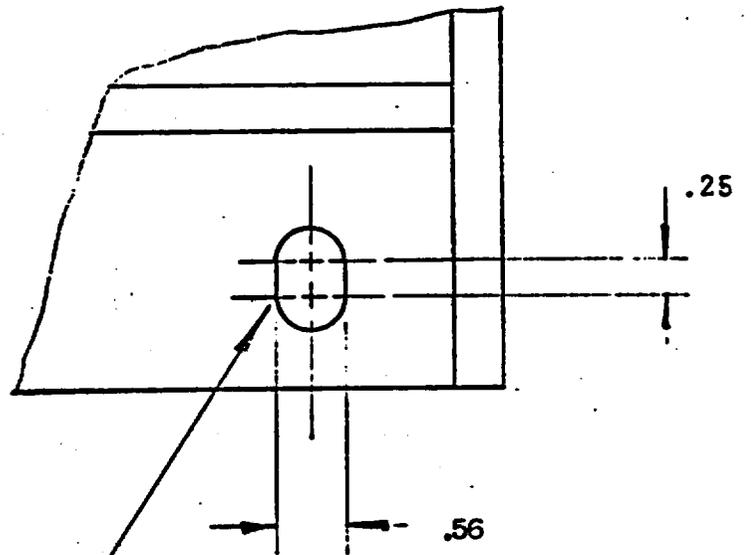
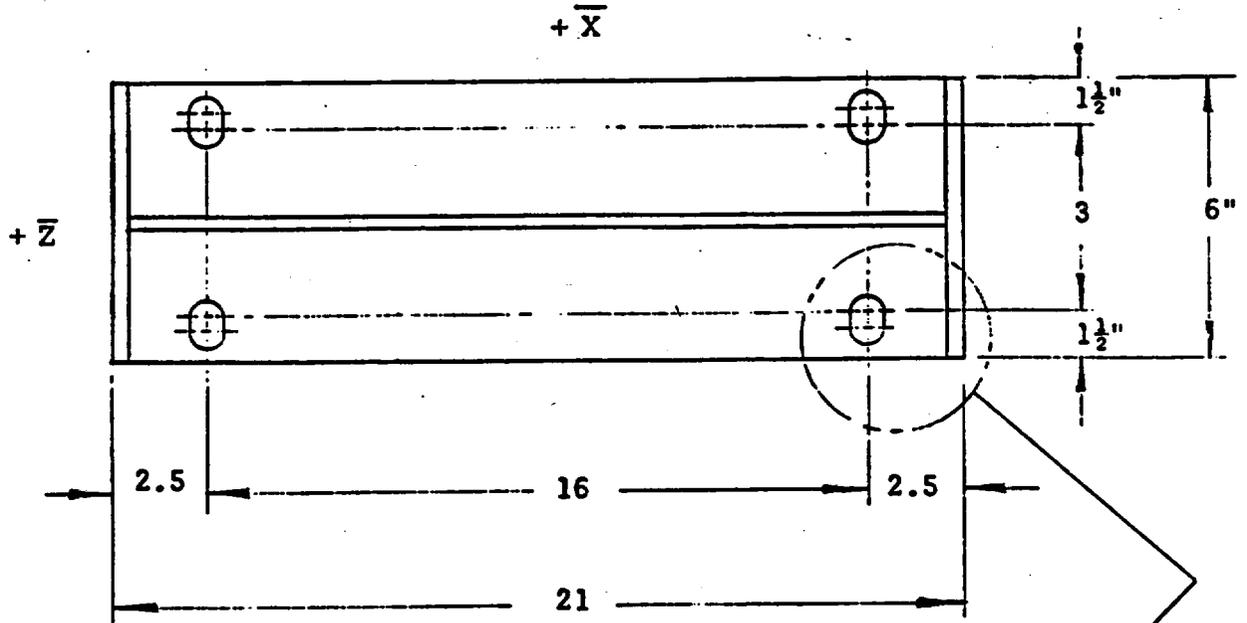
Calculating new effective stress on the bolts.

$$S_e = \sqrt{\frac{1}{2}} \sqrt{(0 - 0)^2 + (0 - 8457)^2 + (8457 - 0)^2 + 6(1094^2 + 6563^2)}$$

$$S_e = 14294 \text{ psi.}$$

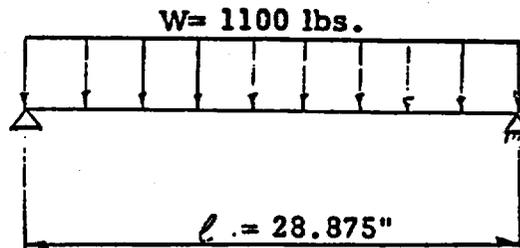
$$M.S. = (78250/14294) - 1 = 4.47$$

Sliding End Base Plate



Slots to take care of expansion of tanks

Analysis of Expansion tank in bending due to a 3 g vertical downward load. The tank is treated as a simply supported beam loaded by a uniform transverse load due to its own weight.



Effective shock loads
on the tank
 $= \sqrt{3^2 + 1^2} = 3.16g$

For a beam with a uniform load, end supports from Ref. 3 ,
Table III, Case 13.

$$\text{Max. } M = 1/8 WL$$

$$M = 1/8 (1100) 3.16 (28.875) = 12546 \text{ in.lbs.}$$

Section Modulus of Tank

$$Z = 0.098 \left(\frac{21^4 - 20^4}{21} \right) = 161 \text{ in.}^3$$

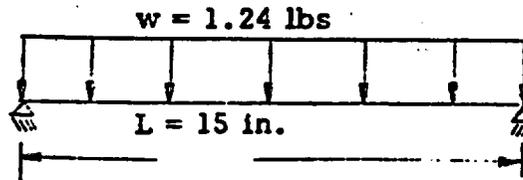
$$S_b = \frac{12546}{161} = 78 \text{ psi.}$$

Allowable stress ($S_{ad} = 0.9 S_{yd}$) for 304 S/S at 340°F from Sect. 1.1
under normal condition and Sect. 1.2 equals to $.90 \times 41667 = 37500 \text{ psi}$

$$\text{M.S.} = \frac{37500}{74} - 1 = 479$$

2.2.5 Expansion Tanks Connecting Lines

Expansion tanks connecting lines during normal transportation are subject to shock loads of 3g longitudinal, 2g vertical and 1g transverse. Largest span between pipe supports is 15 in. Analysis of longitudinal inner connecting lines for 2g vertical and 1g transverse.



effective g's on the pipe. Vertical g's =
 $2 - 1 \text{ Static} = 1g$
 Effective g's $= \sqrt{1^2 + 1^2} = 1.41g$

For a beam with a uniform load, end supports from Ref. 3, Table III, case 13.

$$\text{Max. } M = 1/8 w L$$

$$M = 1/8(1.24)(1.41)(15)$$

$$= 3.27 \text{ in. lbs.}$$

Section Modulus for $\frac{1}{2}$ " pipe sch. 40

$$Z = 0.048 \frac{.840^4 - .622^4}{.840} = .0406 \text{ in.}^3$$

$$S_b = \frac{3.27}{.0406} = 80.5 \text{ psi}$$

S_{all} for 304 S.S. at 340°F from Sect. 1.1, 1.2 is

$$.9 \times 41667 = 37500 \text{ psi}$$

$$\text{M.S.} = \frac{37500}{80.5} = 465$$

Analysis of transverse inner connecting lines for 3g longitudinal and 2g vertical. Largest span between pipe supports is 15 in. Effective g's on the pipe:

$$\text{Vertical g's} = 2 - 1 \text{ Static} = 1g$$

$$\text{Effective g's} = \sqrt{3^2 + 1^2} = 3.16 \text{ g's}$$

For a beam with a uniform load, end supports from Ref. 3, Table III,
case 13.

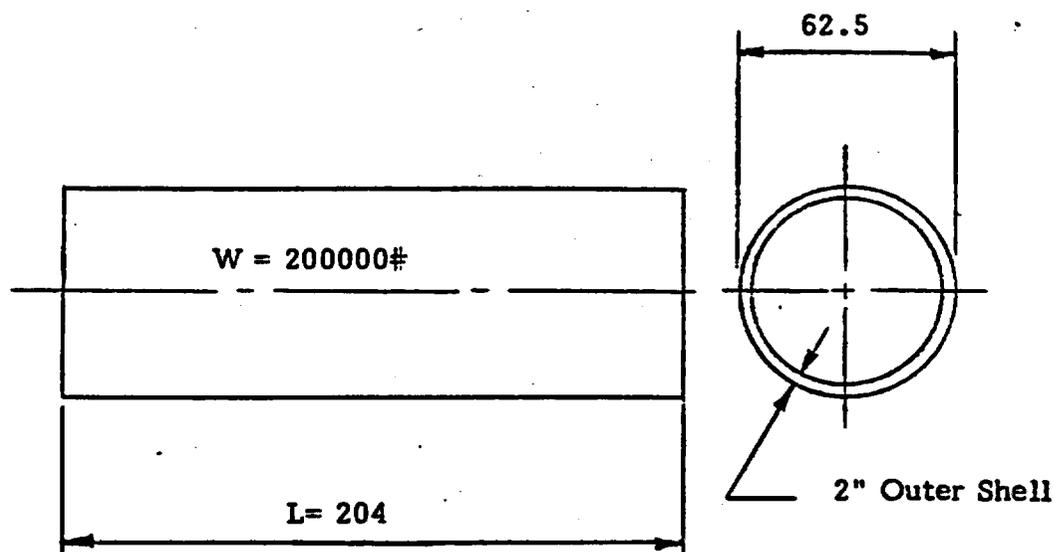
$$\begin{aligned}\text{Max } M &= 1/8 w L \\ &= 1/8 (1.24)(3.16)(15) \\ &= 7.347 \text{ in.lbs.}\end{aligned}$$

$$S_b = \frac{7.347}{.0406} = 181 \text{ psi}$$

$$\text{M.S.} = \frac{37500}{181} - 1 = 206$$

2.3 LOAD RESISTANCE - CASK AS A SIMPLE BEAM 10 CFR 71.32 (a)

Consider the cask as a simple beam with a uniformly distributed load equal to 5 times the total design weight.



Maximum bending moment is:

$$M = \frac{5WL}{8} \quad \text{Ref. 3}$$

$$M = \frac{5(200,000)(204)}{8} = 25.5 \times 10^6 \text{ in. lb.}$$

Maximum bending stress in outer shell is:

$$S = \pm \frac{M}{Z}$$

$$Z = I/C = \frac{\pi}{4 R_o} (R_o^4 - R_i^4) = \frac{\pi}{4(31.25)} (31.25^4 - 29.25^4) =$$

$$Z = 5535 \text{ in.}^4$$

$$S = \frac{25.5 \times 10^6}{5535} = 4610 \text{ psi} < S_{\text{All.}} = 40,000 \text{ psi at room temperature.}$$

2.4 EXTERNAL PRESSURE 10 CFR 71.32 (b)

Packaging shall be adequate to assure that the containment vessel will suffer no loss of contents if subjected to an external pressure of 25 psig. The .75 inch thick by 45 inch diameter inner shell is the containment vessel.

(a) Inner Shell @ 25 psig

$$D_o = 46.5 \text{ in. outside dia.}$$

$$t = .75 \text{ in. thk. of shell}$$

$$L = 180 \text{ in. length of cavity}$$

From Ref.5 Fig. UHA-28.1

$$L/D_o = 180/46.5 = 3.87$$

$$D_o/t = 46.5/.75 = 62$$

$$B = 5,500 \text{ psi @ } 541^\circ \text{ F}$$

Thus, the max. allowable pressure is

$$P = \frac{B}{D_o/t} = \frac{5,500}{62} = 88.71 \text{ psi} > 25 \text{ psi}$$

(b) Top and Bottom Heads @ 25 psig

The loads imposed by the end impact under the hypothetical accident conditions are more severe than the 25 psig external pressure loading. The integrity of the top and bottom heads is analyzed for the limiting condition of impact, see Section 4.0 Hypothetical Accident Conditions.

2.5 Heat 10 CFR 71 Appendix A

Pressures and temperatures for normal operation without auxiliary cooling were calculated assuming the cask is setting in direct sunlight at an ambient temperature of 130°F in still air. The resulting pressures and temperatures were used to evaluate the structural adequacy of the package for normal conditions of transport. The results of this analysis show that the package meets all structural requirements.

2.6 Cold 10 CFR 71 Appendix A

The cask is designed to be operated at -40°F in still air and shade without adverse effects. The neutron shield is filled with a 52.5 - 47.5 volume percent ethylene glycol - water mixture which freezes below -40°F. Since the cask containment vessel is dry, safe operation is not dependent on a minimum decay heat load.

2.7 Vibration

The vibration environment and the g-loadings applicable to flat railcars is reported in "A Survey of Environmental Conditions Incident to the Transportation of Materials" by General American Transportation Corporation, October 1971. (NTIS PB-204 442)

The data extracted herewith show the following g-loads:

| | Longitudinal | Transverse | Vertical |
|-------------------|--------------|------------|----------|
| Peak envelope g's | .50 | .50 | 1.00 |
| Overall rms. g's | .37 | .38 | .79 |

The following excerpts were taken from the above referenced report which was prepared for the Office of Hazardous Materials, Department of Transportation.

Flat Car - Extensive measurements of the vibration environment on a railroad flat car are reported in Reference 13. Data are presented for measurements in the vertical direction, the lateral direction and the longitudinal direction. The data has been replotted to the same format as the previous data and are presented in Figures 18, 19, and 20. Events included in the data are: train leaving switching yards, stopping, crossing intersecting tracks, climbing a hill, going downhill with braking, on level runs at 40 mph, crossing switches, crossing bridges, on rough track, on curves, and in tunnels. Weighting factors were used to account for the probability of occurrence of these events when developing the summarized data. The test car was part of three different train lengths varying in size from 65 to 120 cars.

The measurements indicate that there are two frequency bands in which the highest amplitudes occur, the 0-5 Hz and the 5-10 Hz bands. It is reported that the amplitude distributions in these bands showed little resemblance to vibration type distributions. Most of the peaks in these bands were a result of transient impulses rather than steady state vibration. Above 10 Hz, the vibration levels were below .72 g. In all the frequency bands analyzed, the peaks in the vertical direction were highest.

The transverse direction shows higher levels than the longitudinal, although in the frequency range below 15 Hz, the order is reversed. As a result of this study, it was concluded that the rail environment consists of low level random vibration with a number of repetitive transients superimposed in the low-frequency ranges.

A comparison of the various events based on overall rms values was developed in the report and is shown below.

| Event | Axis | | |
|---------------------------------------|----------------|-----------------|----------------|
| | Long. rms g | Trans. rms g | Vert. rms g |
| Flat Runs 40, 50 mph | 0.25 | 0.19 | 0.35 |
| Up, Down Hills | 0.12 | 0.18 | 0.29 |
| Across Switches & Intersecting Tracks | 0.10 | 0.14 | 0.41 |
| Rough Tracks | 0.04 | 0.04 | 0.19 |
| Around Curves | 0.12 | 0.03 | 0.09 |
| Across Bridges | 0.12 | 0.04 | 0.21 |
| Through Tunnels | 0.12 | 0.17 | 0.28 |
| Overall | 0.37 | 0.38 | 0.79 |

The highest inputs in both the longitudinal and transverse directions resulted from straight flat runs at 40-50 mph. The vertical direction was most severe when crossing switches and other tracks.

A comparison of the frequency spectra for various operating conditions is shown in Figures 22 to 24. Figure 22 shows the effect of speed on the vibration spectra. Figure 23 shows the effect of direction of measurement for a particular event. Figure 24 compares the vibration spectra in the various directions for a transient event. Only the peak and rms values are plotted on these curves. (7)

In a study to determine the effect of height above floor, on the vibration environment, it has been reported (Reference 14) that there is little difference in the levels, at least to a height of six-foot above the cargo floor, which was the limit of the study.

Design conditions for vibration loadings were then established as shown below:

| Vibration (g) | | | |
|---------------------|------|------|------|
| * Peak envelope g's | 0.50 | 1.00 | 0.50 |
| Overall rms g's | 0.37 | 0.79 | 0.38 |

* These peak values are used to evaluate the package integrity under vibration.

XI-2-38C

GENERAL AMERICAN RESEARCH DIVISION
ACCELERATION - G's (0-PEAK)

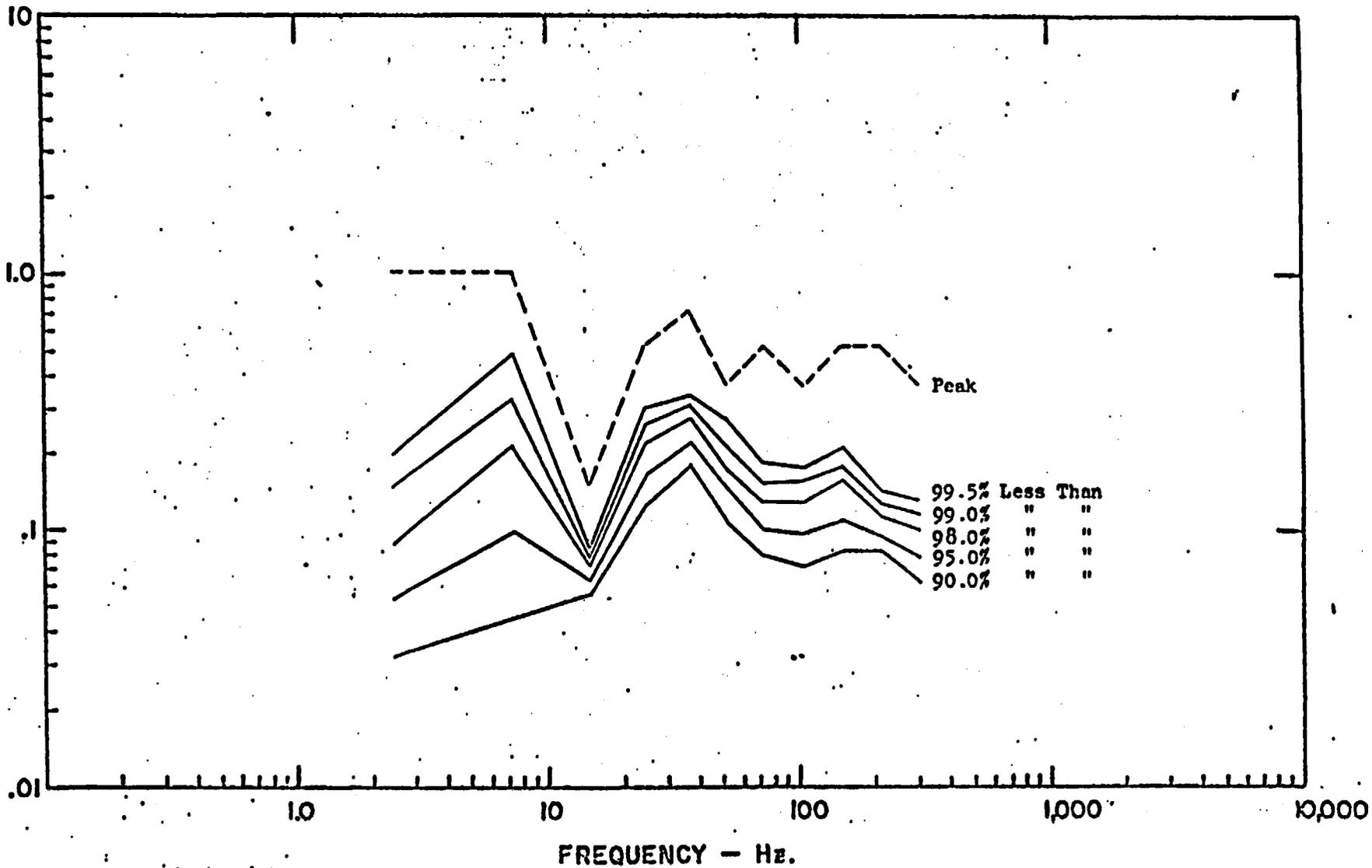


Figure 18. FREQUENCY SPECTRA, RAILROAD, VERTICAL DIRECTION; COMPOSITE OF VARIOUS CONDITIONS

P8-2-388

ACCELERATION - G's (0-PEAK)

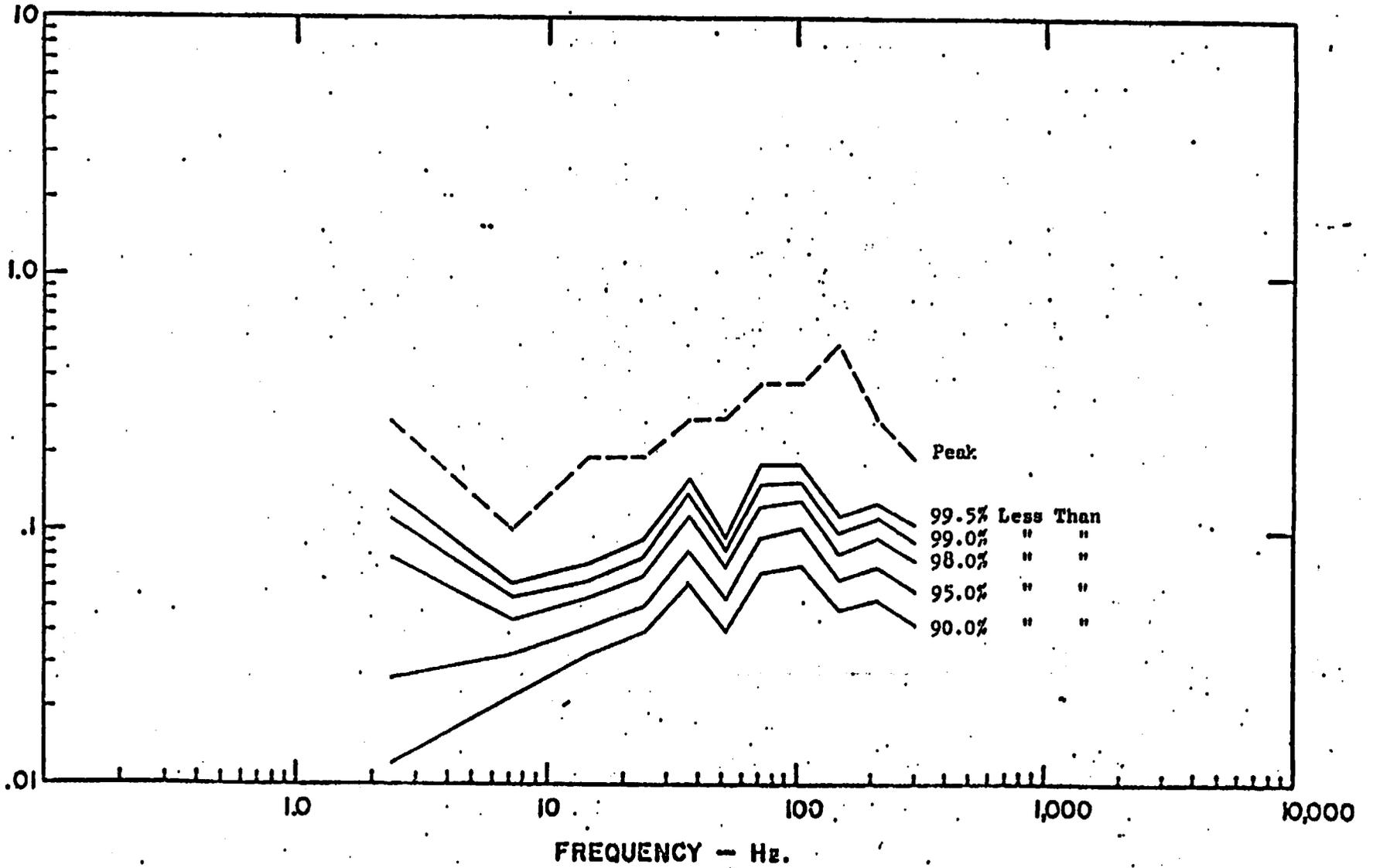


Figure 19 FREQUENCY SPECTRA, RAILROAD, TRANSVERSE DIRECTION, COMPOSITE OF VARIOUS CONDITIONS

1/31/75

XI-2-38e

GENERAL AMERICAN RESEARCH DIVISION
ACCELERATION - G's (0-PEAK)

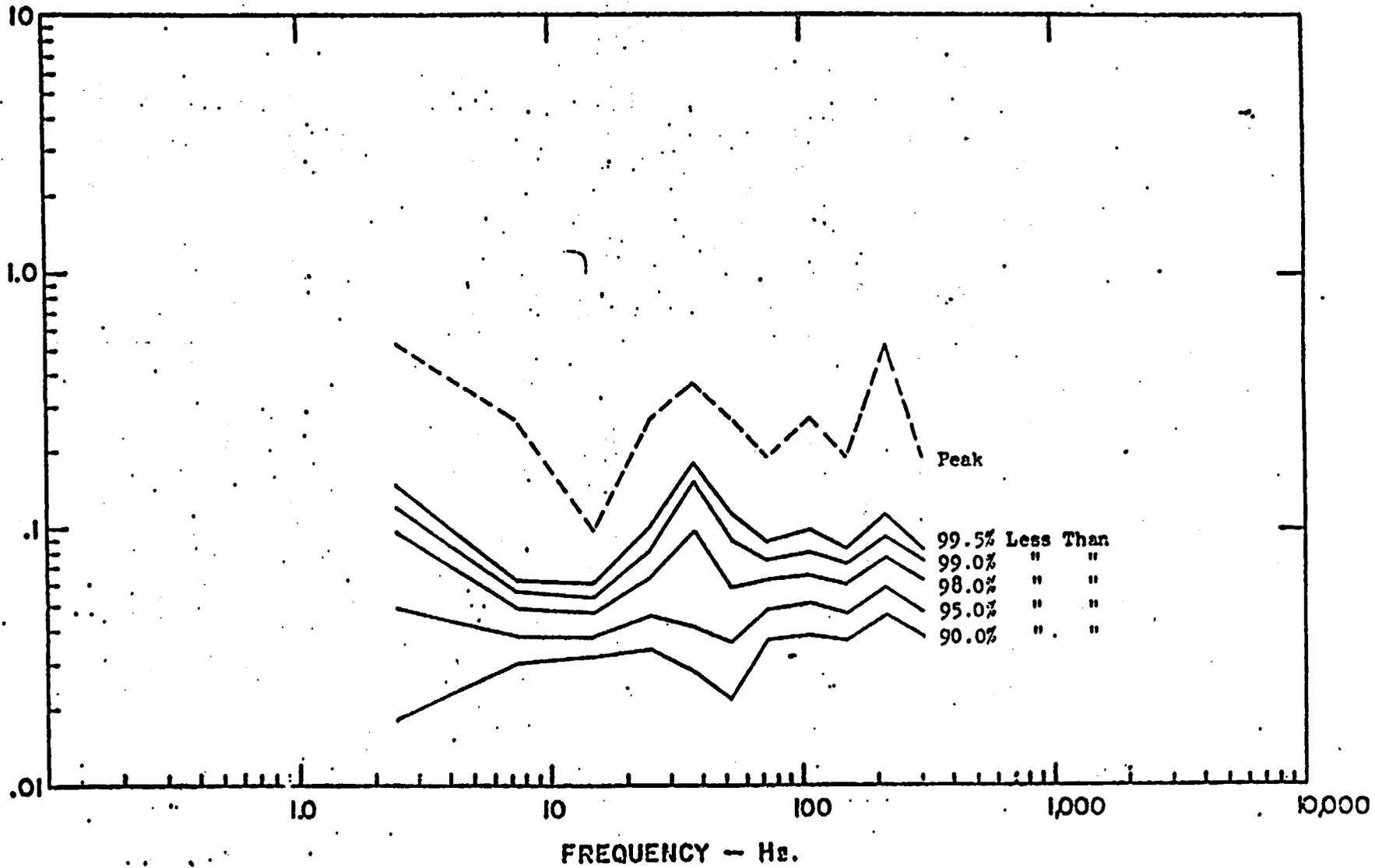


Figure 20 FREQUENCY SPECTRA, RAILROAD, LONGITUDINAL DIRECTION, COMPOSITE OF VARIOUS CONDITIONS

1/31/75

Data On Six Wheel Trucks for Railcar

There are two very favorable factors acting to prevent any excessive resonance forces. First, the three axle truck renders the truck nearly insensitive to any spacing of bolted rail joints, since two of the three axles, and one of the wheels on the third axle, are not subject at any instant to the vertical deflections at the rail joint. The car manufacturers feel that if all railcars were equipped with three axle trucks, that vibration, roll and other similar track disturbances would be practically negligible. Second, each truck has substantial frictional dampers that absorb vibrational energy. The net result is to set a very practical upper limit to the force transmitted to the railcar frame & thence to the cask supports.

The following information is furnished by Standard Car Truck Company, Chicago, Ill., and is applicable to the Buckeye Steel Castings six wheel trucks used on the NLI car.

There are eight spring groups on the two trucks of the rail car. See figure, page XI-2-38j.

| | | |
|------------------------|-------------------------|----------------|
| Gross rail weight | = | 350,000 lbs. |
| Est. unsprung wt. | = | 29,200 lbs. |
| Spring wt. per group | = | 40,100 lbs |
| Spring rate | | 27,730 lbs/in. |
| Spring capacity | | 72,422 lbs/in. |
| Deflection, loaded car | $\frac{40100}{27730}$ = | 1.446 in. |

$$\text{Reserve travel } \frac{72422 - 40100}{27730} = 1.165 \text{ in.}$$

Spring force under each friction wedge

$$\frac{8998 - 1.165 \times 2360}{2} = 3124 \text{ lb.}$$

Normal force between friction wedge and wear plate

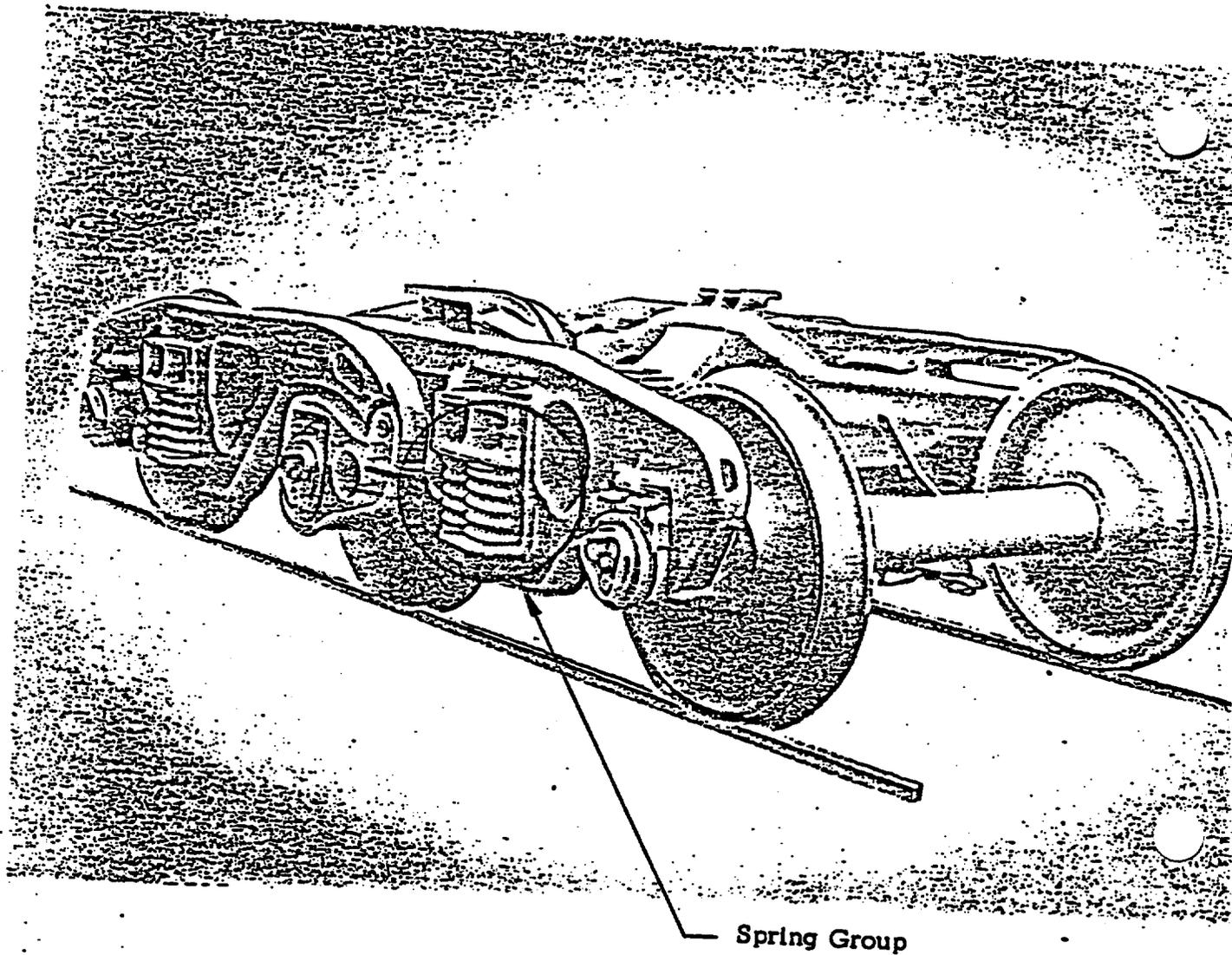
$$3124 \times \tan 55^\circ = 4461 \text{ lbs.}$$

Friction force at 0.25 coefficient of friction

$$8 \times 4461 \times 0.25 = 8923 \text{ lbs. per car}$$

Spring rate for entire car

$$8 \times 27730 = 221840 \text{ lbs/in.}$$



Natural frequency of railcar and cask on truck springs

Spring rate of all springs $K = 221840$ lbs/in.

Mass above springs $\frac{316000}{386.4} = 818.65$

$$f_n = \frac{1}{2\pi} \sqrt{\frac{K}{M}} = \frac{\sqrt{\frac{221840}{818.65}}}{6.283} = 2.62 \text{ c.p.s}$$

Damping in the Trucks.

Coulomb type friction is developed by special wear plates incorporated in the spring assemblies of the trucks of the railcar.

A constant friction force in the trucks for the whole car is $F = 8923$ lbs.

The equivalent viscous damping coefficient is

$$C_{eq} = \frac{4F}{\pi wX} \quad \text{where } w = 2\pi \times 2.62 \text{ at resonance} \\ = 16.462 \text{ rad/sec}$$

$X =$ amplitude of oscillation

$$C_{eq} = \frac{690}{X}$$

For viscous damping, critical damping $= C_c = 2mw$

$$C_c = 2 \times 818.65 \times 16.462 = 26953$$

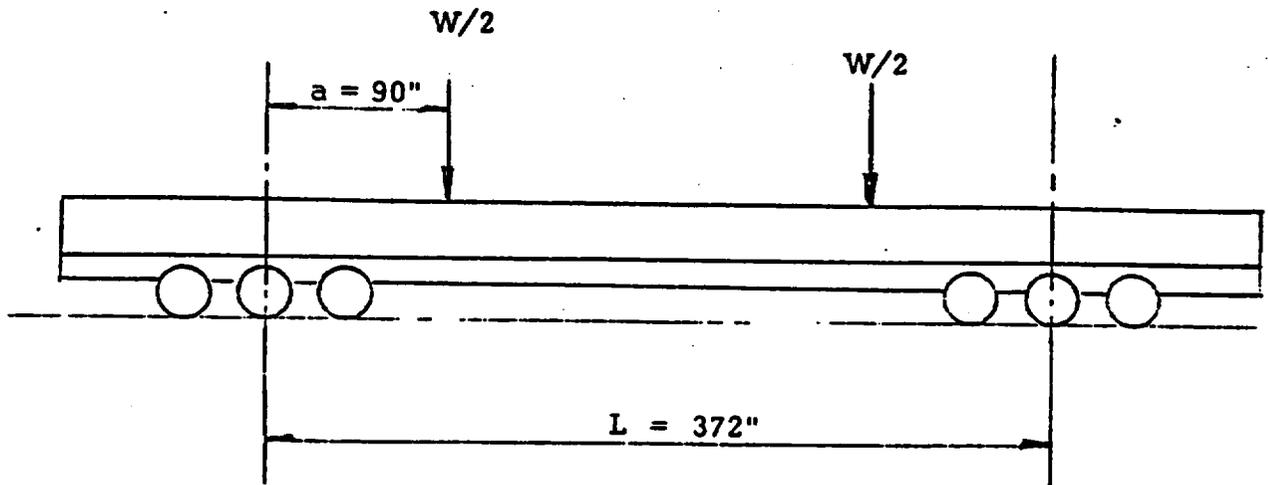
$$\% \text{ critical damping} = \frac{C_{eq}}{C_c} = \frac{690}{26953X} = \frac{1}{39.06X}$$

The amplitude allowable before springs bottom is 1.165 in. $= X$

$$\text{Therefore } \% \text{ critical damping is } \frac{1}{39.06 \times 1.165} = 2.2\%$$

This gives very satisfactory damping.

Natural Frequency of Cask on Railcar Frame



$$f_n = \frac{3.13}{\sqrt{\frac{(W + 0.486 wl) (a^2) (3L - 4a)}{12 EI}}}$$

$$I_{3 \text{ sills}} = 10359 \text{ in}^4 \text{ Ref.-}$$

$$w_{\text{frame}} = 90.694 \text{ lbs/in.}$$

$$wl = 33738 \text{ lbs}$$

$$.486wl = 16400 \text{ lbs}$$

$$a^2 = 8100 \text{ in}^2$$

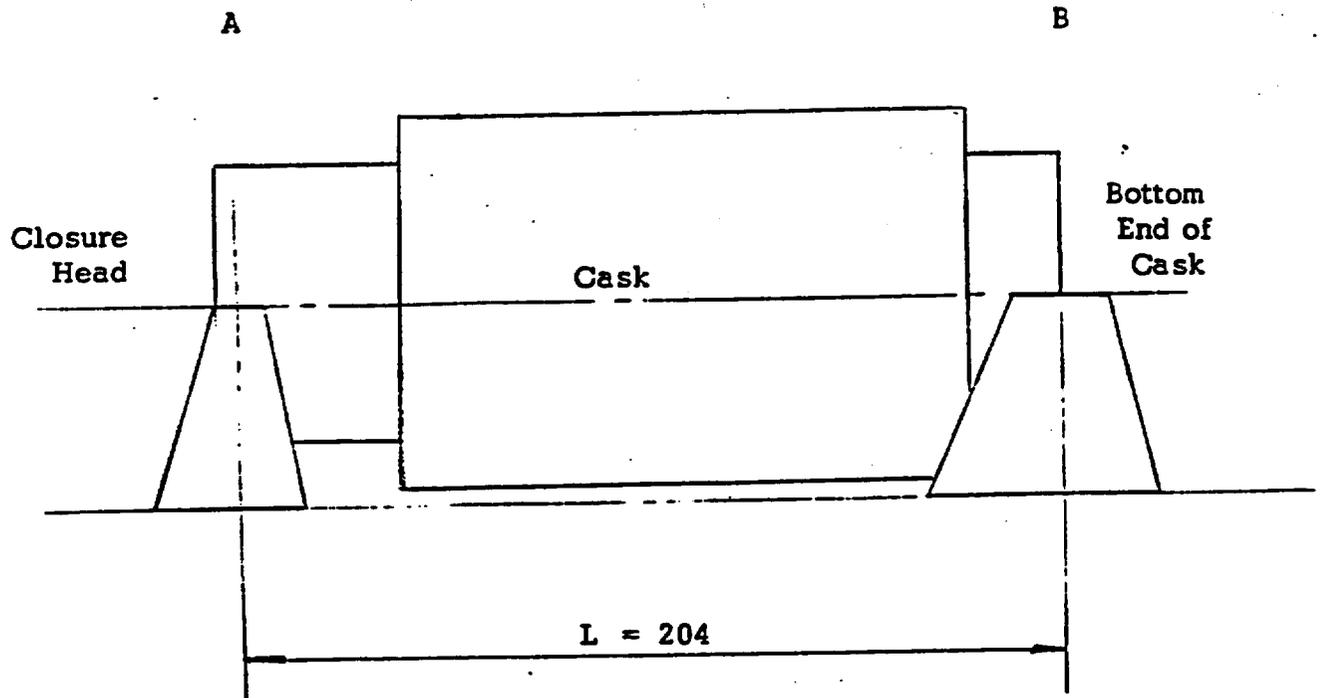
$$3L - 4a = 3 \times 372 - 4 \times 90 = 756 \text{ in.}$$

$$12 EI = 12 (30 \times 10^6) 10359 = 3729240 \times 10^6$$

$$W = 220000 \text{ lbs.}$$

$$W + .486 wl = 236400 \text{ lbs.}$$

$$f_n = \frac{3.13}{\sqrt{\frac{(236400) (8100) (756)}{3729240 \times 10^6}}} = 5.024 \text{ c.p.s.}$$

Natural Frequency of Simply Supported Cask

$$f_n = \frac{9.87}{2} \sqrt{\frac{EIg}{WL^3}} \text{ c.p.s.}$$

| | |
|-----------------------|--|
| <u>EI</u> Inner Shell | $.74906 \times 10^{12}$ |
| Lead | $.6481996 \times 10^{12}$ |
| Outer Shell | 4.6753×10^{12} |
| Water Jacket | $\frac{4.2581 \times 10^{12}}{10.33 \times 10^{12}}$ |

$$g = 386.4 \text{ in/sec}^2$$

$$w = 220000 \text{ lbs.}$$

$$L = 204 \quad L^3 = 8.489664 \times 10^6$$

$$f_n = 1.57 \sqrt{\frac{10.33 \times 10^{12} \times 386.4}{.220 \times 8.489664 \times 10^{12}}}$$

$$= 1.57 \sqrt{2137.127} = 72.58 \text{ c.p.s.}$$

Damping between Cask and A Support

The saddle of A & B supports is lined with low friction plastic material. There is a small amplitude of motion because the cask is fixed longitudinally at the B support and the flexing of the whole car frame in transit causes the saddle areas of supports A and B to move alternately toward each other and away from each other. Thus, the cask experiences a doubled amplitude motion relative to support A, developing coulomb type damping.

The weight on front support A is 110,000 lbs.

Coefficient of friction of brake lining on support is 0.1

Longitudinal frictional force

$$F = 0.1 (110,000) = 11,000 \text{ lbs. under 1 g load Vertical}$$

Equivalent viscous damping is

$$C_{eq} = \frac{4F}{\pi wX} \quad w = 2\pi \times 5.06 = 31.793 \text{ rad/sec.}$$

$$= \frac{4 \times 11,000}{\pi \times 31.793X} = \frac{440.6}{X}$$

For viscous damping, critical damping $C_c = 2mw$

$$C_c = 2 \times 569.36 \times 31.793 \quad m = \frac{220,000}{386.4} = 569.36$$

$$= 36,203$$

$$\% \text{ critical damping} = \frac{C_{eq}}{C_c} = \frac{440.6}{36,203 \times X} = \frac{1}{82.17}$$

For example, the distance X moved under 1 g static load is

$$X = .178 \text{ in.}$$

$$\% C_c = \frac{1}{82.17 \times .178} = \frac{1}{14.63} = .068 = 6.8\%$$

Compared with the damping for the trucks of only 2%, which is satisfactory, the times greater damping applied to the cask is wholly adequate for all normal transportation.

Transmissibility T_f of system springs.

Natural frequencies:

- | | |
|---|-----------|
| 1) car and cask masses on truck springs | 2.62 cps |
| 2) cask mass on car frame elastance | 5.024 cps |
| 3) simply supported cask | 72.58 cps |

Transmissibility factor of force T_f thru a spring element upon the mass attached is dependent upon the ratio $\frac{w}{w_n}$, the forcing frequency divided by the natural frequency of the mass on that spring. Reference is then made to Fig. 2.17 of Ref 36 for finding T_f for any $\frac{w}{w_n}$ ratio.

1) Truck Springs These are subjected to a whole spectrum of loads applied by the truck conditions. w is variable.

$w_n = 2.62$ cps The damping is practical and sufficient. Experience shows .72 g is max transmitted to the bolsters. This value is given on page X1-3-8.

2) Frame Elastance

$$\frac{w}{w_n} = \frac{2.62}{5.06} = .51 \quad C/C_c = .068$$

$T_f = 1.4 \times$ force applied at bolsters

$$= 1.4 \times .72g = 1.008 \text{ g effective on cask supports.}$$

This damping is relatively high and sufficient to prevent undesirable performance at resonance.

3) Cask Flexibility

$$\frac{w}{w_n} = \frac{5.06}{73.25} = .069$$

The acceleration response is obtained by differentiating Eq. (2.36):

$$\frac{\ddot{x}}{F_0/m} = -\frac{\omega^2}{\omega_n^2} R_d \sin(\omega t - \theta) = -R_a \sin(\omega t - \theta) \quad (2.37)$$

The velocity and acceleration response factors defined by Eqs. (2.36) and (2.37) are shown graphically in Fig. 2.13, the former to the horizontal coordinates and the latter to the coordinates having a negative 45° slope. Note that the velocity response factor approaches zero as $\omega \rightarrow 0$ and $\omega \rightarrow \infty$, whereas the acceleration response factor approaches 0 as $\omega \rightarrow 0$ and approaches unity as $\omega \rightarrow \infty$.

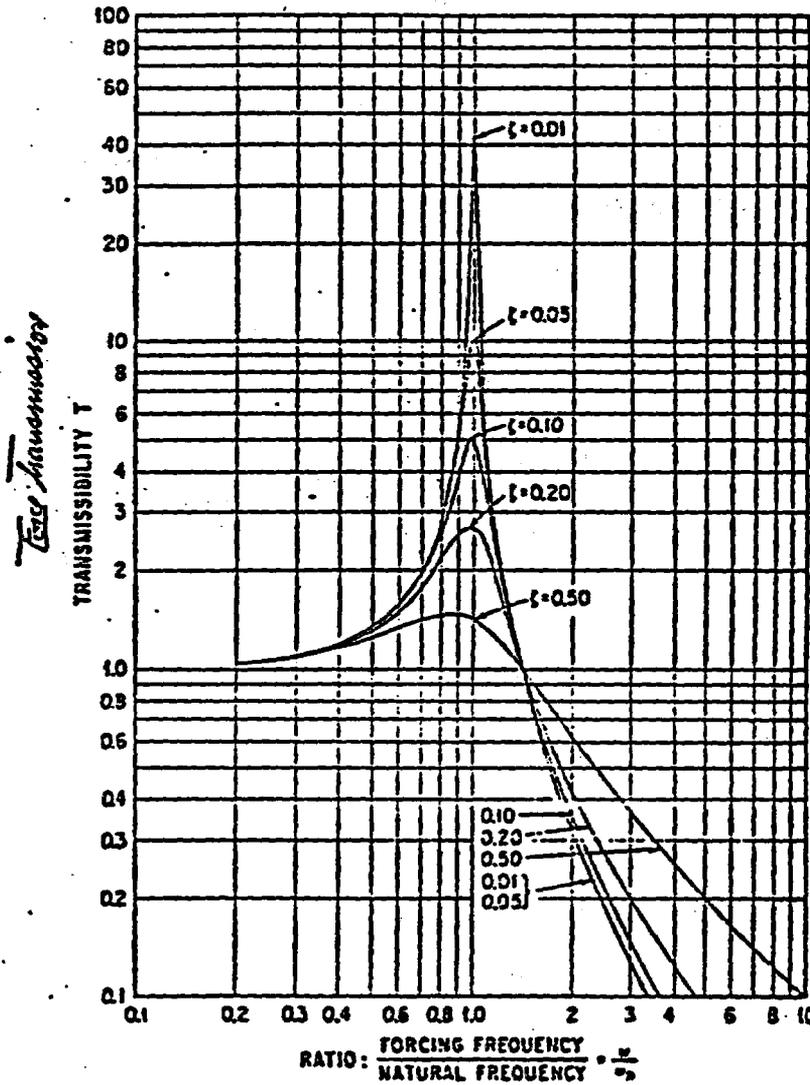


FIG. 2.17. Transmissibility of a viscous-damped system. Force transmissibility and motion transmissibility are identical numerically. The fraction of critical damping is denoted by ζ .

$T_f = 1.0$ for any value of C/C_c of the internal friction of cask.
= $1.0 \times 1.008 \text{ g}$ -- insignificant.

The conclusion is that vibration g's are low and not critical and are sufficiently damped to prevent severe resonance conditions.

Vibration of flexible hose connection between water jacket and expansion tank.

Assembly consists of two disconnect couplings and a 15-inch length of flexible hose, corrugated stainless steel, heavy weight, Anaconda BW-21-1H, 1/2 inch nominal I.D.

The approved minimum length of such hose is 5-1/2 inches, based on vibration tests of a cantilever length of hose as reported in Anaconda Corrugated Hose Bulletin CR (3ED) page CR-14. This is the equivalent, in a simply supported length, of 11 inches minimum. The actual length being 15 inches thus satisfies the design test conditions for vibration.

Vibration of Expansion Tanks

Two lengths of tanks are used, with different spacing between supports. The outer diameter is 21 inches, with 1/2 inch wall. Tanks are filled with water.

$$I = .049(21^4 - 20^4) = 1,689.5 \text{ m}^4$$

$$EI = 4.899 \times 10^{10}$$

| | |
|------------------|-------------------|
| Longer Tanks (2) | Shorter Tanks (2) |
|------------------|-------------------|

$$a = 12.84$$

$$a = 12.81$$

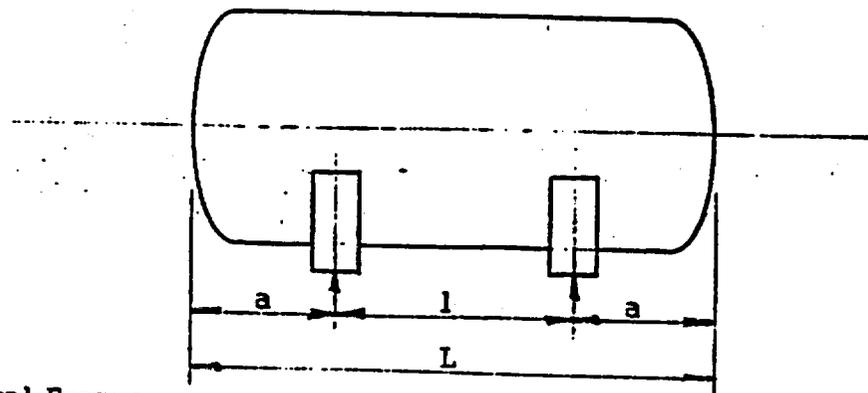
$$l = 28.875$$

$$l = 21.937$$

$$L = 54.562$$

$$L = 47.562$$

Weight = 1100 lbs. Weight = 1000 lbs.



Natural Frequency

$$f_n = \frac{22.4}{2\pi} \sqrt{\frac{EIq}{WL^3}}$$

Ref 36, pg. 1-14

For Longer Tanks

$$f_n = 3.565 \sqrt{\frac{4.899 \times 10^{10} \times 386}{1100 \times 54.562^3}} = 1,160 \text{ cps filled}$$

For Shorter Tanks

$$f_n = 3.565 \sqrt{\frac{4.899 \times 10^{10} \times 386}{1000 \times 47.562^3}} = 1,425 \text{ cps filled}$$

For 500 lbs. Empty Tank - Larger

$$f_n = 3.565 \sqrt{\frac{4.899 \times 10^{10} \times 386}{600 \times 54.562^3}} = 1.29 \times 1,160 = 1,496 \text{ cps empty}$$

In every case, the frequency is so high that excitation forced by the rail car cannot have a transmission ratio more than unity, thereby practically limiting vibration forces to 1g, as indicated for the cask itself.

Vibration Analysis of Expansion Tank Connecting Lines

Largest span between pipe supports of the lines connecting the expansion tanks is 15 inches.

Angular natural frequency for beams of uniform section and uniformly distributed load from Ref. 36 equals to

$$W_n = A \sqrt{\frac{EI}{uL^4}} \text{ Rad./sec.}$$

$E = 26.9 \times 10^6$ psi, modulus of elasticity. Sect. 1.2 for 304 S/S
moment of inertia for $\frac{1}{2}$ " sch. 40 pipe equals to

$$\begin{aligned} I &= .049 (D^4 - d^4) \\ &= .049 (.840^4 - .622^4) \\ &= .01706 \text{ in.}^4 \end{aligned}$$

$L = 15$ in. length of pipe.

$u =$ Mass per unit length of pipe, lb-sec²/in.²

Weight of pipe full of liquid is .0827 lb./in.

$$u = \frac{.0827}{386} = .000214 \text{ lb-sec}^2/\text{in}^2$$

$A = 9.87$ coefficient for a simply support beam.

$$W_n = 9.87 \sqrt{\frac{26.9 \times 10^6 \times .01706}{.000214 \times 15^4}} = 2031 \text{ Rad/sec}$$

$$\text{Converting Rad/sec to cycles/sec} = \frac{2031}{2\pi} = 323 \text{ cycles/sec}$$

The frequency calculated for the largest span between pipe supports is high enough that the piping system will not be excited by the railcar vibration.

2.8 FREE DROP (10 CFR 71, Appendix A)

Packages which weigh more than 30,000 pounds must be evaluated for a free drop through a distance of one foot onto a flat essentially unyielding horizontal surface, striking the surface in a position for which maximum damage is expected.

It is necessary to establish a more credible set of conditions by which to assess the safety of the package for normal conditions of shipment.

Conditions of normal transport should be those conditions experienced by the railroads and considered as normal events in the course of rail shipment. These normal events can be put into two general categories; (1)

Shock associated with normal car handling & (2) vandalism. Shocks

associated with normal car handling result in longitudinal, vertical and

lateral forces being applied to the package which is always in the normal shipping attitude, i.e., horizontal. Therefore in considering a more

credible set of conditions by which to assess the safety of the package,

it must first be established that the position of the package is always

horizontal. It follows, then, that the only position in which the package

could experience a one foot free drop is horizontal. A one foot drop of the

package is not considered to be a credible situation related to normal rail

transport. However, to assess the integrity of the package, the conditions

of assessment must be such as to provide confidence that the package

will not reach the accident mode conditions as a result of normal transport.

Therefore, the package is analyzed for a one foot side drop with the impact

limiters striking the hypothetical unyielding surface.

Section XI-4.3 established the behavior of the impact limiter system. From the graph on page XI-4-25, it can be determined that a one foot side drop would result in a peak g value of 14.55. Using the same analytical methods used to determine structural integrity of the cask in the 30 foot drop accident, we obtain the following component stresses and resulting margins of safety.

The following table, which has been extracted from Sect. 3.8.5, compares the calculated effective stress S_{e2} to the stress allowable $S_{ad} = 0.9 S_{yd}$. The condition analyzed is as follows, normal transport condition, 70 kw decay heat load, 130° F ambient, 16.5 psig internal pressure, 235 psig water jacket pressure, one foot side drop.

| Location | Effective Stress S_{e2} | Temp. | S_{as} $0.9S_{yd}$ |
|----------|---------------------------|-------|-------------------------|
| 1 | 3155 | 268 | 38970 |
| 3 | 3672 | 248 | 39420 |
| 5 | 1796 | 218 | 40094 |
| 7 | 3188 | 316 | 37890 |
| 9 | 4717 | 303 | 38183 |
| 11 | 4806 | 420 | 35910 |
| 13 | 6538 | 359 | 36923 |
| 15 | 15103 | 323 | 37733 |
| 17 | 2113 | 302 | 38205 |
| 19 | 2551 | 290 | 38475 |
| 21 | 1766 | 292 | 38430 |
| 23 | 3710 | 271 | 38902 |
| 25 | 2676 | 241 | 38902 |
| 27 | 523 | 218 | 39577 |
| 29 | 1752 | 260 | 39150 |
| 31 | 412 | 256 | 39240 |
| 33 | 502 | 240 | 39600 |

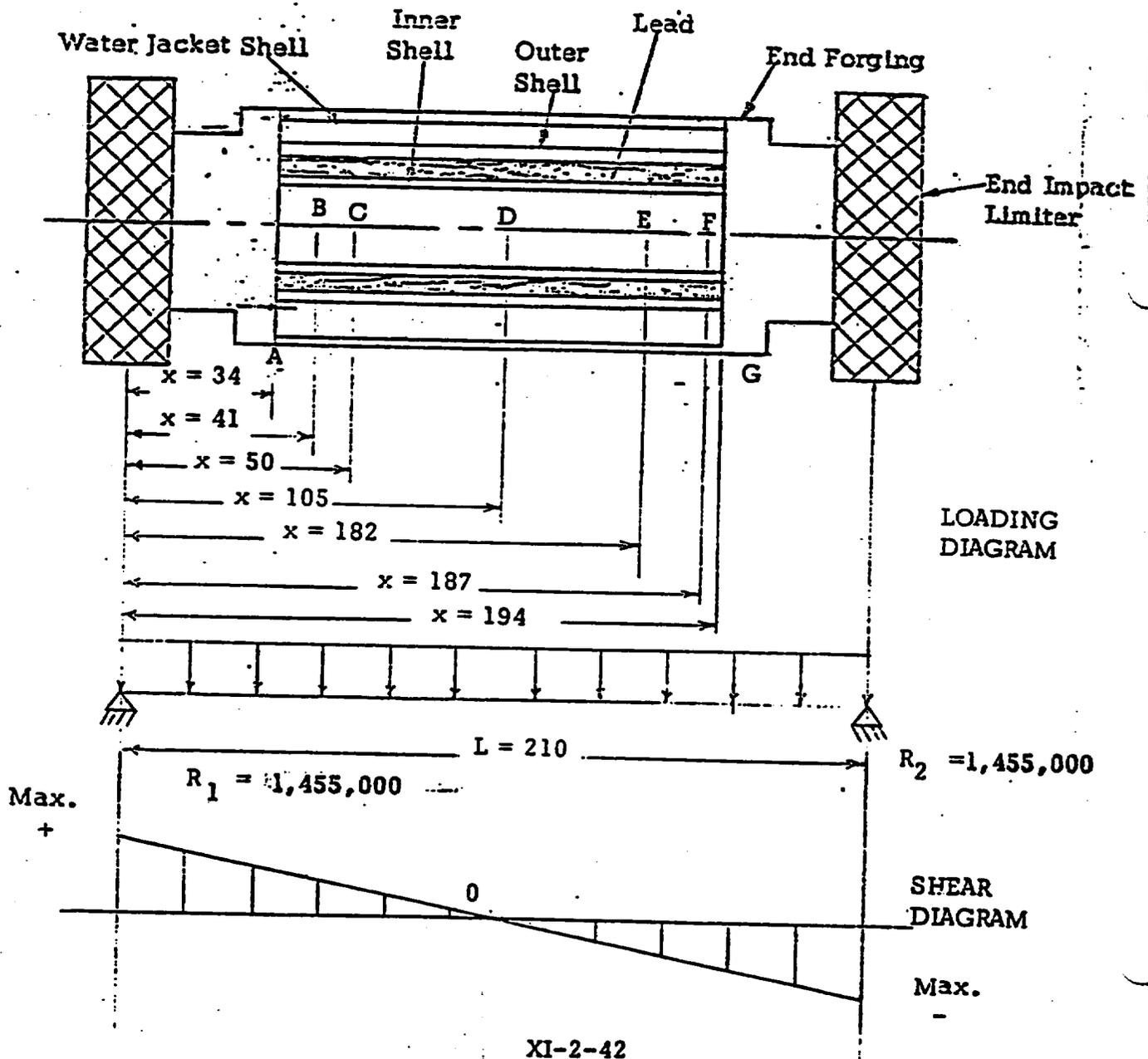
The following table compares the calculated effective stress S_{e2} to the stress allowable $S_{ad} = 0.9 S_{yd}$ for the cold condition case (-40° F isothermal), one foot side drop. The effective stresses for the 1G case were calculated in Sect. 3.8.6.

| Location | 1G x 14.55G | = | S_{e2} | S_{ad} |
|----------|-------------|---|----------|----------|
| 1 | 204 | | 2968 | 69000 |
| 3 | 278 | | 4045 | 69000 |
| 5 | 274 | | 3987 | 69000 |
| 7 | 237 | | 3448 | 69000 |
| 9 | 372 | | 5413 | 69000 |
| 11 | 377 | | 5485 | 69000 |
| 13 | 512 | | 7450 | 69000 |
| 15 | 506 | | 7362 | 69000 |
| 17 | 174 | | 2532 | 69000 |
| 23 | 237 | | 3448 | 69000 |
| 25 | 200 | | 2910 | 69000 |
| 27 | 197 | | 2866 | 69000 |
| 29 | 106 x 14.55 | = | 1542 | 69000 |

The stress range calculations for the above operating ranges were carried out in Sect. 3.8.

Shear Stress Calculation

Calculate maximum shear stress in the shells at points "A", "F", and "G".
 The maximum shear stress is calculated at 90° away from where the maximum bending stress occurs. Calculations are done for the two extremes of the normal transportation conditions: (1) cold condition, -40° F isothermal, no decay heat load; (2) normal conditions, 130° F ambient, 70 kw decay heat 16.45 psig internal pressure, and 235 psig water jacket pressure.



Calculate maximum shear stresses in the inner shell, outer shell and water jacket shell at point "A".

Shear stresses calculated for 81g side impact (Sect. 4.6.2) in the inner shell, outer shell and water jacket shell at point "A" equals to 15401 psi for 14.55g one foot drop shear stresses at point "A" equals to $\frac{15401 (14.55)}{81} = 2766$ psi.

Shear stresses in the outer shell and water jacket at point "F" equals to

$$\begin{array}{l} 17787 \quad \times \quad \frac{14.55}{81} = 3195 \text{ psi} \\ \text{(Sect. 4.6.2)} \end{array}$$

Shear stresses in the inner shell at point "G" equals to $19306 \times \frac{14.55}{81} = 3468$ psi.

Effective stresses S_{e2} in the inner and outer shell at -40°F are as follows.

| | A | F | G |
|--------------------|----------------------|----------------------|----------------------|
| Inner Shell | $S_{e2} = 2766$ psi. | | $S_{e2} = 3468$ psi. |
| Outer Shell | $S_{e2} = 2766$ psi. | $S_{e2} = 3195$ psi. | |
| Water Jacket Shell | $S_{e2} = 2766$ psi. | $S_{e2} = 3195$ psi. | |

Allowable stress ($S_{ad} = 0.9 S_{yd}$) at -40°F for 304 S.S. From Sect. 1.1 under normal conditions and Sect. 1.2 equals to $0.9 \times \frac{2}{3} (115000) = 69000$ psi.

Inner Shell, Outer Shell and

$$\text{Water Jacket Shell at point "A"} \quad MS = \frac{69000}{2766} - 1 = 23.9$$

$$\text{Inner Shell at point "G"} \quad MS = \frac{69000}{3468} - 1 = 18.9$$

Outer Shell and Water Jacket Shell at point "F"

$$MS = \frac{69000}{3195} - 1 = 20.6$$

Calculate effective stresses S_{e2} in the shells at point "A", "F", "G" at normal temperature conditions.

Calculate effective stresses S_{e2} in the inner shell, outer shell and water jacket shell at point "A".

Effective stress S_{e2} in the inner shell at point "A"

$$\sigma_x = 12.1 \text{ psi (Radial Stress) From Sect. 3.8}$$

$$\sigma_y = -1315 \text{ psi (Tangential Stress) From Sect. 3.8}$$

$$\sigma_z = -606 \text{ psi (Axial Stress) From Sect. 3.8}$$

$$T_{yz} = \begin{matrix} .2766 \\ \text{(Sect. 3.8)} \end{matrix} \text{ psi (Shear Stress)}$$

$$T_{xy} = 0 \quad T_{zx} = 176 \text{ (Sect. 3.8)}$$

$$S_{e2} = \sqrt{\frac{1}{2} \left[(12.1 - (-1315))^2 + (-1315 - (-606))^2 + (-606 - 12.1)^2 + 6(2766)^2 \right]}$$

$$= 4936 \text{ psi.}$$

Allowable stress ($S_{ad} = 0.9 S_{yd}$) at 268° F for 304 S.S. From Sect. 1.1

under normal conditions and Sect. 1.2 equals to $0.9 \times 43300 = 38970 \text{ psi}$

$$MS = \frac{38970}{4936} - 1 = 6.89$$

Effective stress S_{e2} in the outer shell at point "A"

$$\sigma_x = -254 \text{ psi (Radial Stress) From Sect. 3.8}$$

$$\sigma_y = -676 \text{ psi (Tangential Stress) From Sect. 3.8}$$

$$\sigma_z = -52 \text{ psi (Axial Stress) From Sect. 3.8}$$

$$T_{yz} = \begin{matrix} 2766 \\ \text{(Sect. 3.8)} \end{matrix} \text{ psi (Shear Stress)}$$

$$T_{xy} = 0 \quad T_{zx} = 223 \text{ (Sect. 3.8)}$$

$$S_{e2} = \sqrt{\frac{1}{2} \sqrt{(-254 - (-676))^2 + (-676 - (-52))^2 + (-52 - (-254))^2 + 6(2766^2 + 223^2)}} \\ = 4838 \text{ psi}$$

Allowable stress ($S_{ad} = 0.9 S_{yd}$) at 248°F for 304 S.S. From Sect. 1.1 under normal conditions and Sect. 1.2 equals to $0.9 \times 43800 = 39420$ psi.

$$MS = \frac{39420}{4838} - 1 = 7.15$$

Effective stress S_{e2} in the water jacket shell at point "A"

$$\sigma_x = -117.5 \text{ psi (Radial Stress) From Sect. 3.8}$$

$$\sigma_y = 590 \text{ psi (Tangential Stress) From Sect. 3.8}$$

$$\sigma_z = 2535 \text{ psi (Axial Stress) From Sect. 3.8}$$

$$T_{yz} = 2766 \text{ psi (Shear Stress)}$$

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

$$S_{e2} = \sqrt{\frac{1}{2} \sqrt{(-117.5 - 590)^2 + (590 - 2535)^2 + (2535 - (-117.5))^2 + 6(2766^2)}} \\ = 5349 \text{ psi.}$$

Allowable stress ($S_{ad} = 0.9 S_{yd}$) at 218°F for 304 S.S. From Sect. 1.1 under normal conditions and Sect. 1.2 equals to $0.9 \times 44550 = 40094$ psi.

$$MS = \frac{40094}{5349} - 1 = 6.43$$

Calculate effective stress S_{e2} in the outer shell and water jacket shell at point "F".

Effective stress S_{e2} in the outer shell at point "F".

$$\sigma_x = -115 \text{ psi (Radial Stress) From Sect. 3.8}$$

$$\sigma_y = -1012 \text{ psi (Tangential Stress) From Sect. 3.8}$$

$$\sigma_z = -175 \text{ psi (Axial Stress) From Sect. 3.8}$$

$$T_{yz} = -3195 \text{ psi (Shear Stress) (Sect. 3.8)}$$

$$T_{xy} = 0 \quad T_{zx} = -259 \text{ psi (Sect. 3.8)}$$

$$S_{e2} = \sqrt{\frac{1}{2} \left[(-115 - (-1012))^2 + (-1012 - (-175))^2 + (-175 - (-115))^2 + 6(3195^2 + 259^2) \right]}$$

$$= 5619 \text{ psi.}$$

Allowable stress ($S_{ad} = 0.9 S_{yd}$) at 241°F for 304 S.S. From Sect. 1.1

under normal conditions and Sect. 1.2 equals to $0.9 \times 43224 = 38902 \text{ psi.}$

$$MS = \frac{38902}{5619} - 1 = 5.92$$

Effective stress S_{e2} in the water jacket shell at point "F"

$$\sigma_x = -117.5 \text{ psi (Radial Stress) From Sect. 3.8}$$

$$\sigma_y = 238 \text{ psi (Tangential Stress) From Sect. 3.8}$$

$$\sigma_z = 2504 \text{ psi (Axial Stress) From Sect. 3.8}$$

$$T_{zx} = 3195 \text{ psi (Shear Stress)}$$

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

$$S_{e2} = \sqrt{\frac{1}{2} \sqrt{(-117.5-238)^2 + (238-2504)^2 + (2504-(-117.5))^2 + 6(3195)^2}}$$

$$= 6057 \text{ psi.}$$

Allowable stress ($S_{ad} = 0.9 S_{yd}$) at 218° F for 304 S.S. From Sect. 1.1 under normal conditions and Sect. 1.2 equals to $0.9 \times 43975 = 39577$ psi.

$$MS = \frac{39577}{6057} - 1 = 5.53$$

Effective stress S_{e2} in the inner shell at point "G"

$$\sigma_x = -8 \text{ psi (Radial Stress) From Sect. 3.8}$$

$$\sigma_y = -46 \text{ psi (Tangential Stress) From Sect. 3.8}$$

$$\sigma_z = -237 \text{ psi (Axial Stress) From Sect. 3.8}$$

$$T_{yz} = 3468 \text{ psi (Shear Stress)}$$

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

$$S_{e2} = \sqrt{\frac{1}{2} \sqrt{(-117.5-238)^2 + (238-2504)^2 + (2504-(-117.5))^2 + 6(3195)^2}}$$

$$\frac{6(3468^2)}{}$$

$$= 6010 \text{ psi}$$

Allowable stress ($S_{ad} = 0.9 S_{yd}$) at 260° F for 304 S.S. From Sect. 1.1 under normal conditions and Sect. 1.2 equals to $0.9 \times 43500 = 39150$ psi.

$$MS = \frac{39150}{6010} - 1 = 5.51$$

Calculate the primary plus secondary stress range in the shells at point (A) loc. 2, 3, 5 at point "F" loc. 26, 27 and at point "G" loc. 30. The procedure for calculating the primary plus secondary stress range is explained in detail in Section 3.8.

Point (A) loc. 2 (Inner Shell) the cycle giving the worse stress range will be: Zero Stress and Normal Transport 70 kw, 130°F ambient, 16.5 psig cavity pressure and 235 psig water jacket pressure. See Sect. 3.8

| Point (A) Loc. 2 (Inner Shell) Condition | STRESS COMPONENTS (PSI) | | | | |
|---|-------------------------|------------|------------|----------|------------------|
| | σ_x | σ_z | σ_y | T_{yz} | T_{xz} |
| 1. Zero Stress | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130°F Ambient | [322 | -44706 | -29619] | 2766 | [-262] Sect. 3.8 |

Given the description of the Loading cycle, the next step is to select one reference point. For this calculation, the zero stress point is selected as the reference.

| Condition | PRELIMINARY STRESS DIFFERENCES (PSI) | | | | |
|--------------------------------------|--------------------------------------|--------------------------|--------------------------|--------------------|--------------------|
| | $\sigma_x - \sigma_{xi}$ | $\sigma_z - \sigma_{zi}$ | $\sigma_y - \sigma_{yi}$ | $T_{yz} - T_{yzi}$ | $T_{xz} - T_{xzi}$ |
| 1. Zero Stress | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130°F Ambient | 322 | -44706 | -29619 | 2766 | -262 |

The principal stresses among the preliminary stress differences are next determined.

| Condition | PRINCIPAL STRESSES (PSI) | | | |
|--------------------------------------|--------------------------|------------|------------|---------------------------------|
| | σ_1 | σ_2 | σ_3 | |
| 1. Zero Stress | 0.0 | 0.0 | 0.0 | From Ref. 3 Table II, Case 7 |
| 2. Normal Transport 130°F Ambient | 323.5 | -29128 | -45199 | |

For each condition, the stress differences between the principal stresses are next determined. The stress differences are tabulated below.

| Condition | STRESS DIFFERENCES (PSI) | | |
|--------------------------------------|--------------------------|-------------------------|-------------------------|
| | $(\sigma_1 - \sigma_2)$ | $(\sigma_2 - \sigma_3)$ | $(\sigma_3 - \sigma_1)$ |
| 1. Zero Stress | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130°F Ambient | 29452 | 16071 | -42523 |

The maximum range for the cycle is then the largest of the stress differences of -42523 psi.

$$\text{Allowable stress } 3S_m = 67342 \quad (\text{Sect. 3.8})$$

$$MS = \frac{67342}{42523} - 1 = .584$$

Point "G" Loc. 30 (Inner Shell). The cycle giving the worse stress range will be -40°F Isothermal and Normal Transport 70 kw, 130°F Ambient, 16.5 psig cavity pressure and 235 psig water jacket pressure. See Sect. 3.8

| Point (G) Loc. 30 (Inner Shell) Condition | STRESS COMPONENTS (PSI) | | | |
|--|-------------------------|------------|------------|----------|
| | σ_x | σ_z | σ_y | T_{yz} |
| 1. -40° F Isothermal | [0.0 | 11675 | 3400 | 0.0 |
| 2. Normal Transport 130° F Ambient | | -6140 | -1467 | 3468 |

Sect. 3.8

Given the description of the Loading Cycle, the next step is to select one reference point. For this calculation, the -40° F Isothermal is selected as the reference.

| Condition | PRELIMINARY STRESS DIFFERENCES (PSI) | | | |
|---------------------------------------|--------------------------------------|--------------------------|--------------------------|--------------------|
| | $\sigma_x - \sigma_{xi}$ | $\sigma_z - \sigma_{zi}$ | $\sigma_y - \sigma_{yi}$ | $T_{yz} - T_{yzi}$ |
| 1. -40° F Isothermal | 0.0 | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130° F Ambient | 0.0 | -17815 | -4867 | 3468 |

The principal stresses among the preliminary stress differences are next determined.

| Condition | PRINCIPAL STRESSES (PSI) | | |
|----------------------|--------------------------|------------|------------|
| | σ_1 | σ_2 | σ_3 |
| 1. -40° F Isothermal | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport | -3996 | 0.0 | -18685 |

Calculate the two principal stresses σ_1 , σ_3 for Normal Transport.

$$\text{From Ref. 3 } \sigma_1, \sigma_3 = \frac{-17815 + (-4867)}{2} + \left[\frac{(-17815 - (-4867))^2}{4} + 3468^2 \right]^{\frac{1}{2}}$$

$$\sigma_1 = -3996 \text{ psi}$$

$$\sigma_3 = -18685 \text{ psi}$$

For each condition, the stress differences between the principal stresses are next determined. The stress differences are tabulated below:

| Condition | STRESS DIFFERENCES (PSI) | | |
|---|--------------------------|-------------------------|-------------------------|
| | $(\sigma_1 - \sigma_2)$ | $(\sigma_2 - \sigma_3)$ | $(\sigma_3 - \sigma_1)$ |
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130 $^\circ\text{F}$ Ambient | -3996 | 18685 | -14689 |

The maximum range for the cycle is the largest of the stress differences of 18685 psi.

$$\text{Allowable stress } 3S_m = 68082 \text{ psi (Sect. 3.8)}$$

$$MS = \frac{68082}{18685} - 1 = 2.64$$

Point "A" Loc. 3 (Outer Shell). The cycle giving the worse stress range will be -40°F Isothermal and Normal Transport 70 kw, 130°F Ambient, 16.5 psig cavity pressure and 235 psig water jacket pressure. See Sect. 3.8

| Point "A" Loc. 3 (Outer Shell) | | STRESS COMPONENTS (PSI) | | | | | |
|---|-----------------|-------------------------|------------|----------|----------------|--------------|--|
| Condition | σ_x | σ_z | σ_y | T_{yz} | T_{xz} | | |
| 1. -40°F Isothermal | [2505 127] | -19246 | -9884 | 0.0 | [769 234] | Sect. 3.8 | |
| 2. Normal Transport 130 $^\circ\text{F}$ Ambient | | 3881 | -5672 | 2766 | | | |

Given the description of the Loading Cycle, the next step is to select one Reference Point. For this calculation, the -40°F Isothermal is selected as the Reference.

| PRELIMINARY STRESS DIFFERENCES (PSI) | | | | | |
|---|--------------------------|--------------------------|--------------------------|--------------------|--------------------|
| Condition | $\sigma_x - \sigma_{xi}$ | $\sigma_z - \sigma_{zi}$ | $\sigma_y - \sigma_{yi}$ | $T_{yz} - T_{yzi}$ | $T_{xz} - T_{xzi}$ |
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130 $^\circ\text{F}$ Ambient | -2378 | 23127 | 4212 | 2766 | -535 |

The principal stresses among the preliminary stress differences are next determined.

| PRINCIPAL STRESSES (PSI) | | | | |
|---|------------|------------|------------|--------------------------------------|
| Condition | σ_1 | σ_2 | σ_3 | |
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 | |
| 2. Normal Transport 130 $^\circ\text{F}$ Ambient | 3817 | 23534 | -2390 | From Ref. 3 Table II Case 7 |

For each condition, the stress differences between the principal stresses are next determined. The stress differences are tabulated below:

| Condition | STRESS DIFFERENCES (PSI) | | |
|--|--------------------------|-------------------------|-------------------------|
| | $(\sigma_1 - \sigma_2)$ | $(\sigma_2 - \sigma_3)$ | $(\sigma_3 - \sigma_1)$ |
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130°F Ambient | -19717 | 25924 | -6207 |

The maximum range for the assumed cycle is then the largest of the stress differences of 25924 psi.

Allowable stress $3S_m = 67807$ (Sect. 3.8)

$$MS = \frac{67807}{25924} - 1 = 1.6$$

Point "A" Loc. 5 (Water Jacket Shell). The cycle giving the worse stress range will be -40°F Isothermal and Normal Transport 70 kw, 130°F Ambient, 16.5 psig cavity pressure and 235 psig water jacket pressure. See Sect. 3.8

| Condition | STRESS COMPONENTS (PSI) | | | |
|--|-------------------------|------------|------------|----------|
| | σ_x | σ_z | σ_y | T_{yz} |
| 1. -40°F Isothermal | 0.0 | -1150 | -443 | 0.0 |
| 2. Normal Transport 130°F Ambient | [-235 | 36467 | 10384] | 2766 |

Sect. 3.8

Given the description of the loading cycle, the next step is to select one Reference Point. For this calculation, the -40°F Isothermal is selected as the Reference.

| Condition | PRELIMINARY STRESS DIFFERENCES (PSI) | | | |
|--|--------------------------------------|--------------------------|--------------------------|--------------------|
| | $\sigma_x - \sigma_{xi}$ | $\sigma_z - \sigma_{zi}$ | $\sigma_y - \sigma_{yi}$ | $T_{yz} - T_{yzi}$ |
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130°F Ambient | -235 | 37617 | 10827 | 2766 |

The principal stresses among the preliminary stress differences are next determined.

| Condition | PRINCIPAL STRESSES (PSI) | | | From Ref. Table II Case 7 |
|--|--------------------------|------------|------------|---------------------------------|
| | σ_1 | σ_2 | σ_3 | |
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 | |
| 2. Normal Transport 130°F Ambient | -235 | 37900 | 10544 | |

For each condition, the stress differences between the principal stresses are next determined. The stress differences are tabulated below:

| Condition | STRESS DIFFERENCES (PSI) | | |
|--|--------------------------|-------------------------|-------------------------|
| | $(\sigma_1 - \sigma_2)$ | $(\sigma_2 - \sigma_3)$ | $(\sigma_3 - \sigma_1)$ |
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130°F Ambient | -38135 | 27356 | 10779 |

The maximum range for the cycle is then the largest of the stress differences of -38135 psi.

Allowable stress $3S_m = 69340$ psi (Sect. 3.8)

$$MS = \frac{69340}{38135} - 1 = .818$$

Point "F" Loc. 26 (Outer Shell). The cycle giving the worse stress range will be -40°F Isothermal and Normal Transport 70 kw, -40°F Ambient, 16.5 psig cavity pressure and 235 psig water jacket pressure. See Sect. 3.8

| Condition | STRESS COMPONENTS (PSI) | | | |
|--|-------------------------|------------|------------|----------|
| | σ_x | σ_z | σ_y | T_{yz} |
| 1. -40°F Isothermal | [0.0 -33 | 23117 | 3398 | 0.0 |
| 2. Normal Transport -40°F Ambient | | -20066 | -6764 | 3195 |

Sect. 3.8

Given the description of the loading cycle, the next step is to select one Reference Point. For this calculation, the -40°F Isothermal is selected as the Reference.

| Condition | PRELIMINARY STRESS DIFFERENCES (PSI) | | | |
|--|--------------------------------------|--------------------------|--------------------------|--------------------|
| | $\sigma_x - \sigma_{xi}$ | $\sigma_z - \sigma_{zi}$ | $\sigma_y - \sigma_{yi}$ | $T_{yz} - T_{yzi}$ |
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport -40°F Ambient | -33 | -43183 | -10162 | 3195 |

The principal stresses among the preliminary stress differences are next determined.

| Condition | PRINCIPAL STRESSES (PSI) | | |
|--------------------------------------|--------------------------|------------|------------|
| | σ_1 | σ_2 | σ_3 |
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport -40°F Ambient | -9856 | 0.0 | -43489 |

For simplification of the analysis σ_x for normal transport is 0.0, since the -33 psi is a very small stress.

From Ref. 3, Table II, Case 5

$$\sigma_1, \sigma_3 = \frac{-43183 + (-10162)}{2} + \left[\frac{(-43183 - (-10162))^2}{2} + 3195^2 \right]^{\frac{1}{2}}$$

$$\sigma_1 = -9856 \text{ psi}$$

$$\sigma_3 = -43489 \text{ psi}$$

For each condition, the stress differences between the principal stresses are next determined. The stress differences are tabulated below:

| Condition | STRESS DIFFERENCES (PSI) | | |
|--------------------------------------|--------------------------|-------------------------|-------------------------|
| | $(\sigma_1 - \sigma_2)$ | $(\sigma_2 - \sigma_3)$ | $(\sigma_3 - \sigma_1)$ |
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport -40°F Ambient | -9856 | 43489 | -33633 |

The maximum range for the cycle is the largest of the stress differences of 43489 psi .

Allowable stress $3S_m = 68199$ psi (Sect. 3.8)

$$MS = \frac{68199}{43489} - 1 = .568$$

Point "F" Loc. 27 (Water Jacket Shell). The cycle giving the worse stress range will be -40°F Isothermal and Normal Transport 70 kw, 130°F Ambient, 16.5 psig cavity pressure and 235 psig water jacket pressure. See Sect. 3.8

Point "F" Loc. 27 (Water Jacket Shell) STRESS COMPONENTS (PSI)

| Condition | σ_x | σ_z | σ_y | T_{yz} |
|---|------------|------------|------------|----------|
| 1. -40°F Isothermal | [0.0 | -2953 | 48 | 0.0 |
| 2. Normal Transport 130 $^{\circ}\text{F}$ Ambient | | | | 3195 |

Sect. 3.8

Given the description of the loading cycle, the next is to select one Reference Point.

For this calculation the -40°F Isothermal is selected as the reference.

PRELIMINARY STRESS DIFFERENCES (PSI)

| Condition | $\sigma_x - \sigma_{x1}$ | $\sigma_z - \sigma_{z1}$ | $\sigma_y - \sigma_{y1}$ | $T_{yz} - T_{yz1}$ |
|---|--------------------------|--------------------------|--------------------------|--------------------|
| 1. -40°F Isothermal | 0.0 | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130 $^{\circ}\text{F}$ Ambient | -235 | 45743 | 10106 | 3195 |

The principal stresses among the preliminary stress differences are next determined.

| Condition | PRINCIPAL STRESSES (PSI) | | |
|---------------------------------------|--------------------------|------------|------------|
| | σ_1 | σ_2 | σ_3 |
| 1. -40° F Isothermal | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130° F Ambient | -235 | 46027 | 9821 |

For each condition, the stress differences between the principal stresses are next determined. The stress differences are tabulated below:

| Condition | STRESS DIFFERENCES (PSI) | | |
|---------------------------------------|--------------------------|-------------------------|-------------------------|
| | $(\sigma_1 - \sigma_2)$ | $(\sigma_2 - \sigma_3)$ | $(\sigma_3 - \sigma_1)$ |
| 1. -40° F Isothermal | 0.0 | 0.0 | 0.0 |
| 2. Normal Transport 130° F Ambient | -46262 | 36206 | 10056 |

The maximum range for the cycle is then the largest of the stress differences of -46262 psi.

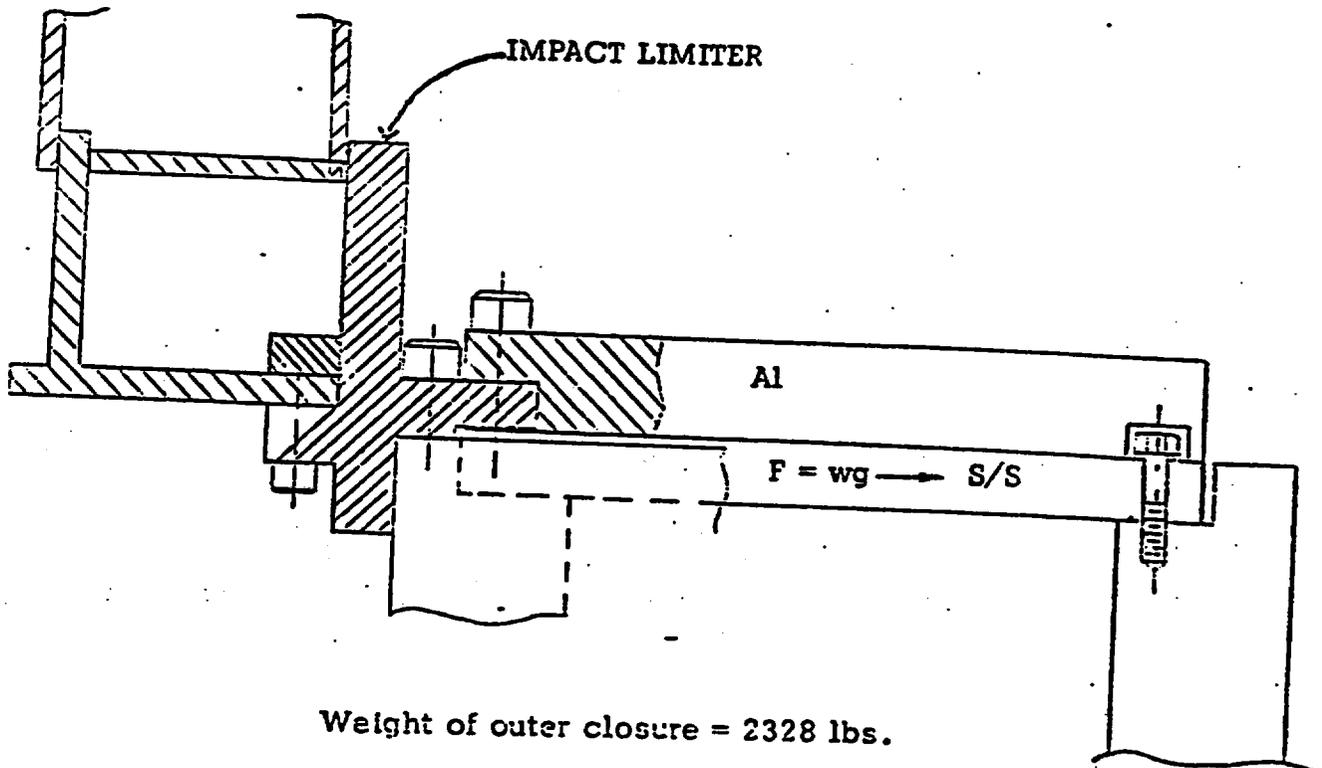
Allowable $3S_m = 69340$ psi (Sect. 3.8)

$$MS = \frac{69340}{46262} - 1 = .498$$

2.8.1 One Foot Drop Outer Closure Head

During the side impact the outer closure head is subjected to an acceleration of 14.55 g in the plane of the closure.

Integrity of outer closure must be maintained.



Weight of outer closure = 2328 lbs.

Lateral force on closure $F = 14.55 \times 2328 = 33872$ lbs.

Analysis of bolts in shear:

Outer closure is bolted down with 28 $1\frac{1}{2}$ in. dia. bolts.

Yield strength of bolts at 368°F is 85,000 psi (Sect. 1.2).

Ultimate tensile strength at 368°F is 130,000 psi (Sect. 1.2)

Shear area of $1\frac{1}{2}$ -8 bolts at the shear plane is 0.9408 in.²

$$\text{Shear Stress } S_s = \frac{F}{28A} = \frac{33872}{28 \times 0.9408} = 1286 \text{ psi}$$

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

Effective Stress

$$S_{e2} = \sqrt{\frac{1}{2} \left[(0 - 0)^2 + (0 - 12,158.)^2 + (12,158 - 0)^2 + 6(1286)^2 \right]}$$

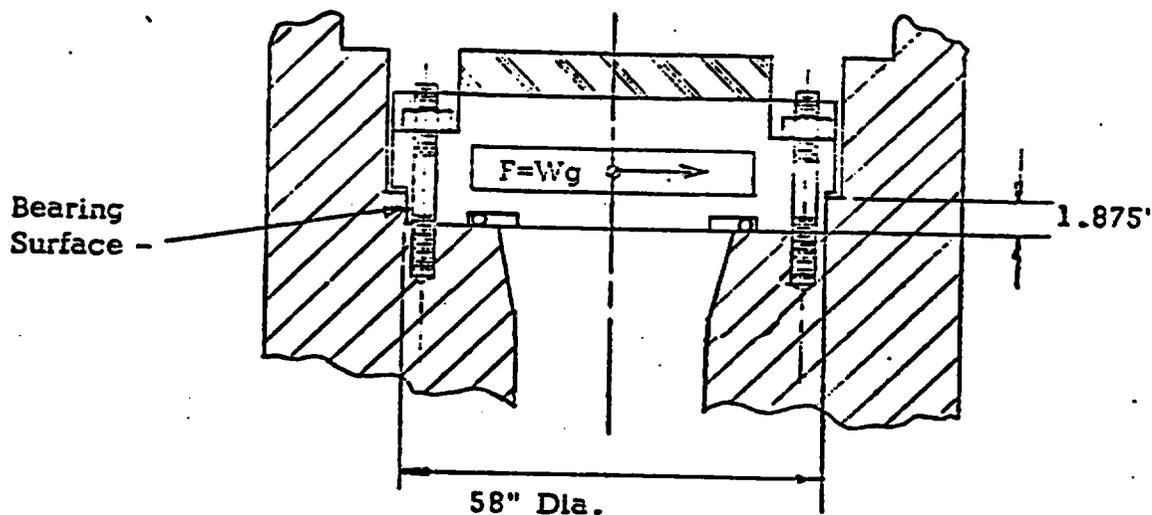
$$S_{e2} = 12360 \text{ psi.}$$

Allowable stress ($S_a = .75 S_y$) for the outer closure bolts under normal condition from Sect. 1.1 and Sect. 1.2 equals $.75 \times 85000 = 63750 \text{ psi.}$

$$\text{M.S.} = (63750/12360) - 1 = 4.16.$$

One Foot Drop Inner Closure

During the side impact the inner closure is subjected to an acceleration of 14.55g in the plane of the closure. Integrity of inner closure must be maintained to provide the desired containment of the cask contents.



Design Weight of Inner Closure = 7400 lbs.

Lateral force on closure, $F = (14.55) (7400)$

$F = 107670$ lbs

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.

XI-2-42r

Calculating bearing stress on the 304 S.S.
forging

$$S_b = \frac{P}{A} = \frac{107670}{108.75} = 990 \text{ psi.} \quad A = 1.875 \times 58 = 108.75 \text{ in}^2$$

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

$$M.S. = \frac{55620}{990} - 1 = 55$$

The second category of conditions experienced in normal transport is vandalism. The area of concern here is damage caused by objects which are thrown, dropped or fired at the package.

Thrown objects would be those things which are easily grasped in one hand and propelled for short distances. These would be objects weighing approxi-

mately five pounds. The package mounted on the rail car is completely enveloped by a personnel barrier which is constructed of expanded metal and braced by appropriate structural shapes. Penetration of the envelope from objects thrown at the side or ends of the personnel barrier is virtually impossible.

Dropped objects are those objects which could be handled by two people. The object could be lifted and dropped from an overpass striking the top of the cask. The maximum weight of the object is assumed to be 150 pounds and for the convenience of analysis it is assumed to be in the form of a six inch diameter solid steel bar having slightly rounded edges. The drop height is considered to be 15 feet. The areas which could be struck by an object dropped from above are the neutron shield jacket and the expansion tanks which are located along side of the cask. All other critical components are protected by being located essentially in the shadow of the cask body and the expansion tanks. The primary concern in evaluating the system integrity is that there is no breach or penetration of a component. Local deformation such as bent cooling fins are to be expected, however, there will be no significant reduction in the effectiveness of the package. The following analysis evaluates the various component resistance to puncture. The energy produced by a 15 foot drop of a 150 lb. cylindrical bar is

$$E = WH$$

$$E = 150 \times 180 = 27,000 \text{ in-lbs.}$$

(penetration energy)

Neutron Shield Jacket -

The energy required to punch out a 6" Dia. plug under pure shear (Conservative

approach for hemispherical end) from Ref. 16, page 13-25

$$U = Kft = Kr A t = Kr \pi Dt^2$$

$r = .75 (61000)$ at $350^{\circ}\text{F} = 45750$ psi tensile shear stress

$$A = \pi Dt$$

$$D = 6 \text{ in}$$

$t = .75$ in. thickness of jacket shell

$K = 39$ Penetration to Fracture percent.

$$U = .39 (\pi) (45750) (6) (.75^2)$$

$U = 189182$ in-lb. (Puncture Resistance)

Expansion Tanks

$$U = Kft = Kr A t = Kr \pi Dt^2$$

$r = .75 (61000)$ at $350^{\circ}\text{F} = 45750$ psi tensile shear stress

$$A = \pi Dt$$

$$D = 6$$

$t = .500$ in. thickness of expansion tank shell

$K = 39$ Penetration to Fracture percent

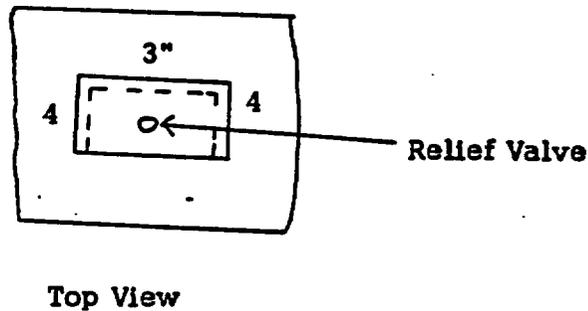
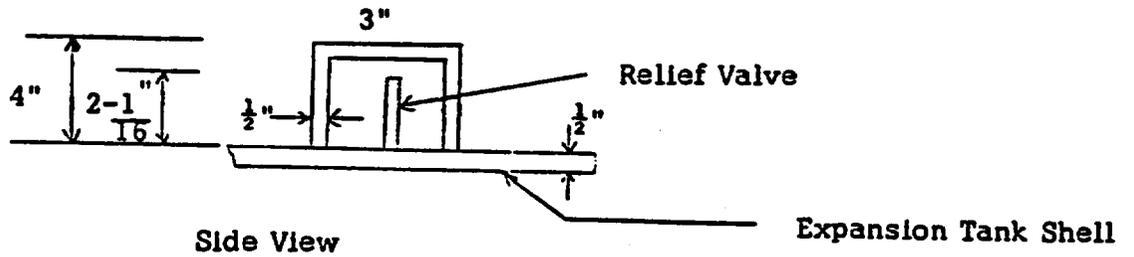
$$U = .39 (\pi) (45750) (6) (.5^2)$$

$= 84081$ in-lbs (Puncture Resistance)

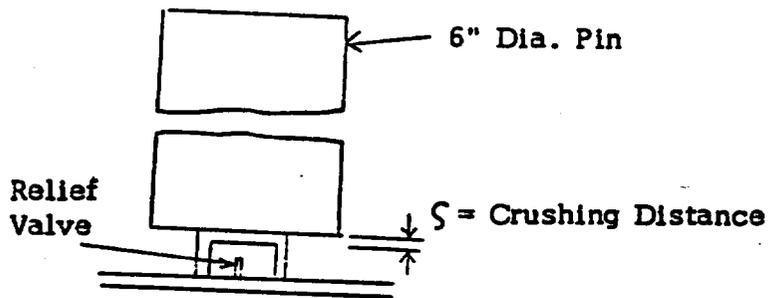
Since the energy required to punch out a 6 inch plug is much higher than the energy produced by the 15 ft. drop of the 150 lb. cylindrical steel bar, the

bar will not penetrate the .75 inch thick jacket shell or the .500 inch thick expansion tank shell.

Relief Valve Box on Expansion Tank.



Condition No. 1



$$KE = \sigma_y A S$$

$$\sigma_y = 40667 \text{ psi}$$

Dynamic Yield Strength at
350°F (Sect. 1.2)

$$A = 11 (.5)$$

$$= 6.5 \text{ in}^2$$

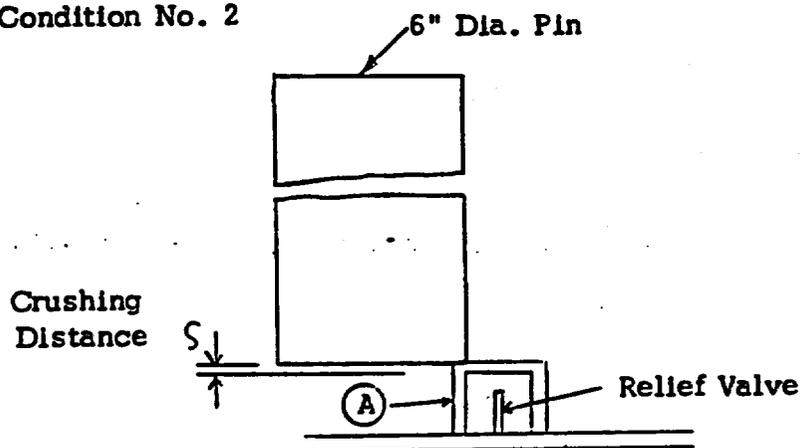
$$27000 = 40667 (6.5)$$

$$= \frac{27000}{264336}$$

$$= .102 \text{ in.}$$

The crushing distance of .102 inches will not cause the top plate to come into contact with the relief valve.

Condition No. 2



$$KE = \sigma_y AS$$

$$\sigma_y = 40667 \text{ psi}$$

$$27000 = 40667 (2)$$

$$A = 4 (.5)$$

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

$$S = \frac{27000}{81334} = .332"$$

The crushing distance of .332 inches will not cause the top plate to come into contact with the relief valve.

Check for shearing of .50 inch expansion tank shell by plate "A".

Puncture Resistance $U = K r l t^2$

$K = .39$

$r = .75 (61000) \text{ at } 350^{\circ}\text{F}$

$= 45750 \text{ Psi tensile}$

shear stress

$l = \text{total shear}$

length

$U = .39 (45750) (9) (.5^2)$

$= 40146 \text{ in-lbs}$

$= (4) (2) + (.5) (2)$

$= 9"$

$t = \text{shell thickness}$

$.5"$

Plate "A" will not penetrate

thru the shell since the puncture resistance

energy is much higher than the energy produced by the 15 ft. drop of the 150 lb. cylindrical steel bar.

Expansion tank drain valve, gage, piping and water jacket relief valve are not vulnerable to the drop object since they are protected by the cask body and expansion tanks.

The final assessment of package integrity considers the effects on the package of a fired object. A fired object is considered to be 30 caliber softnose bullet as used for deer hunting. Field tests were performed to determine penetrating power of such a bullet. Stainless steel test plates .75 inches, .50 inches and .25 inches thick were used as the target plates. One round was fired at each plate from a distance of 100 yards.

The .25 inch and .50 inch thick plates were penetrated. The neutron shield expansion tanks are made of .50 inch thick plate. Penetration of these tanks under normal conditions of transport is not desirable. A second field test was set up to determine an effective means of protecting the neutron shield expansion tanks. Two plates were attached to a common base plate with their surfaces parallel to each other and spaced approximately 1 inch apart (See Figure 1). Again, one round was fired at the test piece from a distance of 100 yards. The bullet passed through the .25 inch plate spending most of its penetrating power and flattened against the .50 inch thick plate making a crater approximately .10 inches deep. The tests satisfactorily demonstrated that placement of the .25 inch plate in front of the .50 inch plate provides adequate protection for the .50 inch plate. The local reduction of approximately .10 inches in the .50 inch plate thickness still provides a metal thickness three times greater than the required (0.112 pg. x 1-3-87) by the expansion tank design pressure. These findings were incorporated into the package design by providing .25 inch thick steel panels in the lower portion of the personnel barrier. The location of these panels relative to the neutron shield expansion tank system can be seen on drawings 70654F and 70665F.

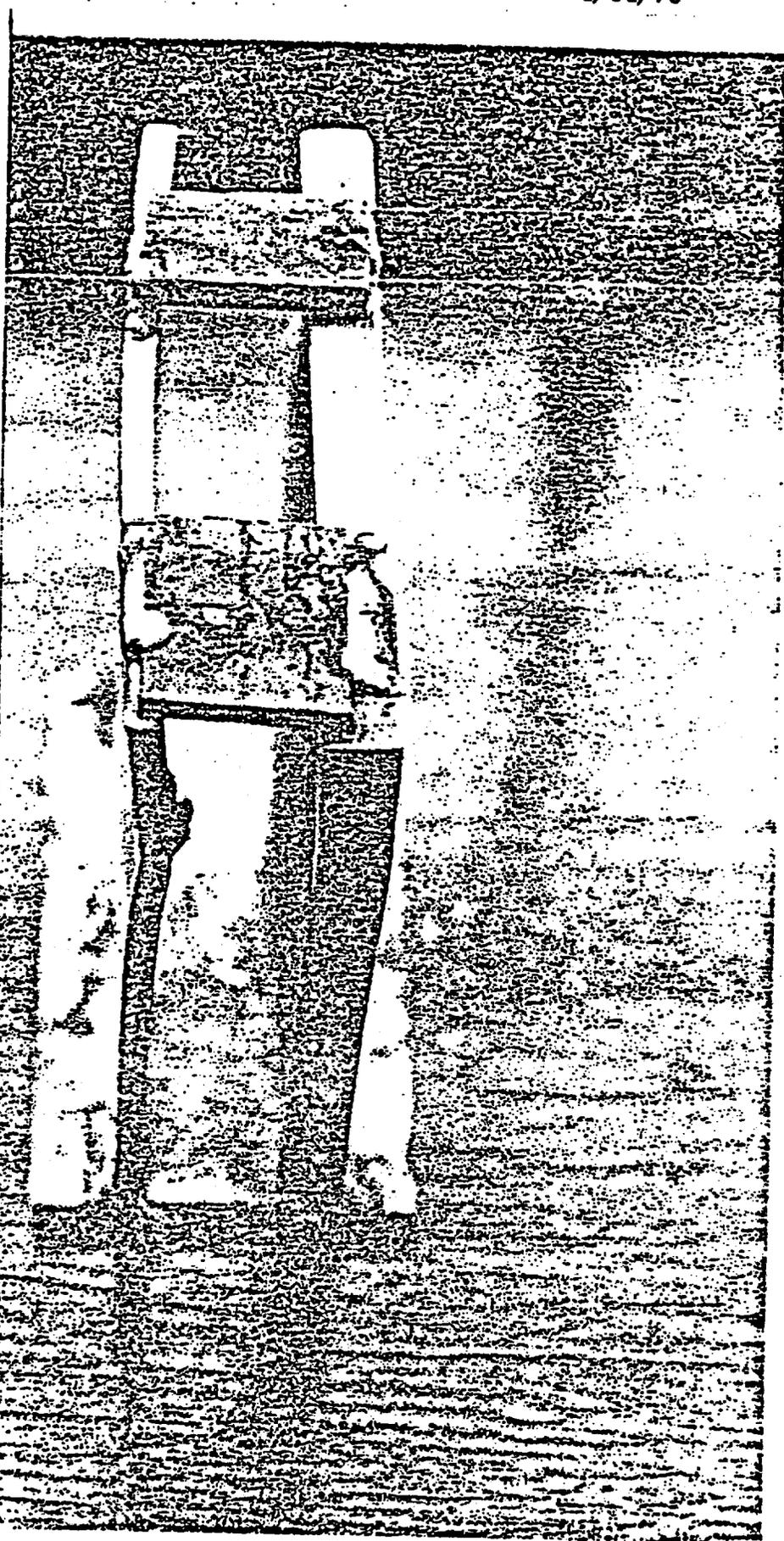
The test of the .75 inch thick plate was satisfactory. The plate suffered only partial penetration resulting in a metal thickness of .50 inches. The water jacket shell is .75 inches thick. The reduction in wall thickness in a local area is not sufficient to cause a break

of the water jacket shell. The effectiveness of the package will not be reduced as a result of being subjected to the final object.

Test Set-up.

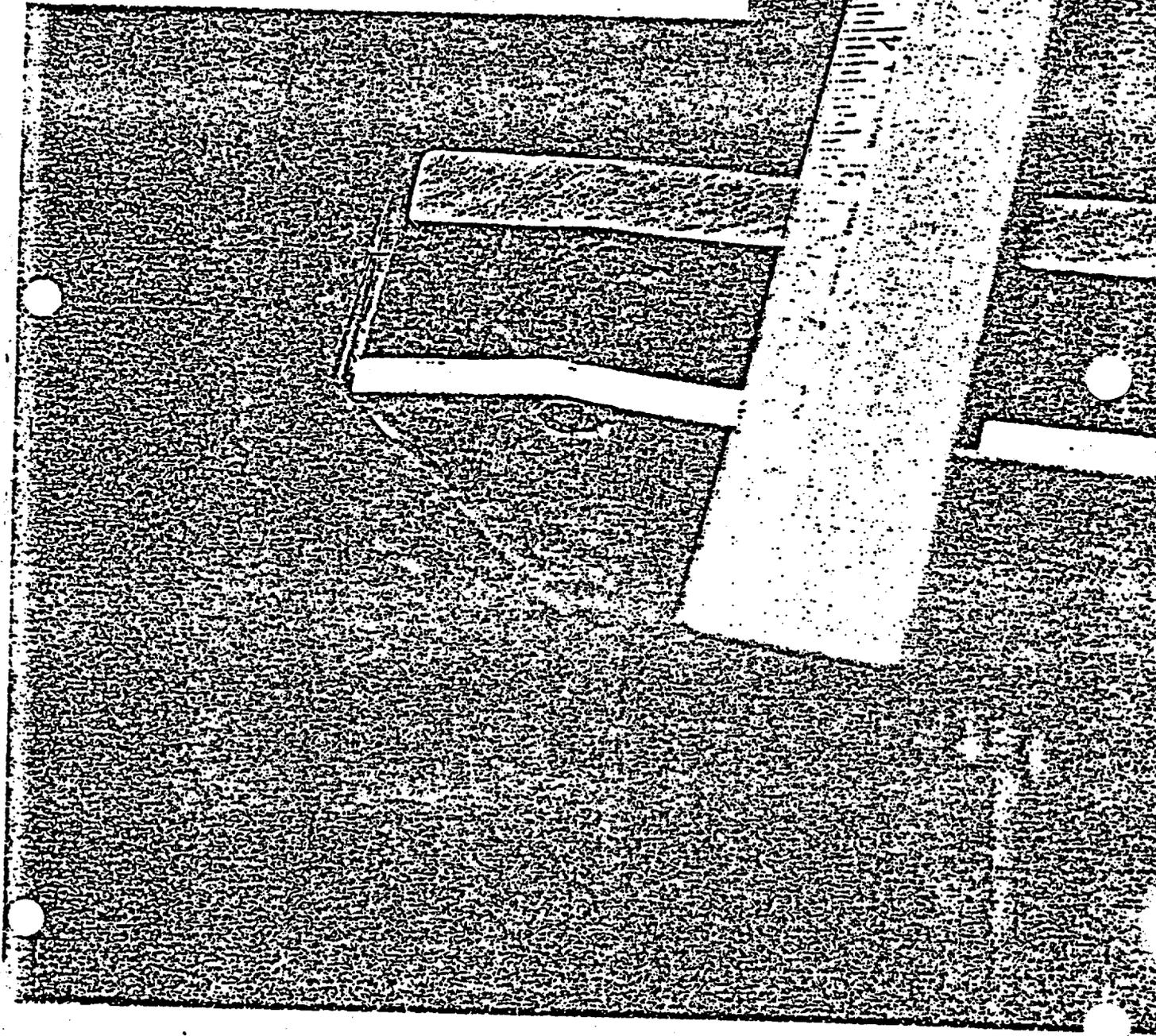
**.25 inch steel plate
placed approximately
1 inch in front of a
.50 inch stainless
steel plate**

Fig. 1



The crater seen in the .50 inch
plate is approximately .10 inches
deep.

Fig. 2 -



1/31/75

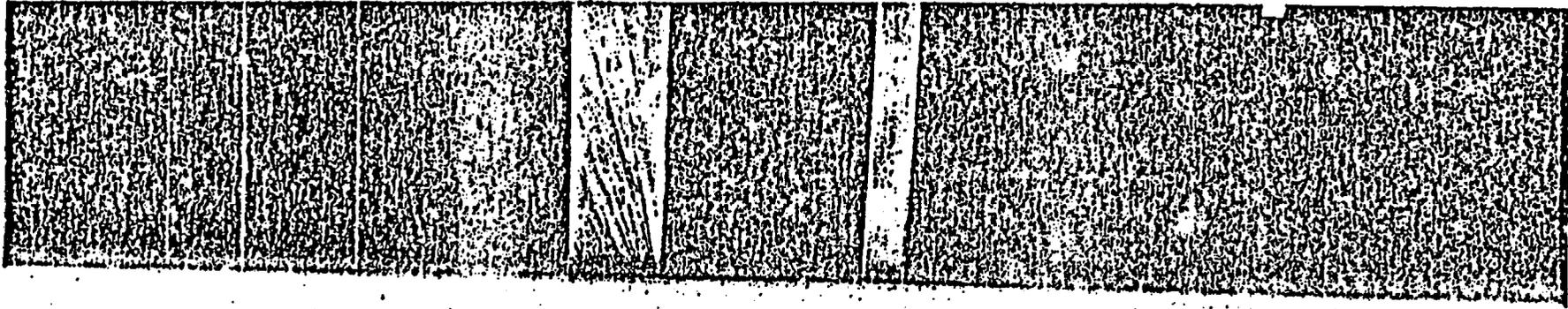
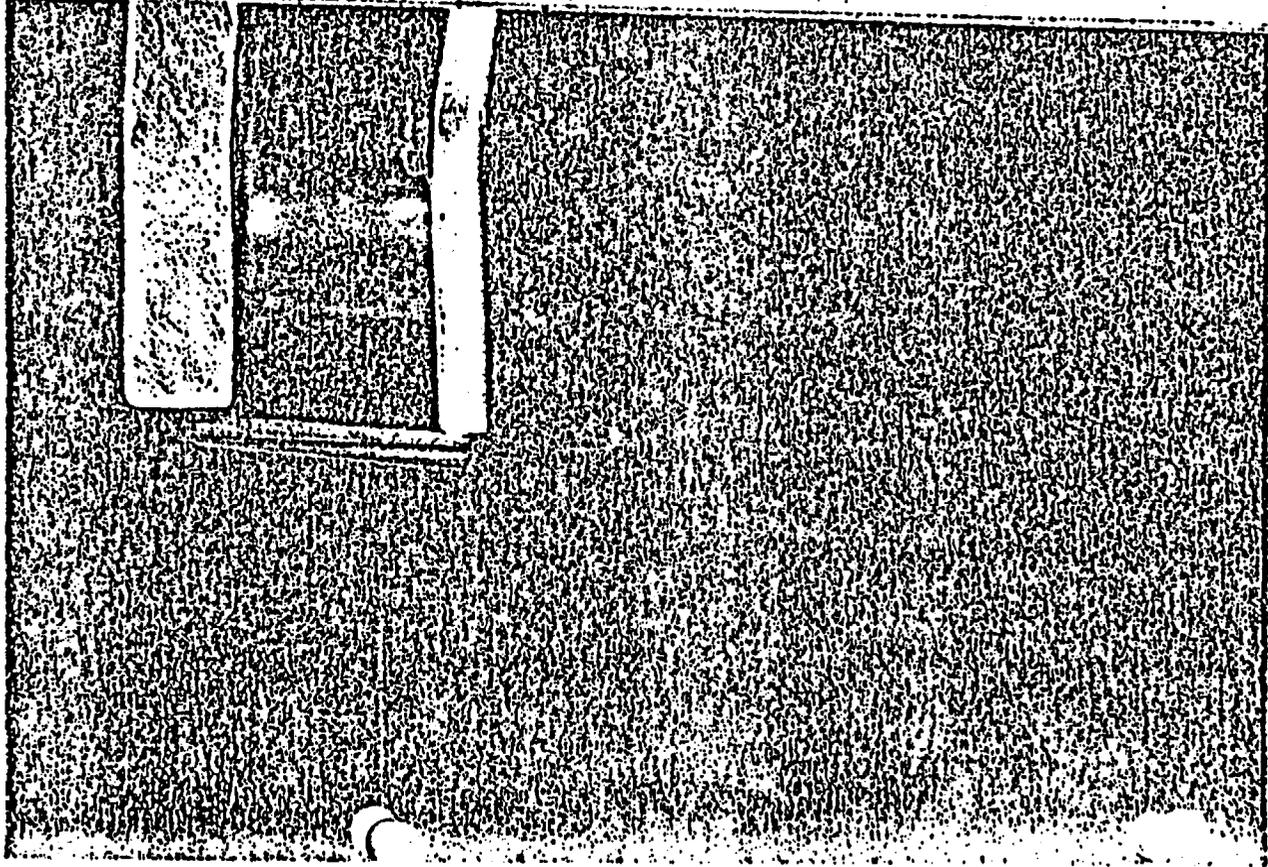


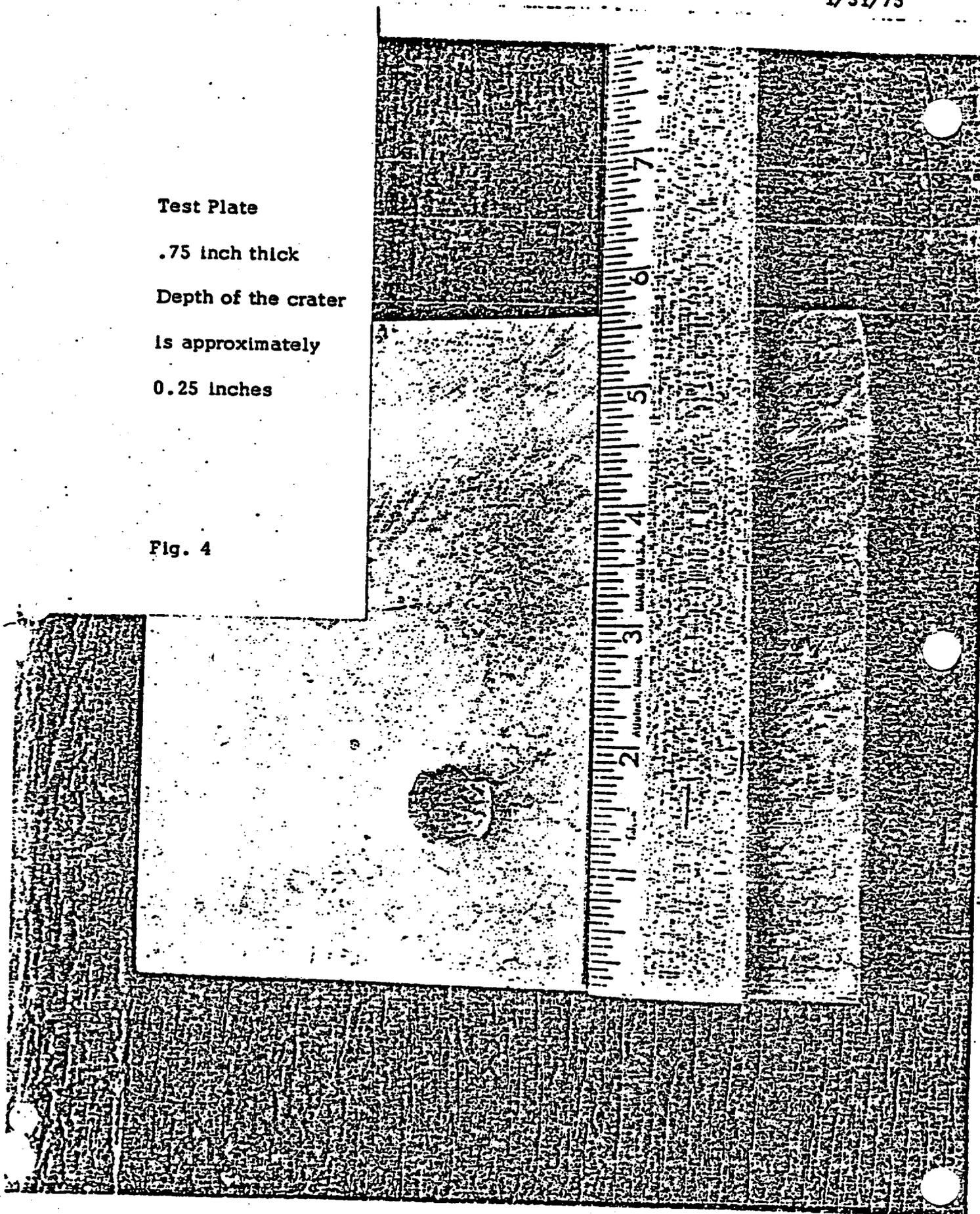
Fig. 3



XI-2-49d

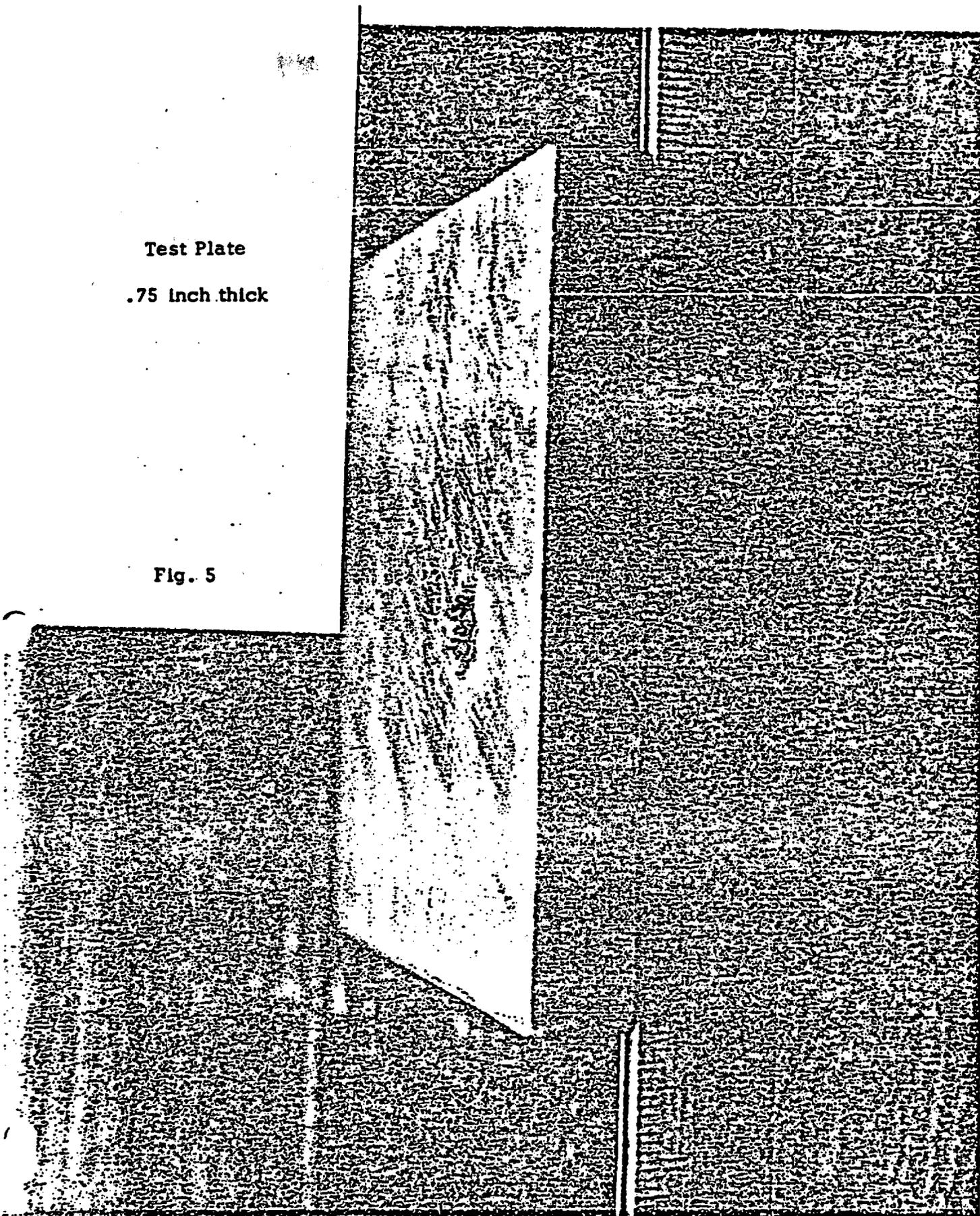
Test Plate
.75 inch thick
Depth of the crater
is approximately
0.25 inches

Fig. 4



Test Plate
.75 inch thick

Fig. 5



2.9 Penetration 10 CFR 71 , Appendix A

Impact of the hemispherical end of a vertical steel cylindrical bar $1\frac{1}{4}$ inches in dia. and weighing 13 lbs. dropped from a height of 40 inches onto exposed surface of the cask, which is expected to be most vulnerable to puncture. The following items are considered most vulnerable :

| | |
|---|--------------|
| Water Jacket Shell | t = .75 in. |
| Expansion Tanks | t = .50 in. |
| Expansion tank relief valve cover | t = .50 in. |
| Cover plate for closure head cavity drain | t = .375 in. |

The energy produced by a 40 inch drop of a 13 lb. steel cylinder bar is:

$$E = WH$$

$$W = 13 \text{ lb.}$$

$$H = 40 \text{ in.}$$

$$E = 13 (40) = 520 \text{ in.-lb.}$$

The energy required to punch out a $1\frac{1}{4}$ in. dia. plug under pure shear (conservative approach for hemispherical end, from Ref 16).

$$U = k F t = k \tau A t = k \tau \pi D t^2$$

Calculating for the thinnest of the items considered vulnerable ($t = .375$):

$$\tau = .6(59,500) = 35,700 \text{ psi tensile shear stress}$$

$$A = \pi D t$$

$$D = 1.25 \text{ in.}$$

$$t = .375 \text{ in. plate thickness}$$

$$k = .39 \text{ penetration to fracture percent}$$

$$U = .39\pi (35,700) (1.25) (.375)^2$$

$$U = 7,687 \text{ in. lb.}$$

Since the energy required to punch out a $1\frac{1}{4}$ inch plug is much higher than the energy produced by the 40 inch drop of the 13 lb. cylindrical steel bar, the bar will not penetrate the .375 inch thick plate. Therefore all items having a plate thickness greater than .375 can be considered structurally adequate.

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3.0 Normal Conditions of Transport

3.1 Design Conditions

The cask is designed to operate over the following range of conditions.

(1) Temperature Conditions

A- 70KW decay heat load, 130°F ambient.

B- 70KW decay heat load, -40°F ambient.

C- 40KW decay heat load, -40°F ambient.

D- 70°F isothermal

E- -40°F isothermal

F- Cool down and unloading. Operations associated with unloading the fuel assemblies. Temperatures are those associated with the unloading operations and under appropriate ambient temperature conditions.

(2) Loading Conditions

A- One-foot free fall.

B- Shock and vibration as experienced in normal transportation.

C- Pressure in cask cavity.

D- Pressure in water jacket.

Pressures for the various cask components for both PWR and BWR fuel loads are tabulated below. Pressure in the containment vessel may result from either the initial charge of helium or from a combination of helium and fission gas released from the fuel.

PRESSURE SUMMARY IN THE CASK

70KW Decay Heat Load, 130°F Ambient

| Containment Vessel Pressure | | |
|-----------------------------|------------|------------|
| | PWR Fuel | BWR Fuel |
| Helium | 16.45 psig | 16.45 psig |
| Helium plus Fission Gas | 63.15 psig | 72.9 psig |
| Water Jacket Pressure | | |
| Static | 235 psig | 235 psig |

70KW Decay Heat Load, -40°F Ambient

| Containment Vessel Pressure | | |
|-----------------------------|------------|------------|
| | PWR Fuel | BWR Fuel |
| Helium | 16.45 psig | 16.45 psig |
| Helium plus Fission Gas | 63.15 psig | 72.9 psig |
| Water Jacket Pressure | | |
| Static | 33 psig | 33 psig |

40KW Decay Heat Load, -40°F Ambient

| Containment Vessel Pressure | | |
|-----------------------------|------------|------------|
| | PWR Fuel | BWR Fuel |
| Helium | 16.45 psig | 16.45 psig |
| Helium plus Fission Gas | 63.15 psig | 72.9 psig |
| Water Jacket Pressure | | |
| Static | 14 psig | 14 psig |

70°F Isothermal

Cavity pressure and water jacket pressure equal 0-psig.

-400F Isothermal

Cavity pressure and water jacket pressure equal 0-psig.

Cool Down

| | Containment Vessel Pressure | |
|--------|-----------------------------|----------|
| | PWR Fuel | BWR Fuel |
| Static | 233 psig | 233 psig |
| | Water Jacket Pressure | |
| Static | 0-psig | 0-psig |

TEMPERATURE SUMMARY AT CASK MID-PLANE

70KW decay heat load, 130°F Ambient

| <u>Cask Location</u> | <u>Temperature (°F) - BWR & PWR Fuel</u> |
|-------------------------|--|
| Water Jacket | 323 |
| Neutron Shield (Liquid) | 341 |
| Outer Shell | 359 |
| Lead Shield | 389 |
| Inner Shell | 420 |
| Aluminum Basket | 466 |
| Fuel Elements | 659 |
| Absorber Sleeves | 499 |
| Basket Supports | 438 |
| Helium | 659 |

(Above temperatures are averages throughout each component)

| <u>Cask Location</u> | <u>Temperature (°F)</u> | |
|----------------------|-------------------------|----------|
| | PWR Fuel | BWR Fuel |
| Fuel Element | 833 (A) | 655 (A) |
| Aluminum Basket | 542 (B) | 542 (B) |

(A) average hot element temperatures; (B) maximum hot point temperatures.

These values are used to calculate thermal expansion of cask contents.

TEMPERATURE SUMMARY AT CASK ENDS

70KW decay heat load, 130°F Ambient

| <u>Cask location</u> | <u>Temperature (°F) - BWR & PWR Fuel</u> |
|-----------------------------|--|
| Aluminum cover plate | 250 |
| Outer Closure, seal & Bolts | 250 |
| Ricorad on closure | 300 |
| Inner closure | 350 |
| Inner closure seal & bolts | 325 |
| Bottom head | 325 |
| Ricorad on bottom head | 300 |

Loss of auxiliary cooling results in higher temperatures and larger temperature differences throughout the cask than when cooling is present. Consequently, pressures in the containment vessel and neutron shield and stresses due to differential thermal expansion increase; material properties are also affected. Hence, design conditions have been based on no auxiliary cooling as the normal mode of transport.

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The stress analysis of the cask shells and water jacket depends among other factors on the temperature difference between the shells. The steady state temperature distribution across the shell was used for all cases. The steady state condition has the highest temperature levels and also the greatest temperature difference between shells.

The cask dryout transient for the cask shells and water jacket is shown in Pg. 3-7 . This figure shows the response of the cask after fuel is loaded. The cask is vertical and there is no mechanical cooling. The cask cavity is also dry. Pg. 3-7 shows that the ΔT across the shells is increasing throughout the transient. Although steady state is not reached in this analysis, it is very apparent that the ΔT across the shells will continue to increase and the steady state ΔT will be larger than any point in the transient.

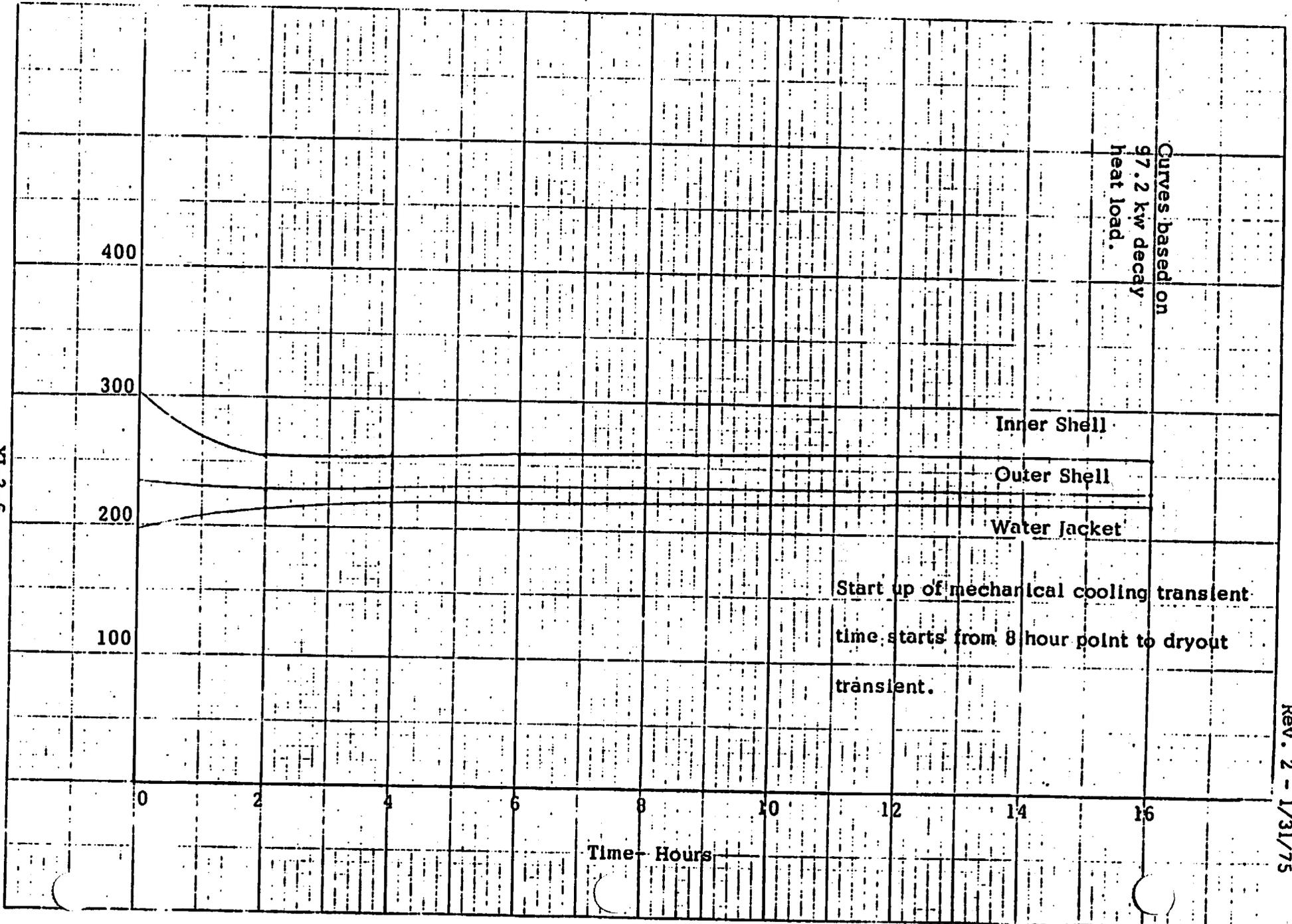
Fig. on Pg. 3-6 shows the response of the cask shells to mechanical cooling. The mechanical cooling starts 8 hours after the beginning of the dryout transient.

The ΔT across the shells during mechanical cooling will be less since most of the heat is being removed by the cooling water.

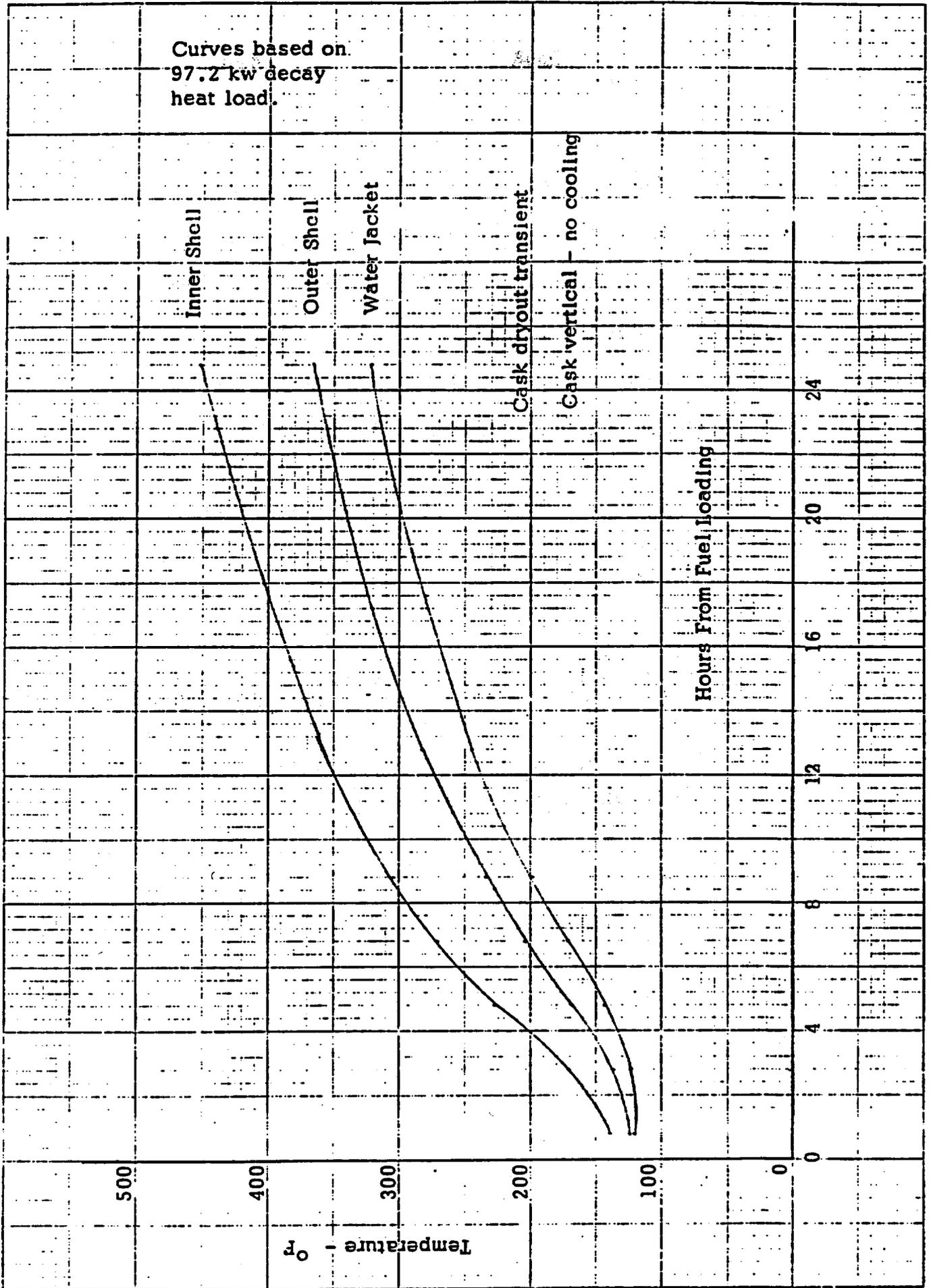
Besides the fire accident transient the other transient of interest is the loss of coolant transient. Since the metal diffusivity has not changed, the shells will respond in the same manner as the dryout transient and the steady state ΔT will still be the highest.

The fire accident transient is shown in Fig. VIII-6 and VIII-7. Again these figures show that the highest ΔT occurs at steady state condition except for the water jacket at the end of the fire.

XI-3-6



Curves based on
97.2 kw decay
heat load.



During transport the railcar and cask will experience the various shocks and vibrations normally incident to the railroad environment. The effect of these on the cask will be minimized through the use on the railcar of softly-sprung friction-controlled trucks and long travel hydraulic cushion gears at the couplers. A comprehensive review of the published information on railroad operations was carried out (References 36, 42, and 44). Design conditions for loadings imposed on the cask by the railcar were then established as shown below:

| <u>Loading Type</u> | <u>Loading Direction and Level</u> | | |
|---------------------|------------------------------------|----------|-----------|
| | Longitudinal | Vertical | Tranverse |
| Shock (g) | 3 | 2 | 1 |
| Vibration (g) | | | |
| * Peak envelope g's | 0.50 | 1.00 | 0.50 |
| Overall rms g's | 0.37 | 0.79 | 0.38 |

* These peak values are used to evaluate the package integrity under vibration (See Sect. 2.7)

3.2

Summary of Results

Tabulation of Stress vs. Allowables

| | Combined psi | All. psi | M.S. |
|---------------------------------|-----------------|-------------|-------|
| Cask Shells (Sect. 3.8) | See Section 3.8 | | |
| Bottom Head (Sect. 3.9) | | | |
| (1) Inner Plate | 810.48 | 35700 | 43 |
| (2) Center Plate | 835 | 33600 | 39.2 |
| (3) Outer Plate | 3146 | 35700 | 10.35 |
| Inner Head (Sect. 3.10) | | | |
| (1) Inner Plate | 1625 | 35700 | 20.9 |
| (2) Center Plate | 5019 | 33600 | 5.69 |
| (3) Outer Plate | 5963 | 35700 | 4.98 |
| Outer Closure Head (Sect. 3.11) | 1591 | 38800 | 24 |
| Water Jacket Expansion Tank | | | |
| (1) Shell (Sect. 3.13) | 4817.5 | 19575 | 3.06 |
| (2) Head (Sect. 3.13) | 7082 | 19575 | 1.76 |

3.3 Determination of Pressures in Containment Vessel and Water Jacket

The normal operating pressure (no credit for auxiliary cooling) is based on the average fuel rod temperature, under normal conditions of transport. The inner container is initially filled with helium at an assumed gas temperature of 68° F and one atmosphere pressure.

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

3.3.1 Containment Vessel Pressure based on PWR Type Fuel

Table 1 shows design parameters of the four common types of fuel rods. The values shown except for V_1 were obtained or calculated from data in the fuel PSARs. V_1 was calculated from the following equation :

$$V_1 = \frac{P_0 T_1 V_0}{T_0 P_1} + \frac{1.36E f M R T_1}{P_1} \quad (1) \text{ Ref. 22}$$

It is possible that future elements will be pressurized to a higher pressure than that shown in Table (1) (P_0). Therefore a new value of P_1 will be calculated using equation (1) with an initial fill pressure of 515 PSIA for PWR type fuel. This assumes that the initial geometry of the rods and the volume V_1 does not change. The total amount of gas in the rods will then be calculated. It will then be assumed that all the rods release the gas to the inner container. The pressure of the inner container will be calculated for normal conditions of transport and for the fire accident condition.

From equation (1)

$$P_1 = \frac{P_0 T_1 V_0}{T_0 V_1} + \frac{1.36E f M R T_1}{V_1}$$

Using the B&W element data from Table 1 since this results in slightly higher cask pressures, P_1 is calculated as follows:

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

$$P_1 = 3305 \text{ PSIA}$$

TABLE I

FUEL ROD GAS PRESSURE DESIGN PARAMETERS

| | <u>CE</u> | <u>B&W</u> | <u>Westinghouse</u> | <u>GE</u> |
|---|-----------------------|-----------------------|-----------------------|-----------------------|
| Internal Rod Pressure (P_1), psia | 2620 | 2765 | 2115 | 1100 |
| Gas Temperature in Reactor (T_1), ° R | 1100 | 1100 | 1100 | 1010 |
| Initial Fill Temperature (T_0), ° R | 530 | 530 | 530 | 530 |
| Initial Fill Pressure (P_0), psia | 15 | 365 | 365 | 15 |
| Initial Fill Volume (V_0), in. ³ | 1.73 | 1.534 | 1.58 | 4.482 |
| Fraction Released (f) | 0.31 | 0.23 | 0.21 | 0.35 |
| Burnup, (E) GWD/MTU | 37.3 | 42.2 | 42.0 | 37.4 |
| Uranium Loading (M) MTU | 2.25×10^{-3} | 2.19×10^{-3} | 2.21×10^{-3} | 4.04×10^{-3} |
| Free Gas Volume in Reactor (V_1), in. ³ | 0.627 | 0.889 | 1.14 | 2.76 |

This is the pressure that will develop in the fuel pins at the end of life condition if the initial fill pressure is 515 PSIA. From this pressure and the temperature and volume from Table 1, the number of Mols of gas in the fuel assembly can be calculated.

$$PV = NRT$$

$$N = \frac{3305 (.889)}{12 (1545) (1100)} = 1.44 \times 10^{-4} \text{ mols/fuel pin}$$

One PWR fuel assembly has 208 rods and there are 10 fuel assemblies in the cask.

$$N_F = 1.44 \times 10^{-4} (208) (10) = .299 \text{ mols}$$

The fuel cavity will be filled with helium before shipment if it is assumed that the helium fill gas temperature is 68°F and 14.7 psia. The amount of helium in the fuel cavity would be:

For PWR basket -

$$N_h = \frac{14.7 (144) (76.8)}{528 (1545)} = .199 \text{ mols}$$

The total number of mols in the containment vessel if all fuel rods rupture equals :

$$N_T = .299 + .199 = .498 \text{ mols}$$

The pressure in the containment vessel can now be calculated for normal conditions of transport.

Fuel temperature at normal conditions is 659 avg. (See 3.1)

$$P = \frac{.498 (1545) (659 + 460)}{76.8 (144)} = 77.85 \text{psia} \quad 63.15 \text{psig}$$

If fuel rods do not fail, pressure is based on fuel temperature of 659°F avg.

The containment vessel initially filled with helium at a temp. of 68°F and

one atmosphere pressure $P = \frac{(659 + 460) (14.7)}{(68 + 460)} = 31.15 \text{psia} \text{ or } 16.45 \text{psig}$

3.3.2 Containment Vessel Pressure based on BWR Type Fuel

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

$$P_1 = \frac{P_0 T_1 V_0}{T_0 V_1} + \frac{1.36 \epsilon M R T_1}{V_1}$$

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

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

Number of mols of gas in one fuel pin is:

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

One BWR fuel assembly has 49 rods and there are 24 fuel assemblies in the cask.

$$N_f = 2.56 \times 10^{-4} (49) (24) = .301 \text{ mols}$$

Number of mols of helium in containment vessel

$$N_h = \frac{14.7(144)(64)}{528(1545)} = .166 \text{ mols}$$

Total number of mols in containment vessel

$$N_T = .301 + .166 = .467 \text{ mols}$$

The pressure in the containment vessel can now be calculated for the normal conditions of transport.

Fuel temperature at normal conditions is 659° avg. (See 3.1)

$$P = \frac{.467 (1545) (659 + 460)}{64 (144)} = 87.6 \text{ psia or } 72.9 \text{ psig.}$$

If fuel rods do not fail, pressure is based on fuel temperature at 659° average. The containment vessel initially filled with helium at a temperature of 68° F and one atmosphere pressure.

$$P = \frac{(659 + (460) (14.7))}{68 + 460} = 31.15 \text{ psia or } 16.45 \text{ psig.}$$

Summary of Internal pressure conditions

Normal Operating Pressure

| | |
|--------------------|------------|
| @ PWR Temperatures | 16.45 psig |
| @ BWR Temperatures | 16.45 psig |

Normal Operating Pressure-Fuel Failure

| | |
|----------|------------|
| PWR Fuel | 63.15 psig |
| BWR Fuel | 72.9 psig |

3.3.3 Determination of pressure in Water Jacket

The bulk ethylene glycol and water temp is 340° F (See Sect. 3.1). The saturation pressure at this temperature is 85.3 psig. Water jacket design temperature is 340° F, resulting in a pressure of 148.3 psig. (See Sect. 3.13). Relief valve setting is 220 psig. ± 5%. The water jacket is designed for a pressure of 235 psig.

3.4 Thermal Stress Analysis of Aluminum Basket

Introduction

The objective of this analysis is to assess the plastic deformation of the aluminum basket for the initial steady state normal condition without auxiliary cooling. The plastic deformation results from local thermal stresses in the basket exceeding the yield stress of Type 1180 aluminum.

To perform the analysis, an elastic-plastic finite element computer program is utilized for a plane stress geometry. Only the aluminum basket and the inner stainless shell are modeled for this stress analysis. The absorber sleeves, lead, outer shell, etc., are not modeled here because their effect may be represented by appropriate boundary conditions on the basket and inner shell.

Program Description and Input

The computer model consists of constant strain (elastic-plastic) elements for the aluminum basket and isoparametric (elastic) elements for the stainless inner liner. In making the finite element model, the element mesh was chosen fine enough to accurately determine the strain gradients in the critical regions of the basket. With respect to this, a minimum of two elements through the thickness of the ligamented areas was judged adequate. Also, a finer mesh in the vicinity of the basket hole corners was utilized.

The effect of the lead in the steady state normal condition without auxiliary cooling is included in the model as a pressure of 20 psig acting on the inner shell. In the hot condition therefore, the lead provides very little restraint to radial expansion of the inner shell.

The ANSYS finite-element program (Ref. 1) uses an incremental ("initial strain") technique to obtain an elastic-plastic solution. The loading is applied in increments, and at each incremental level an elastic solution is first done. Subsequent iteration of the solution at each incremental level corrects the elastic solution for the plasticity effects. In this method, the plasticity lags the applied loading, and the predicted stresses are higher than the true stresses. The difference between predicted and true stresses is made arbitrarily small by performing a sufficient number of iterations at each incremental loading level.

The convergence of the iterative plastic solution is measured by the "plastic convergence" factor λ_i printed out at each iteration i . The "plastic convergence" is defined as (Ref. 2)

$$\lambda_i = \frac{\epsilon_{\text{plastic}, i} - \epsilon_{\text{plastic}, i-1}}{\epsilon_{\text{elastic}, i}}$$

Since the elastic strains $\epsilon_{\text{elastic}}$ are generally much smaller than the plastic strains, a value of $\lambda = .05$, for example, represents a very accurate solution.

The plastic deformation is considered time-independent in this analysis. The increments of (thermal) load are chosen as a computational convenience and do not correspond to transient temperature distributions intermediate to the steady state.

Yielding is predicted from the von Mises criterion. The elastic principal stresses (σ_1, σ_2) are combined to give

$$\text{SIGEFF} = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \sigma_2}$$

Whenever SIGEFF exceeds the yield stress, yielding occurs.

The Prandtl-Reuss flow relations are used to relate the stress components to the incremental components of plastic strain:

$$\frac{\delta \epsilon_x^P}{\sigma'_x} = \frac{\delta \epsilon_y^P}{\sigma'_y} = \frac{\delta \epsilon_z^P}{\sigma'_z} = \frac{\delta \gamma_{xy}^P}{\tau'_{xy}} = \frac{\delta \gamma_{yz}^P}{\tau'_{yz}} = \frac{\delta \gamma_{zx}^P}{\tau'_{zx}} = \frac{3}{2} \delta \delta$$

The primed stresses are the deviatoric stresses

$$\sigma'_x = \sigma_x - \frac{\sigma_x + \sigma_y + \sigma_z}{3}$$

and

$$\tau'_{xy} = \tau_{xy}$$

The proportionality factor $\delta \delta$ varies from point to point in the model; its significance is that the plastic solution follows the history of straining at each point in the body.

Upon reverse loading, the stress-strain curve is taken to be the same shape as the virgin stress-strain curve, but offset to account for the strain due to previous plastic deformation.

The stress-strain data used for 1180 aluminum is given in Table I. Because there is no sharply defined yield point, the most accurate way to specify the σ - ϵ is to include values for small strains as well as the usual larger strains. Also, in order to make the first elastic increment at 800°F compatible with the slope implied by $E = 4.0 \times 10^6$, selection of a first interval of $\epsilon = 10^{-4}$ was necessary. When the strains exceed $\epsilon = 10^{-4}$, deviation from purely elastic behavior is registered by the computer program and the value of σ is found from the σ - ϵ curve at the appropriate (interpolated) temperature.

The program combines the (x,y,z) components of strain to find an equivalent uniaxial strain "EPGEN" which can be used with the σ - ϵ curve. EPGEN is found by summing the elastic and plastic strains and combining according to:

$$\text{EPGEN} = \frac{\sqrt{(\epsilon_x - \epsilon_y)^2 + (\epsilon_y - \epsilon_z)^2 + (\epsilon_z - \epsilon_x)^2 + \frac{3}{2} \gamma_{xy}^2}}{\sqrt{2} (1 + \text{POSGEN})}$$

where

$$\text{POSGEN} = 0.5 - (0.5 - \nu) \frac{E_{\text{secant}}}{E_{\text{elastic}}}$$

is the effective Poisson's ratio for the element. A value of 0.33 represents elastic deformation for aluminum, and the maximum

value of 0.50 represents pure plastic flow.

The equivalent stress obtained from the uni-axial strain is called "SIGE" in the program.

The steady state temperatures in the normal condition without auxiliary cooling were obtained from the PWR and BWR thermal solutions in Section VIII. Since the steady state condition is characterized by the highest temperatures and the lowest yield stresses, the maximum plastic deformation is expected for this condition.

The plastic calculation requires the thermal load to be applied incrementally. The initial condition is taken to be uniform temperature at 70°F. Plastic solutions are obtained for intermediate temperature distributions scaled linearly between the initial uniform distribution and the final steady state distribution. The results at each intermediate level are used as initial conditions for the next level. In this manner, the severe nonlinearities imposed by the plastic behavior are made more tractable. This method of incremental loading is not intended to simulate transient heatup behavior, but is merely a computational device.

Results

Figures 1 through 4 show the element and node numbering for the PWR and BWR geometries.

Plots of the radial gap "USEP" between the aluminum basket and the inner shell versus angular location are given in Figures 5 and 6. With the assumption of a 105 mil cold gap, neither the PWR nor the BWR basket contacts the inner shell in the hot condition. In this model, the lead does not have sufficient restraint to prevent the inner shell from displacing radially. Although the aluminum basket expands in the hot condition, there is enough corresponding expansion of the inner shell so that contact does not occur.

The maxima of "USEP" in Figures 5 and 6 correspond to the corners of the baskets nearest the perimeter. The average gap in both cases is about 35 mil.

The plastic solution shows the total strains to be relatively small, of the order of 0.2%. In Table II, a list of representative elements having high strains is given. The largest strains occur near the corners of the basket holes and in the innermost ligaments. In these regions, a small amount of plastic flow is sufficient to reduce the "elastic" stresses by a large factor. An equivalent strain of .00417 in Element 409 of the BWR was the maximum strain computed in both models. Element 409 is at a corner, and the high shear strain results from a stress concentration effect. The results in the vicinity of the basket corners are conservative, since the corners are sharp in the computer model, but have a 1/4" radius in the actual basket.

Figures 7 and 8 show the iso-stress plots of the equivalent stress for the PWR and BWR baskets, respectively. The large gradients in the vicinity of the basket corners are evident in these computer plots.

In order to determine the cyclic heatup-cooldown behavior of the aluminum basket, the computer models were unloaded quasi-statically back to a uniform 70°F temperature. The results of the quasi-static unloading calculation showed that the aluminum basket experienced elastic shakedown after the first cycle. That is, a certain amount of permanent plastic deformation remains after the first unloading, but the residual stresses in the unloaded condition are sufficient to prevent further plastic deformation during any subsequent heatup or cooldown.

The amount of plastic "set" after the first unloading is quite small; the maximum element set strain was of the order of 0.1%, which is well within acceptable limits in terms of functional operability.

Figure 9 shows the computer printout for a typical PWR element. The upper printout is for the hot steady state condition, and the lower is for the unloaded condition at 70°F. In the hot condition, the plastic strains are displayed in the row labeled "PLASTIC." In the unloaded condition, these hot plastic strains are entered into the "O.SHIFT" row, and the new cold plastic

strains are in the "PLASTIC" row.

The cold plastic strains are in the opposite direction to the hot plastic strains, so that the net plastic strain at the end of the cycle is considerably smaller than the hot plastic strain. Further, the stresses in the unloaded condition (i.e., the residual stresses) are substantial and opposite in sign to the stresses in the loaded condition. The aluminum basket will therefore not yield on the next heatup.

In examining these data, it is important to remember that the σ - ϵ curves are very different in the hot and cold conditions. This explains why "SIGEFF" is about twice as much in the unloaded case as in the hot condition.

Conclusions

For the assumptions regarding material properties and boundary conditions employed in this analysis, the ANSYS plastic finite element solution shows the strains in the aluminum basket to be less than 0.42% in the normal steady state without auxiliary cooling. The maximum strain is therefore two orders of magnitude less than the total strain to failure at 700°F given in Section 1.2.

With the 105 mil cold gap, the aluminum basket and the inner liner do not contact each other at the steady state temperature.

When the cask is cooled down to room temperature in a slow, uniform manner, plastic set of the order of 0.1% remains. The existence of the residual stress field in the cold condition assures that the aluminum basket will experience totally elastic behavior during subsequent slow heatup-cooldown cycles.

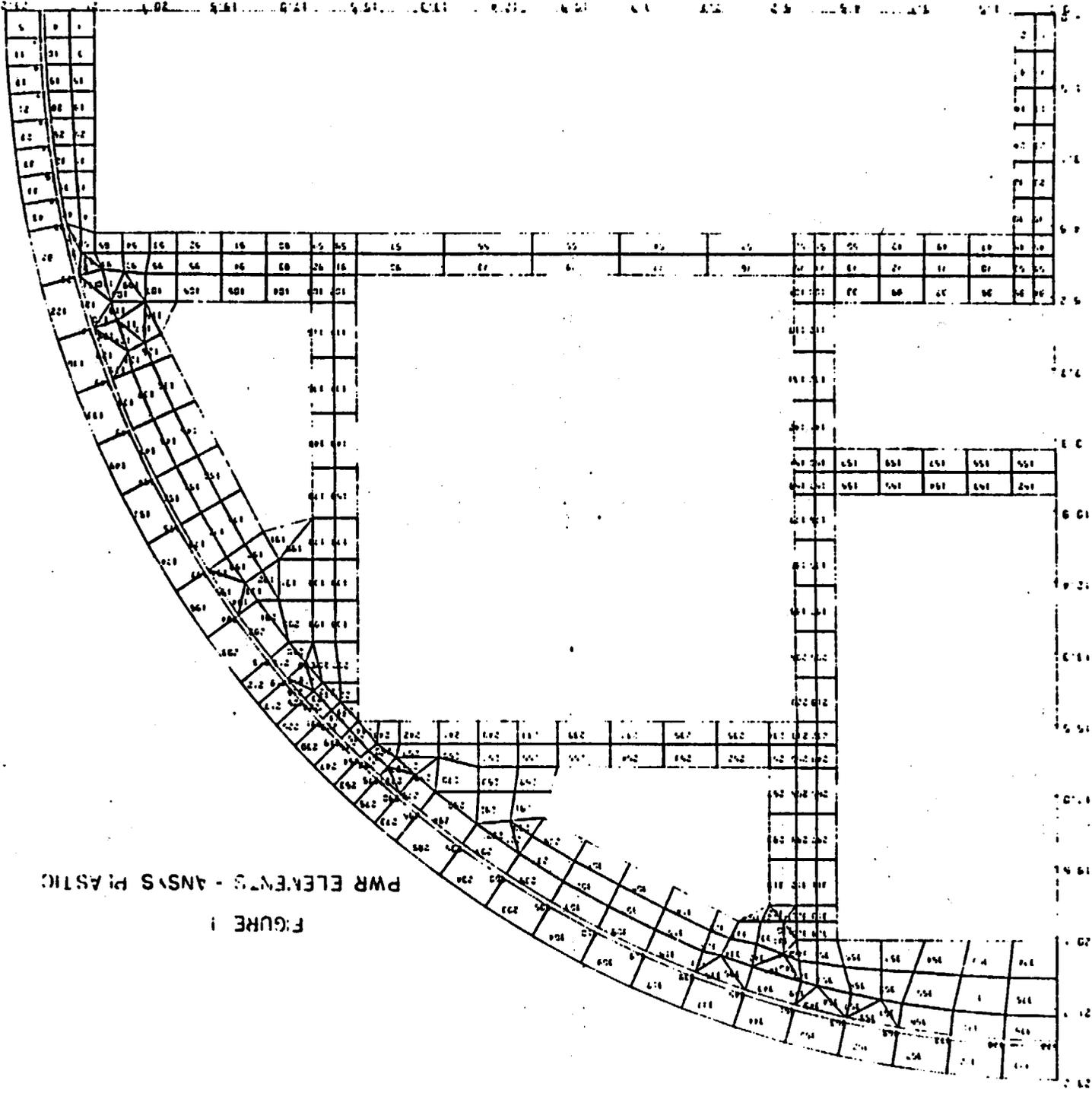


FIGURE 1
PWR ELEMENTS - ANSYS PLASTIC

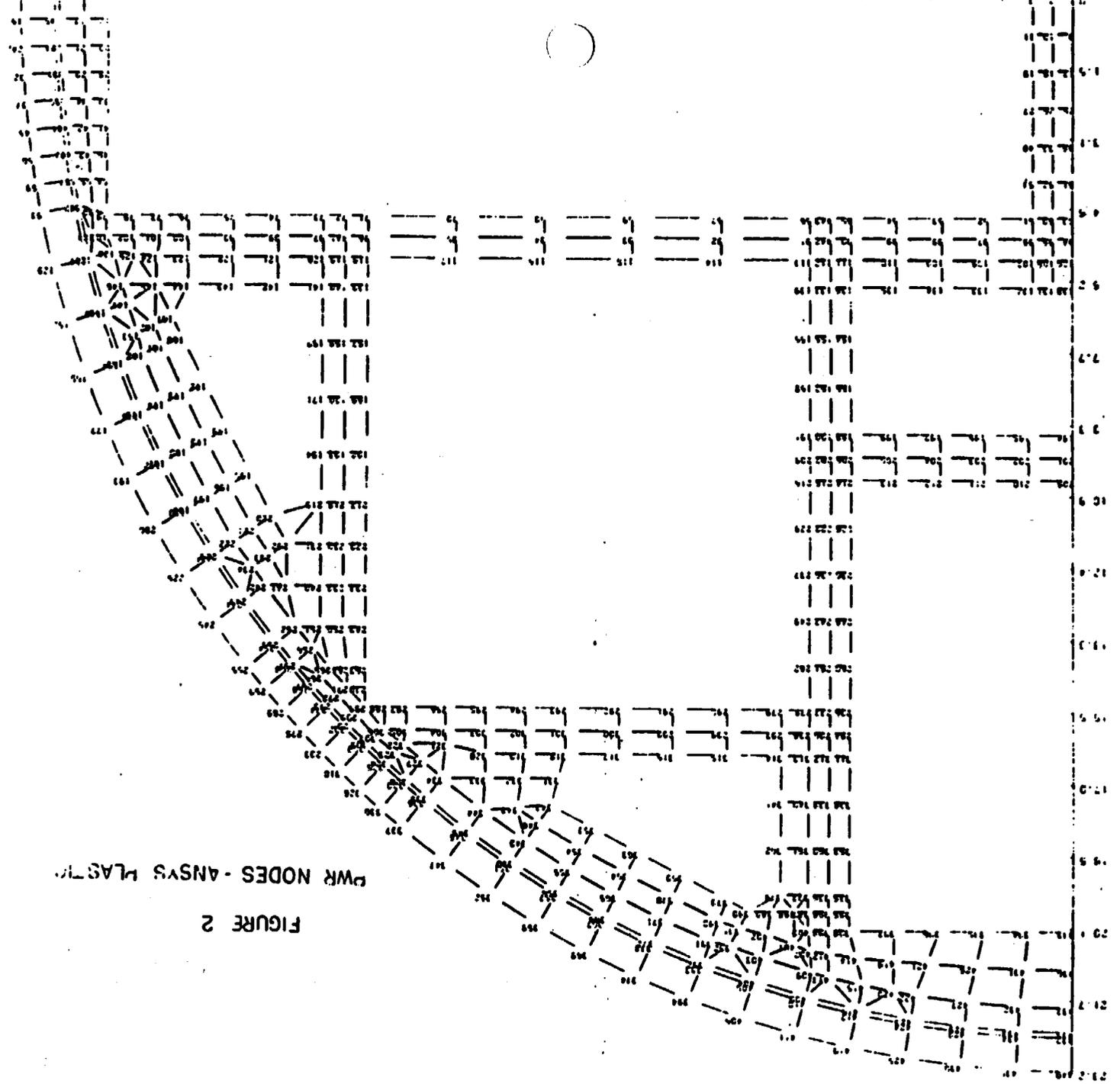
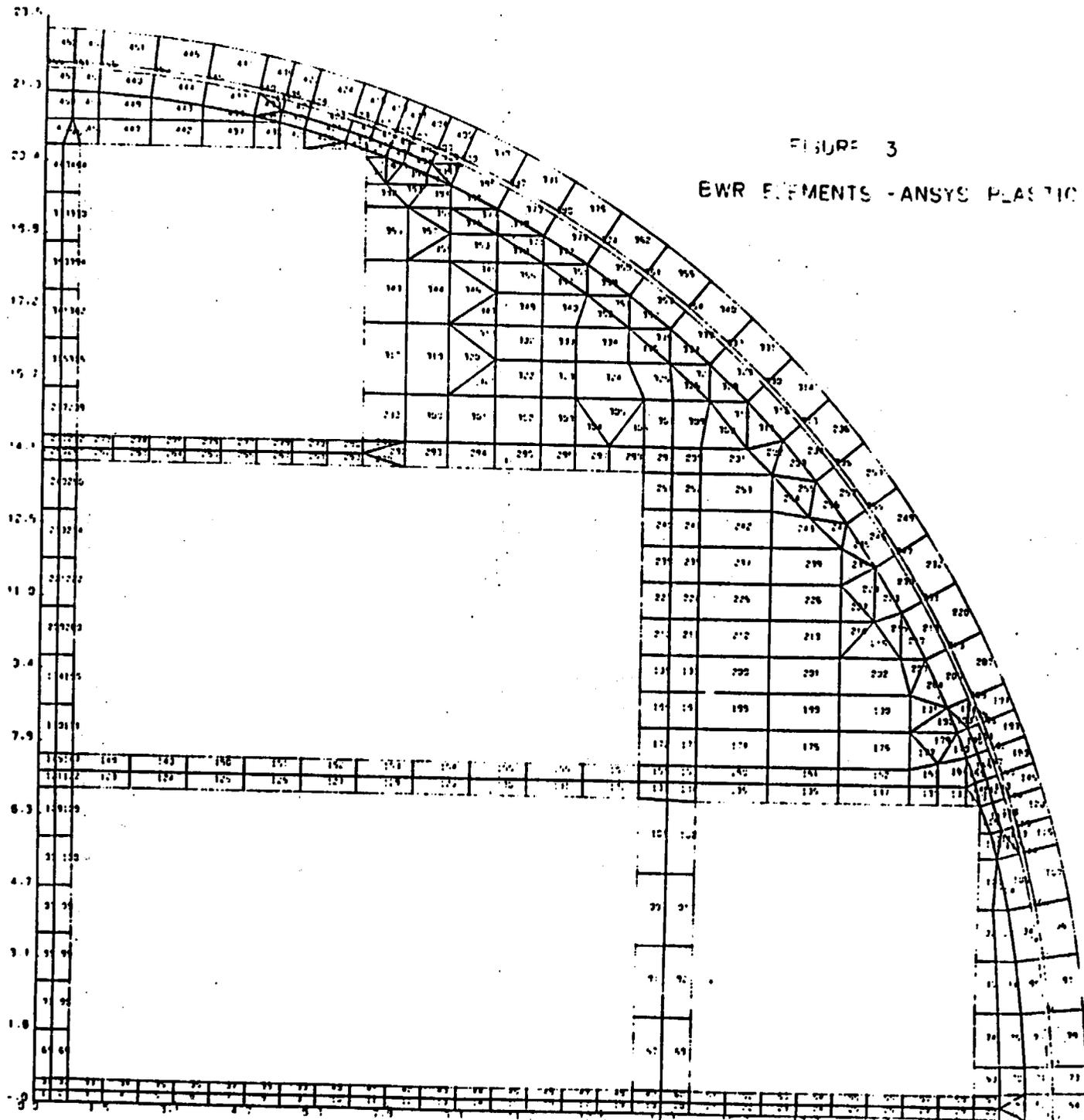


FIGURE 2
PWR NODES - ANSYS PLASTIC

XI-3-27



REV. 1 - 0/1/13

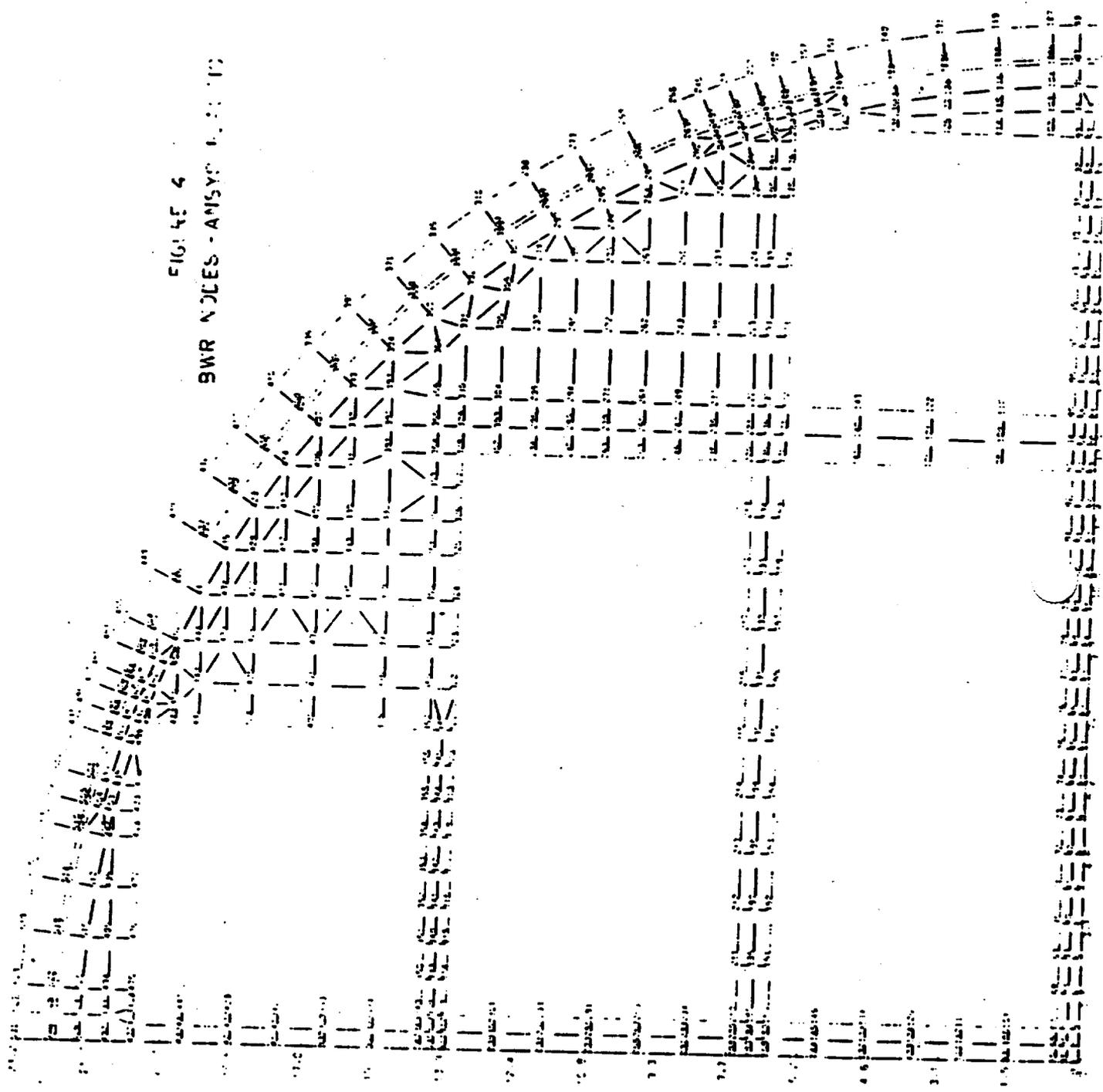


FIGURE 4
 SWR POINTS - ANSYS (1)

FIGURE 5
PWR GAP FOR NORMAL STEADY
STATE W/O AUXILIARY COOLING

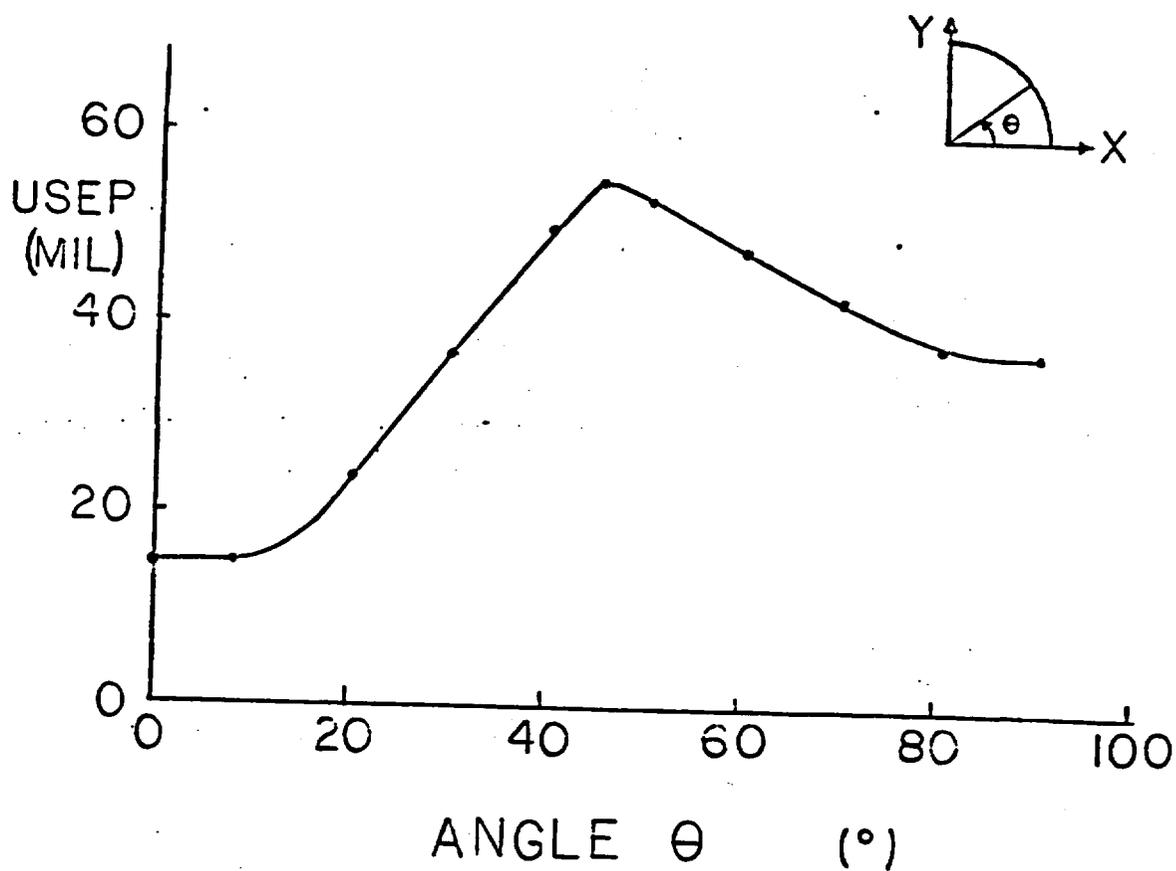


FIGURE 6
BWR GAP FOR NORMAL STEADY
STATE W/O AUXILIARY COOLING

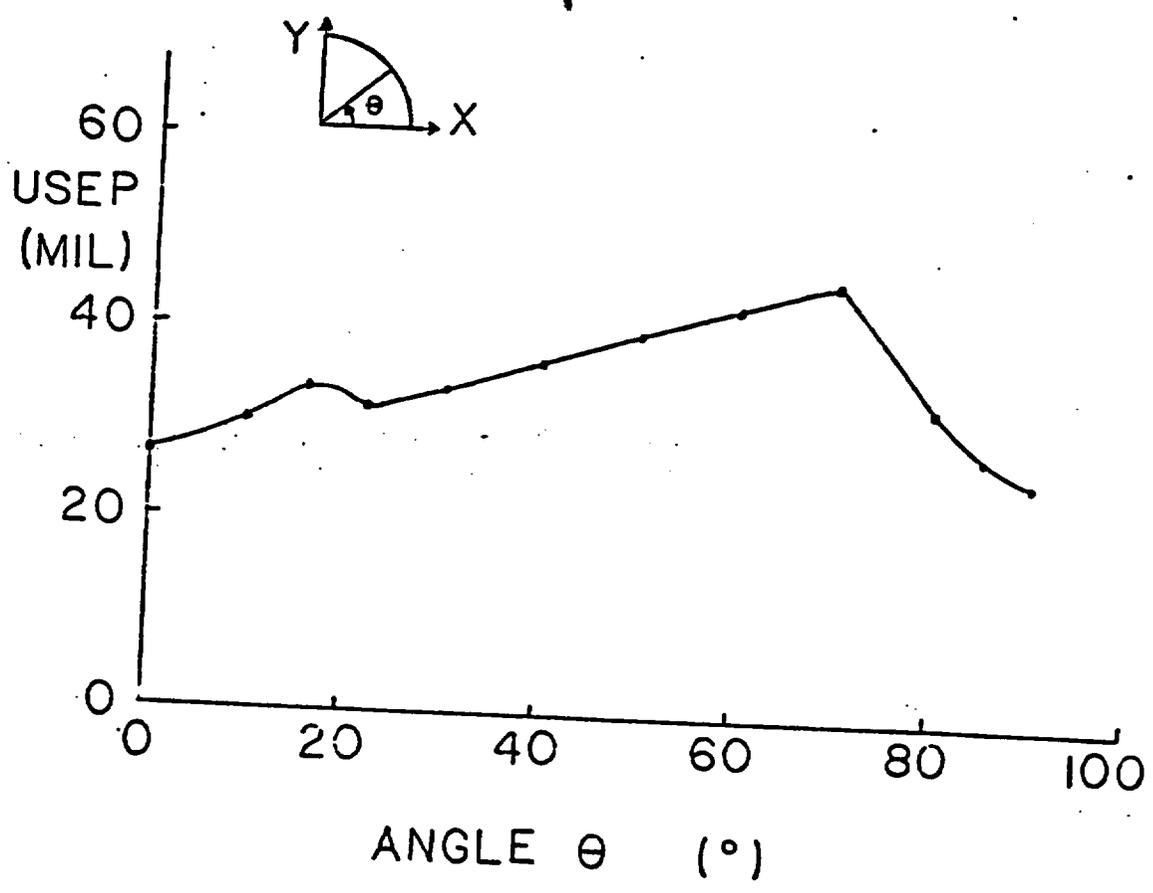
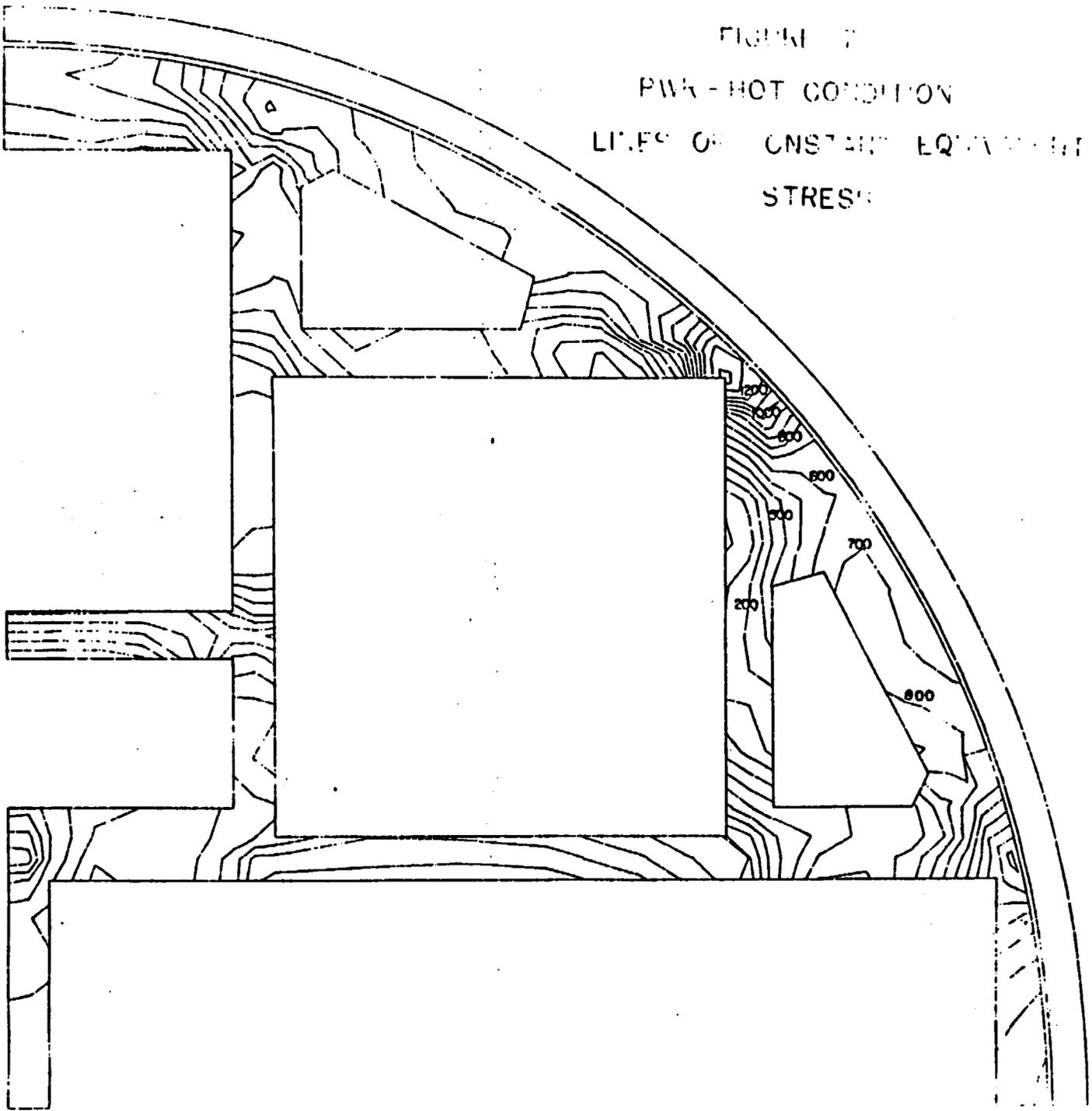
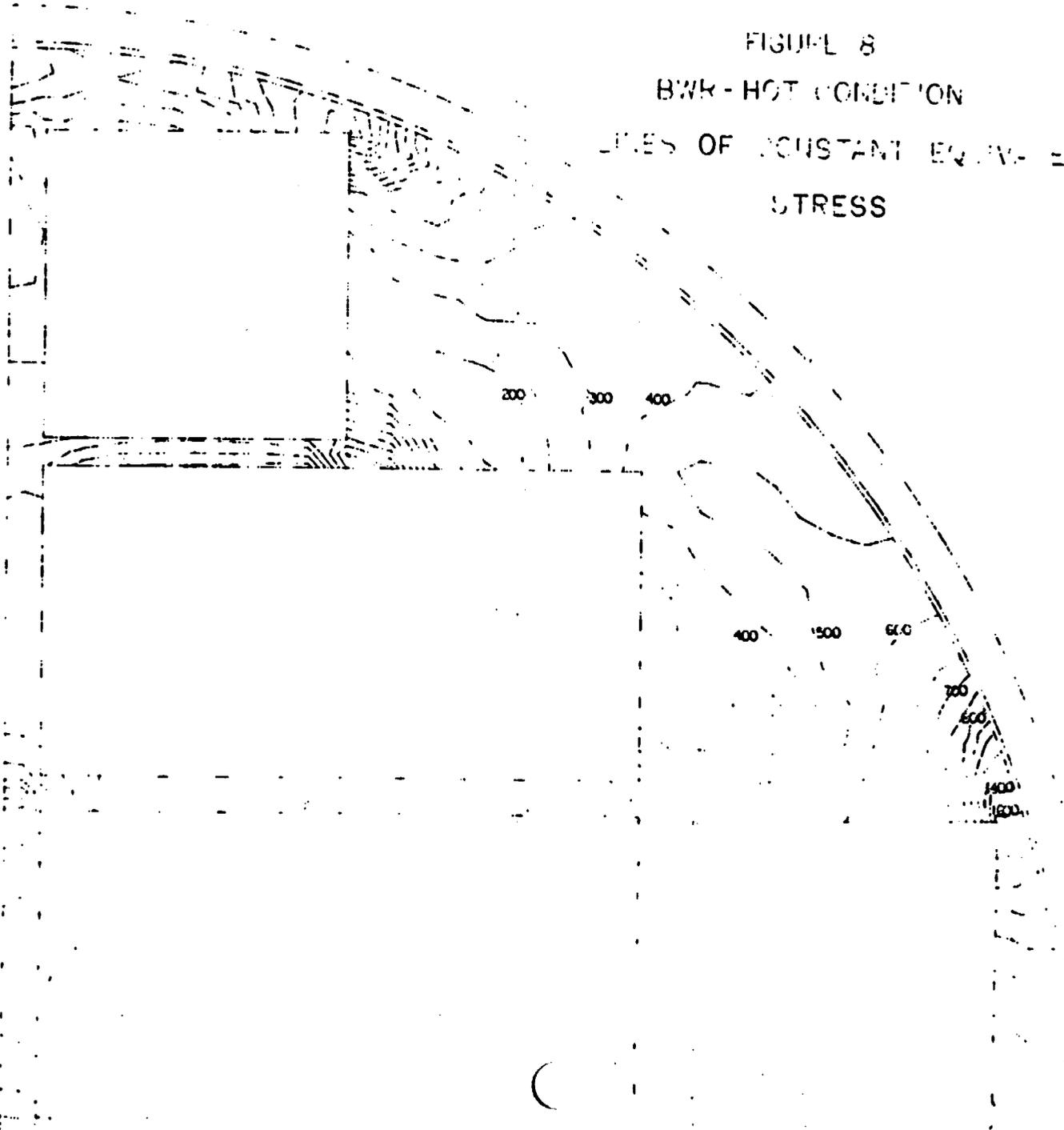


FIGURE 7
PWR - HOT CONDITION
LINES OF CONSTANT EQUIVALENT
STRESS



XI-3-31

FIGURE 8
BWR - HOT CONDITION
LINES OF CONSTANT EQUIVALENT
STRESS



XI-3-32

Typical PWR Element Printout - ANSYS Program

Figure 9

ELEM 67 NODES= 104 81 -0 MATERIAL = 1 AREA= .0846 STRESS INT.= 3015. SIGEFF= 2728.
 X= 21.70 Y= 4.90 I= 70. XY STR= 258. -1838. 1083.
 ELASTIC .0000865 -.0001924 .0002081 .0000502 PLASTIC .0005554 -.0009089 .0014686 .0003535
 THERMAL 0.000000 0.000000 0.000000 0.000000 SWELL 0.000000 0.000000 0.000000 0.000000
 CREEP 0.000000 0.000000 0.000000 0.000000 0.SHIFT -.0007216
 PUGEN= .0015210 PUSGEN= .4696 SIGE= 2723.27 FLUENCE= 0.
 .0009365 -.0018747 -.0002149

ROOM TEMPERATURE UNLOADED CONDITION

ELEM 67 NODES= 104 00 81 -0 MATERIAL = 1 AREA= .0846 STRESS INT.= 1556. SIGEFF= 1347.
 X= 21.70 Y= 4.90 I= 581. XY STR= .436. -631. -474.
 ELASTIC -.0000897 .0000858 -.0002439 .0000019 PLASTIC -.0007216 .0009365 -.0018747 -.0002149
 THERMAL .0072612 .0072612 0.000000 0.000000 SWELL 0.000000 0.000000 0.000000 0.000000
 CREEP 0.000000 0.000000 0.000000 0.000000 0.SHIFT 0.000000
 PUGEN= .0016545 PUSGEN= .4792 SIGE= 1347.37 FLUENCE= 0.
 .0009365 -.0018747 -.0002149

HOT STEADY STATE CONDITION

TABLE I
1180 ALUMINUM DATA

| Temperature | $\epsilon = .0001$ | $\epsilon = .0005$ | .0020 | .01 | .10 | E |
|-------------|--------------------|--------------------|-------|------|------|--------------------|
| 70°F | $\sigma = 1000$ | 2240 | 2950 | 4500 | 7500 | 10.0×10^6 |
| 500°F | 850 | 1330 | 1670 | 2200 | 2360 | 8.5×10^6 |
| 600°F | 650 | 1140 | 1420 | 1850 | 1890 | 6.5×10^6 |
| 700°F | 500 | 940 | 1180 | 1550 | 1590 | 5.0×10^6 |
| 800°F | 400 | 750 | 940 | 1200 | 1280 | 4.0×10^6 |

3.5 Axial Thermal Expansion

3.5.1 PWR Basket Arrangement

The Table on Pg. XI-3-37 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.

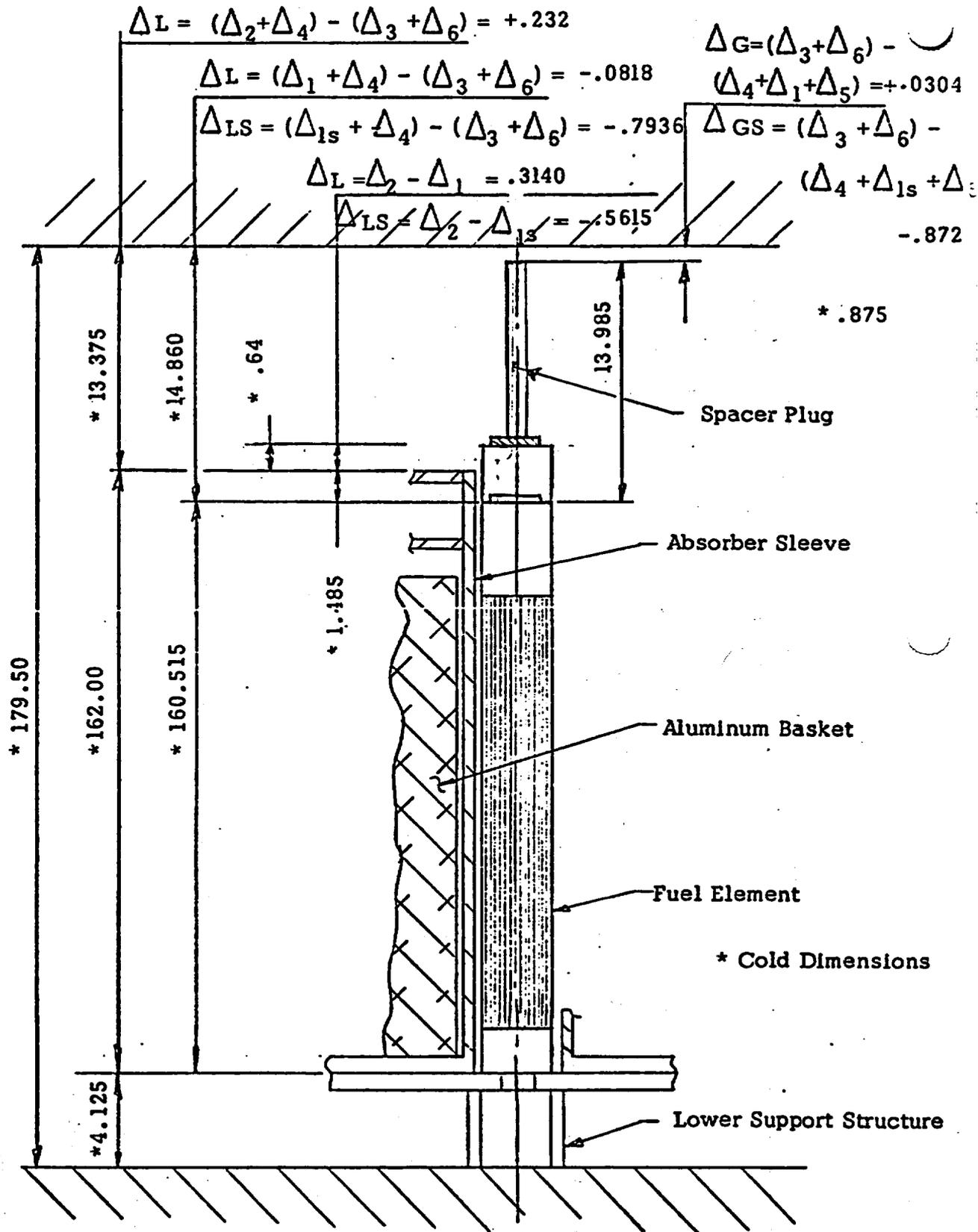
It can be seen from the figure on Pg. XI-3-37 that under normal operating conditions without auxiliary cooling, minimal gaps exist between the spacer plug and the underside of the closure head and the top of the fuel basket and the spacer plug. This arrangement provides adequate fuel support yet does not impose thermal expansion forces on the internal structure or the containment vessel.

PWR Temperatures and relative expansions: Normal Conditions

| Item | Length | Temp. ° F | ΔT ° F | (mean) α in/in/°F | Material | Expansion $\Delta L = L (\Delta T)\alpha$ |
|-------------------------|---------|--------------|-------------------|-----------------------------|------------|--|
| Fuel Element (1)(3)(4) | 160.515 | 833 | 765 | 10.08×10^{-6} | S.S. 304 | $\Delta_{1s} = 1.2377$ |
| Fuel Element (1)(3)(4) | 160.515 | 833 | 765 | * 2.95×10^{-6} | Zlr. 4 | $\Delta_1 = .3622$ |
| Absorber Sleeve | 162.00 | 499 | 431 | 9.70×10^{-6} | S.S. 304 | $\Delta_2 = .6762$ |
| Containment Shell (2) | 156.00 | 353 | 285 | 9.47×10^{-6} | S.S. 304 | $\Delta_3 = .4210$ |
| Lower Support Structure | 4.125 | 438 | 370 | 9.67×10^{-6} | S.S. 304 | $\Delta_4 = .01476$ |
| Spacer Plug | 13.985 | 438 | 370 | 13.75×10^{-6} | Al.6061-T6 | $\Delta_5 = .07886$ |
| Top Forging | 11.50 | 410 | 342 | 9.62×10^{-6} | S.S. 304 | $\Delta_6 = .0378$ |

- (1) Fuel element length is to the point where spacer compression legs pickup hard point of the fuel element.
- (2) Calculations for containment shell are based on axial expansion of 2" outer shell, a length of 156" is used.
- (3) The guide thimbles control fuel growth due to temperature.
- (4) Irradiated length

* See Ref. 30 , A maximum value is used for axial expansion to insure a conservative calculation of Thermal growth.



3.5.2 BWR Basket Arrangement

The Table on Page XI-3-39a 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-3-39b that under normal operating conditions without auxiliary cooling, 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.

| | Length in. | Temp. ° F. | ΔT ° F. | (mean) α in/in ° F. | Material | Elongation |
|------------------------------------|---------------|---------------|--------------------|-------------------------------|---------------|------------------------|
| Fuel Element (4) | 152.925 | 655 | 587 | 9.88×10^{-6} | S.S. Type 304 | $\Delta_{1s} = 0.9449$ |
| Fuel Element (1)(3) | 157.890 | 655 | 587 | * 2.95×10^{-6} | Zr.-4 | $\Delta_1 = 0.2745$ |
| Fuel Element (2) | 5.035 | 655 | 587 | 9.88×10^{-6} | S.S. Type 304 | $\Delta_1^1 = 0.0292$ |
| Absorber Sleeve | 167 | 499 | 431 | 9.7×10^{-6} | S.S. Type 304 | $\Delta_2 = 0.698$ |
| Containment Shell | 156 | 353 | 285 | 9.56×10^{-6} | S.S. Type 304 | $\Delta_3 = 0.425$ |
| Top Forging | 11.5 | 410 | 342 | 9.6×10^{-6} | S.S. Type 304 | $\Delta_4 = 0.0378$ |
| Lower Support Structure | 3 | 438 | 370 | 9.64×10^{-6} | S.S. Type 304 | $\Delta_5 = 0.0107$ |
| Spacer Plug | 8.9 | 438 | 370 | 13.75×10^{-6} | Al.5052-H32 | $\Delta_6 = 0.0458$ |
| Difference In Spacer Arm Length | 3.615 | 438 | 370 | 13.75×10^{-6} | Al.5052-H32 | $\Delta_6^1 = 0.0184$ |

- (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

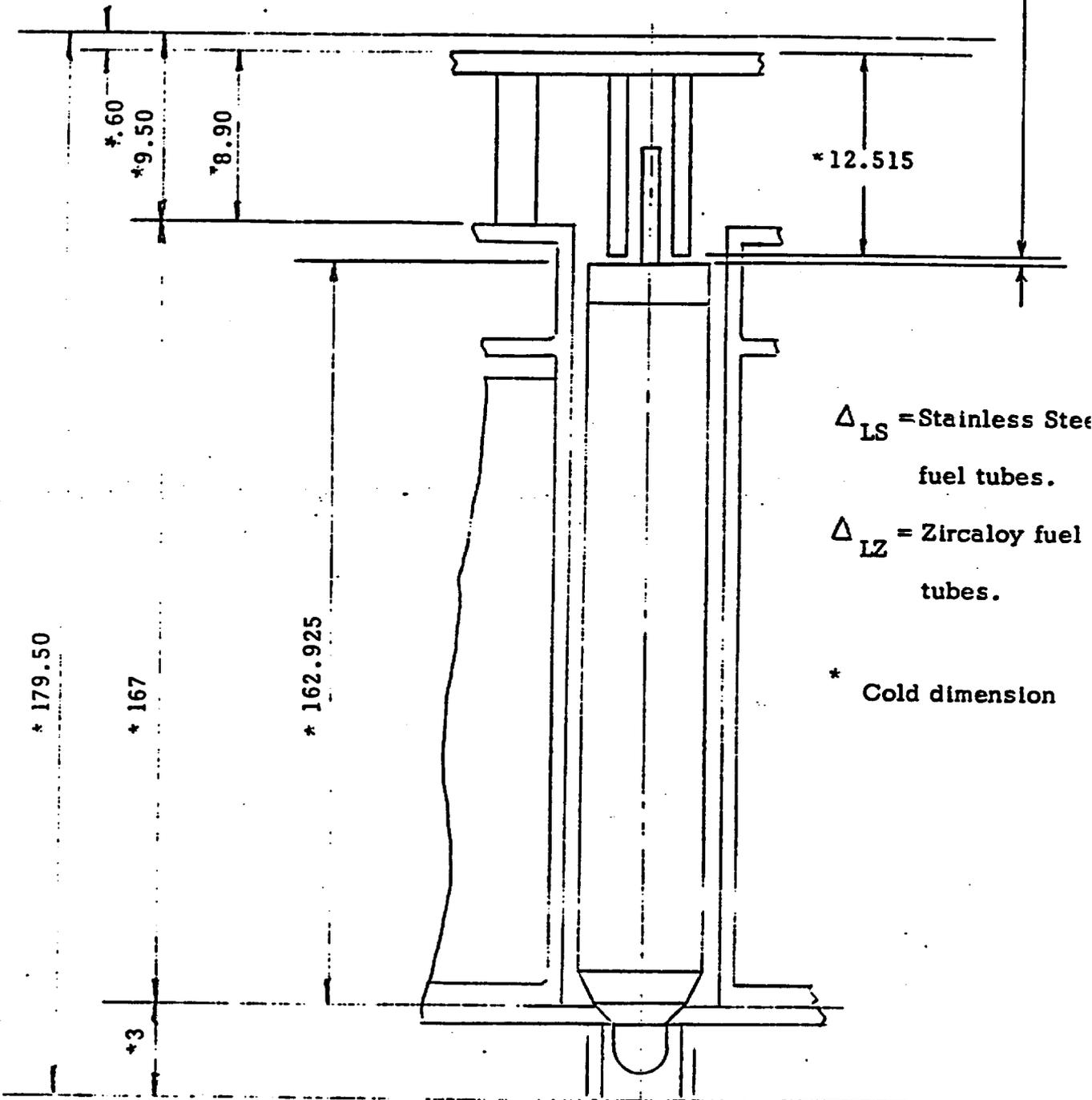
* See Ref. 30. A maximum value is used here in order to obtain a conservative elongation value.

BWR - Axial Expansion

$$\Delta_L = (\Delta_3 + \Delta_4) - (\Delta_2 + \Delta_5 + \Delta_6) = -.2917$$

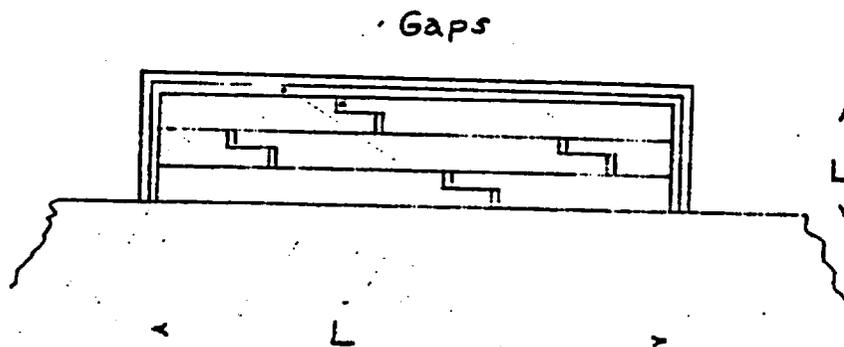
$$\Delta_{LS} = \Delta_2 - (\Delta_6^1 + \Delta_{1s}^1) = -.2650$$

$$\Delta_{LZ} = \Delta_2 - (\Delta_6^1 + \Delta_1^1 + \Delta_1^1) = +.3759$$



3.6 Thermal Expansion of Solid Neutron Shielding

RICORAD PPC-V will be used as solid neutron shielding material in the cask. RICORAD PPC-V has excellent high temperature stability—no melting, sag or slump at temperatures up to 600°F. Its coefficient of linear thermal expansion is $(1.51) 10^{-4}$ in/in/°F. Gaps will be provided to accommodate thermal expansion (Section 1.2) of the shielding as shown in the following sketch:



Gap size will be established as follows:

$$\text{Gap length} = \frac{(1.51) 10^{-4} L (T-68)}{\text{NO. of Gaps}}$$

Where

T = average shield temperature in °F

3.7 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 3.8.

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XI-3-43

XI-3-44

XI-3-45

XI-3-46

XI-3-47

XI-3-48