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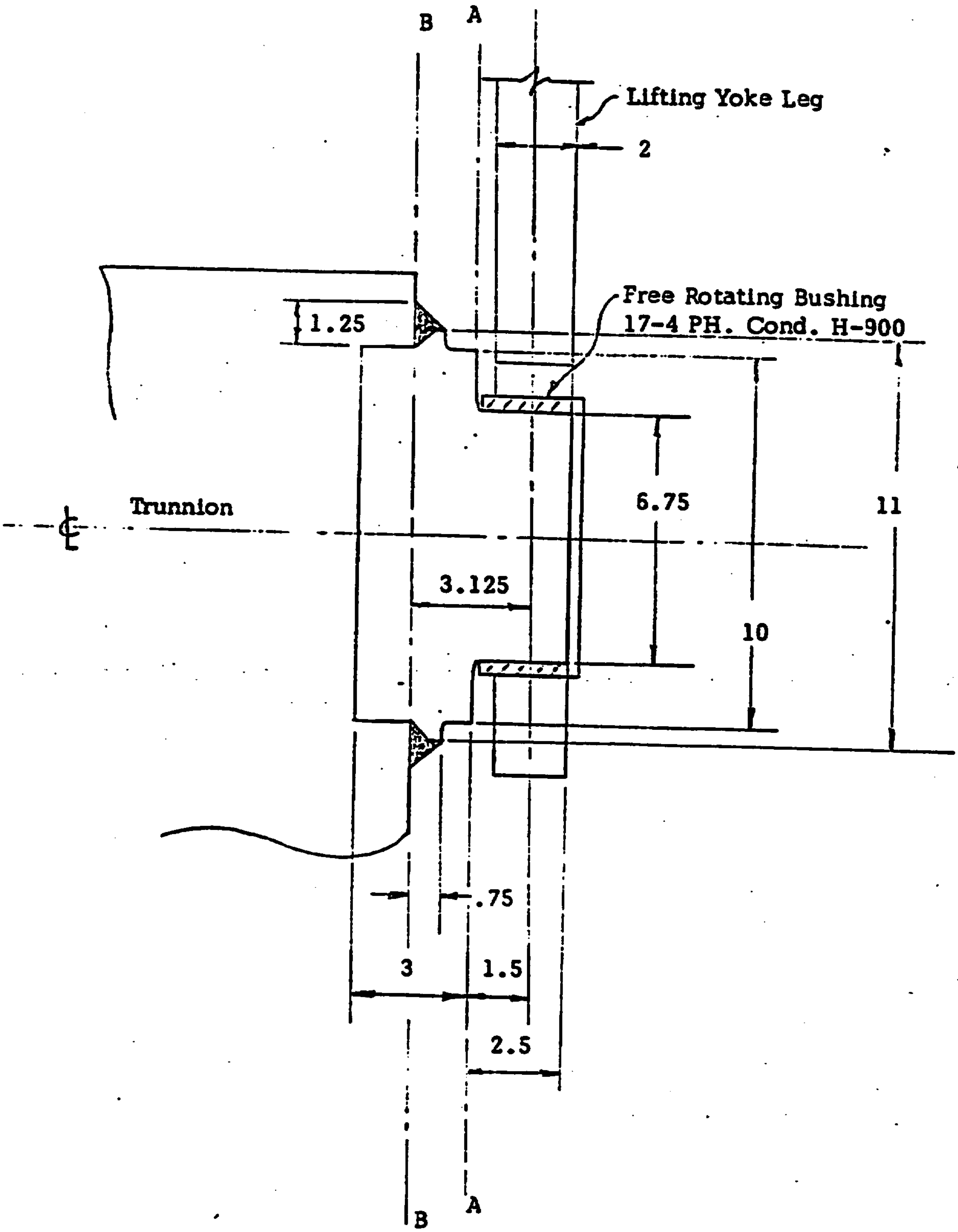


TRUNNION DETAIL

Section B-B Weld - 45° conical throat

Throat area of weld = $t \pi d$

$$= (.707 \times 1.25)\ \pi\ (10 + \frac{1.25}{2}) = 29.5 \text{ in.}^2$$

Z of throat area of weld

$Z = \pi r^2 t$ of thin cylinder

$$= \quad A\ (\frac{r}{2}) = 29.5\ (\frac{10.625}{4}) = 78.36 \text{ in.}^3$$

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

$$S_t = \frac{11964}{2} + \sqrt{5982^2 + 2777^2}$$

$$= 5982 + 6595 = 12{,}577 \text{ psi}$$

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

Section A-A 6.75 dia. trunnion section at root

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

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

Bending Moment $M_B = 300{,}000\ (1.5) = 450{,}000$ 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

$$S_t = \frac{14930}{2} + \sqrt{7465^2 + 8384^2} = 18691 \text{ psi.}$$

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

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.

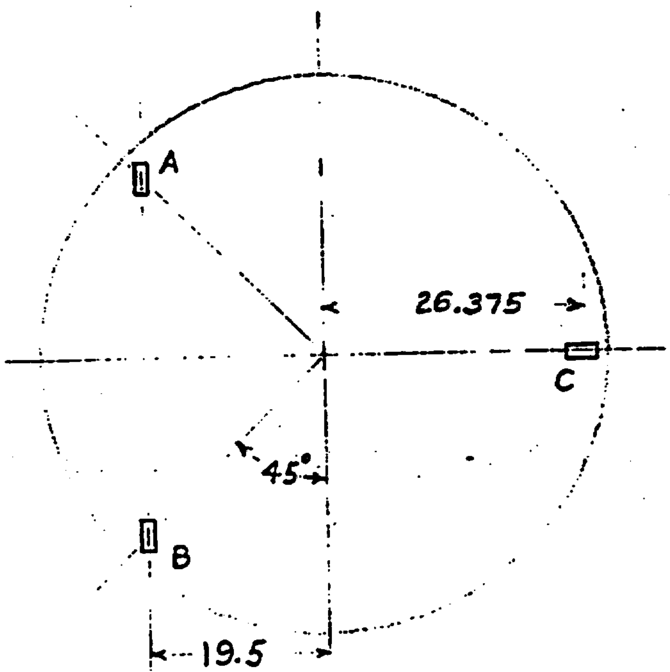
The eye bolts are 3/4" (shank) diameter, each rated at a strength of 23,400 lbs. which is approximately five times the design load carried by one eye bolt taking half the design load.

2.1.3 Inner Closure Lifting Lugs

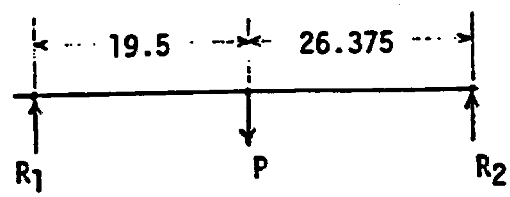
Inner closure design weight is 7400 lb., (sect. VII).

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:

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

$R_1 = 22200 - 9437 = 12763$ lb.

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

Eye bolt C, $F_C = \underline{9437}$ 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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2.2 Tie-Downs

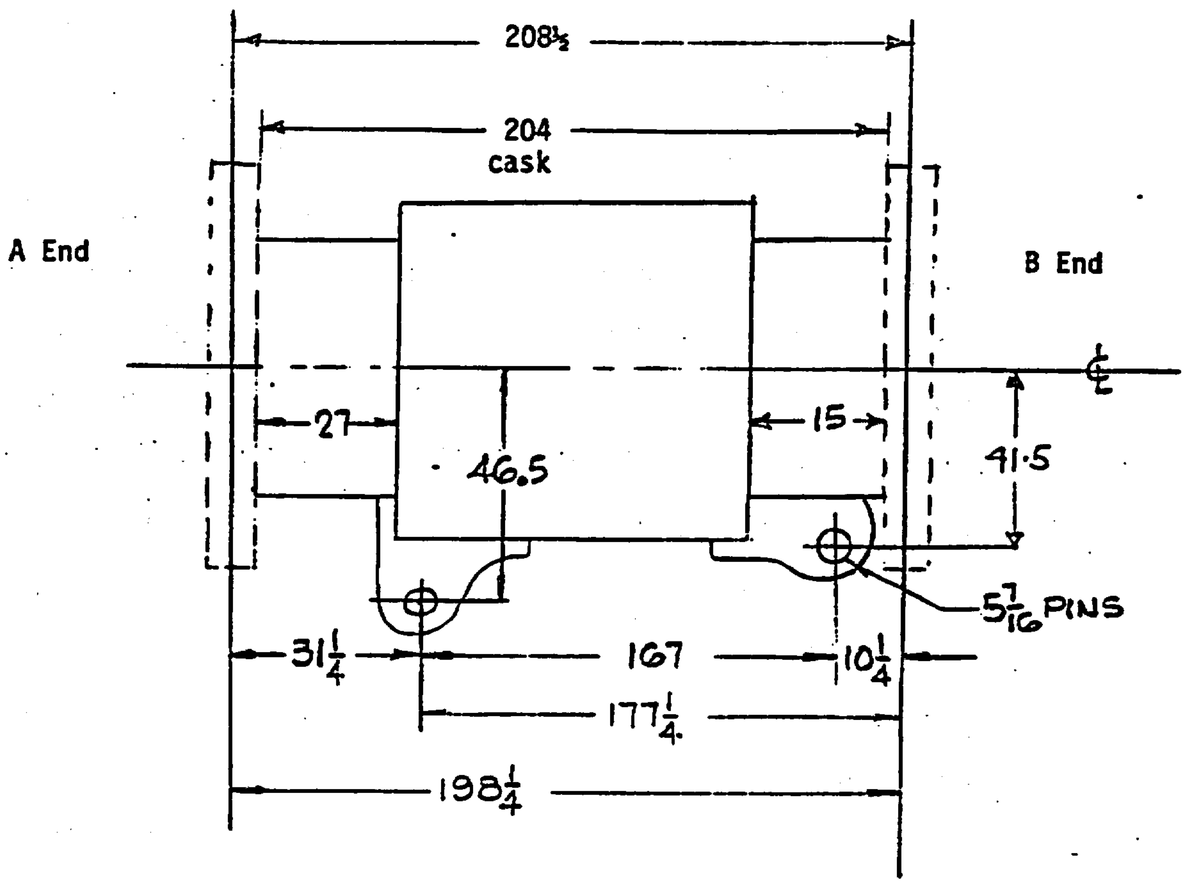
The tie-down system consists of (1) cask lugs with mating plate anchorages welded to the railcar center sill to take vertical upward and longitudinal loads, and (2) V saddles to take vertical downward and lateral loads.

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 45^0 flat saddle bearing plates (which form a 90^0 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 x 220,000 = 2,200,000 lbs at rear tie down.

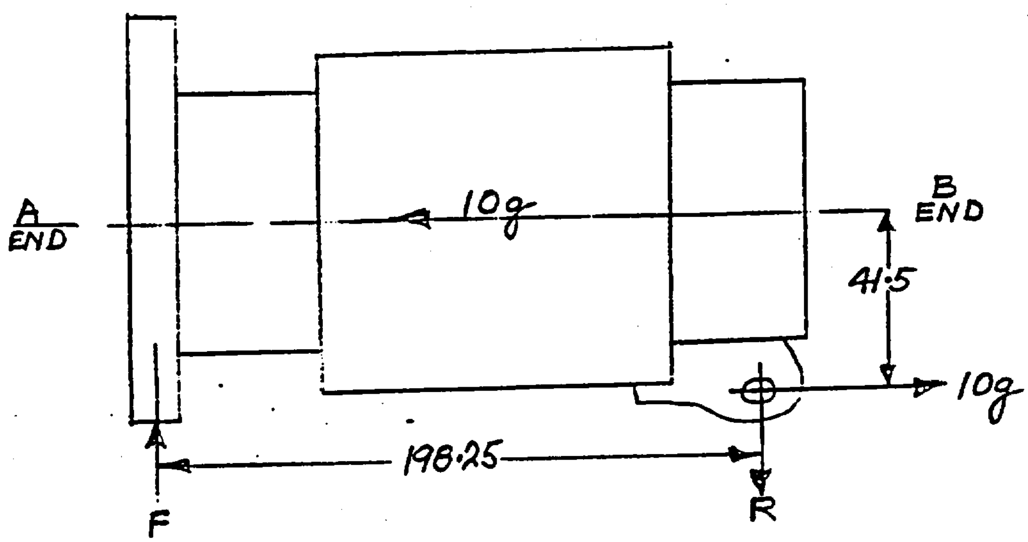
5g Transverse = $\frac{5 \times 220,000}{2}$ = 550,000 lbs at front and rear saddles

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

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 foreward on cask



Pitching couple at F and R

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

Verticals for rear tie down at R

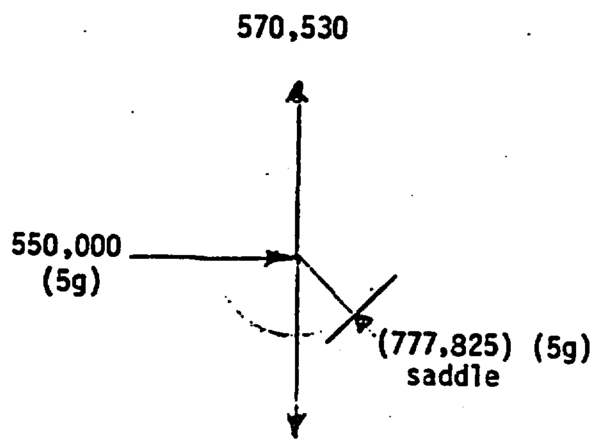
↑ 460,530 pitching 10g
↑ 220,000 2g V
↓ 110,000 1g static wt.

↑ 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



R = 1,120,530 Lug
Rear Tie-down plane (R)
A End Impact

Rear tie-down cask lug and pin, net loads

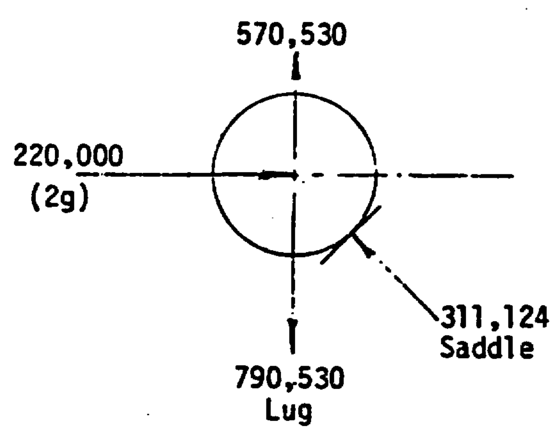
1,120,530 lbs. ↑ vertical, tension

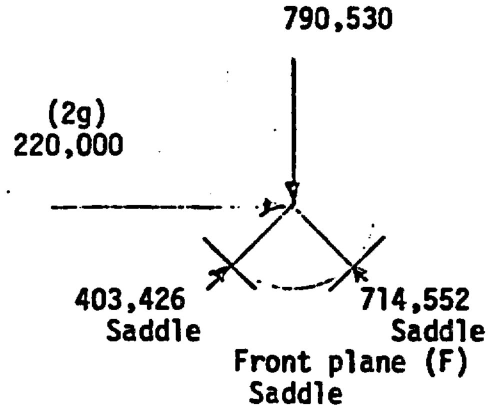
2,200,000 lbs. ← toward A end.

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.



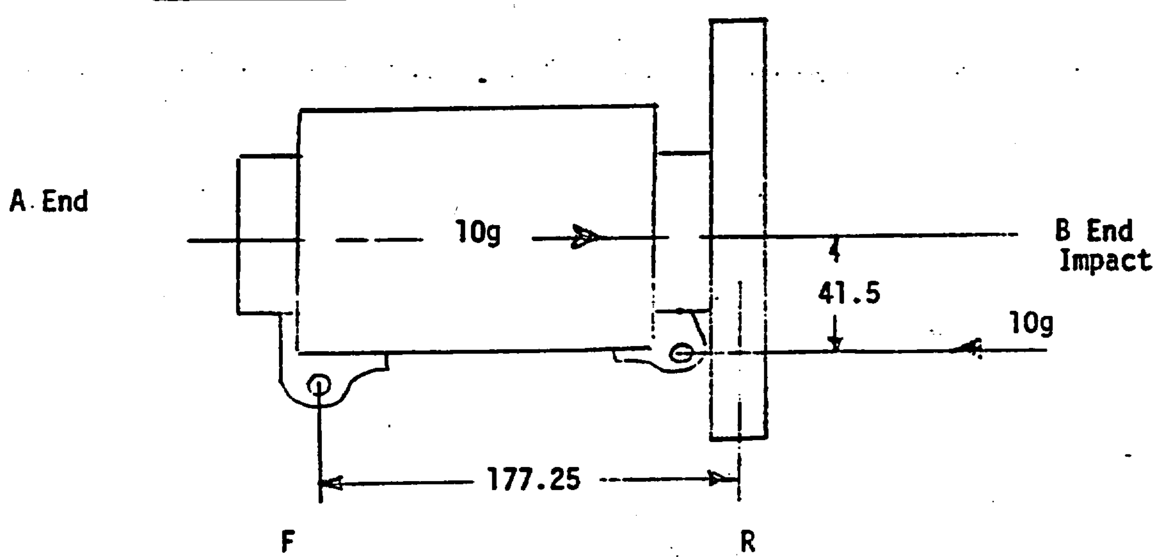


↓ 460,530 pitching
↓ 220,000 2g V
↓ 110,000 1g static wt.
↓ 790,530 net compression

A End Impact

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

B end Impact



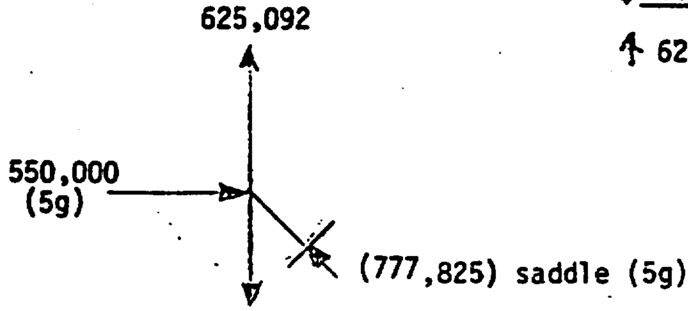
Pitching couple at F and R.

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

For front tie-down at F

↑ 515,092 pitching
↑ 220,000 2g V
↓ 110,000 1g Static

↑ 625,092 net tension



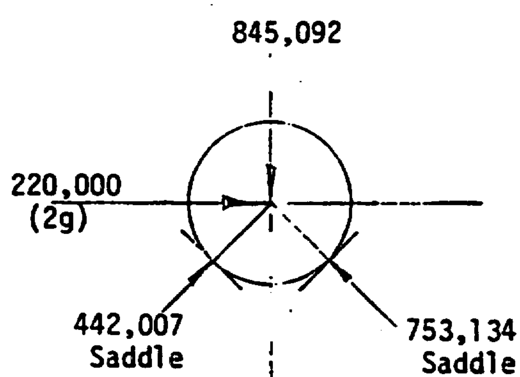
F = 1,175,092 Lug
Front Tie down Plane (F)

B End Impact

For Rear Saddle at R, with 2g transverse

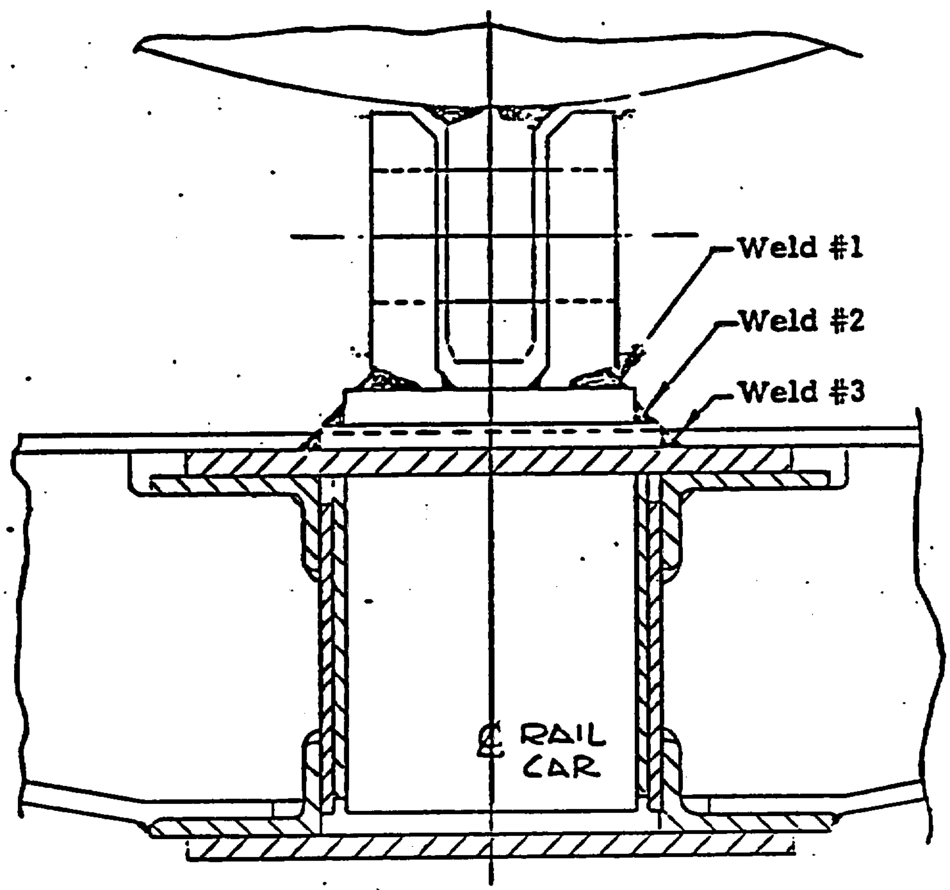
↓ 515,092 pitching
↓ 220,000 2g V
↓ 110,000 1g static wt.

↓ 845,092 net compression



Rear Saddle Plane (R)
B End Impact

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

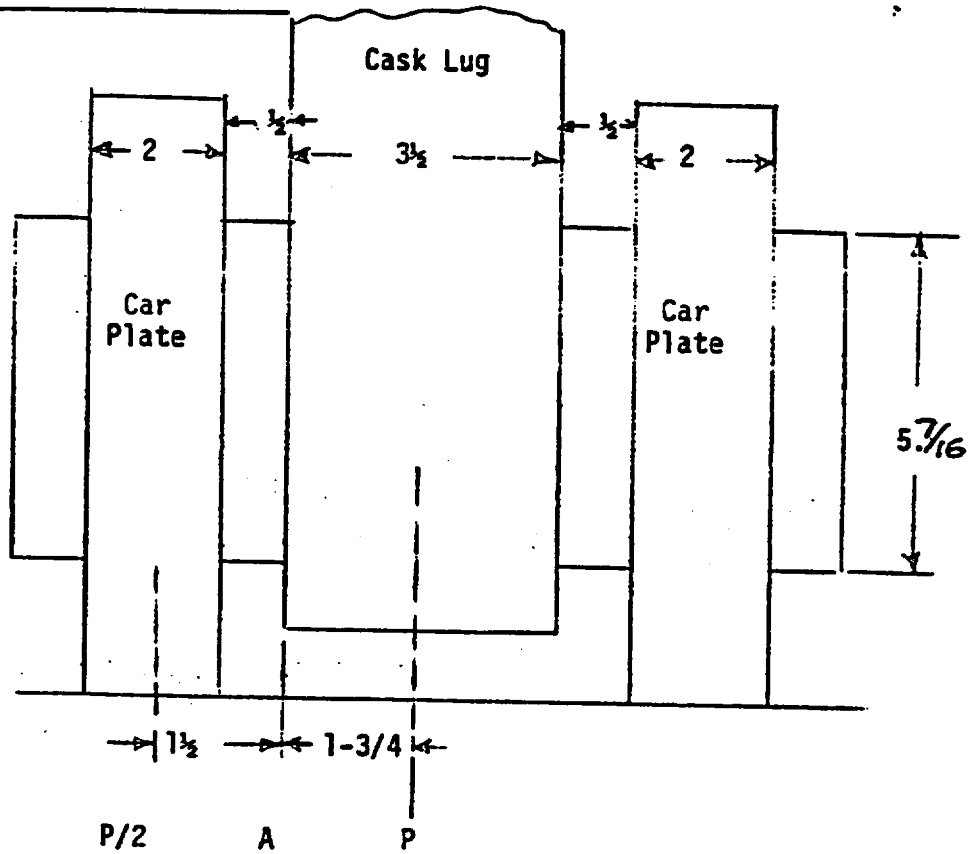


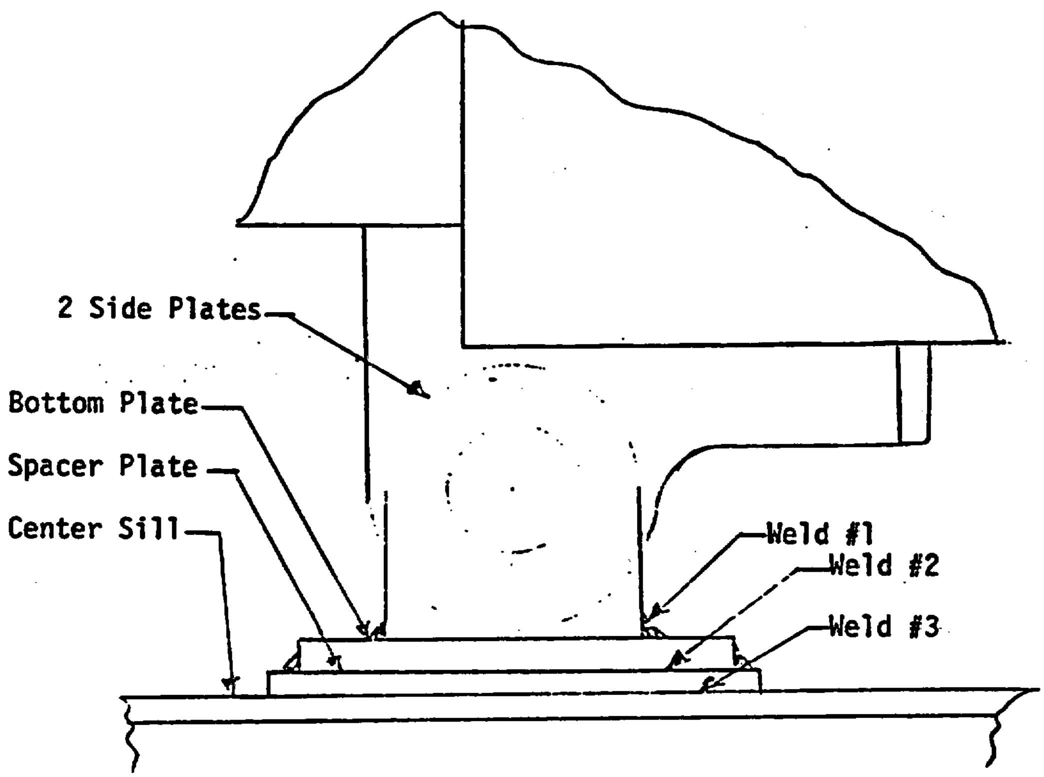
Front Tie Down

"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

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

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

Pin dia = 5-7/16 in.

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

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

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

Pin stresses at section A

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

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

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

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

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

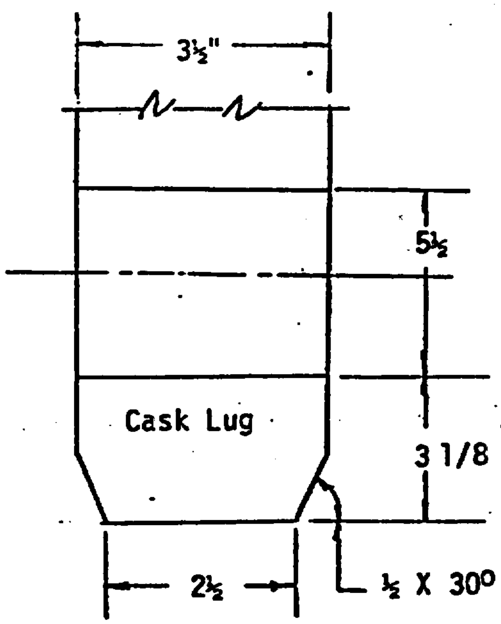
Pin stresses at center P

$M_B = P/2 \times 3\frac{1}{4} - P/2 \times 7/8 = P/2 \times 2.375 = 1{,}395{,}422$ in lbs.

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

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

2.2.1.3 Front Cask Lug Analysis



Section through C.L. of pin

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

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

Hoop stress across area - tension

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

$$\text{M.S.} = \frac{9 \times 1000,000}{55,933} - 1 = .609$$

Shear tearout at 40° from C.L.

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

$$S_s = \frac{1.175,092}{24.625} = 47,719 \text{ psi}$$

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

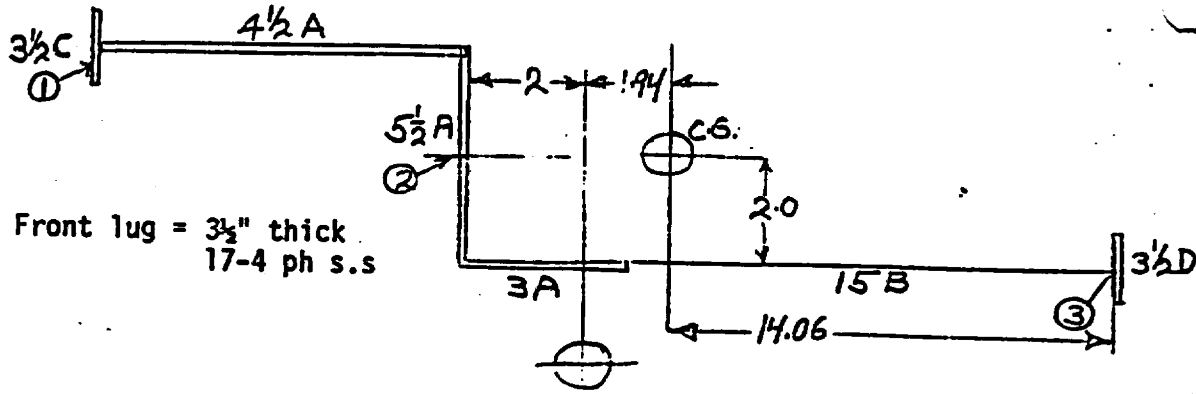
Tension area thru diameter

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

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

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

2.2.1.4 Front cask lug - Weld strength



Front lug = 3½" thick
17-4 ph s.s

Weld A = 1-1/4" J + 1" fillet on 2 sides of 3½ lug

Weld B = 1" fillet on 2 sides

Weld C = 1" fillet across end of lug on cask

Weld D = 2 x 3/4" fillets at end of lug, normal to cask

Throat areas of welds

A = 2 x 1½" = 3"

B = 2 x .707" = 1.414"

C = .707"

D = 2 x (3/4 x .707) = 1.06"

Weld material has substantially same physical as 304 S.S base material of cask, so values of latter are used.

(75,000 UTS and 30,000 YP)

Therefore, throat area of weld is critical, rather than interface areas of contact with cask or lug.

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

Weld	Area (A)	X	AX	Y	AY
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	C
3½D	3½ x 1.06 = 3.71	22½	83.47	0	0
	66.39	x 8.44	=560.25	2.00	133.21
		- 4.50			
		3.94			

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

$$+ \frac{2 \times 5.5(1\tfrac{1}{2})^3}{12} + 16.5\,(3.94)^2 + \frac{3 \times 3^3}{12} + 9.0\,(2.44)^2$$

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

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

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

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

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

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Eccentricity of vertical load = 1.94"

P_1 = 1,175,092 lbs (XI-2-16)

$T = P_1 \times ecc = 1{,}175{,}092 \times 1.94 = 2{,}279{,}678$ 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)

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

$S_s = 655 \times 14.2 = 9301$ psi $\uparrow$ compression

S_T = Uniform tension stress over all welds

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

Net stress = 17,700 - 9301 = 8399 psi tension

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

Stress at point (2)

$\rho = 3.94$

$S_s = 655 \times 3.94 = 2581$ psi $\downarrow$

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

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

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

Stress at point (1)

$$\rho = \sqrt{8.44^2 + 3.5^2} = 9.14"$$

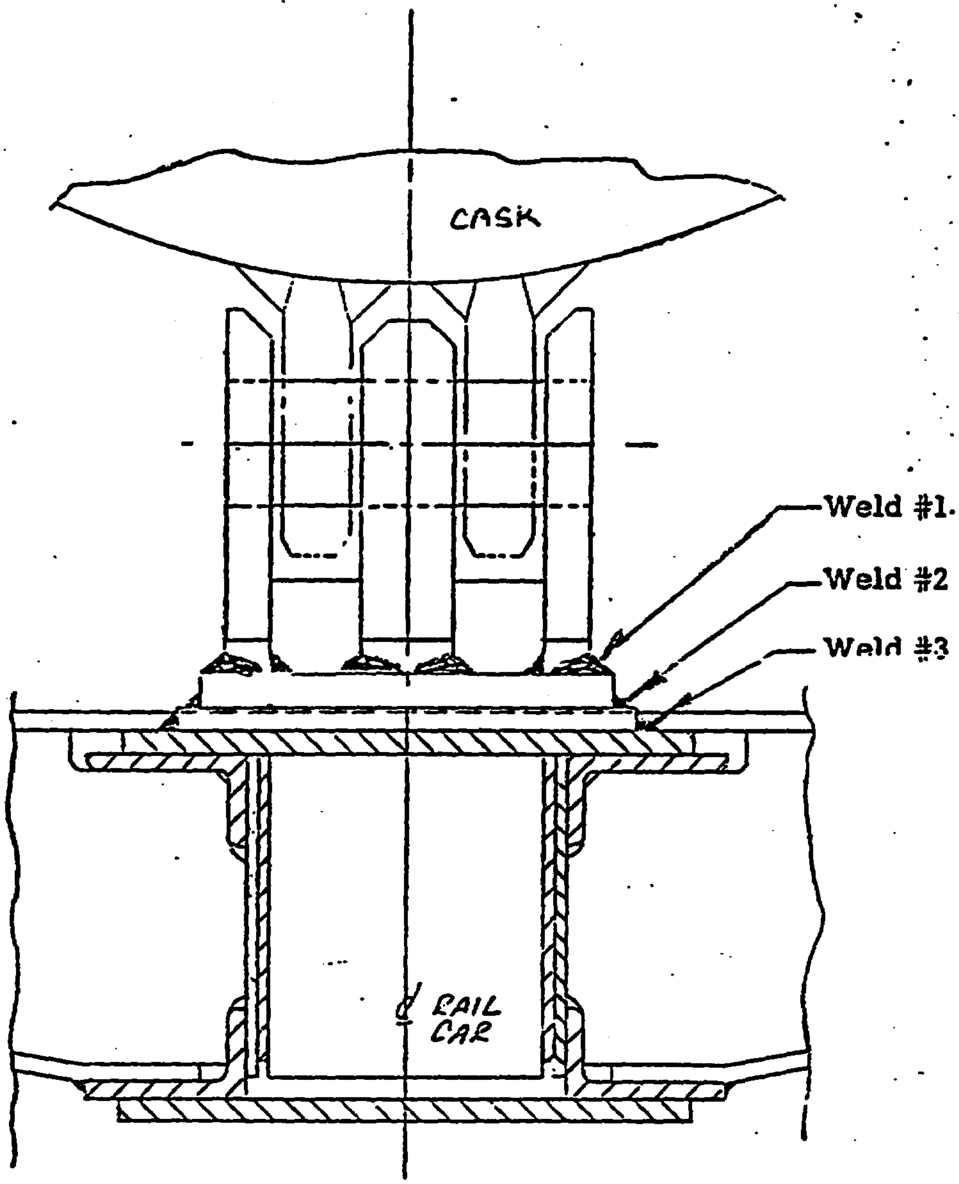
$S_s = 655 \times 9.14 = 5989$ psi $\downarrow$

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

Net stress = 17,700 + 5985 = 23,682 psi $\downarrow$

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

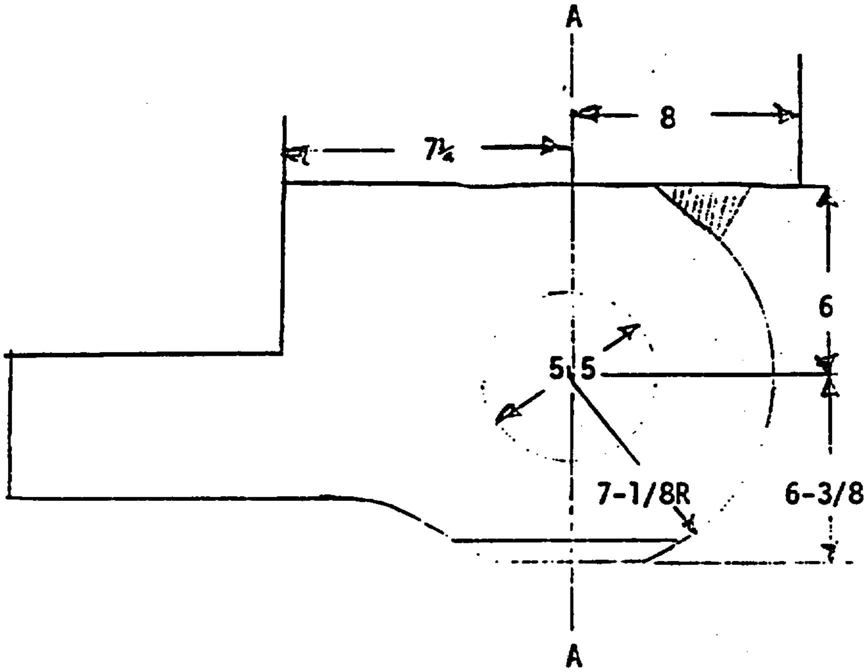
2.2.2 Tie down - bottom end of cask ("B" End)



Rear Tie Down

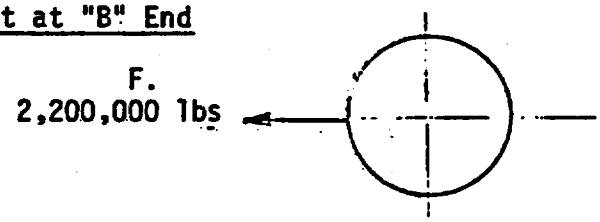
"B" End

Bottom of Cask

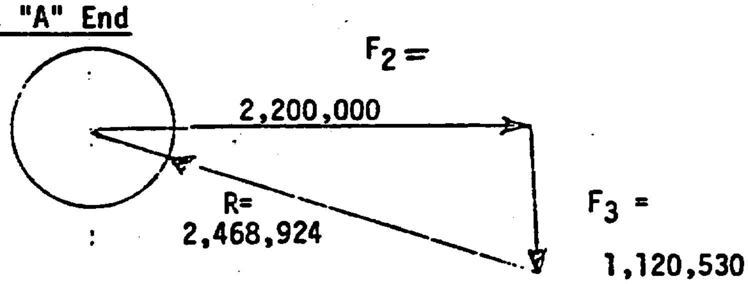


Rear Cask Lug (B End)

For impact at "B" End

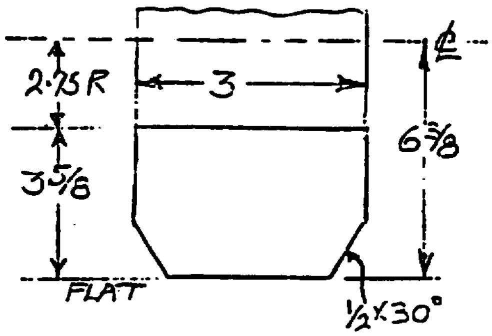


For impact at "A" End



2.2.2.1 Rear Cask Lugs - B - End

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



Min. area at vertical section =

(3 x 3-5/8) - ½ x .866 = 10.44 in.^{2}

Max tension load = 2,468,924 lbs

on both lugs (Impact at "A" End)

Hoop stress thru section A = $\dfrac{2,468,924}{(12.375-5.5)3x2}$ = 59,852 psi

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

Shear tear - out at 40° to load line

Area = 2 (3 x 4.66) = 27.96"^{2}

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

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

Bearing stress at P

Bearing area 2 lugs = 2 x 5.5 x 3 = 33in^2

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

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

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

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

$Z \approx .098\ (5.4375)^3 = 15.755\ in^3$

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

Load distribution

Fig. 1

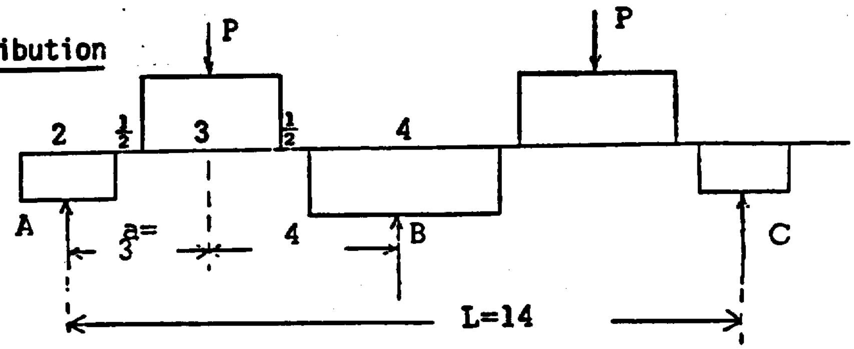


Fig. 2

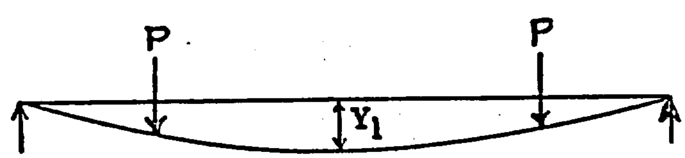


Fig. 3

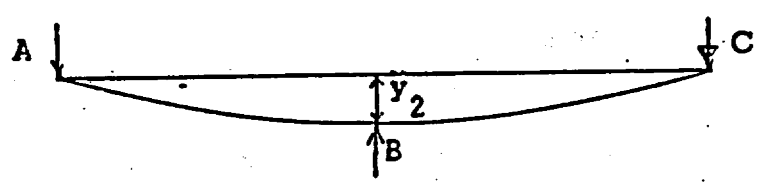


Fig. 2 Shows center deflection y_1, caused by two loads (P) on simply supported beam (support B removed)

Fig. 3 shows support force B required to return center point to save 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.

From Fig. 2 $Y_1 = \frac{Pa}{24EI}(3L^2 - 4a^2)$

$a = 3$ $L = 14$

$Y_1 = \frac{P \times 3}{24EI}(3 \times 14^2 - 4 \times 3^2) = \frac{69P}{EI}$

From fig. 3 $Y_2 = Y_1 = \frac{BL^3}{48EI} = \frac{B \times 14^3}{48EI} = \frac{57.1666B}{EI}$

Equating $69P = 57.1666B$

$B = 1.207P$

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

But max. $P = 1,234,462$ lbs

$B = 1,489,000$ lbs

$A = C = 489,464$ lbs

Moment at P $= 489,464 \times 3 - \frac{1,234,462}{2} \times .75 = 1,005,469$ in lbs

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

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

Shear at P $= 489,464 - \frac{1234464}{2} = -127,768$ lbs

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

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

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

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

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

2.2.2.3 <u>Welds of lugs to cask</u>

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.

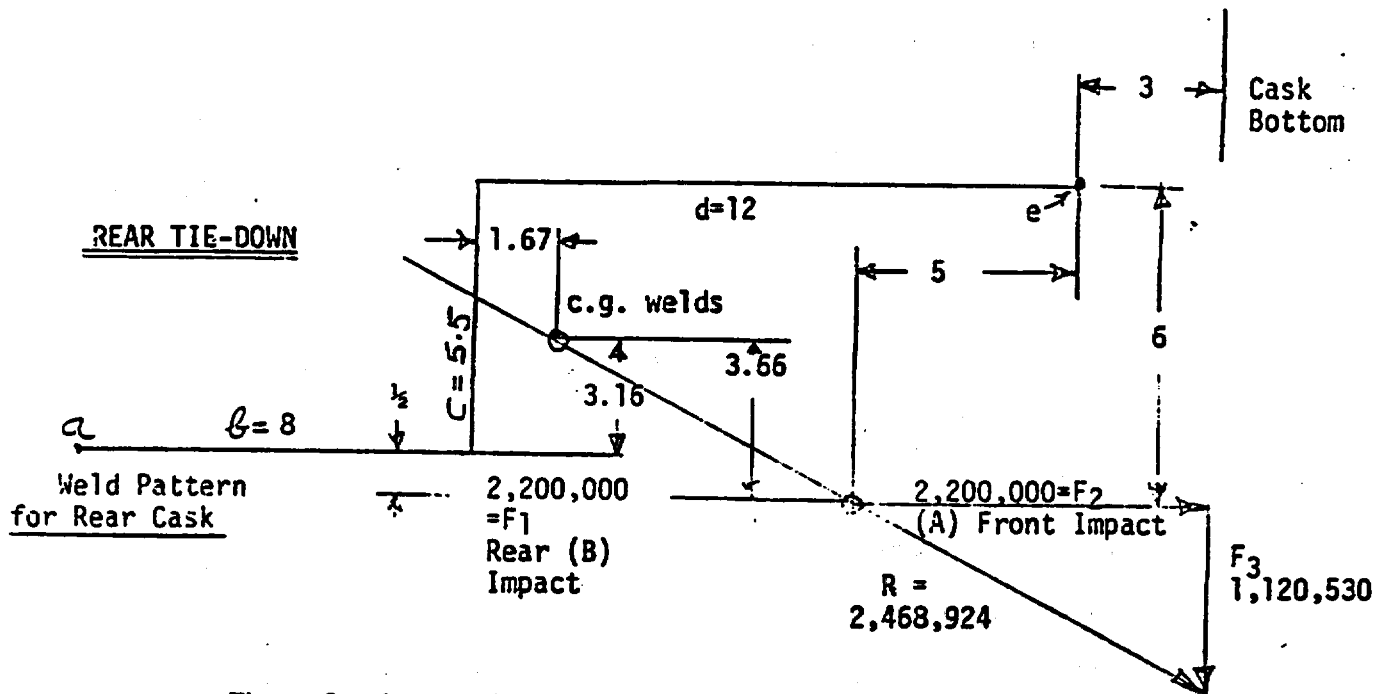
There are two conditions of applied loads acting through the center of the pin. When the force applied at the cask C.G. acts toward the B end of the railcar, the rear tie-down is loaded only by the 10g L force while the vertical forces are taken in compression on the rear saddle itself. When the force 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.)

<u>For an impact at (A) end</u>, the total resultant load of 2,468,924 lbs. acts almost through the c.g. of the welds, and so regligible 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

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



These loads are for 2 lugs 3" thick

Throat widths - single lug. (w)

a 1½" fillet w = 1.5 x .707 = 1.06 in.

b, c, d each 2 (1½ fillets + 1/2 penetration)

w = 2 x 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	3.75	20	75	5.5	20.625
		83.43	($\bar{9}$.67)	807	(3.16)	264

Polar moment of inertia I_p for each lug

$$= \sum \left(\frac{wl^3}{12} + wl\,x^2 + \frac{lw^3}{12} + wly^2\right) \quad \text{where } wl = A$$

$$= \sum A \left(\frac{w^2}{12} + \frac{l^2}{12} + x^2 + y^2\right)$$

$$I_p = 3.18\left(\frac{1.06^2}{12} + \frac{3^2}{12} + 9.67^2 + 3.16^2\right) = 24\left(\frac{3^2}{12} + \frac{8^2}{12} + 5.67^2 + 3.16^2\right)$$

$$+ 16.5\left(\frac{3^2}{12} + \frac{5.5^2}{12} + 1.67^2 + .41^2\right) + 36\left(\frac{3^2}{12} + \frac{12^2}{12} + 4.33^2 + 2.34^2\right)$$

$$+ 3.75\left(\frac{1.25^2}{12} + \frac{3^2}{12} + 10.33^2 + 2.34^2\right)$$

$$= 331.769 + 1157.228 + 102.76 + 1331.082 + 423.993$$

$$= 3346.858$$

For both lugs; $2I_p = 6693.7 \text{ in.}^4$

$2A = 166.86 \text{ in.}^2$

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.

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

Torque due to eccentricity is

T = 2,200,000 x 3.66 = 8,052,000 in. lbs.

$T/I_p = \frac{8,052,000}{6693.7}$ = 1202.9 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 in.

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

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

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

M.S. = $\frac{.9 \times 30,000}{25925}$ -1 = .04 conservative because compressive stress.

For midpoint of (c)

ρ = app. 1.67 $\uparrow S_s$ = 1.67 x 1202.9 = 2.009 psi (shear)

$S_c = \sqrt{13,185 + 3 \times 2009^2}$ = 13,636 psi (compression)

M.S. = $\frac{.9 \times 30,000}{13.636}$ -1 = .98

For point (e)

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

$\downarrow S_t$ = 10.59 x 1202.9 = 12,741 psi tension

$\leftarrow S$ = 13,185 psi shear

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

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

2.2.3 Railcar tie-downs and saddles   Accident analysis

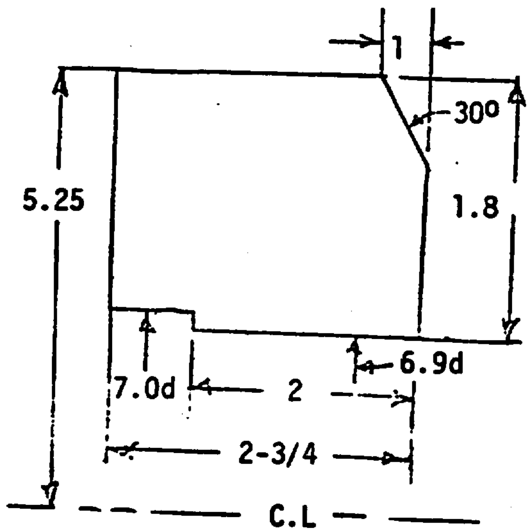
This analysis deals with those parts of the tie-down system which are attached to the railcar rather than the cask body itself. The same applied loadings are used as in the previous analyses of the cask members, namely 10g longitudinal, 5g lateral and 2g vertical. The difference in this analysis is that, 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 analy To prove that in an accident, the failure would be thru the railcar attachment welds, a final summary of M.S. valves is made, comparing these railcar members with the cask ratings - for the accident condition only.

Material properties	- 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.

$$\text{Area} = (1.8 \times 2.75) - (3/4 \times 05) - \left(\frac{1.732}{2}\right) = 4.047$$

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

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

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

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

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

Tension on diameter

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

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

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

Bearing $S_{BR} = \frac{587,546}{6.9 \times 2} = 42,576$ 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 ↑ impact at B end

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

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

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 = $2 \times 1 \times .707 \times 20 = 28.28\text{in}^2$ - throat of weld

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

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

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

Area = $2 \times 1 \times 20 = 40\text{in.}^2$

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

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

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)

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

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

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

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

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

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

Moment arm = 6-1/8 + 1½ = 7-5/8 in. height.

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

$$M = 7.625F$$

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

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

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

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

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

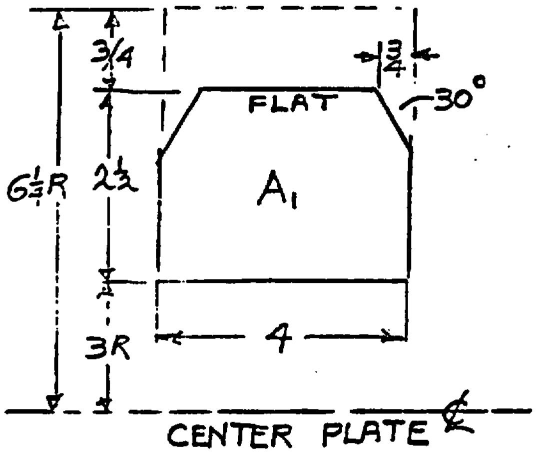
Longitudinal force to break away weld #2

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

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

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

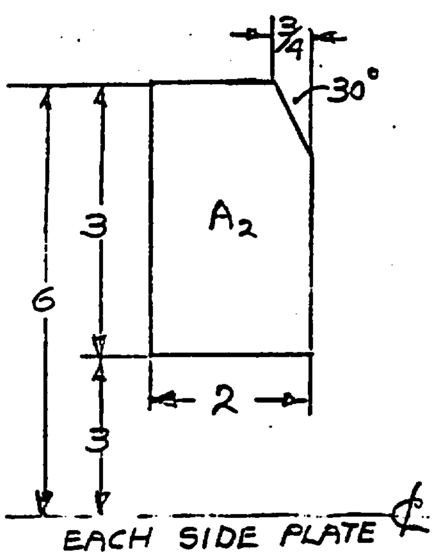
$M.S. = \dfrac{115,000}{82943} - 1 = .394$

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

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

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

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



Load on each side plate = 489,464 lbs.

Hoop stress at top - flatted

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

$S_t = \dfrac{489.464}{2 \times 5.5125} = 44,396$ psi

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

Shear tear out at 40° to 45° load line

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

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

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

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 railcar

Weld 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 = 255in^2

(2 x 42½ + 16) of 1" fillet = 101 x .70 throat = 71.4 in^2

Total - weld #1 = 326.4 in^2

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 = 85.55in^2

Weld 3 Spacer plates to railcar center sill (2)

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

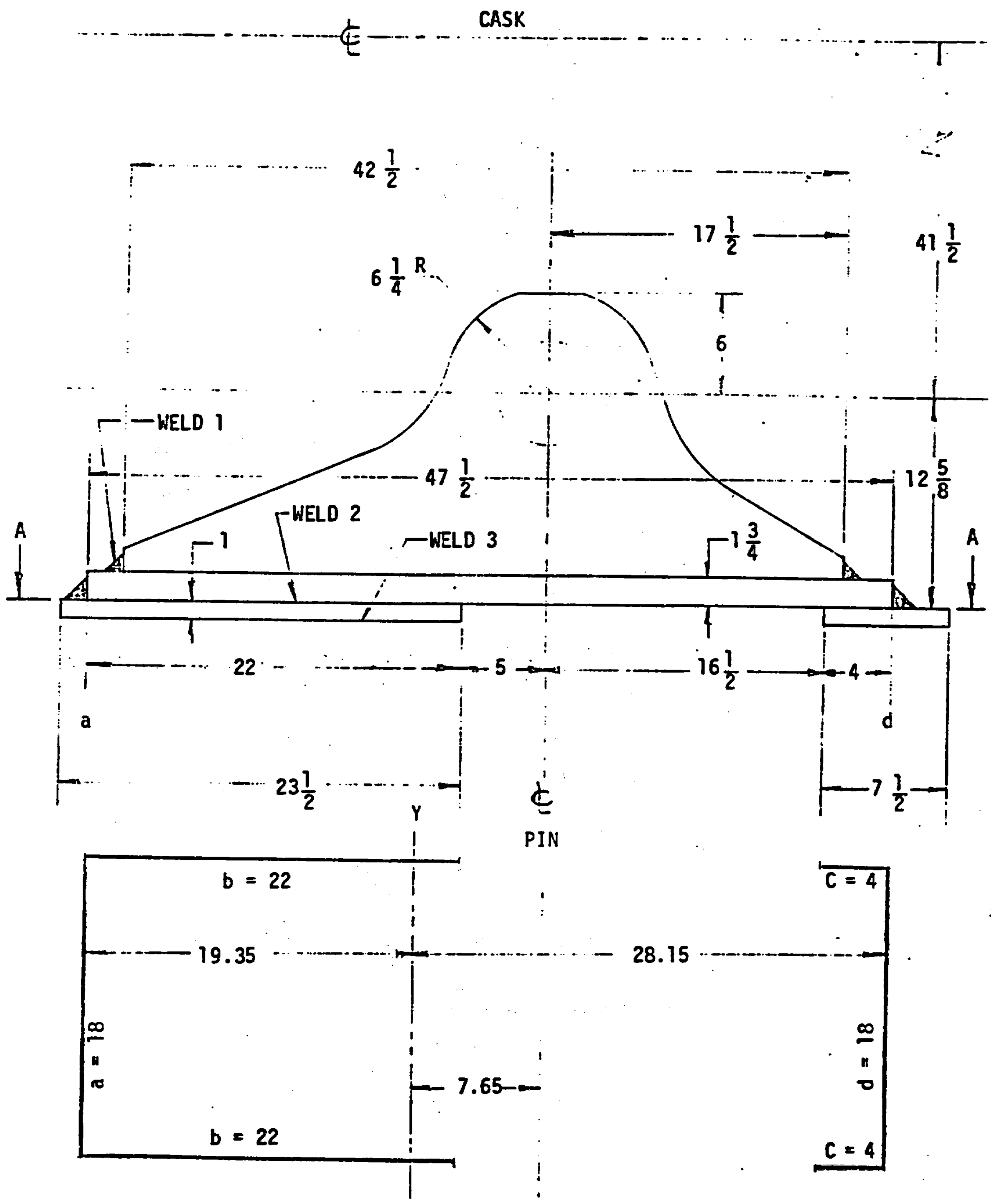
Weld 75,000 psi UTS.

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

142" x 1¼" throat = 177.5in^2

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 - <u>Weld 2</u>

a	18	x	0	=	0
2b	2x22	x	11	=	484.
2c	2x4	x	45.5	=	364
d	18	x	47.5	=	855
	88	x	(19.35)	=	1703 in^2

$$Iyy = 18 \times 19.35^2 + 2 \times \frac{22^3}{12} + 2 \times 22 \times 8.35^2 + \frac{2x4^3}{12} + 2x4x26.15^2 + 18x28.15$$

$= 31327.in^3$ for line welds

Weld throat = 1-3/8" x .707 = .972in

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

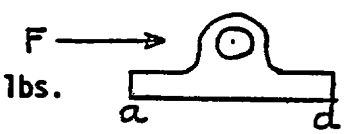
$Z_a = \frac{31327}{19.35} = 1619\ in^2$ for line welds x .972" = 1574 in^3

Area=19.35 in^2

weld length = 88", weld throat area = 85.54 in^2

<u>For impact at rear (B) end</u> - Loads on tie-down at rear - <u>weld 2</u>

F_1 = 2,200,000 lbs.



M = 2,200,000 x 12 5/8 = 27,775,000 in lbs.

This puts tension at <u>a</u> & compression at <u>d.</u>

There is no vertical load in this case.

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

at <u>a</u> max tension $S_{\ell} = \frac{M}{1574} = 17646$ psi

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

$= 47914$ psi

$M.S. = \frac{75,000}{47,914} - 1 = .565$ at <u>a</u>

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

F_1 = 2,200,000 lbs. ←

V_1 = 1,120,530 lbs. ↑

R = 2,468,924 ↖

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

This puts tension at D and compression at A

Stress at D (Sac. Weld #2)

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

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

4 Total tension = 33,338 + 13,099 = 46,437 psi

Shear stress K = 25,719 psi

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

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

The weld is between T-1 plates.

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

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

2.2.3.5 <u>Saddle bearing stresses</u> - Impact limiters on support structure

The compression loads on a 45° 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 <u>impact</u> structure is 28 x 1.5 = <u>42 in²</u>

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

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

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

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 consistant 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

Impact at	Member	Stress	M.S. (.9 y.p.)
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)

Impact at	Member	Stress	M.S. (U.T.S)
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 develope only when impact is at the B end.

Satisfactory M.S. valves 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. valves 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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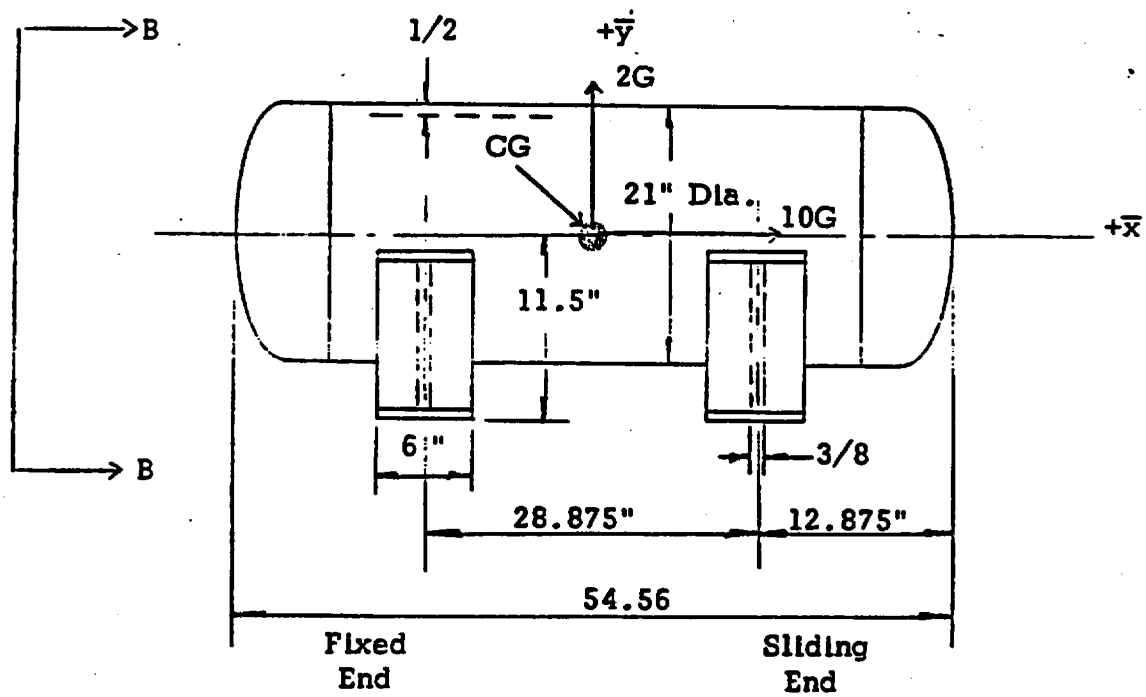
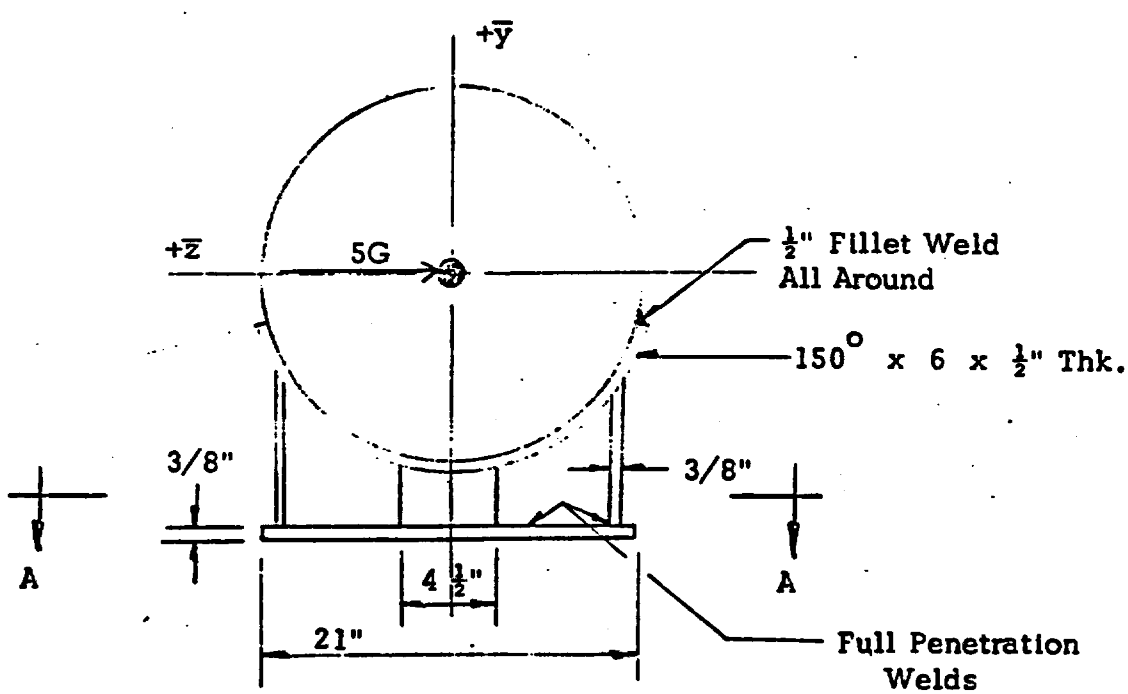
2.2.4 Expansion Tank Tie-Down



All Materials are 304 S.S.



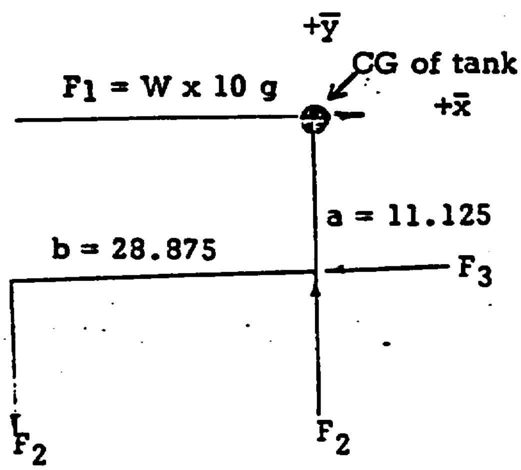
VIEW B-B

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



Support Reactions

$W = 1100$ lbs.

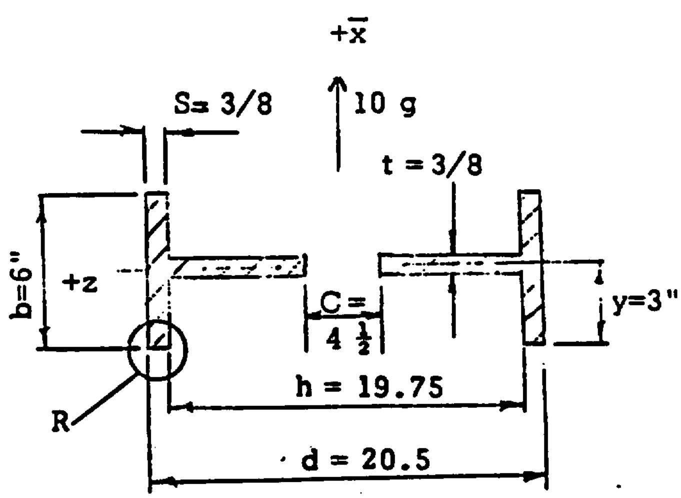
$F_1 = 1100 \times 10$

$= 11000$ lbs.

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

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

$F_2 = 4238$ lbs.



Section "A-A"

Area $A = \left[b(d-c)\right] - \left[h-c\ (b-t)\right]$

$= \left[6(20.5-4.5)\right] - \left[19.75-4.5\ (6-.375)\right]$

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

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

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

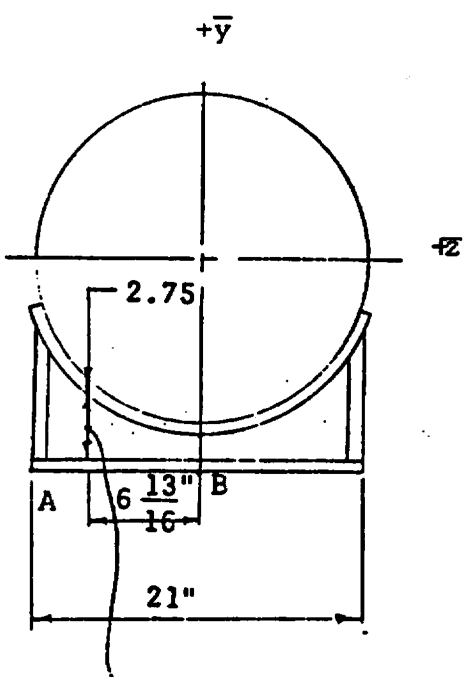
$$= \frac{11000}{10.22}$$

$$= 1076 \text{ psi}$$

$F_1 = 11000$ 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

$$M = 1100\ (10\ g)\ (2.75)$$

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

Section modulus for Section A-A

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

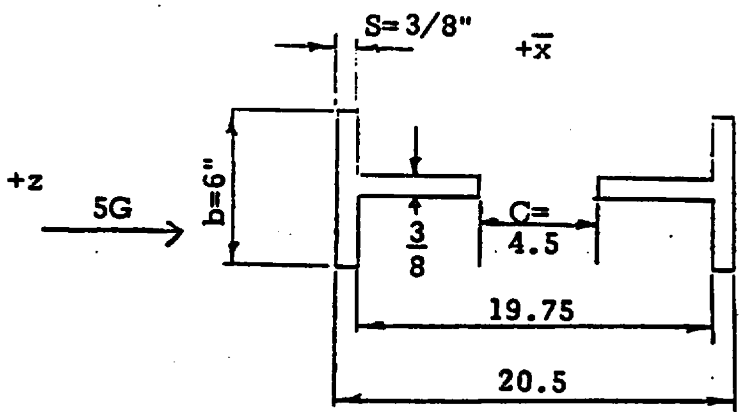
$$= \frac{2(.375)(6^3) + (19.75-4.5)(.375^3)}{6 \times 6}$$

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

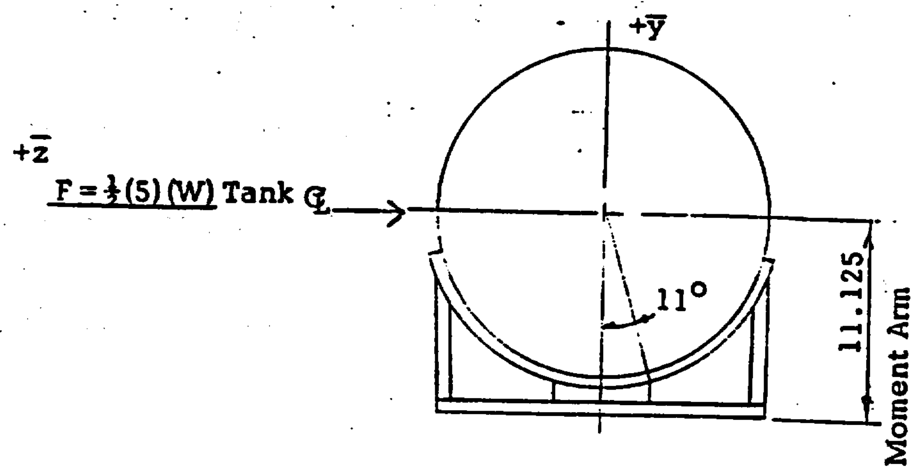
Bending stress $S_b = \frac{M}{Z}$

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

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



Section "A-A"



$F = \frac{1}{2}(5)(1100)$

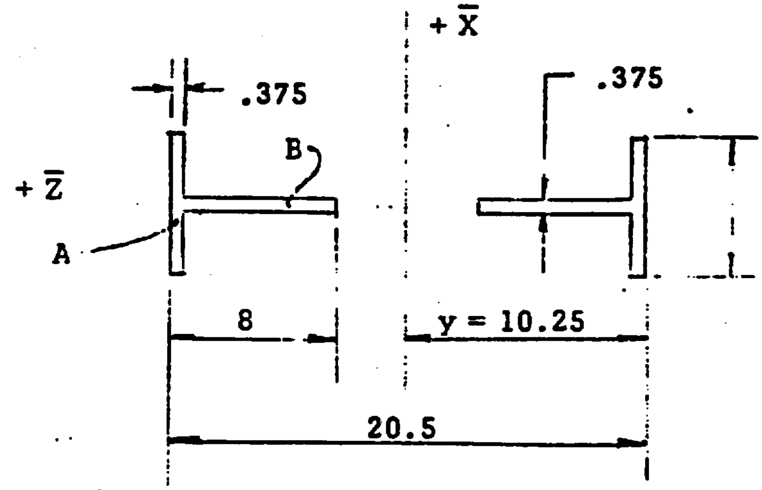
$= 2750$ lbs.

(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 in^4	d, in.	Ad^2, 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"

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

$$Z = 67.66$$

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

$$S_s = \frac{F}{A} \qquad F = 4238 \text{ lbs.}$$

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

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

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)

$$St = \frac{F}{A} \qquad F = \frac{1100}{2}$$

$$= \frac{550}{10.22} = 54 \text{ psi} \qquad = 550 \text{ lbs.}$$

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

Calculate effective stress S_{e2} through Section "A-A" at point "R" (See page 23,24)

$\sigma_x = 0$ psi

$\sigma_y = 415 + 6692 + 452 + 54 = 7613$ psi

$\sigma_z = 0$ psi

$T_{xy} = 1076$ psi, $T_{yz} = 415$ psi, $T_{zx} = 0$ 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 x 41660 =37500 psi

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

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

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

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

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

$= 0.01544$

$t = .5$ in.

$K = 0.01544$

$P = 7380.37$ lbs.

$R_{10g} = 4238$ lbs.

$R_{5g} = 1492.37$ lbs.

$R_{3g} = 1650$ lbs.

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

$= 1650$ lbs.

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

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

$R_{5g} = 1492.37$ lbs.

$P = 1650 + 1492.37 + 4238$

$= 7380.37$ lbs.

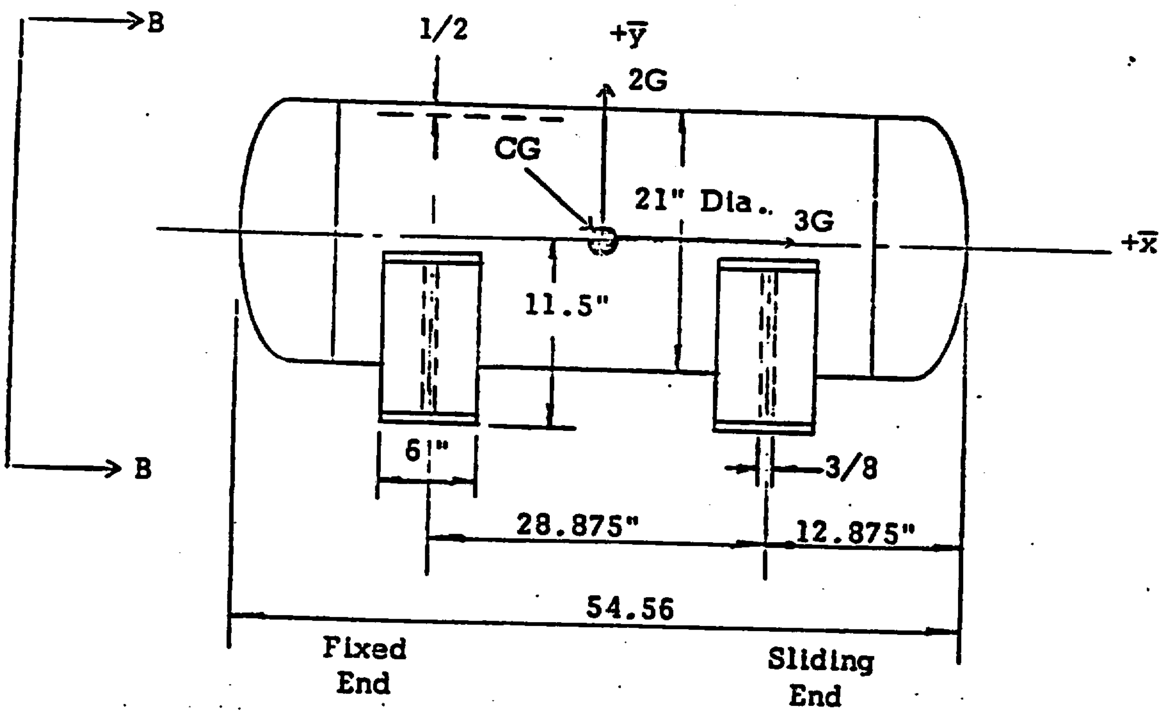
$$S\ max = 0.01544 \left(\frac{7380.37}{.5^2} \right) \log_e \frac{10.5}{.5} = 603 \text{ psi.}$$

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 $0.9 \times 21500 = 19350$ psi.

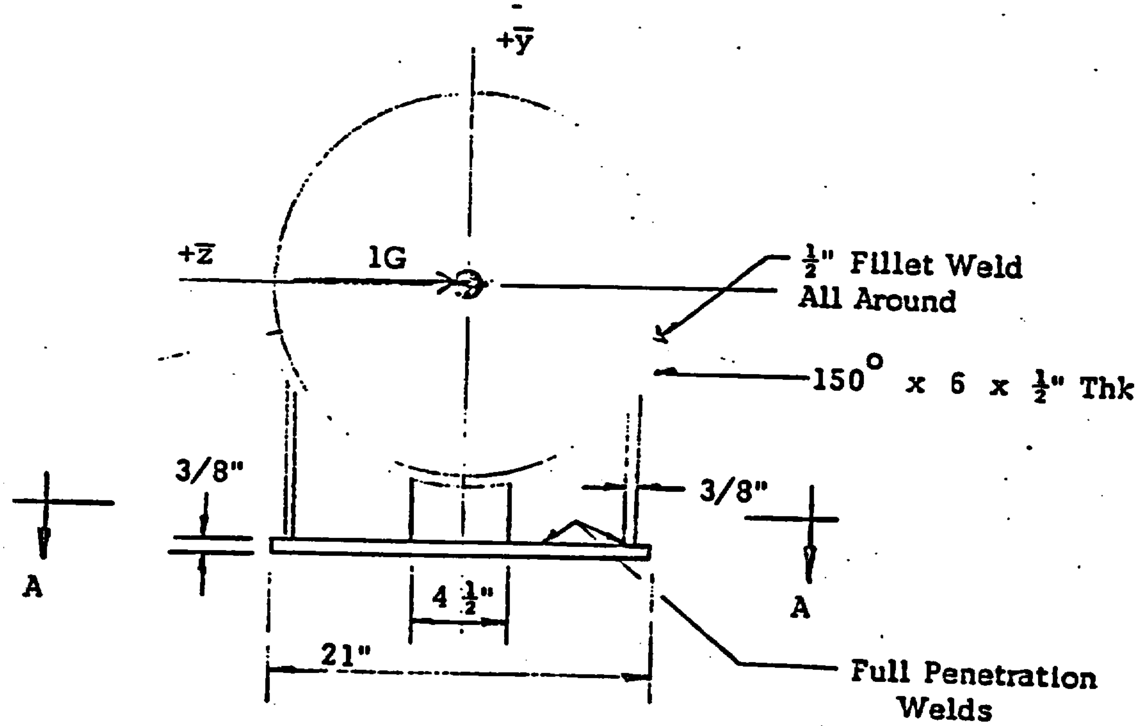
$$M.S. = \frac{19350}{603} - 1 = 31.08$$

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

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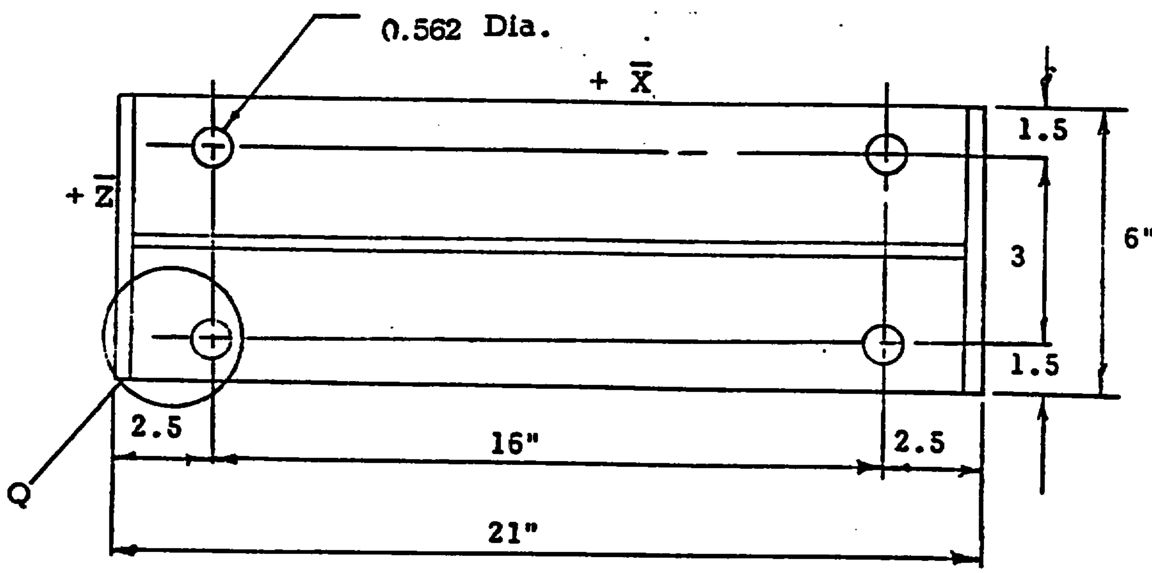


Fixed End Tie-down Base Plate (Sect. A-A) pg XI-2-29a

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 in.^{2}

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

<u>For 1g lateral loadings</u>

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

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

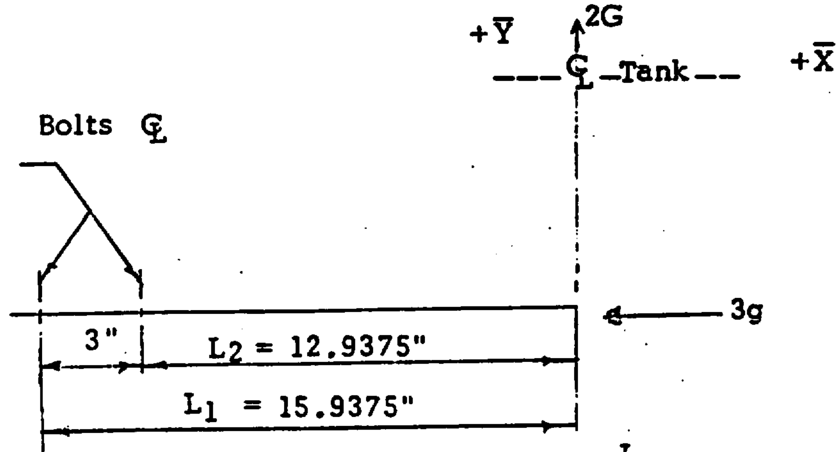
$$8\,F_2 = 1100\ (1)$$

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

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

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

For 3 g Longitudinal Loading



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

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

$F_3 = 533$ lbs.

$4\,F_4 = 1100\,(3)$

$F_4 = 825$ 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

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

$8\,F_5 = 1100\,(1)$

$F_5 = 137.5$ lbs.

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

$\sigma_x = 0$ psi

$\sigma_y = 1395 + 3756 + 969 = 6120$ psi

$\sigma_z = 0$ psi

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

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

$= 13048$ psi

Allowable stress ($S_{as} = .75\ S_{ys}$) under normal condition from Sect. 1.1 and Sect. 1.2 equals 0.75 (105000) = 78750 psi; $S_{e1} = 13048 < S_{All} = 78750$ 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.

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

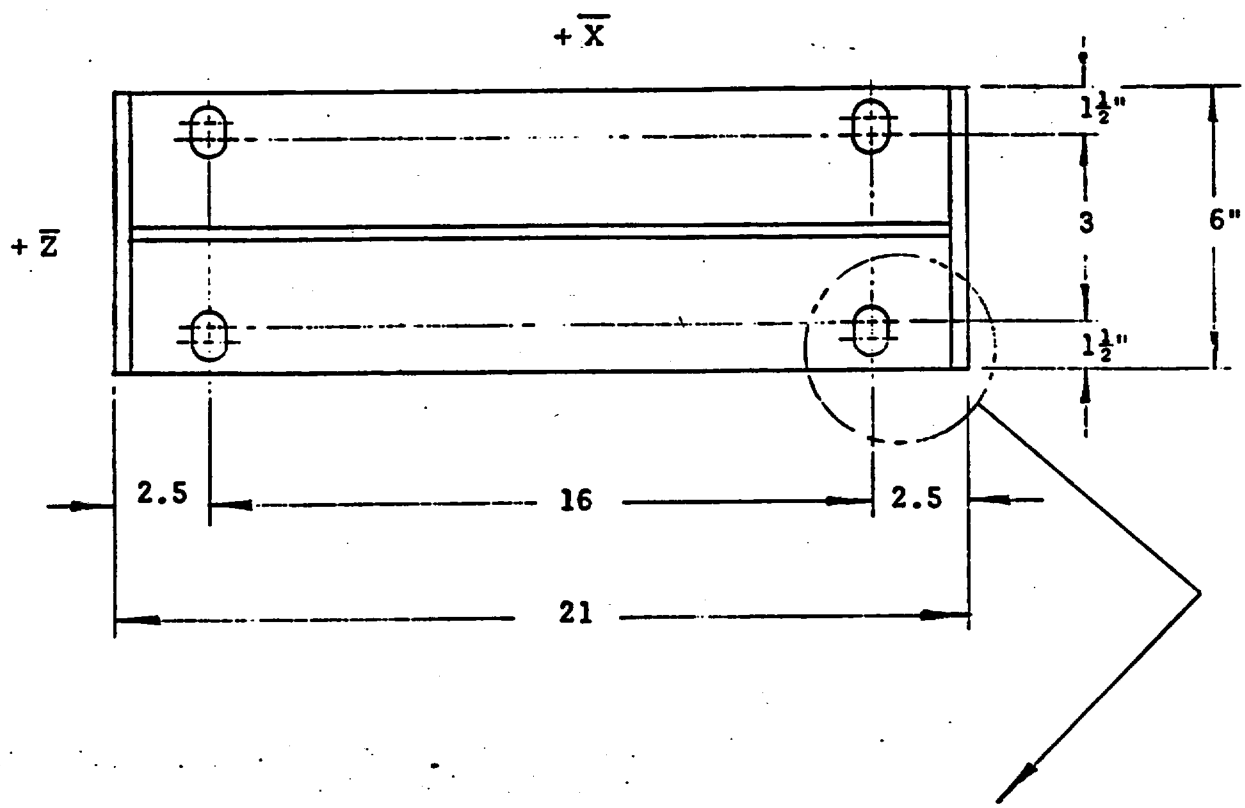
Calculating new effective stress on the bolts.

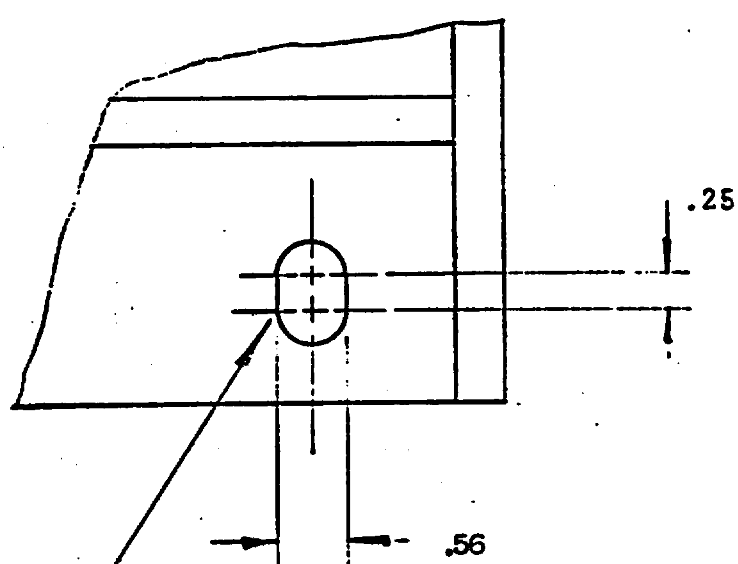
$$S_e = \sqrt{\frac{1}{2}}\ \sqrt{(0-0)^2 + (0-8457)^2 + (8457-0)^2 + 6(1094^2 + 6563^2)}$$

$S_e = 14294$ psi.

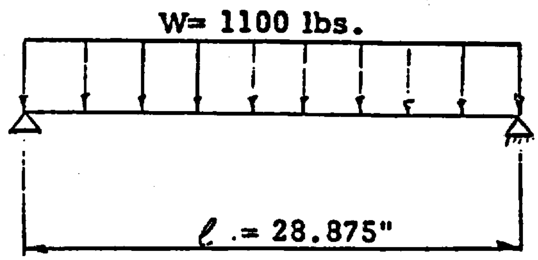
M.S. = (78250/14294) -1 = 4.47

Sliding End Base Plate





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

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

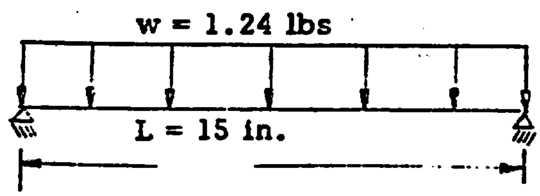
$$S_b = \frac{12546}{161} = 78 \text{ psi.}$$

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 x 41667 = 37500 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 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 ½" pipe sch. 40

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

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

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:

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

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

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

$$= 1/8\ (1.24)(3.16)(15)$$

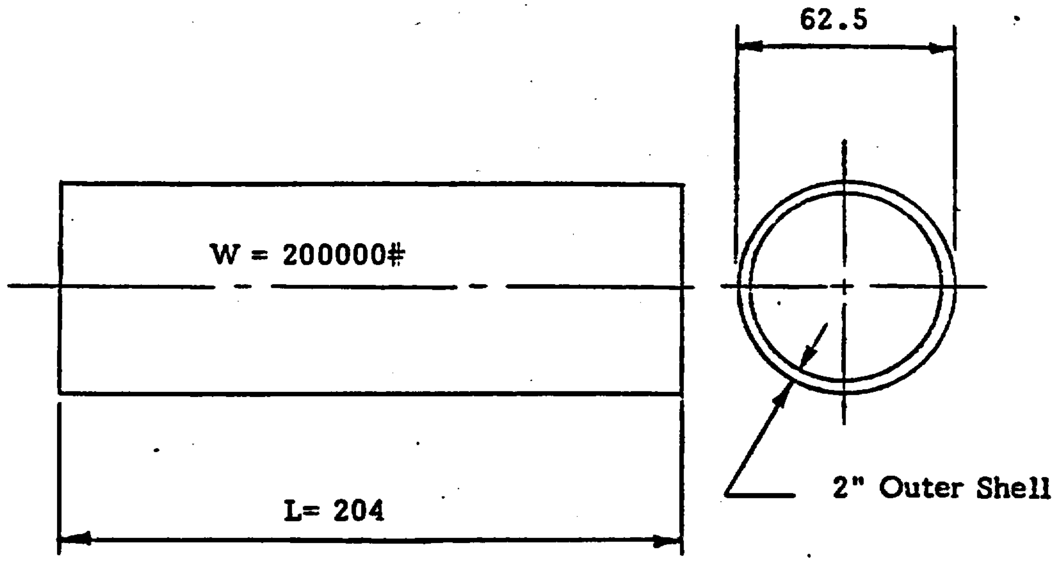
$$= 7.347 \text{ in.lbs.}$$

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

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

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:

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

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

Maximum bending stress in outer shell is:

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

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

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

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

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) <u>Inner Shell @ 25 psig</u>

D_o = 46.5 in. outside dia.

t = .75 in. thk. of shell

L = 180 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 psi @ 541° 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) <u>Top and Bottom Heads @ 25 psig</u>

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 where 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 affects. 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 (7) 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.

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.

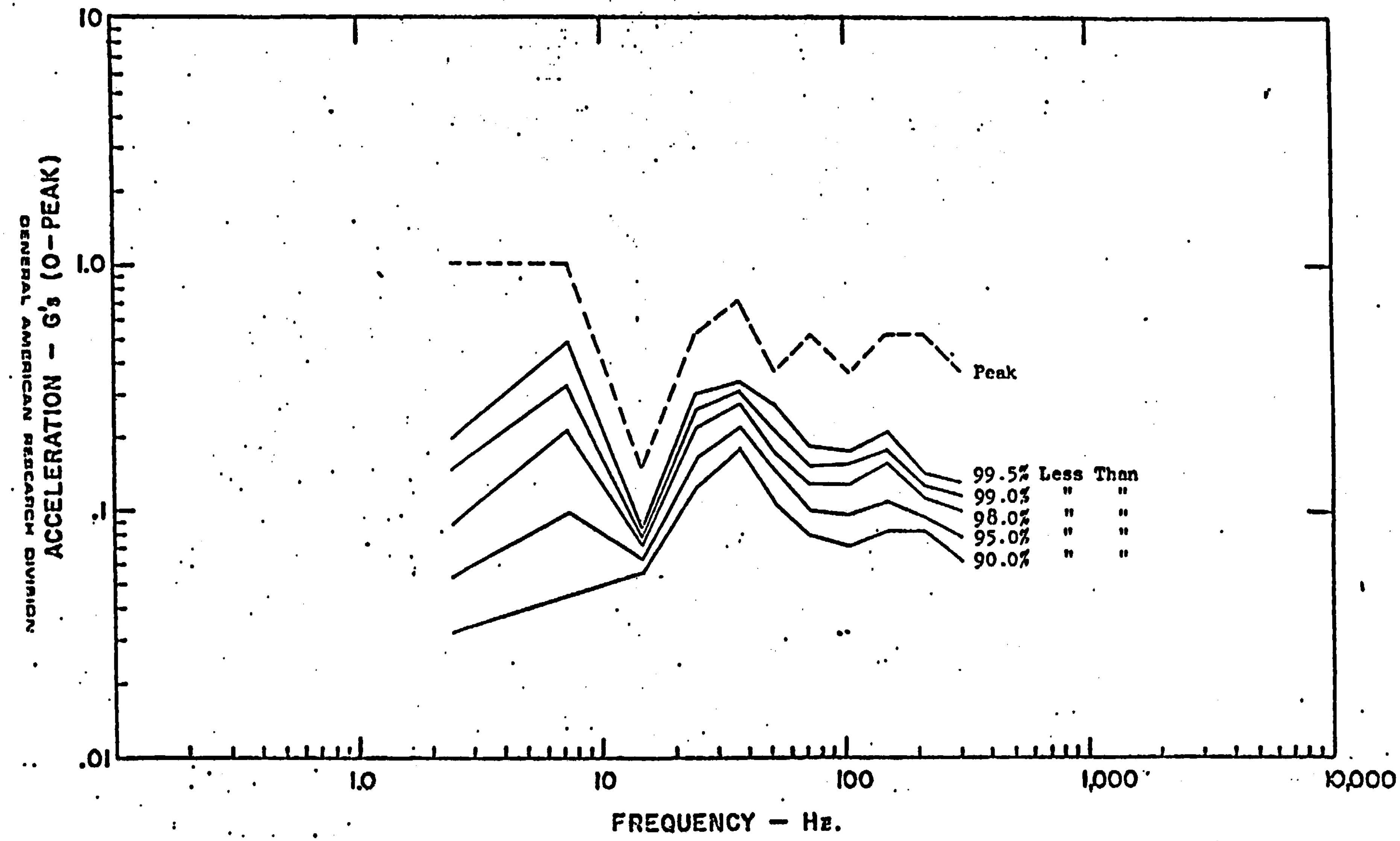


Figure 18 . FREQUENCY SPECTRA, RAILROAD, VERTICAL DIRECTION, COMPOSITE OF VARIOUS CONDITIONS

GENERAL AMERICAN RESEARCH DIVISION

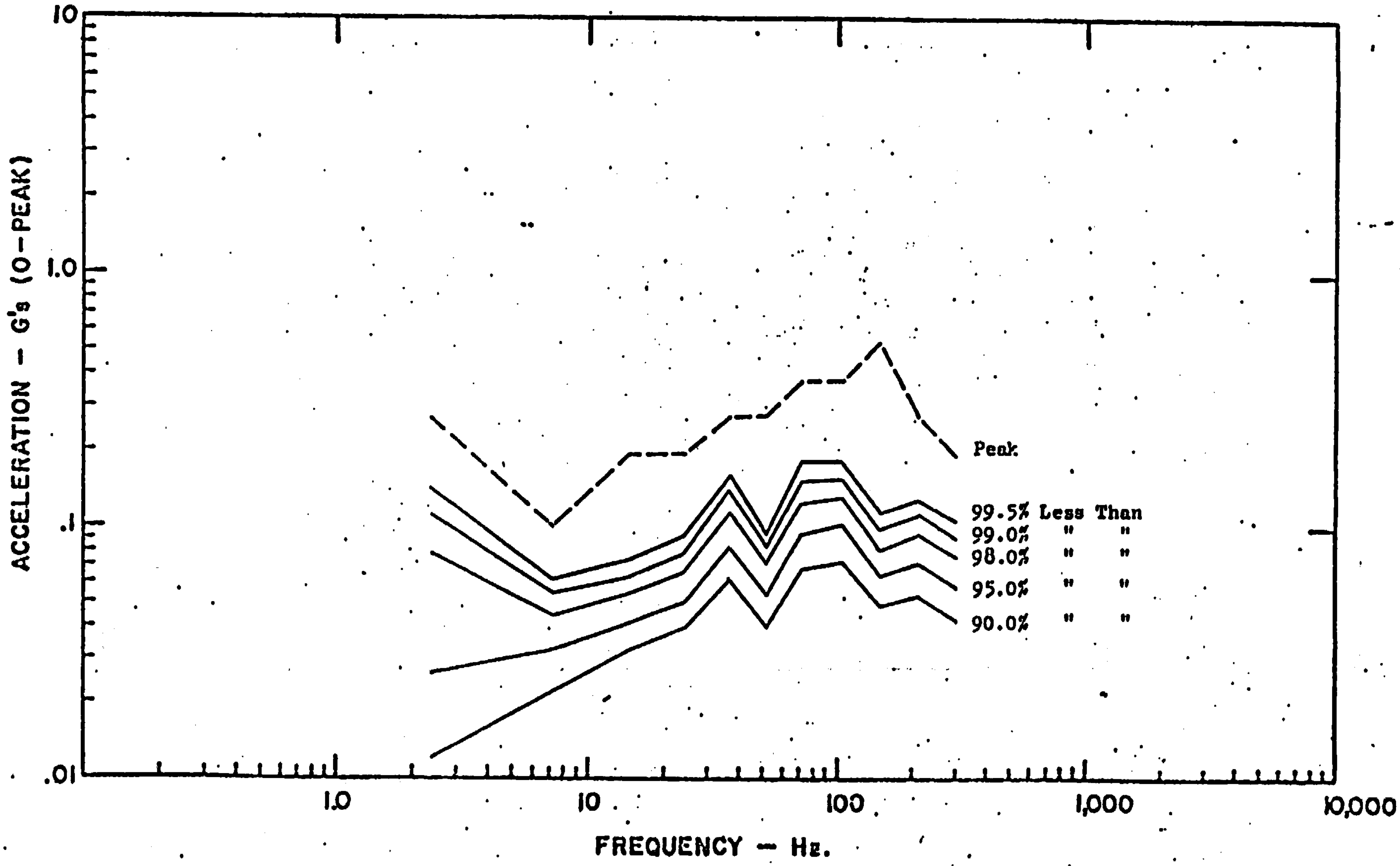


Figure 19 FREQUENCY SPECTRA, RAILROAD, TR(ERSE DIRECTION, COMPOSITE OF VARIOUS CONDITIONS

P83-2-38

1/31/75

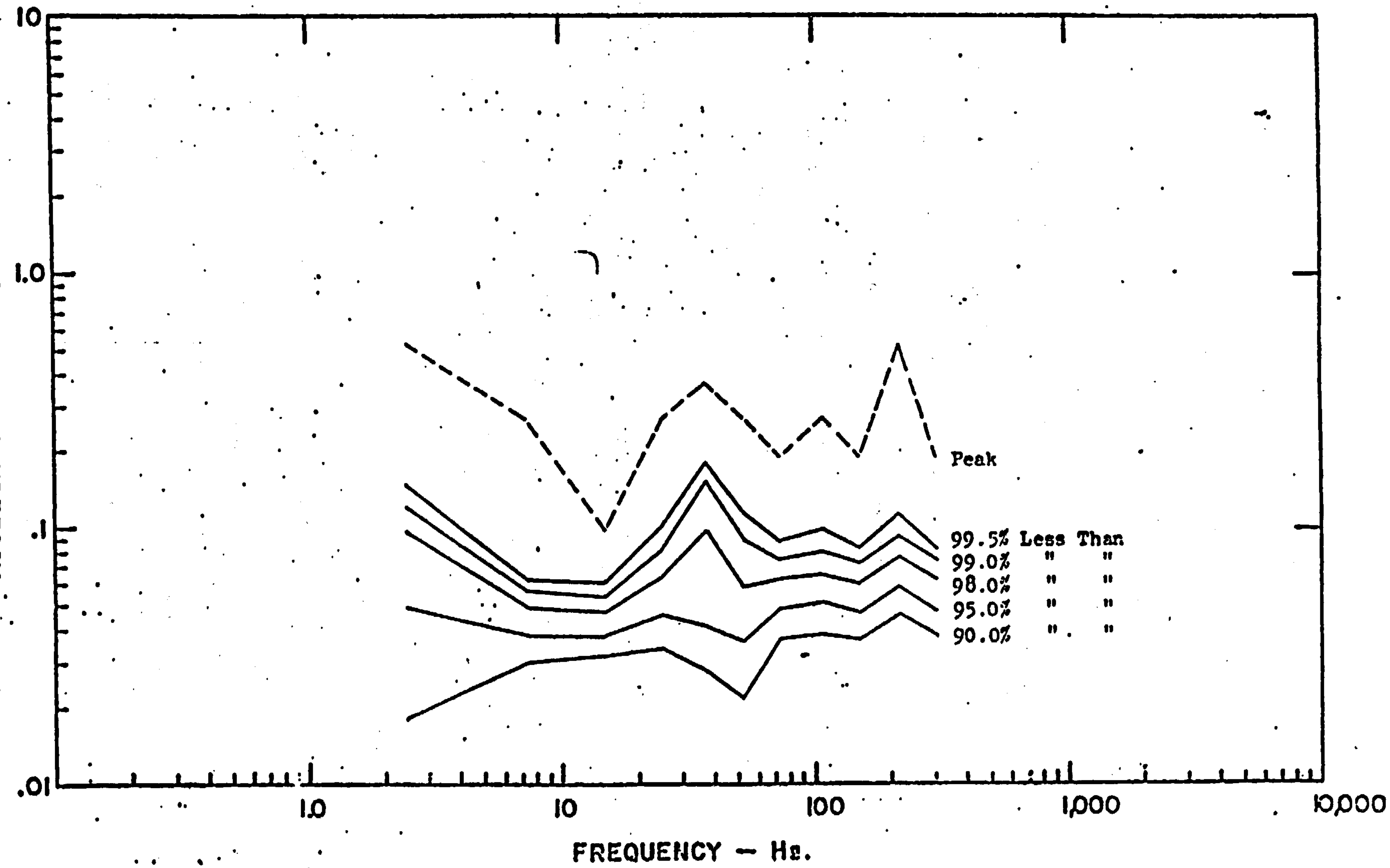


Figure 20 FREQUENCY SPECTRA, RAILROAD, LONGITUDINAL DIRECTION, COMPOSITE OF VARIOUS CONDITIONS

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.

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

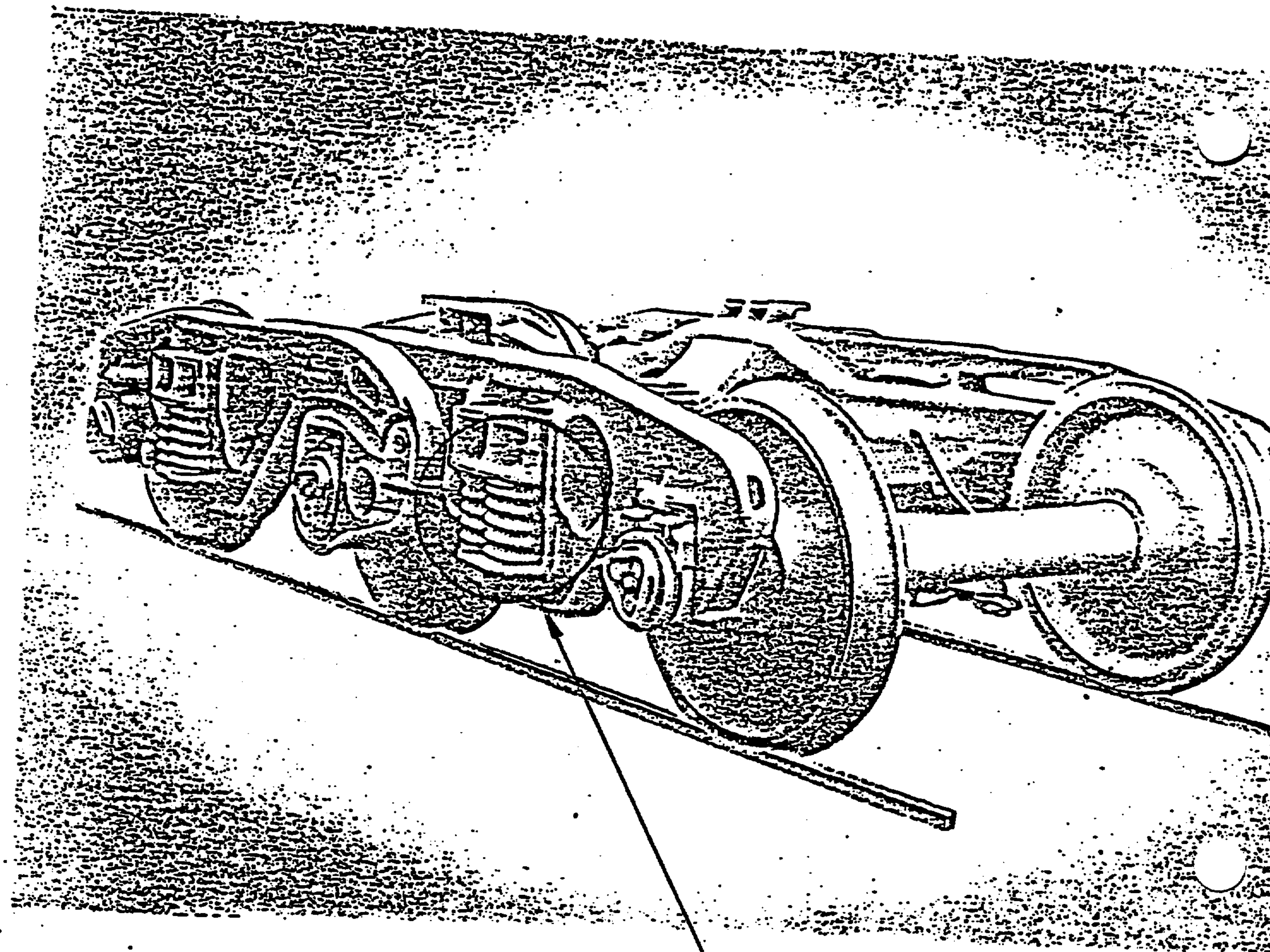
$$3124 \times \tan 55^{o} = 4461 \text{ lbs.}$$

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

$$C_{eq} = \frac{4F}{\pi w X} \qquad \text{where} \quad w = 2\pi \times 2.62 \text{ at resonance}$$

$$= 16.462 \text{ rad/sec}$$

X = amplitude of oscillation

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

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

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

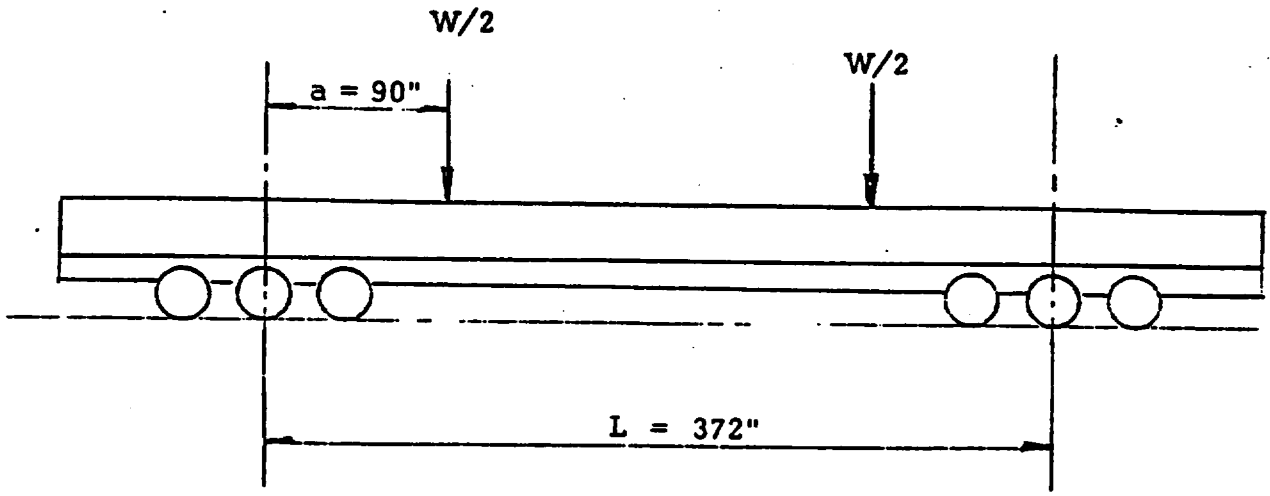
$$\%\text{ critical damping} = \frac{C_{eq}}{C_c} = \frac{690}{26953X} = \frac{1}{39.06X}$$

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

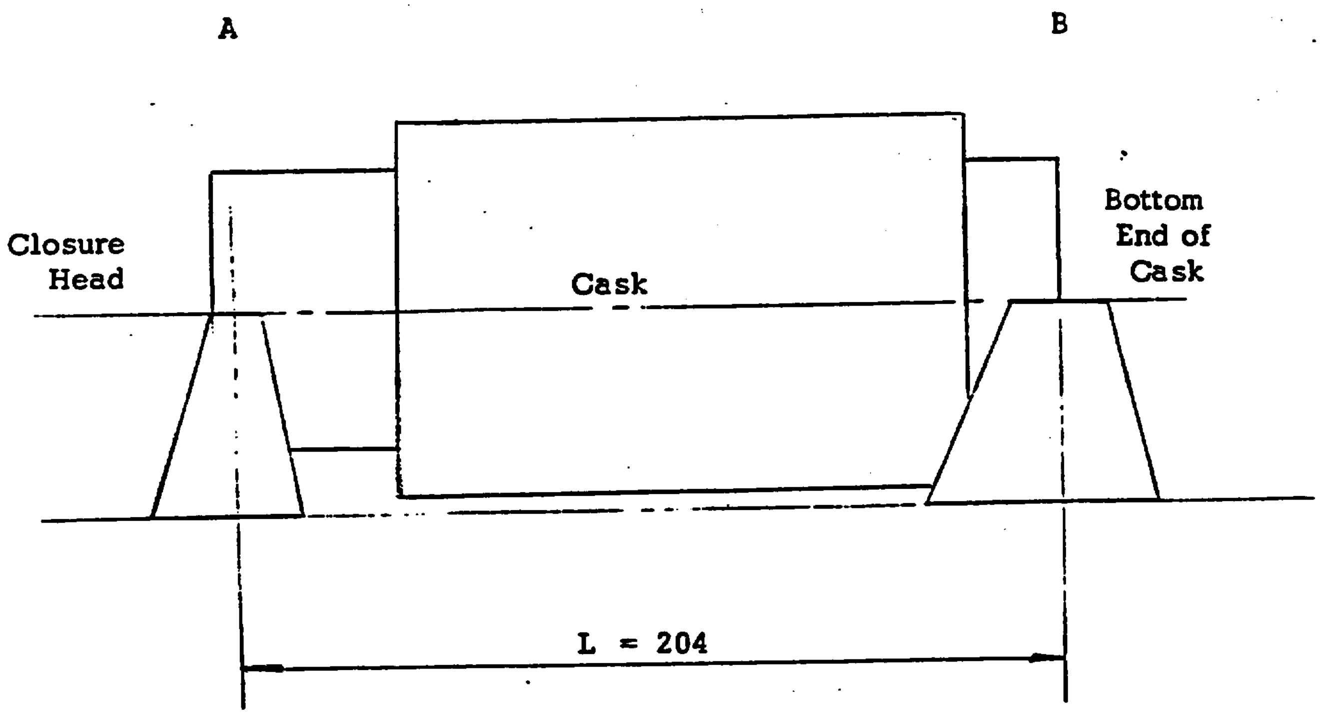


$$f_n = \frac{3.13}{\sqrt{\dfrac{(W + 0.486\ wl)\ (a^2)\ (3L - 4a)}{12\ E\ I}}}$$

$I_{3\ sills}$	=	$10359\ in^4$ Ref.-
w_{frame}	=	90.694 lbs/in.
wl	=	33738 lbs
.486wl	=	16400 lbs
a^2	=	$8100\ in^2$
3L - 4a	=	3 x 372 - 4 x 90 = 756 in.
12 E I	=	$12\ (30 \times 10^6)\ 10359 = 3729240 \times 10^6$
W	=	220000 lbs.
W + .486 wl	=	236400 lbs.

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

Natural Frequency of Simply Supported Cask



$$f_n = \frac{9.87}{2}\sqrt{\frac{E\,I\,g}{WL^3}} \quad \text{c.p.s.}$$

EI Inner Shell	$.74906 \times 10^{12}$
Lead	$.6481996 \times 10^{12}$
Outer Shell	4.6753×10^{12}
Water Jacket	4.2581×10^{12}
	10.33×10^{12}

$g = 386.4$ in/sec^2

$w = 220000$ lbs.

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

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

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

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

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

Equivalent viscous damping is

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

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

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

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

$$= 36,203$$

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

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

$X = .178$ 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 4 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 $\frac{w}{wn}$, 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 $\frac{w}{wn}$ 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

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

$T_f = 1.4$ x force applied at bolsters

$= 1.4 \times .72g = 1.008$ 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):

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

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 $\omega \to 0$ and $\omega \to \infty$, whereas the acceleration response factor approaches 0 as $\omega \to 0$ and approaches unity as $\omega \to \infty$.

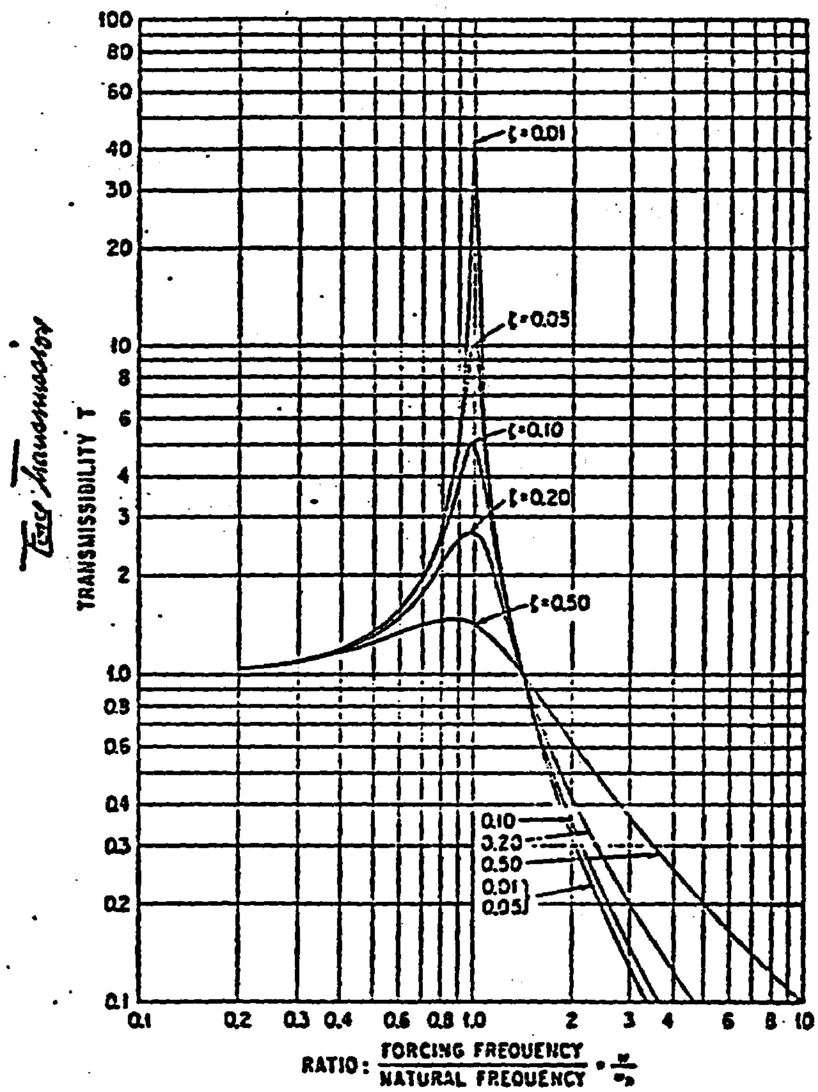


FIG. 2.17. Transmissibility of a viscous-damped system. Force transmissibility and motion transmissibility are identical numerically. The fraction of critical damping is denoted by ζ.

T_f = 1.0 for any value of C/C_c of the internal friction of cask.

= 1.0 x 1.008 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.

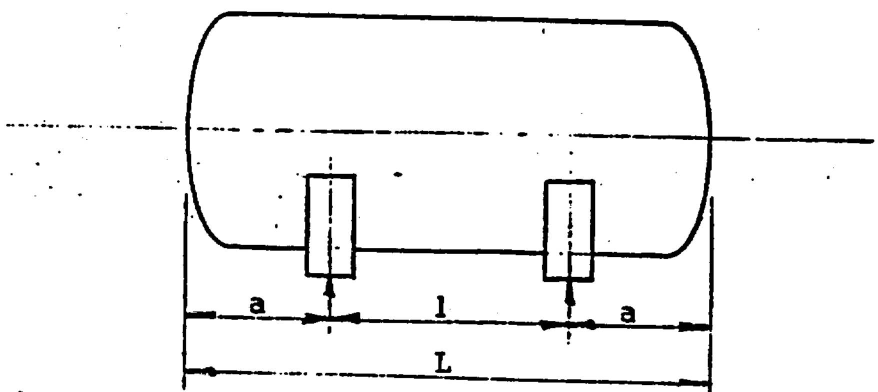
Vibration of Expansion Tanks

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.

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

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

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

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

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

<u>For 500 lbs. Empty Tank - Larger</u>

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

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 1 g, as indicated for the cask itself.

<u>Vibration Analysis of Expansion Tank Connecting Lines</u>

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

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

$E = 26.9 \times 10^6$ psi, modulus of elasticity. Sect. 1.2 for 304 S/S

moment of inertia for ½" sch. 40 pipe equals to

$$I = .049\ (D^4 - d^4)$$

$$= .049\ (.840^4 - .622^4)$$

$$= .01706 \text{ in.}^4$$

$L = 15$ in. length of pipe.

u = Mass per unit length of pipe, $\text{lb-sec}^2/\text{in.}^2$

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.

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

Converting Rad/sec to cycles/sec $= \frac{2031}{2\pi} = 323$ 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 Se_2 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

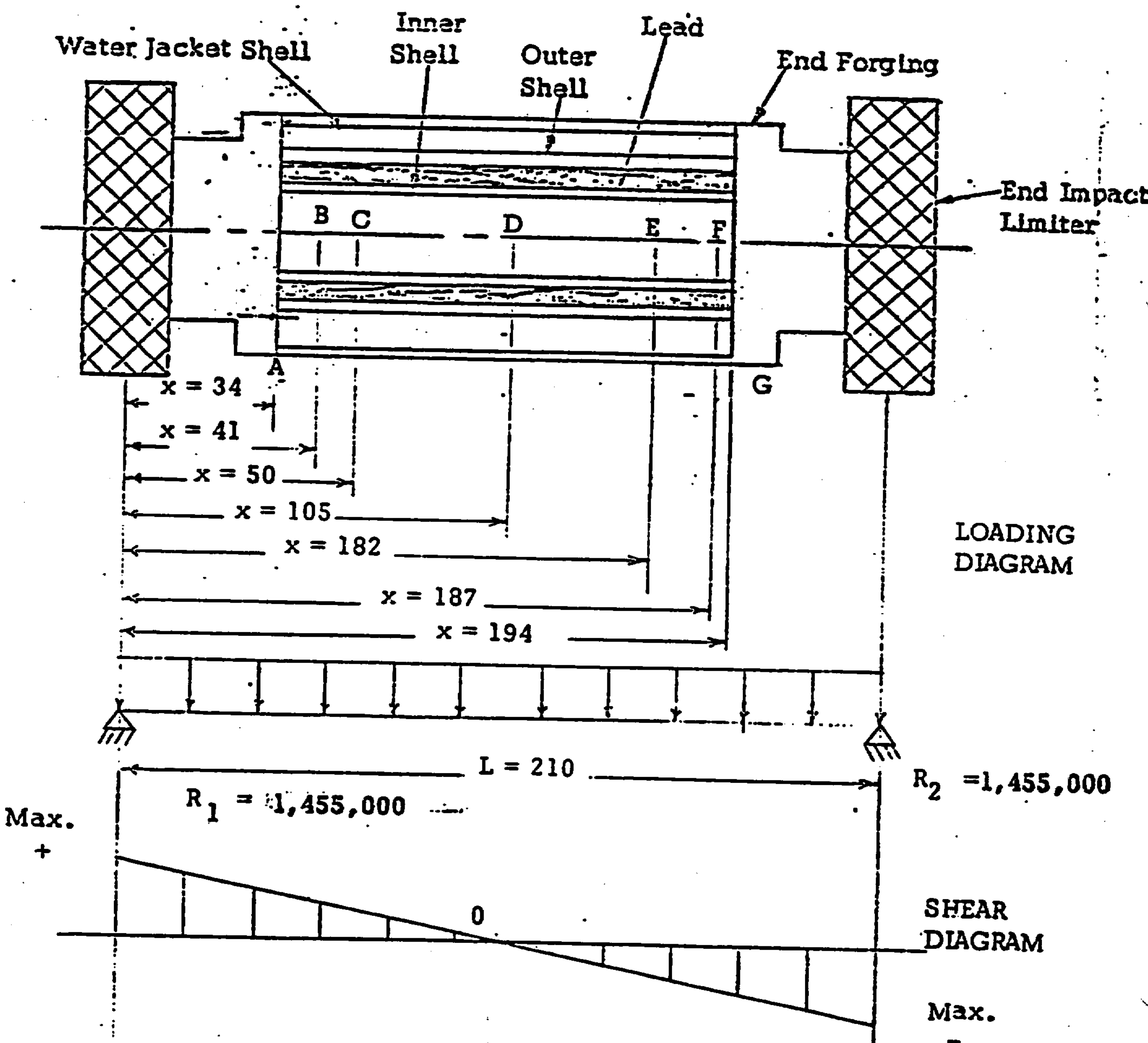
The following table compares the calculated effective stress S_{e2} to the stress allowable $S_{ad} = 0.9\ S_{yd}$ for the cold condition case (-40^{o}F isothermal), one foot side drop. The effective stresses for the 1G case were calculated in Sect. 3.8.6.

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

$$\underset{\text{(Sect. 4.6.2)}}{17787} \times \frac{14.55}{81} = 3195 \text{ psi}$$

Shear stresses in the inner shell at point "G" equals to $\underset{\text{(Sect. 4.6.2)}}{19306} \times \frac{14.55}{81} =$ 3468 psi.

Effective stresses S_{e2} in the inner and outer shell at $-40^{\circ}F$ 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 $-40^{\circ}F$ 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 Water Jacket Shell at point "A" $\quad MS = \frac{69000}{2766} - 1 = 23.9$

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"

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

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

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

T_{yz} = 2766 psi (Shear Stress) (Sect. 3.8)

T_{xy} = 0 T_{zx} = 176 (Sect. 3.8)

$$S_{e2} = \sqrt{\tfrac{1}{2}}\sqrt{(12.1 - (-1315))^2 + (-1315 - (-606))^2 + (-606-12.1)^2 + 6(2766^2 + {}^2)}$$

= 4936 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 x 43300 = 38970 psi

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

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

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

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

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

T_{yz} = 2766 psi (Shear Stress)

T_{xy} = 0 T_{zx} = 223 (Sect. 3.8)

$$S_{e2} = \sqrt{\tfrac{1}{2}}\sqrt{(-254 - (-676))^2 + (-676 - (-52))^2 + (-52 - (-254))^2 + 6\,(2766^2 + 223^2)}$$

$= 4838$ 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 x 43800 = 39420 psi.

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

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

σ_x = -117.5 psi (Radial Stress) From Sect. 3.8

σ_y = 590 psi (Tangential Stress) From Sect. 3.8

σ_z = 2535 psi (Axial Stress) From Sect. 3.8

T_{yz} = 2766 psi (Shear Stress)

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

$$S_{e2} = \sqrt{\tfrac{1}{2}}\sqrt{(-117.5 - 590)^2 + (590 - 2535)^2 + (2535-(-117.5))^2 + 6\ (2766^2)}$$

$= 5349$ 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 x 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".

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

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

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

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

T_{xy} = 0 T_{zx} = -259 psi (Sect. 3.8)

$$S_{e2} = \sqrt{\tfrac{1}{2}}\sqrt{(-115-(-1012))^2 + (-1012-(-175))^2 + (-175-(-115))^2 + 6\,(3195^2 + 259^2)}$$

= 5619 psi.

Allowable stress (S_{ad} = 0.9 S_{yd}) at 241 ^{o}F for 304 S.S. From Sect. 1.1 under normal conditions and Sect. 1.2 equals to 0.9 x 43224 = 38902 psi.

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

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

σ_x = -117.5 psi (Radial Stress) From Sect. 3.8

σ_y = 238 psi (Tangential Stress) From Sect. 3.8

σ_z = 2504 pis (Axial Stress) From Sect. 3.8

T_{zx} = 3195 psi (Shear Stress)

T_{xy} = T_{yz} = 0

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

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

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 x 43975 = 39577 psi.

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

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

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

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

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

T_{yz} = 3468 psi (Shear Stress)

T_{xy} = T_{zx} = 0

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

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

$$= 6010 \text{ psi}$$

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 x 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) STRESS COMPONENTS (PSI)

Condition	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 Sect. 3.8	-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.

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. 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.

PRINCIPAL STRESSES (PSI)

Condition	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.

STRESS DIFFERENCES (PSI)

Condition	$(\sigma_1-\sigma_2)$	$(\sigma_2-\sigma_3)$	$(\sigma_3-\sigma_1)$
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.

Allowable stress $3S_m$ = 67342 (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) STRESS COMPONENTS (PSI)

Condition	σ_x	σ_z	σ_y	T_{yz}
1. -40°F Isothermal	[0.0	11675	3400]	0.0
2. Normal Transport 130°F Ambient	[0.0	-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.

PRELIMINARY STRESS DIFFERENCES (PSI)

Condition	$\sigma_x-\sigma_{xi}$	$\sigma_z-\sigma_{zi}$	$\sigma_y-\sigma_{yi}$	$T_{yz}-T_{yzi}$
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.

PRINCIPAL STRESSES (PSI)

Condition	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.

From Ref. 3 Table II Case 5 $\sigma_1, \sigma_3 = \frac{-17815 + (-4867)}{2} + \left[\left(\frac{-17815 - (-4867)}{2}\right)^2 + 3468^2\right]^{\frac{1}{2}}$

$\sigma_1 = -3996$ psi

$\sigma_3 = -18685$ psi

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

STRESS DIFFERENCES (PSI)

Condition	$(\sigma_1-\sigma_2)$	$(\sigma_2-\sigma_3)$	$(\sigma_3-\sigma_1)$
1. -40°F Isothermal	0.0	0.0	0.0
2. Normal Transport 130°F Ambient	-3996	18685	-14689

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

Allowable stress $3S_m$ = 68082 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. $-40^{\circ}F$ Isothermal	2505	-19246	-9884	0.0	769	Sect. 3.8
2. Normal Transport $130^{\circ}F$ Ambient	127	3881	-5672	2766	234	
		Sect. 3.8				

Given the description of the Loading Cycle, the next step is to select one Reference Point. For this calculation, the $-40^{\circ}F$ 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. $-40^{\circ}F$ Isothermal	0.0	0.0	0.0	0.0	0.0
2. Normal Transport $130^{\circ}F$ 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. $-40^{\circ}F$ Isothermal	0.0	0.0	0.0	
2. Normal Transport $130^{\circ}F$ 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:

STRESS DIFFERENCES (PSI)

Condition	$(\sigma_1-\sigma_2)$	$(\sigma_2-\sigma_3)$	$(\sigma_3-\sigma_1)$
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

Point "A" Loc. 5 (Water Jacket Shell) STRESS COMPONENTS (PSI)

Condition	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 Sect. 3.8	10384]	2766

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

PRELIMINARY STRESS DIFFERENCES (PSI)

Condition	$\sigma_x-\sigma_{xi}$	$\sigma_z-\sigma_{zi}$	$\sigma_y-\sigma_{yi}$	$T_{yz}-T_{yzi}$
1. $-40^{o}F$ Isothermal	0.0	0.0	0.0	0.0
2. Normal Transport $130^{o}F$ Ambient	-235	37617	10827	2766

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

PRINCIPAL STRESSES (PSI)

Condition	σ_1	σ_2	σ_3	
1. $-40^{o}F$ Isothermal	0.0	0.0	0.0	
2. Normal Transport $130^{o}F$ Ambient	-235	37900	10544	From Ref. Table II Case 7

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

STRESS DIFFERENCES (PSI)

Condition	$(\sigma_1-\sigma_2)$	$(\sigma_z-\sigma_3)$	$(\sigma_3-\sigma_1)$
1. $-40^{o}F$ Isothermal	0.0	0.0	0.0
2. Normal Transport $130^{o}F$ 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

Point "F" Loc. 26 (Outer Shell) STRESS COMPONENTS (PSI)

Condition	σ_x	σ_z	σ_y	T_{yz}
1. -40°F Isothermal	0.0	23117	3398	0.0
2. Normal Transport -40°F Ambient	-33	-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.

PRELIMINARY STRESS DIFFERENCES (PSI)

Condition	$\sigma_x-\sigma_{xi}$	$\sigma_z-\sigma_{zi}$	$\sigma_y-\sigma_{yi}$	$T_{yz}-T_{yzi}$
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.

PRINCIPAL STRESSES (PSI)

Condition	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

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

$\sigma_1 = -9856$ psi

$\sigma_3 = -43489$ psi

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

STRESS DIFFERENCES (PSI)

Condition	$(\sigma_1-\sigma_2)$	$(\sigma_z-\sigma_3)$	$(\sigma_3-\sigma_1)$
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^{o}F Isothermal and Normal Transport 70 kw, 130^{o}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^{o}F Isothermal	0.0	-2953	48	0.0
2. Normal Transport 130^{o}F Ambient	-235	42790	10154	3195

Sect. 3.8

Given the description of the loading cycle, the next is to select one Reference Point. For this calculation the -40^{o}F Isothermal is selected as the reference.

PRELIMINARY STRESS DIFFERENCES (PSI)

Condition	$\sigma_x-\sigma_{xi}$	$\sigma_z-\sigma_{zi}$	$\sigma_y-\sigma_{yi}$	$T_{yz}-T_{yzi}$
1. -40^{o}F Isothermal	0.0	0.0	0.0	0.0
2. Normal Transport 130^{o}F Ambient	-235	45743	10106	3195

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°F Ambient	-235	46027	9821

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

STRESS DIFFERENCES (PSI)

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

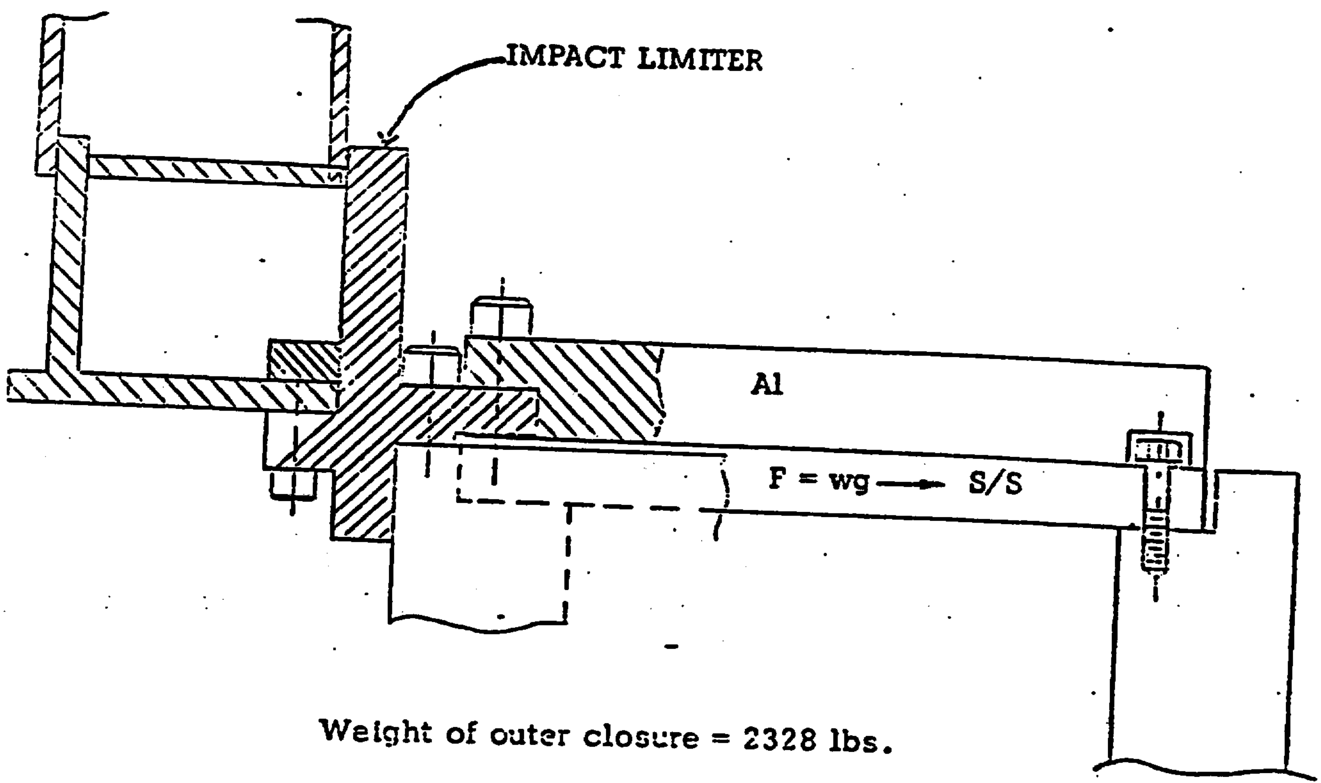
The maximum range for the cycle is then the largest of the stress differences of -46262 psi.

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 x 2328 = 33872 lbs.

Analysis of bolts in shear:

Outer closure is bolted down with 28 $1\frac{1}{4}$ 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 $1\frac{1}{4}$-8 bolts at the shear plane is 0.9405 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

$$S_{e2} = \sqrt{\tfrac{1}{2}} \sqrt{(0-0)^2 + (0-12{,}158)^2 + (12{,}158-0)^2 + 6(1286)^2}$$

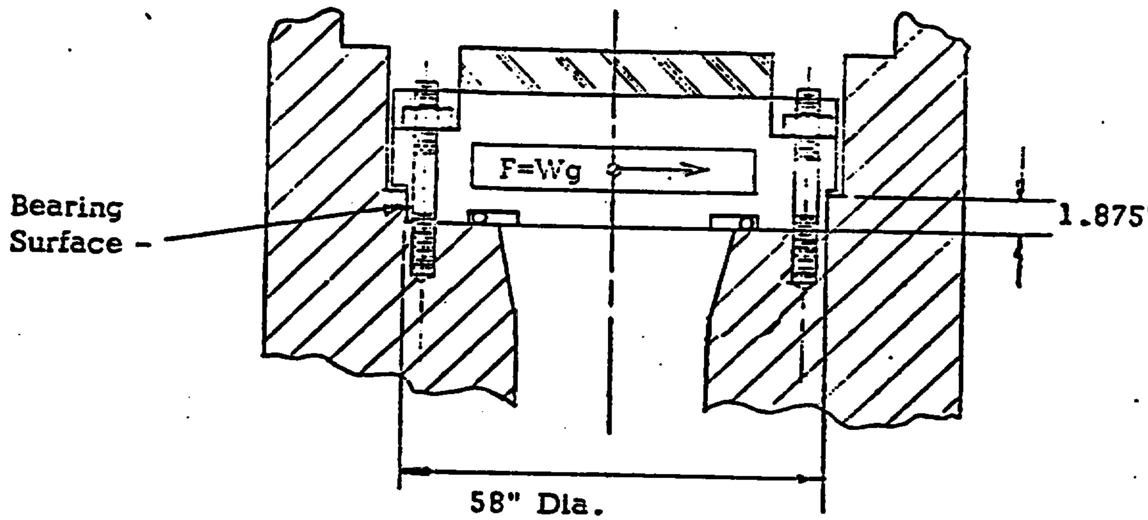
$$S_{e2} = 12360 \text{ psi.}$$

Allowable stress ($S_a = .75\ S_y$) for the outer closure bolts under normal condition from Sect. 1.1 and Sect. 1.2 equals 0.75 x 85000 = 63750 psi.

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.

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 catagory 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 approximately 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°F = 45750 psi tensile shear stress

$A = \pi Dt$

D = 6 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°F = 45750 psi tensile shear stress

$A = \pi Dt$

D = 6

t = .500 in. thickness of expansion tank shell

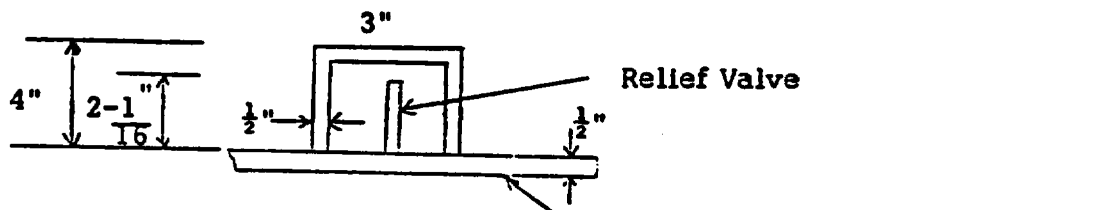
K = 39 Penetration to Fracture percent

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

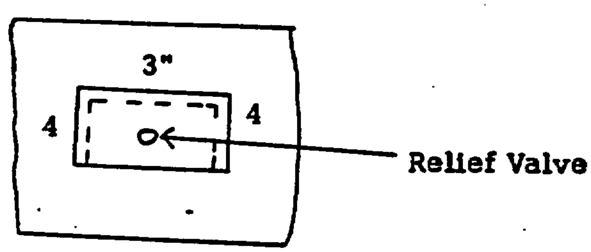
= 84081 in-lbs (Puncture Resistance)

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.

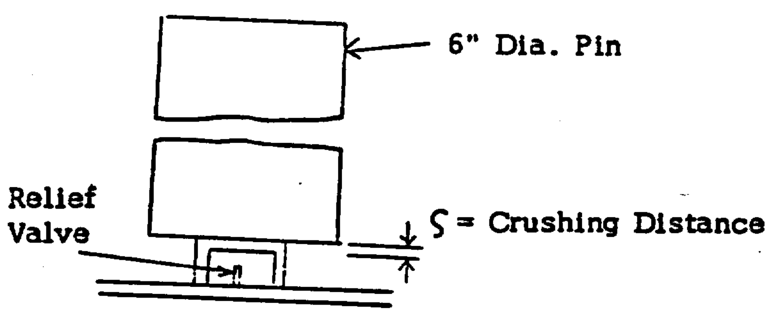


Side View



Top View

Condition No. 1



$KE = \sigma_y \ A \ \varsigma$

σ_y = 40667 psi

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

$A = 11\ (.5)$

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

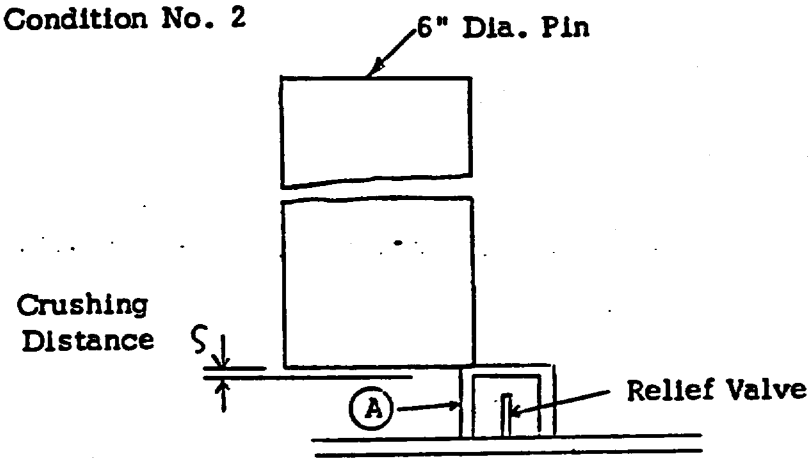
$27000 = 40667\ (6.5)$

$= \frac{27000}{264336}$

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

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

Condition No. 2



$KE = \sigma_y\ A\varsigma$

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

$27000 = 40667\ (2)$

$A = 4\ (.5)$

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

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

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 $U = K\,r\,l\,t^2$

$K = .39$

$r = .75\ (61000)$ at 350°F

$= 45750$ Psi tensile shear stress

l = total shear length

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

$= 9"$

t = shell thickness

.5"

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

$= 40146$ in-lbs

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.

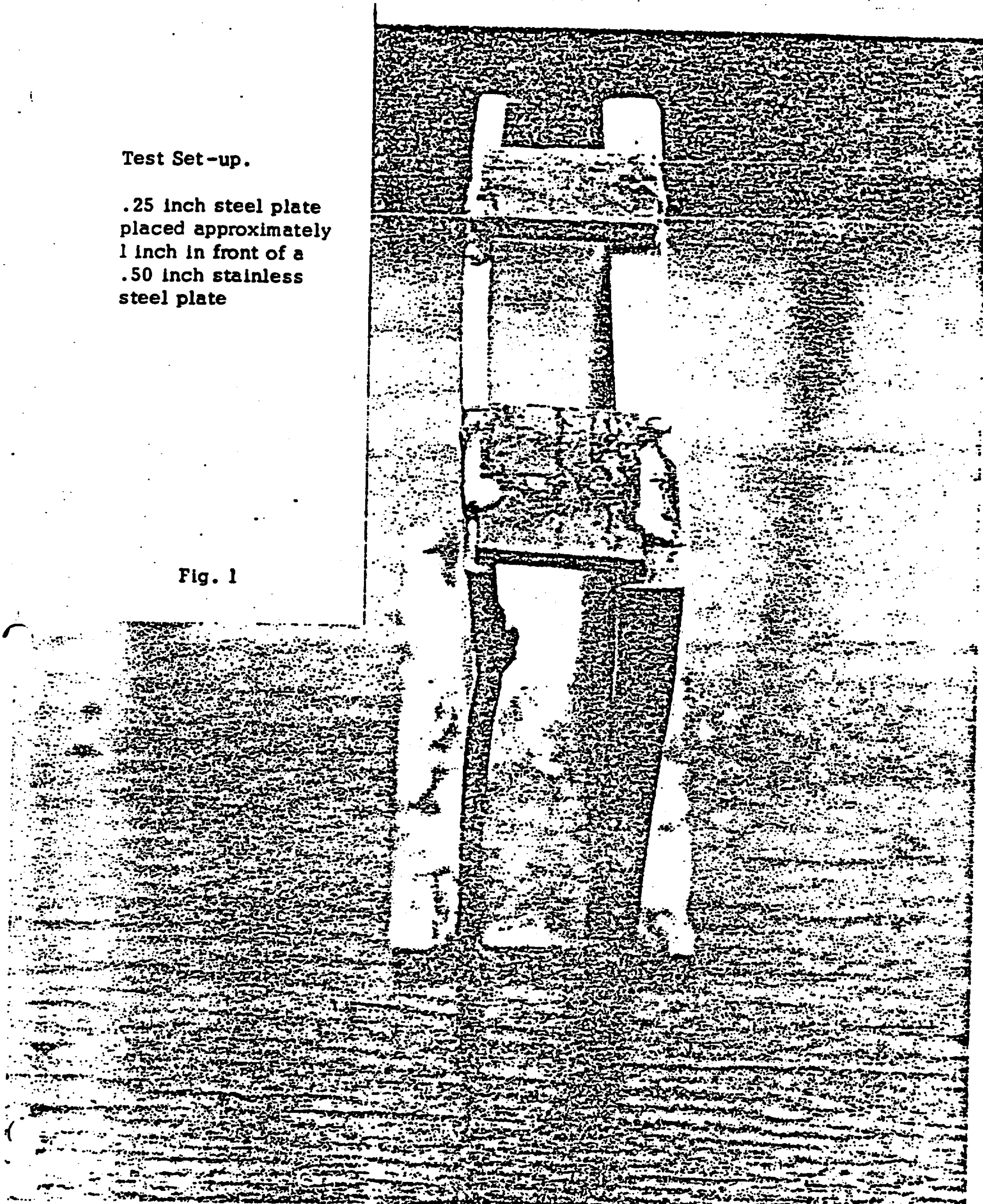
The .25 inch and .50 inch thick plates were penetrated. The neutron shield expansion tanks are made of .50 inch thick plate. Penetration of these tanks under normal conditions of transport is not desirable. A second field test was set up to determine an effective means of protecting the neutron shield 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

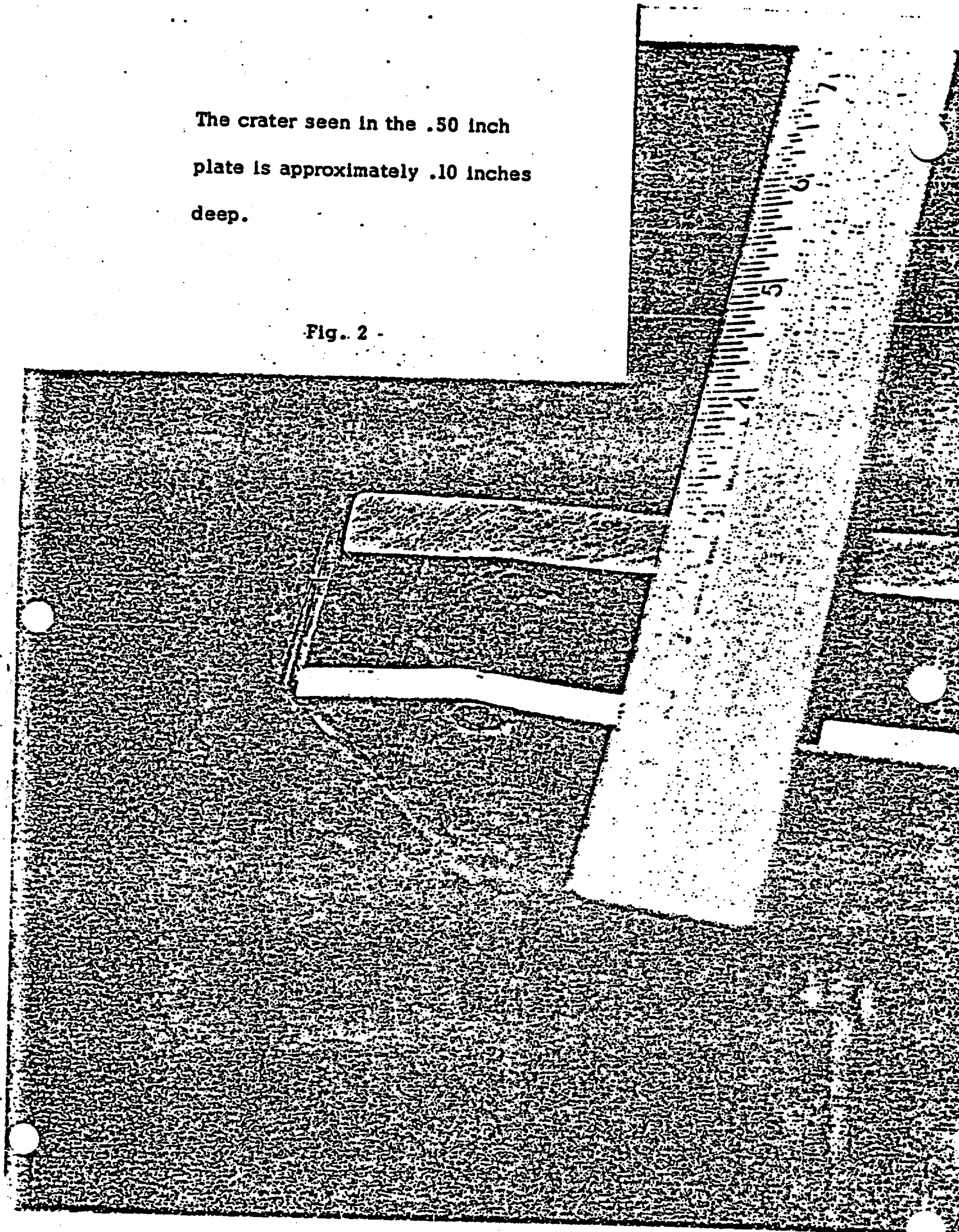
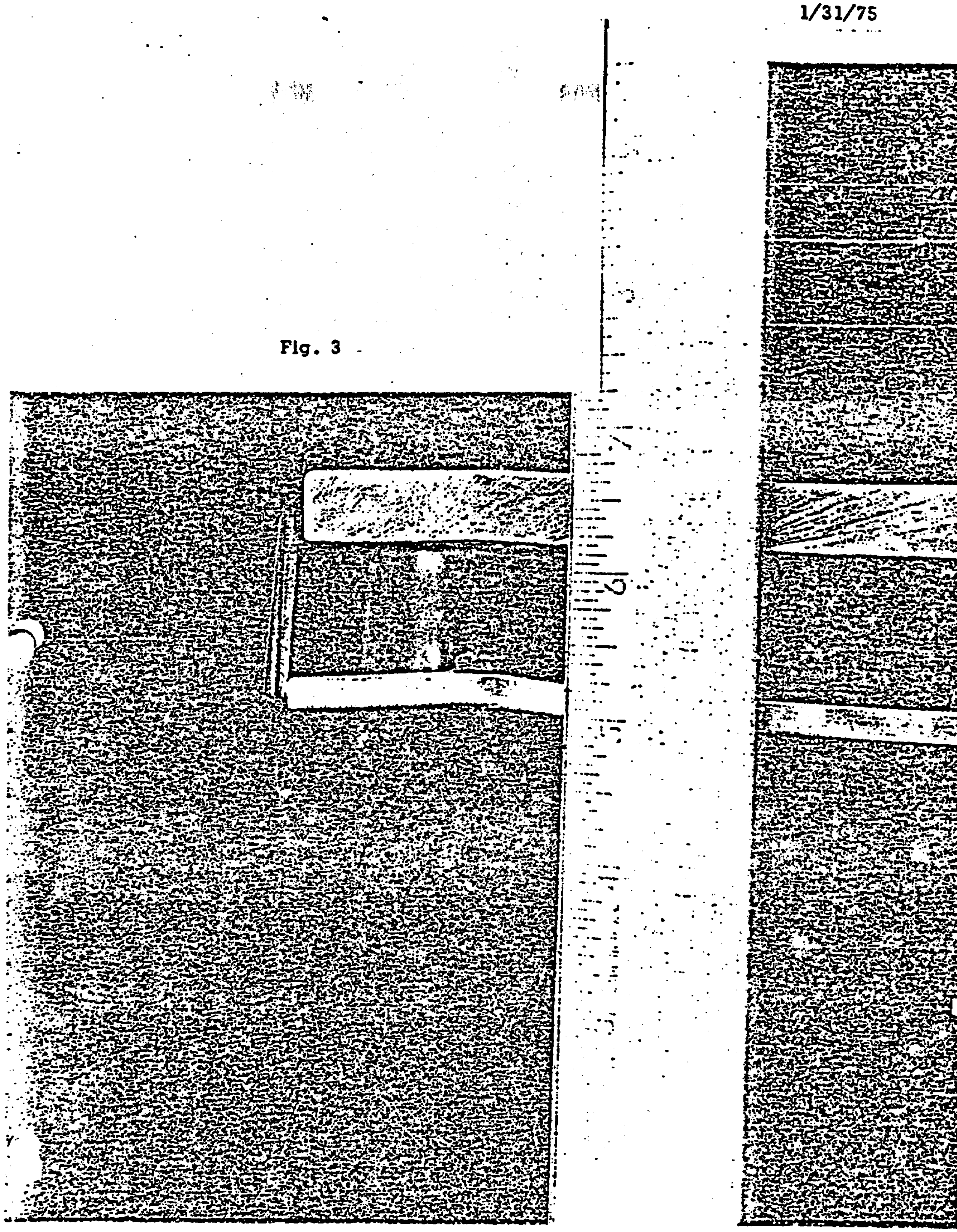


Fig. 3



Test Plate

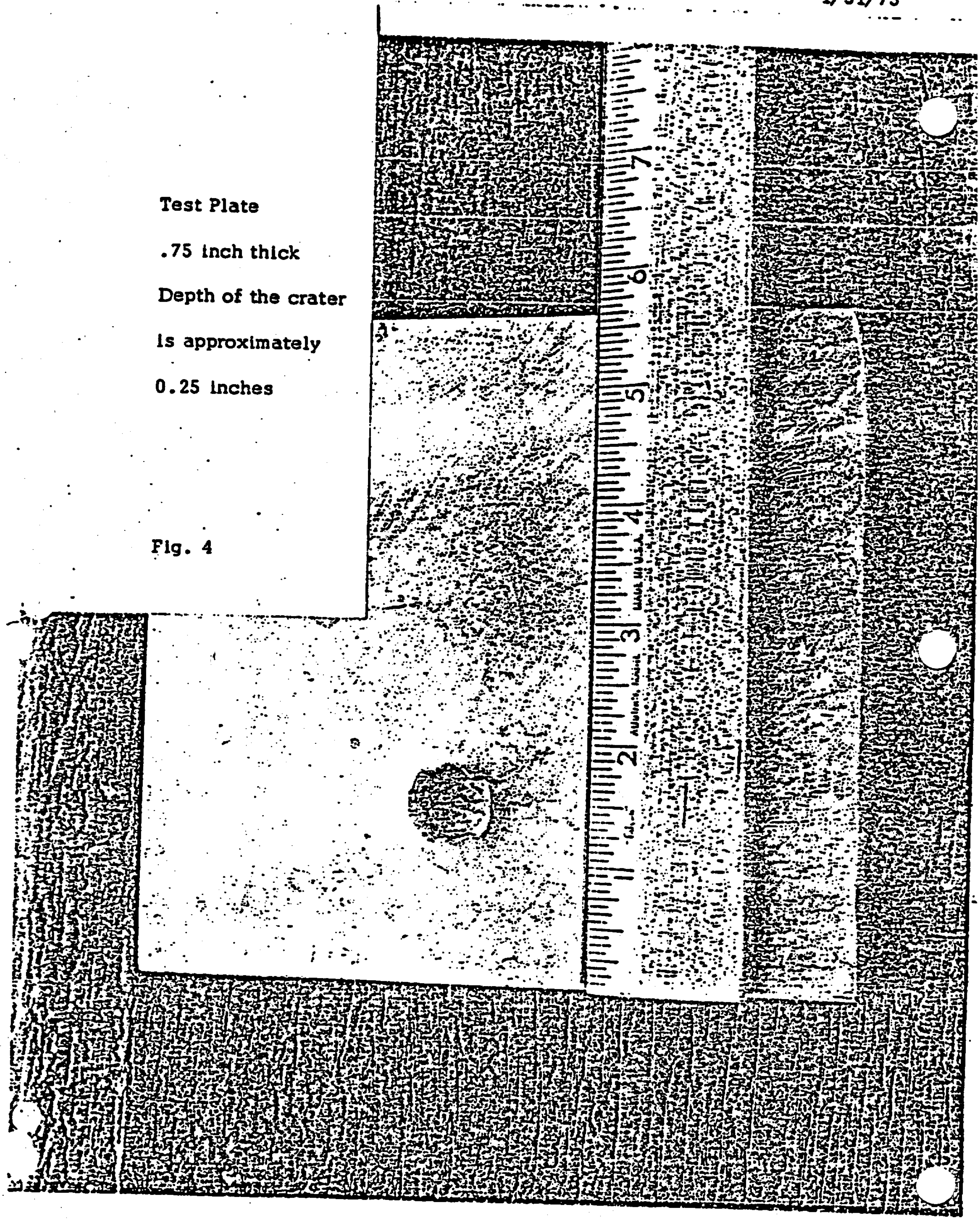
.75 inch thick

Depth of the crater

is approximately

0.25 inches

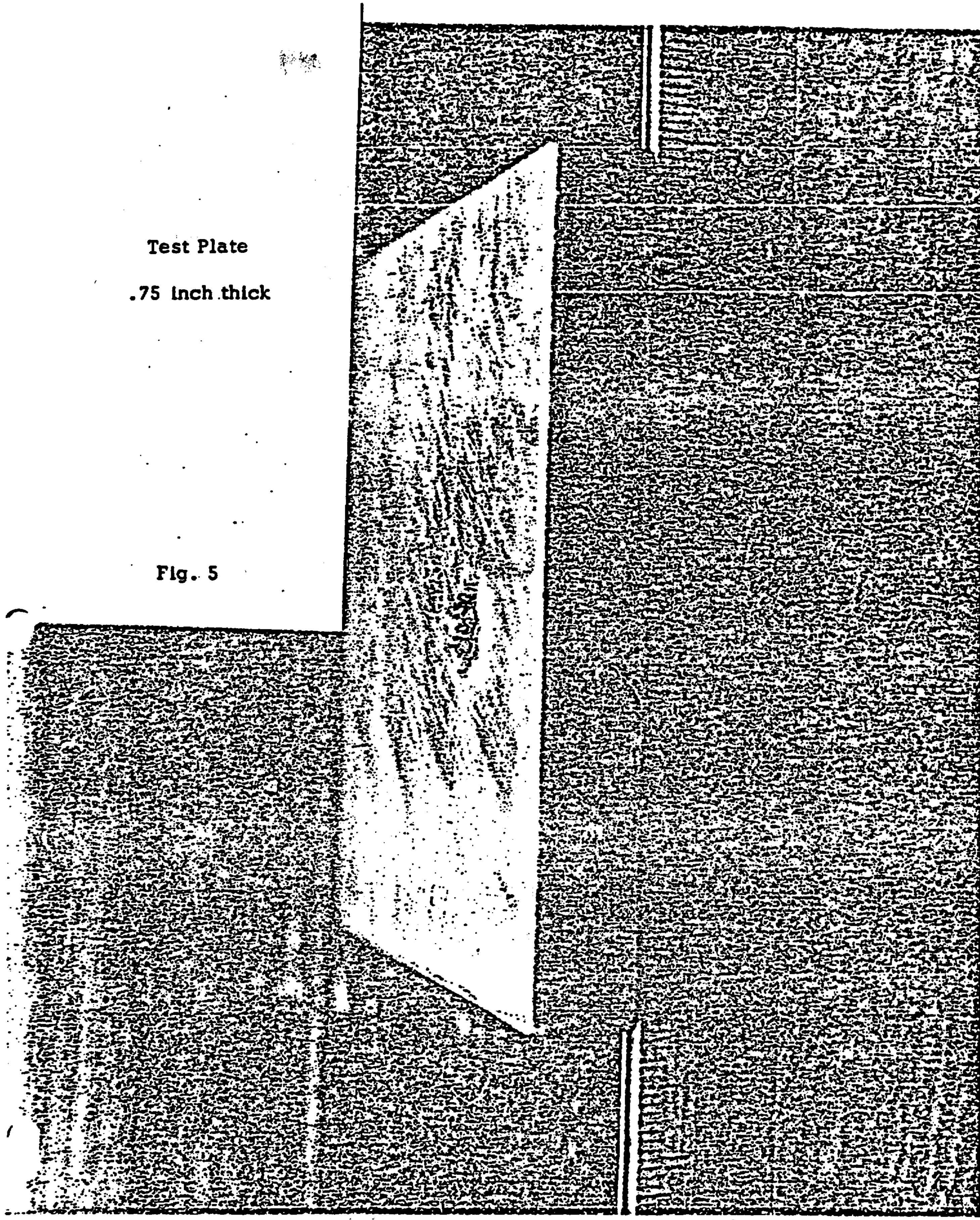
Fig. 4



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 lb.

H = 40 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 thinest of the items considered vulnerable ($t = .375$):

τ	=	.6(59,500) = 35,700 psi tensile shear stress
A	=	π D t
D	=	1.25 in.
t	=	.375 in. plate thickness
k	=	.39 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.

-40°F 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

Cask Location	Temperature (°F) - BWR & PWR Fuel
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)

Cask Location	Temperature (°F)	
	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

Cask location	Temperature (°F) - BWR & PWR Fuel
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.

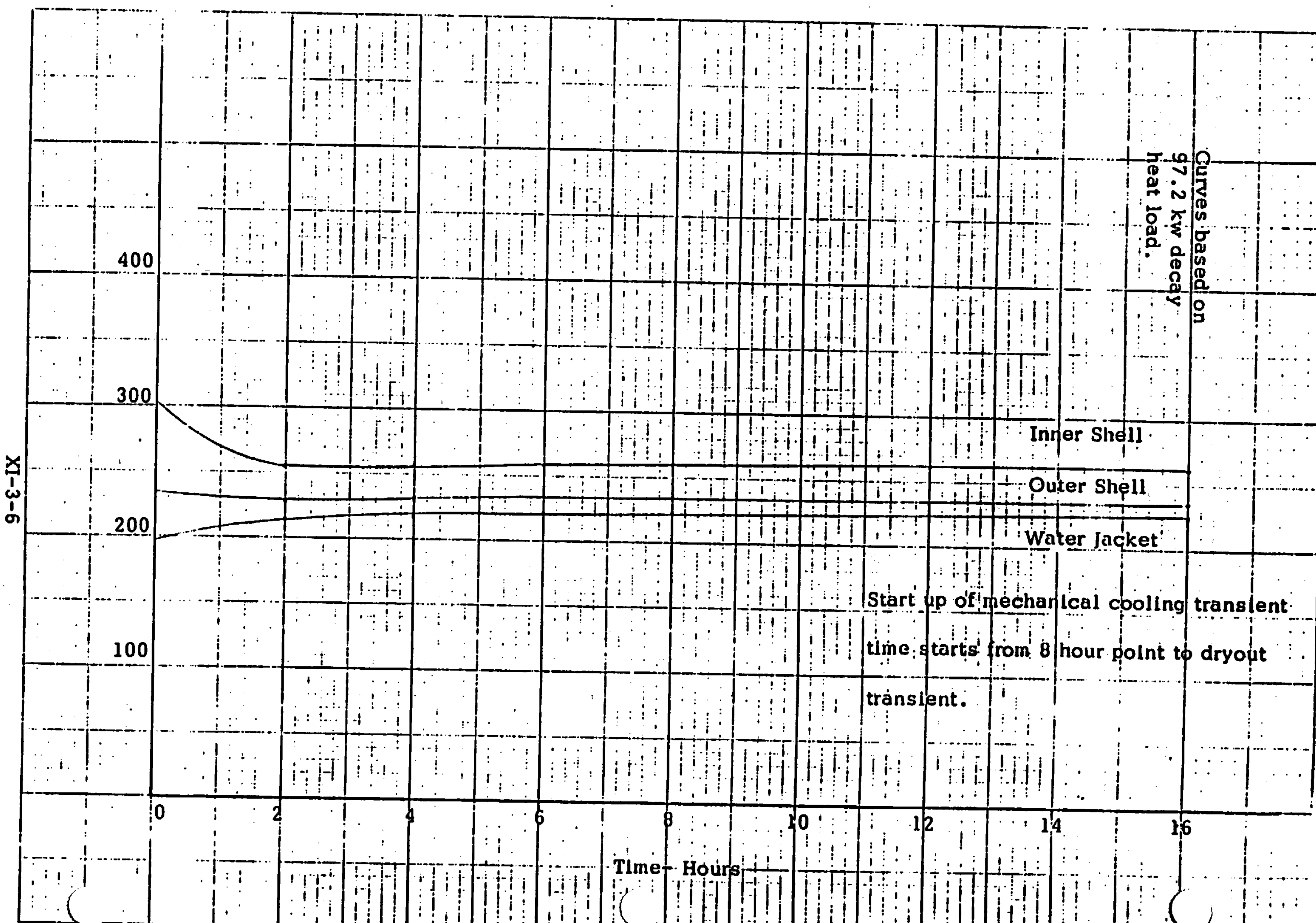
The cask dryout transient for the cask shells and water jacket is shown in Pg. 3-7 . This figure shows the response of the cask after fuel is loaded. The cask is vertical and there is no mechanical cooling. The cask cavity is also dry. Pg. 3-7 shows that the ΔT across the shells is increasing throughout 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.

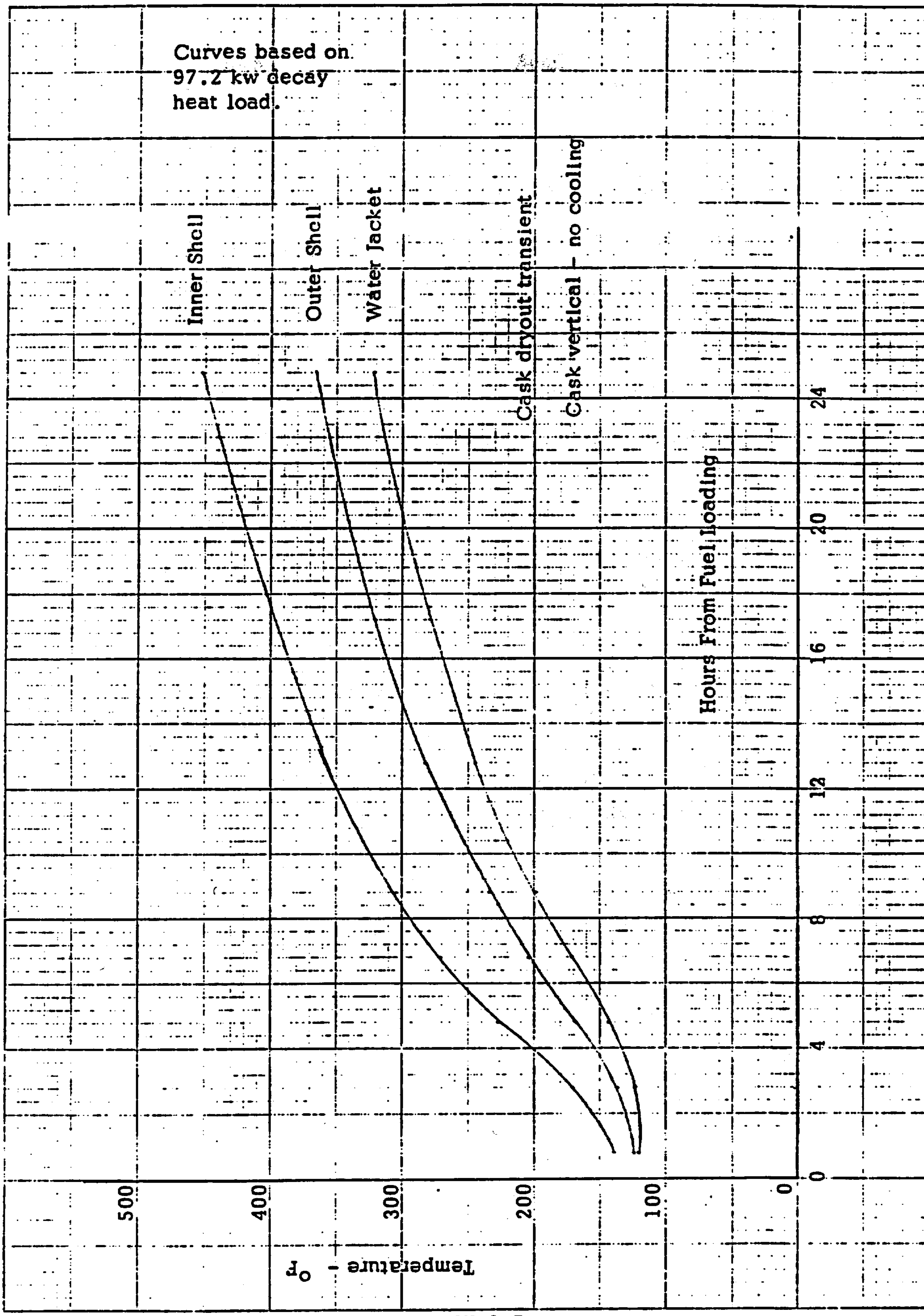
Fig. on Pg. 3-6 shows the response of the cask shells to mechanical cooling. The mechanical cooling starts 8 hours after the beginning of the dryout transient.

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 difusivity 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.


400
300
200
100
0
2
4
6
8
10
12
14
16
Time- Hours
Inner Shell
Outer Shell
Water Jacket
Curves based on 97.2 kw decay heat load.
Start up of mechanical cooling transient time starts from 8 hour point to dryout transient.


Curves based on
97.2 kw decay
heat load.
Inner Shell
Outer Shell
Water Jacket
Cask dryout transient
Cask vertical - no cooling
Hours From Fuel Loading
0
4
8
12
16
20
24
Temperature - °F
500
400
300
200
100
0

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:

Loading Type	Loading Direction and Level		
	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 accomodate 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 :

$$V_1 = \frac{P_0 T_1 V_0}{T_o P_1} + \frac{1.36 EfMRT_1}{P_1} \qquad \text{(1) Ref. 22}$$

It is possible that future elements will be pressurized to a higher pressure than that shown in Table (1) (P_o). Therefore a new value of P_1 will be calculated using equation (1) with an initial fill pressure of 515 PSIA for PWR type fuel. This assumes that the initial 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)

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

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

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

This is the pressure that will 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.

$$N_P = 1.44 \times 10^{-4}\ (208)\ (10) = .299 \text{ mols}$$

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 : $N_T = .299 + .199 = .498$ 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 $P = \frac{(659 + 460)\ (14.7)}{(68 + 460)} = 31.15 \text{psia or } 16.45 \text{psig}$

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:

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

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

P_1 = 1740 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)

$$\lambda_i = \frac{\varepsilon_{plastic,\ i} - \varepsilon_{plastic,\ i-1}}{\varepsilon_{elastic,\ i}}$$

Since the elastic strains $\varepsilon_{elastic}$ are generally much smaller than the plastic strains, a value of $\lambda = .05$, for example, respresents 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:

$$\frac{\delta\varepsilon_x^p}{\sigma_x'} = \frac{\delta\varepsilon_y^p}{\sigma_y'} = \frac{\delta\varepsilon_z^p}{\sigma_z'} = \frac{\delta\gamma_{xy}^p}{\tau_{xy}'} = \frac{\delta\gamma_{yz}^p}{\tau_{yz}'} = \frac{\delta\gamma_{zx}^p}{\tau_{zx}'} = \frac{3}{2}\,\delta\beta$$

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 $\delta\beta$ 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^{o}F compatible with the slope implied by $E = 4.0 \times 10^{6}$, selection of a first interval of $\varepsilon = 10^{-4}$ was necessary. When the strains exceed $\varepsilon = 10^{-4}$, 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,τ) 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:

$$\mathrm{EPGEN} = \frac{\sqrt{(\varepsilon_x-\varepsilon_y)^2 + (\varepsilon_y-\varepsilon_z)^2 + (\varepsilon_z-\varepsilon_x)^2 + \frac{3}{4}\gamma_{xy}^2}}{\sqrt{2} \quad (1 + \mathrm{POSGEN})}$$

where

$$\mathrm{POSGEN} = 0.5 - (0.5 - \nu)\,\frac{E_{secant}}{E_{elastic}}$$

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 lableled "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.

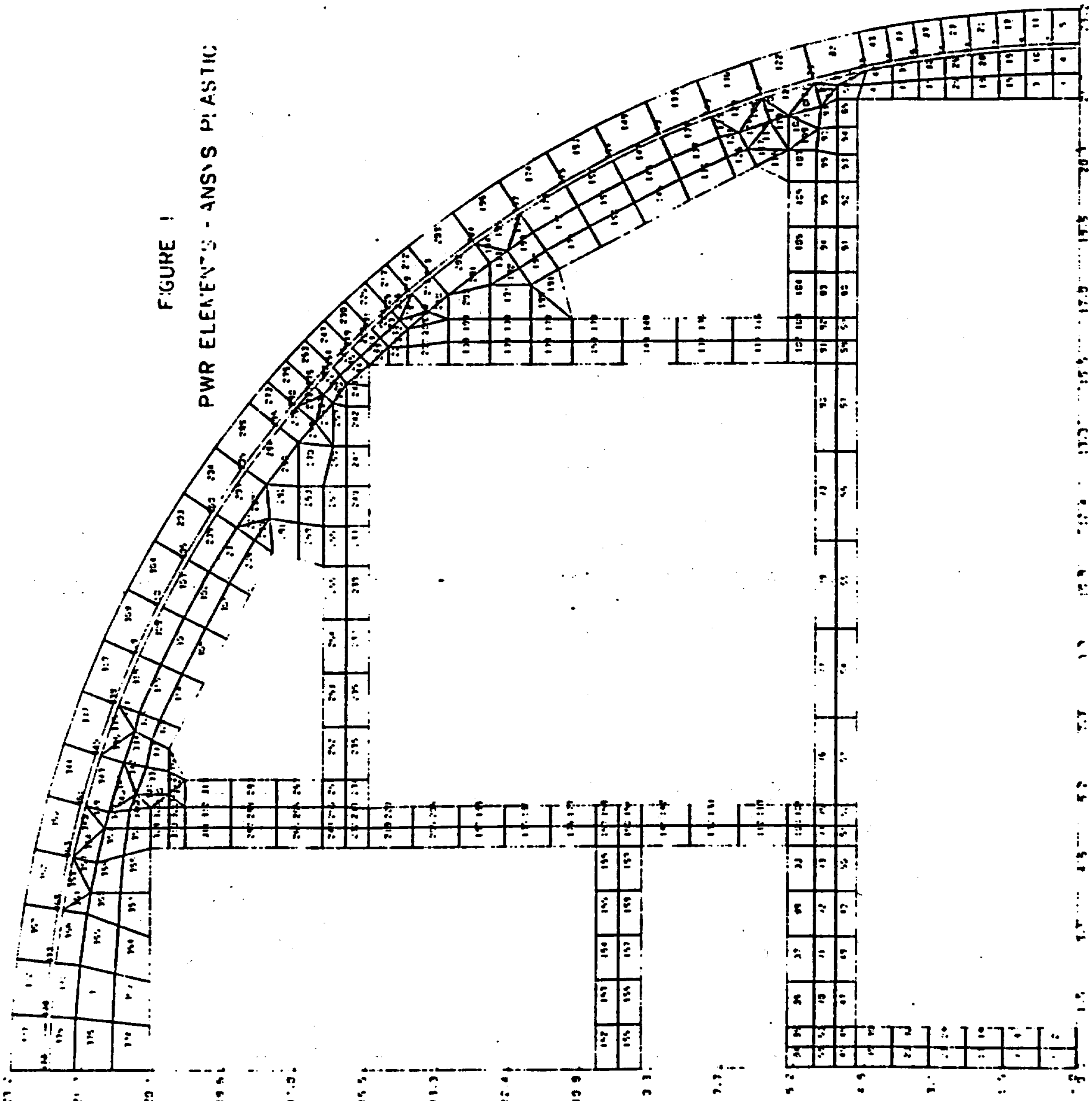
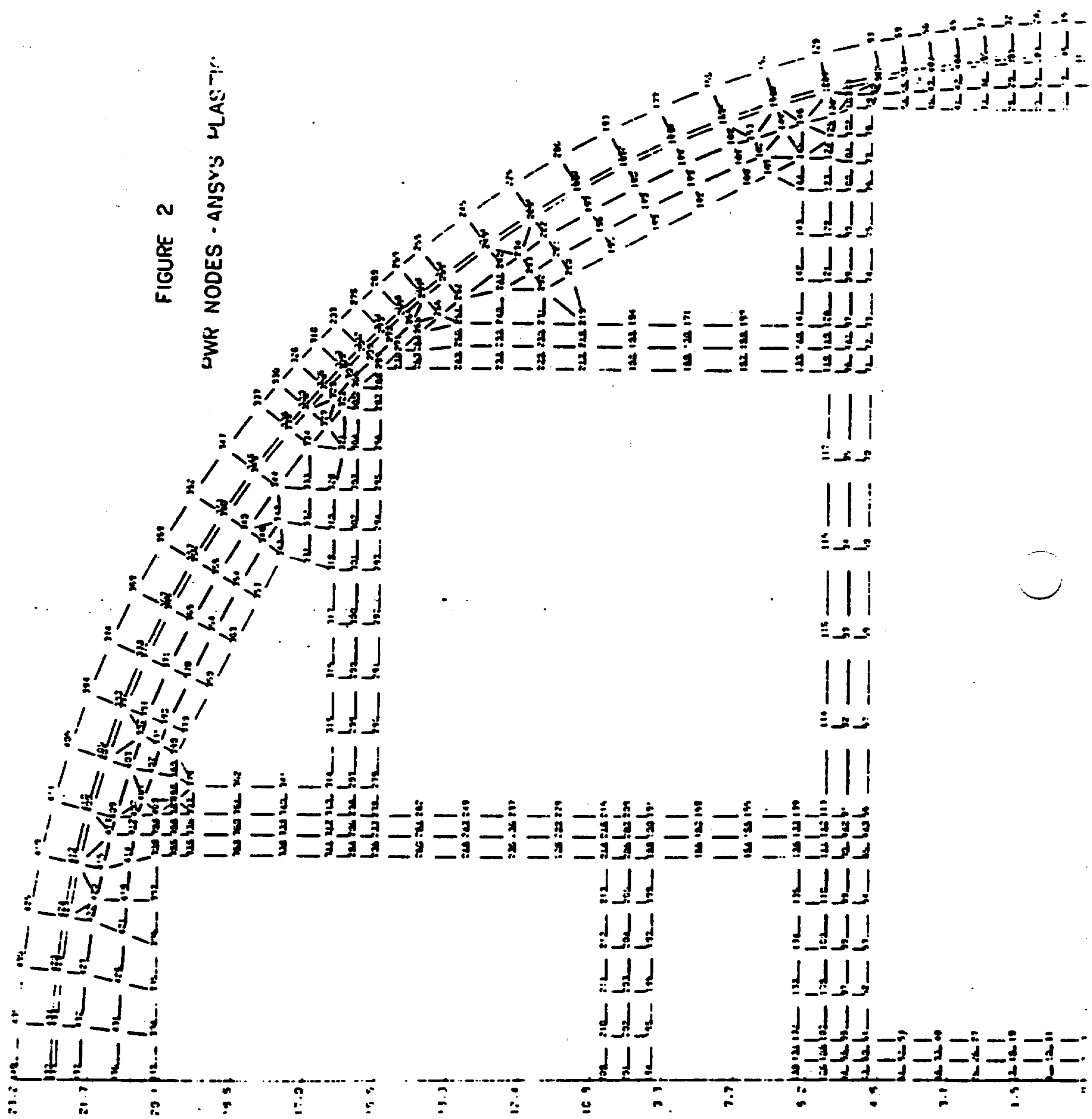


FIGURE 1

PWR ELEMENTS - ANSYS PLASTIC

FIGURE 2

PWR NODES - ANSYS PLASTIC



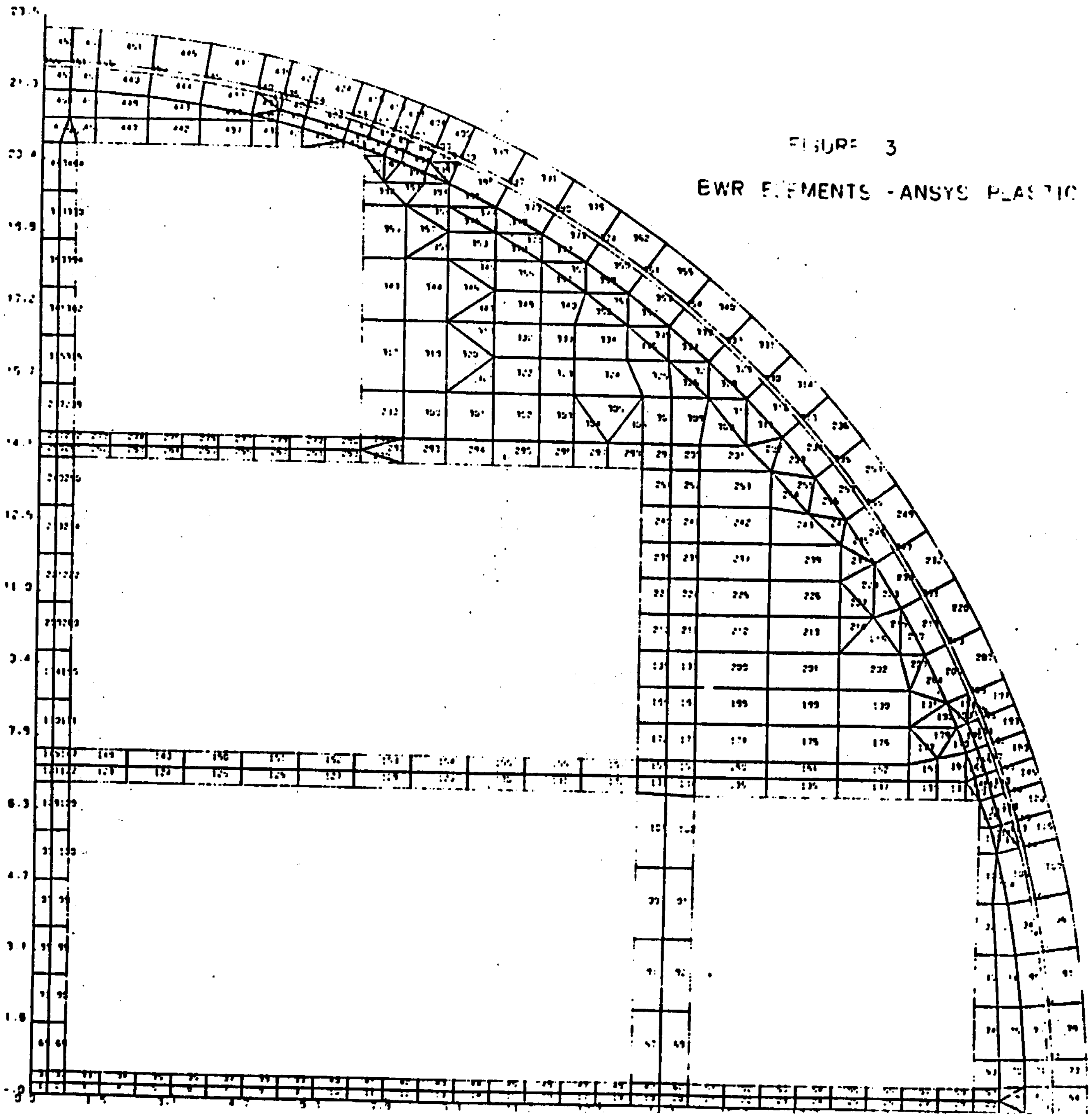


FIGURE 3

EWR ELEMENTS - ANSYS PLASTIC

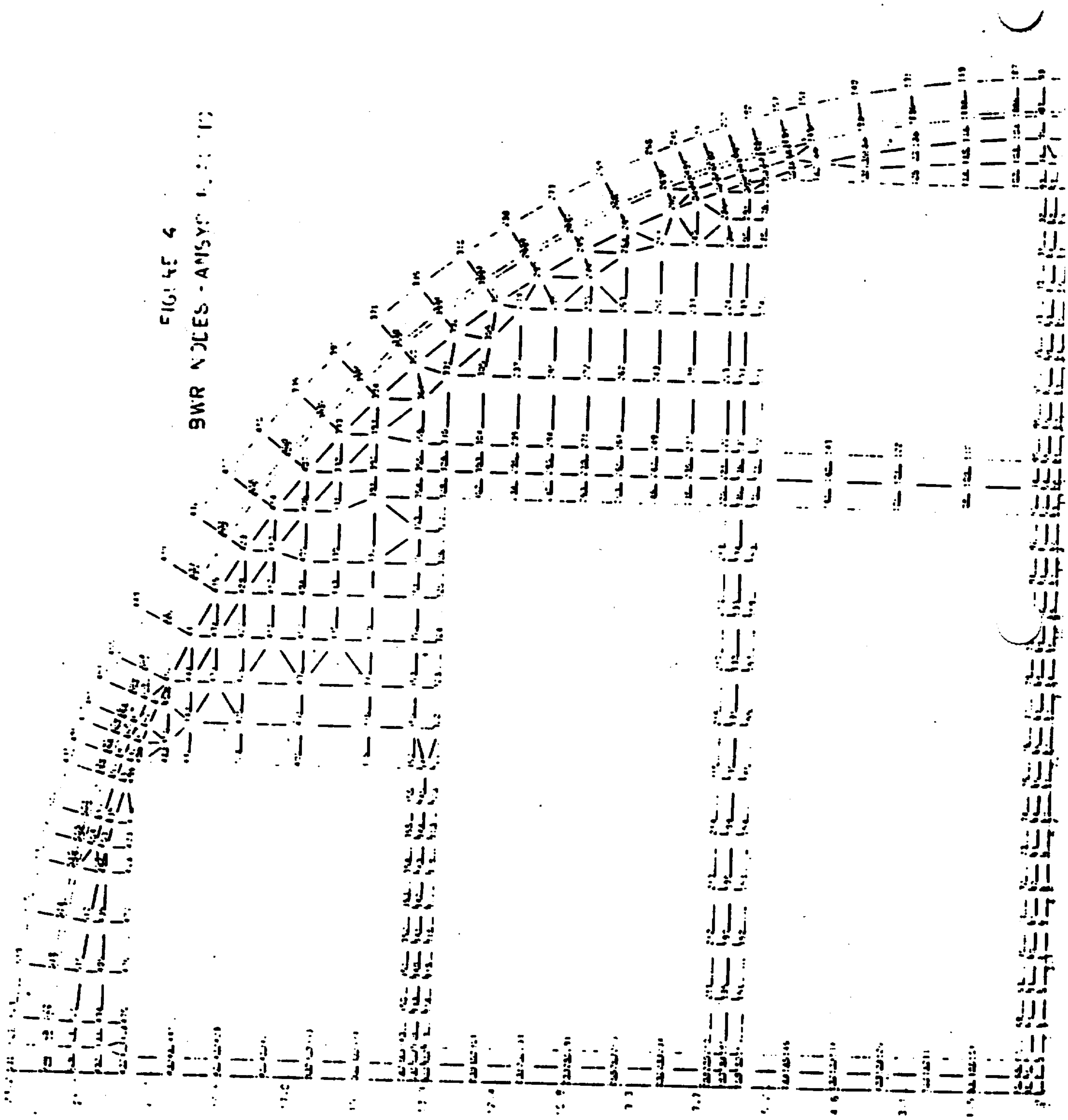


FIGURE 4

BWR NODES - ANSYS [illegible]

FIGURE 5

PWR GAP FOR NORMAL STEADY STATE W/O AUXILIARY COOLING

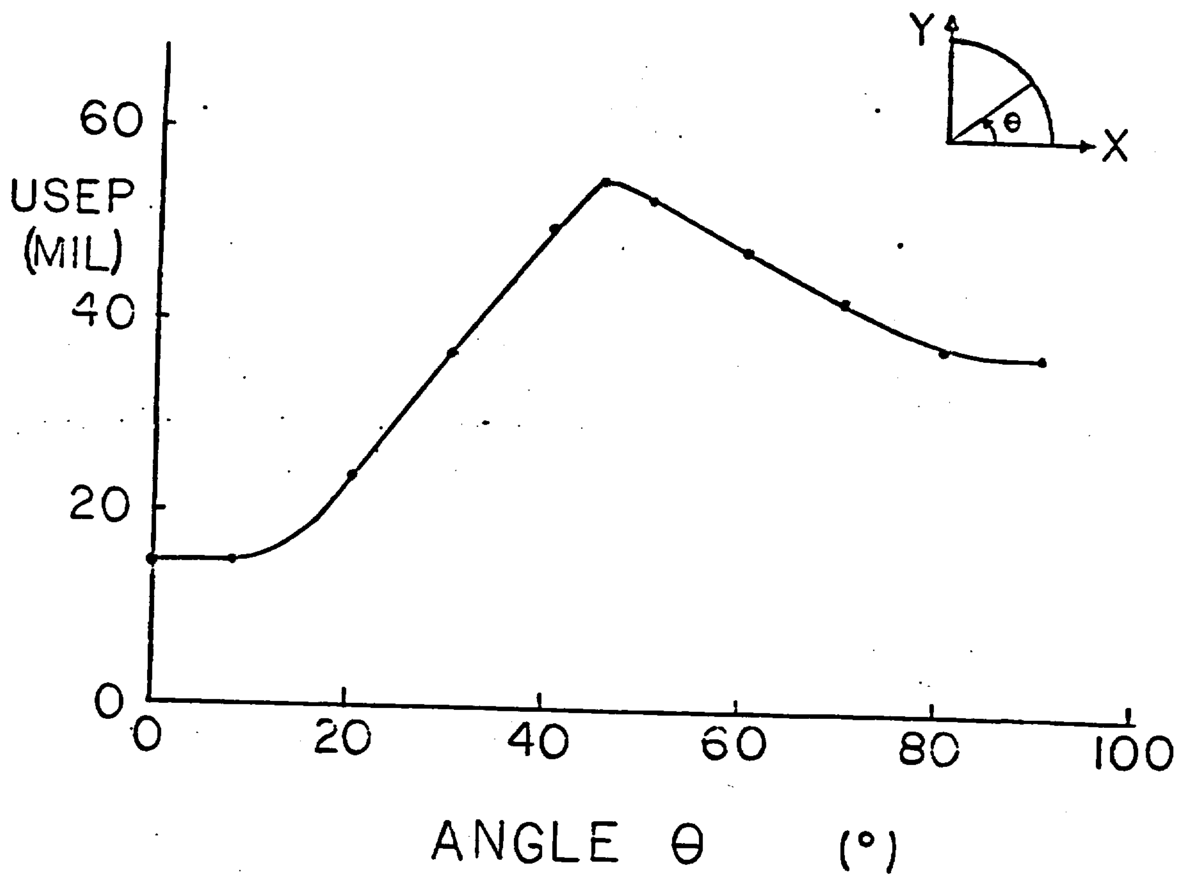
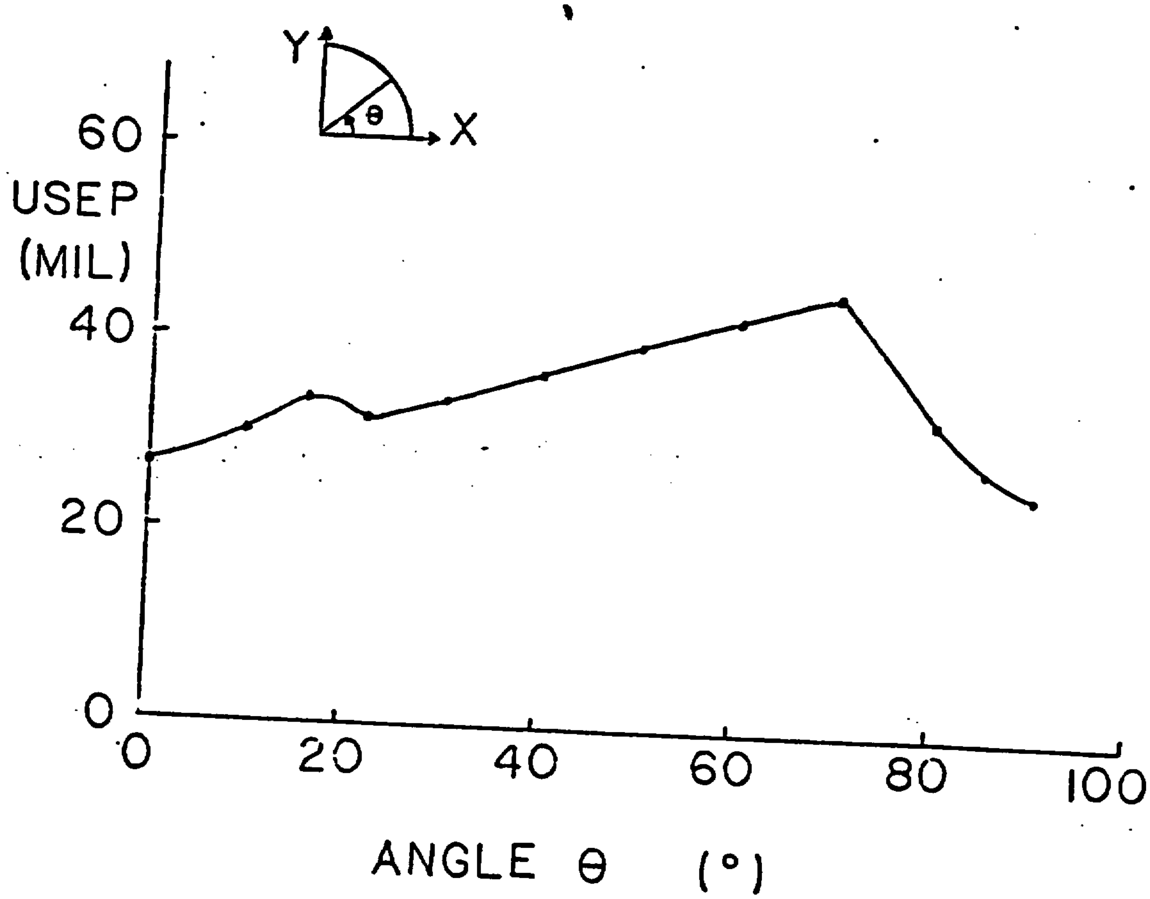


FIGURE 6
BWR GAP FOR NORMAL STEADY STATE W/O AUXILIARY COOLING



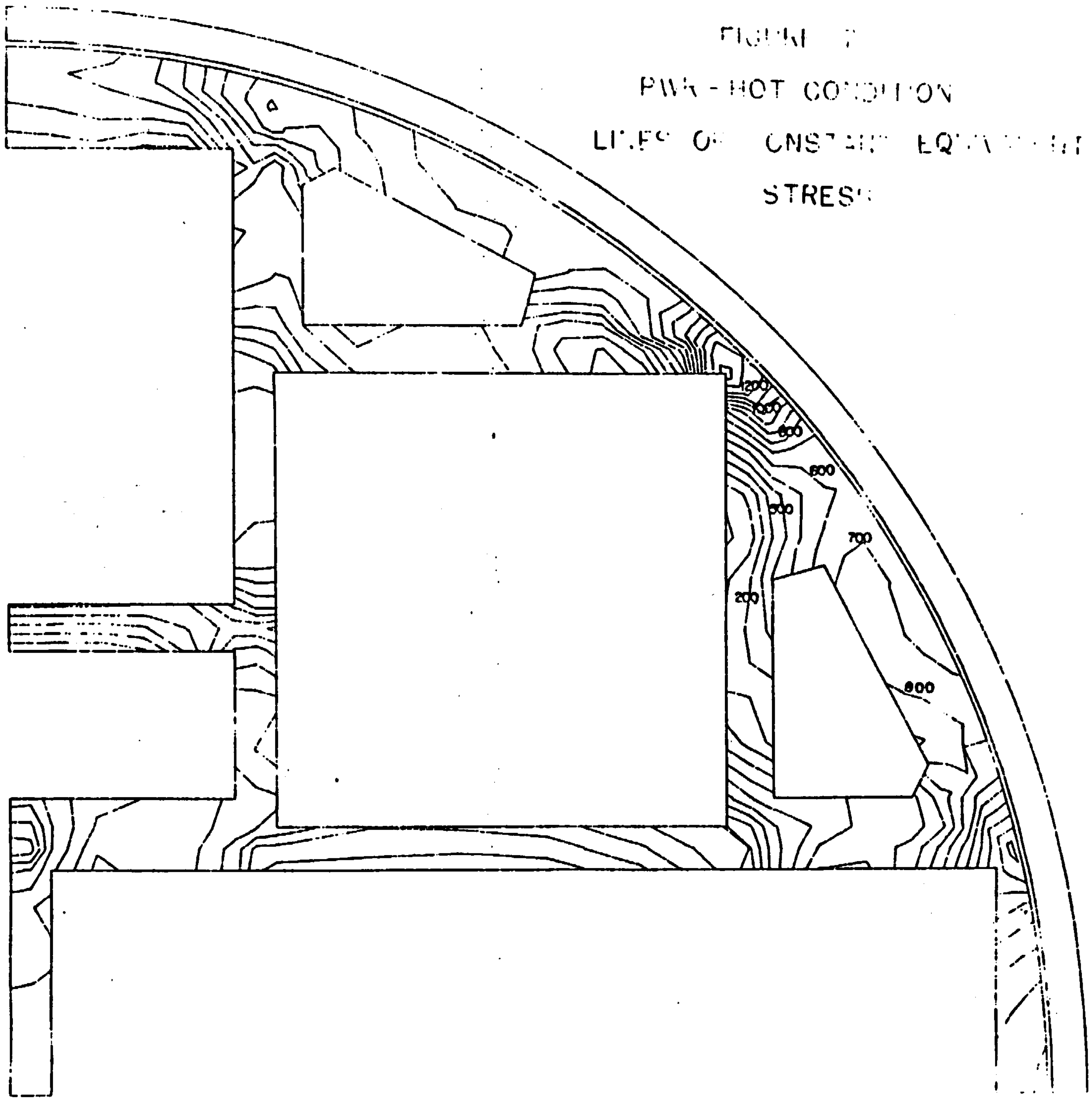


FIGURE 7

PWR - HOT COND ON

LINES OF CONSTANT EQ

STRESS

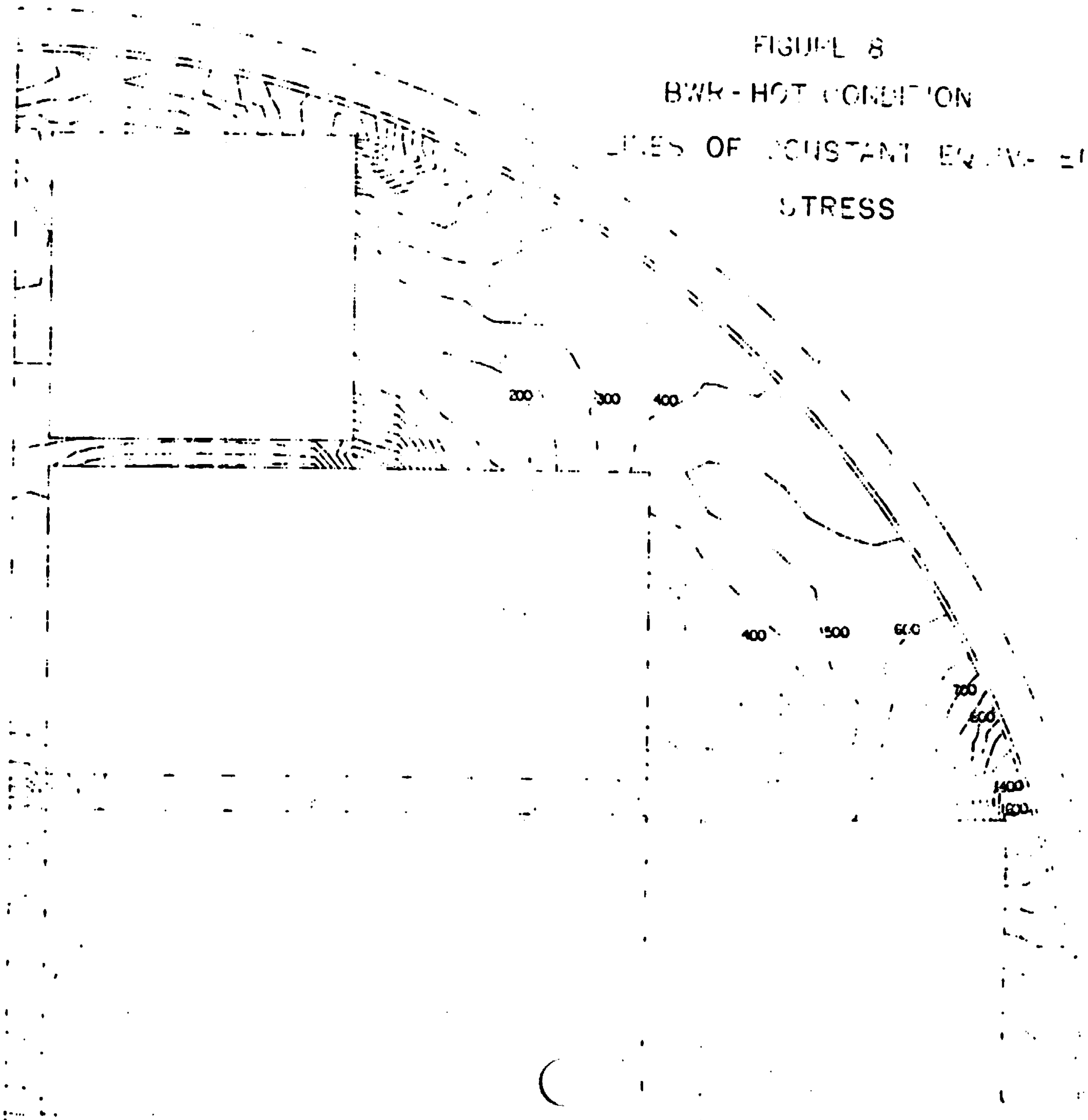


FIGURE 8

BWR - HOT CONDITION

LINES OF CONSTANT EQUIVALENT STRESS

HOT STEADY STATE CONDITION

```
ELEM   67  NODES=  104   80   81   -0   MATERIAL = 1  AREA=   .0846  STRESS INT.=   1556.  SIGEFF=   1347.
X= 21.70 Y=  4.90 T= 581. XY STR=   -474.    434.   -631.         0. MAX-MIN STR=    758.   -798.    778. A=-27.1
 ELASTIC  -.0000897  .0000858 -.0002439  .0000019  PLASTIC  -.0007216  .0009365 -.0018747 -.0002149
 THERMAL   .0072612  .0072612 0.0000000 0.0000000  SWELL    0.0000000 0.0000000 0.0000000 0.0000000
   CREEP  0.0000000 0.0000000 0.0000000 0.0000000  0.SHIFT  0.0000000 0.0000000 0.0000000 0.0000000
  EPGEN=   .0016545   POSGEN=   .4792   SIGE=   1347.37   FLUENCE= 0.
```

ROOM TEMPERATURE UNLOADED CONDITION

```
ELEM   67  NODES=  104   80   81   -0   MATERIAL = 1  AREA=   .0846  STRESS INT.=   3015.  SIGEFF=   2728.
X= 21.70 Y=  4.90 T=  70. XY STR=    258.  -1838.   1083.         0. MAX-MIN STR=    717.  -2297.   1507. A= 67.0
 ELASTIC   .0000865 -.0001924  .0002881  .0000502  PLASTIC   .0005554 -.0009089  .0014686  .0003535
 THERMAL  0.0000000 0.0000000 0.0000000 0.0000000  SWELL    0.0000000 0.0000000 0.0000000 0.0000000
   CREEP  0.0000000 0.0000000 0.0000000 0.0000000  0.SHIFT  -.0007216  .0009365 -.0018747 -.0002149
  EPGEN=   .0015210   POSGEN=   .4696   SIGE=   2723.27   FLUENCE= 0.
```

Figure 9

Typical PWR Element Printout - ANSYS Program

TABLE I

1180 ALUMINUM DATA

Temperature	ε = .0001	ε = .0005	.0020	.01	.10	E
70°F	σ = 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

Table II

REPRESENTATIVE ELEMENTS HAVING HIGH STRAINS
NORMAL STEADY STATE CONDITION WITHOUT
AUXILIARY COOLING

	Element	EPGEN	Largest Strain Component	Direction of Largest Component	SIGE (psi)
	1	.0008285	.0006301	Y	897
PWR	67	.0016545	.0018747	XY	1347
	220	.0010913	.0007968	XY	1117
	. . . . .	. . . .	. . . .	. . . .	. . . .
	103	.0011044	.0009068	Y	1291
BWR	143	.0021771	.0018953	Y	1559
	410	.0017504	.0015368	X	1505

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	Δ_{1s} = 1.2377
Fuel Element (1)(3)(4)	160.515	833	765	* 2.95×10^{-6}	Zir. 4	Δ_1 = .3622
Absorber Sleeve	162.00	499	431	9.70×10^{-6}	S.S. 304	Δ_2 = .6762
Containment Shell (2)	156.00	353	285	9.47×10^{-6}	S.S. 304	Δ_3 = .4210
Lower Support Structure	4.125	438	370	9.67×10^{-6}	S.S. 304	Δ_4 = .01476
Spacer Plug	13.985	438	370	13.75×10^{-6}	Al.6061-T6	Δ_5 = .07886
Top Forging	11.50	410	342	9.62×10^{-6}	S.S. 304	Δ_6 = .0378

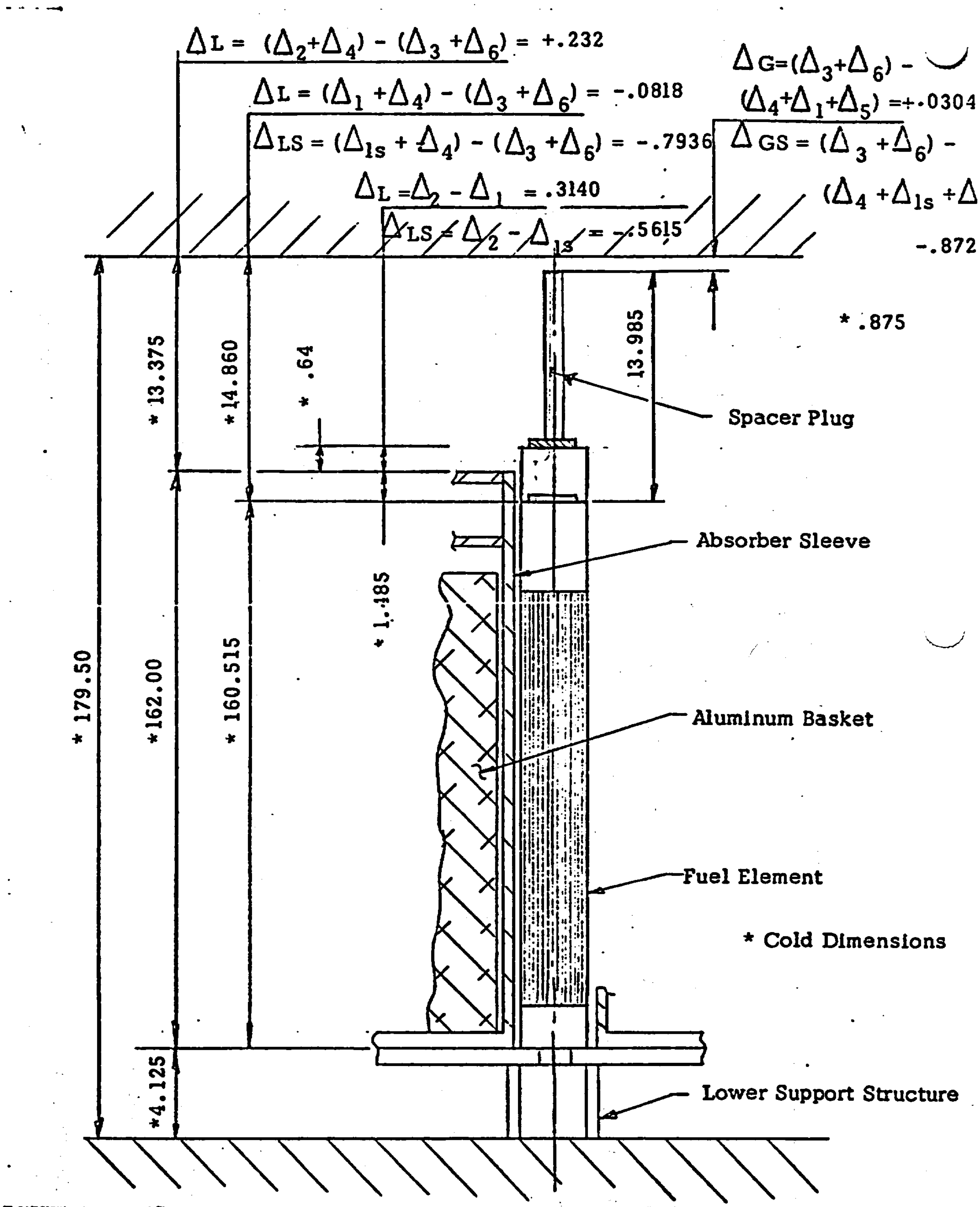
(1) Fuel element length is to the point where spacer compression legs pickup hard point of the fuel element.

(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.


ΔL = (Δ2+Δ4) - (Δ3 +Δ6) = +.232
ΔL = (Δ1 +Δ4) - (Δ3 +Δ6) = -.0818
ΔLS = (Δ1s + Δ4) - (Δ3 +Δ6) = -.7936
ΔL = Δ2 - Δ1 = .3140
ΔLS = Δ2 - Δ1s = -.5615
ΔG=(Δ3+Δ6) - (Δ4+Δ1+Δ5) =+.0304
ΔGS = (Δ3 +Δ6) - (Δ4 +Δ1s +Δ
-.872
* .875
* 13.375
* 14.860
* .64
13.985
Spacer Plug
Absorber Sleeve
* 1.485
* 179.50
*162.00
* 160.515
Aluminum Basket
Fuel Element
* Cold Dimensions
*4.125
Lower Support Structure

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.	Δ T °F.	(mean) α in/in °F.	Material	Elongation
Fuel Element (4)	162.925	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

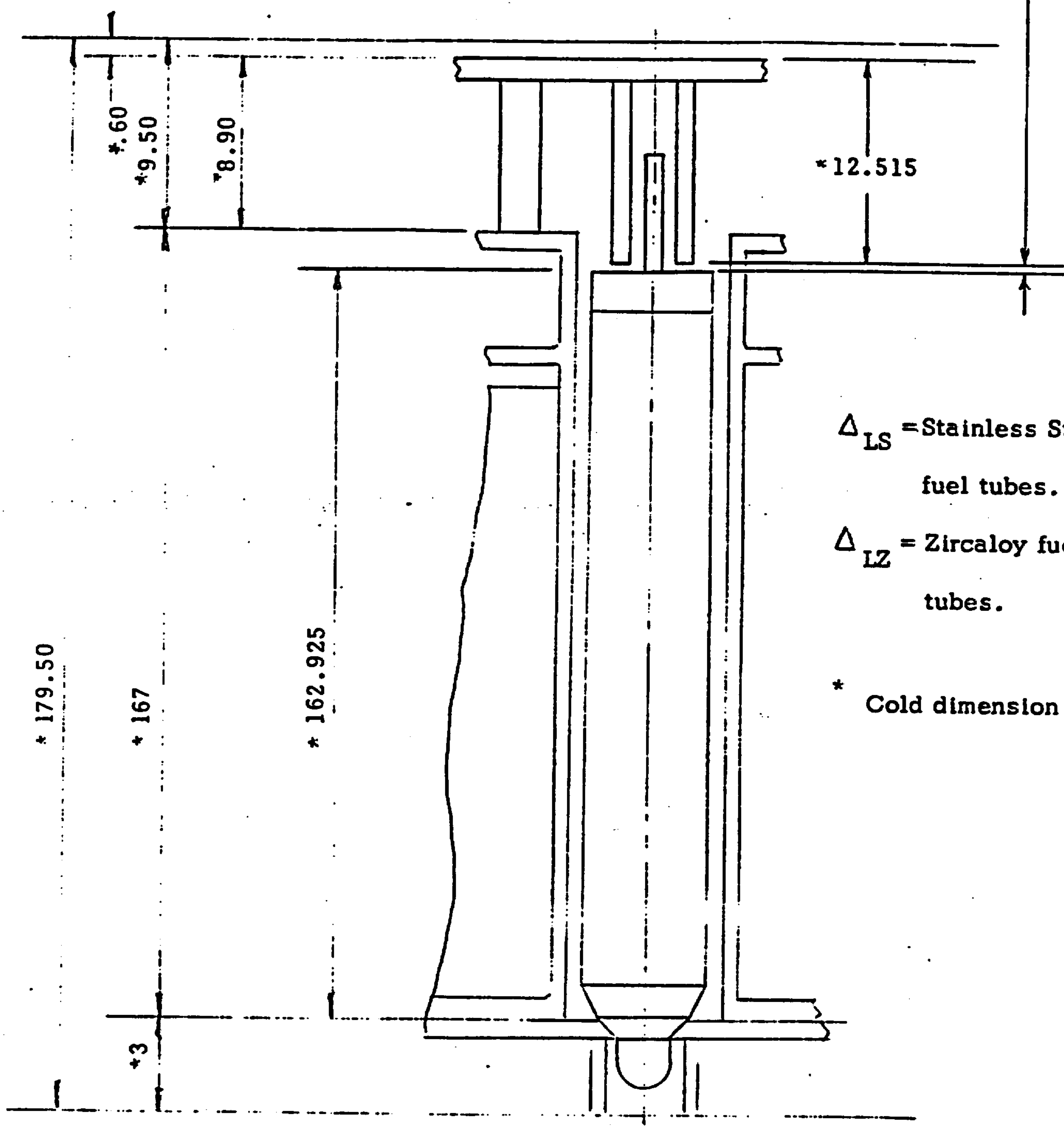
* See Ref. 30. A maximum value is used here in order to obtain a conservative elongation value.

BWR - Axial Expansion

$$\Delta_L = (\Delta_3 + \Delta_4) - (\Delta_2 + \Delta_5 + \Delta_6) = -.2917$$

$$\Delta_{LS} = \Delta_2 - (\Delta_6^1 + \Delta_{1s}) = -.2650$$

$$\Delta_{LZ} = \Delta_2 - (\Delta_6^1 + \Delta_1 + \Delta_1^1) = +.3759$$



Δ_{LS} = Stainless Stee[l] fuel tubes.

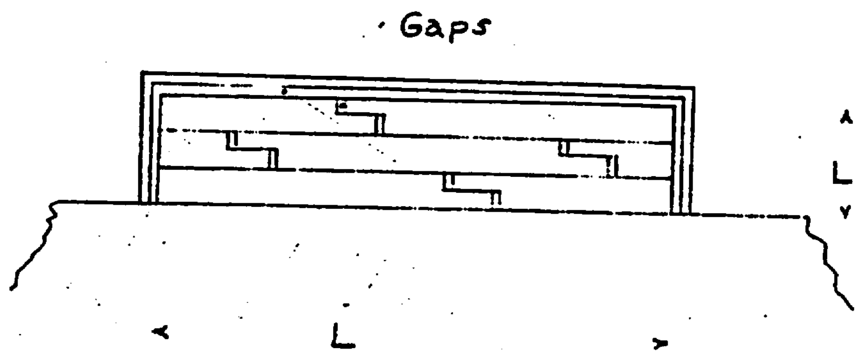
Δ_{LZ} = Zircaloy fuel tubes.

* Cold dimension

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 coeffecient 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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