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Buckling Assessment of Hanford C Farm Tank Bottom Liner for Vacuum loading

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U.S. Department of Energy Contract DE-AC27-99RL14047

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Abstract: This document provides a documented basis for the retrieval operations of C Farm Tanks as tanks approach empty condition with particular application to tanks C-104 and C-106.

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Buckling Assessment of Hanford C Farm Tank Bottom Liner for Vacuum Loading

Prepared for the U.S. Department of Energy
Assistant Secretary for Environmental Management

CH2MHILL
Hanford Group, Inc.
Richland, Washington

Contractor for the U.S. Department of Energy
Office of River Protection under Contract DE-AC06-99RL14047

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Buckling Assessment of Hanford C Farm Tank Bottom Liner for Vacuum Loading

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Yes	No	N/A	
[X]	[ ]	[ ]	Problem completely defined.
[X]	[ ]	[ ]	Necessary assumptions explicitly stated and supported.
[ ]	[ ]	[X]	Computer codes and data files documented.
[X]	[ ]	[ ]	Data used in calculations explicitly stated in document.
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1.0 PURPOSE

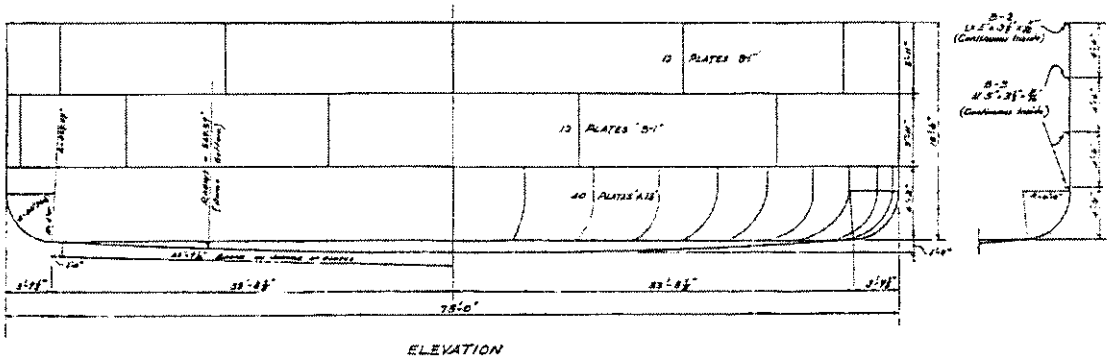
Determine allowable vacuum to prevent bottom uplift (buckling) of carbon steel liner in Series-100, C Farm tanks as tanks are emptied with particular emphasis to lead retrieval tanks 241-C-104 and -106.

2.0 BACKGROUND

The Hanford Site, Series-100, C Farm tanks are 75-foot diameter single-shell tanks (SSTs) with a waste storage capacity of 530,000 gallons. The tanks were constructed in 1943-44 and are identical in design to the B, T, and U Farm tanks and similar to the BX Farm tanks that were constructed in 1946-47 (except nominal wall thickness of steel liner bottom which is 3/8 inches for BX Farm tanks and 1/4 inch for B, C, T, and U Farm tanks).

The SSTs are steel-lined, reinforced concrete vault type underground waste storage tanks. A steel liner covers the bottom, lower knuckle, and cylindrical wall of the concrete vault structure. The steel liner bottom and cylindrical wall was constructed from 1/4-inch thick ASTM A7-39 welded steel plates . The bottom 4-foot radius lower knuckle was constructed from 5/16-inch thick formed welded steel plates. The steel liner bottom is a shallow inverted spherical shell with a one-foot rise from center to tangent with knuckle. The dished bottom liner rests on a 2-inch thick grout layer, a 3-ply asphalt waterproofing layer, and a 6-inch thick reinforced concrete foundation which is supported by the soil. The liner cylindrical wall is circumferential stiffened by angles welded to the inner surface of the liner at approximately 4 ft 6 in. vertical intervals. A 3-ply asphalt waterproofing layer separates the liner from the concrete cylindrical wall. An elevation and cross-sectional view of the steel liner is shown in Figure 1.

Figure 1. Steel Liner for B, C, T, and U Tank Farms. (Drawing D-3)



Current OSD-T-151-00013 (2000) limits for SSTs require that the minimum pressure relative to atmosphere (internal vacuum) not exceed 1 inch water gauge (w.g.) minus the stored waste height, not to exceed -9 inches w.g. Hence, if the tank contains 6 inches of waste, the minimum pressure limit is -5 inches w.g. The minimum pressure limit is provided to prevent buckling of the tank bottom. The upper limit (-9 in. w.g.) on the minimum pressure guards against buckling of the liner cylindrical wall and excessive loading of the concrete dome of the tank.

If active ventilation is required during retrieval activities this limit is difficult to maintain as the waste level decreases and the tank approaches empty. In particular, because the bottom of these SSTs are dished the depth of the waste varies radially from the center of the tank. As the tank is emptied the outer portion of the tank bottom is uncovered while the center portion is covered by waste. Hence, the OSD limits against buckling the bottom liner are reassessed herein in an effort to provide less restrictive constraints during retrieval operations, in particular as the tank approaches empty (bounding case). Note that original design specification and drawings (D-2) specify that liquid is stored at atmospheric pressure. Subsequent use of active ventilation on some tanks, such as 241-C-106 due to high heat load, required that vacuum not exceed liquid level as discussed above.

3.0 ASSUMPTIONS AND CONSIDERATIONS

1. Tank empty (bounding case), no hydrostatic load on bottom of tank steel liner.
2. Model bottom portion of steel liner as shallow spherical cap under uniform net external load due to internal vacuum (discounting self-weight of bottom liner).
3. Neglect any bonding force between grout layer and steel liner.
4. Construction imperfections in spherical shape of bottom liner are considered.
5. Reduction in liner thickness from uniform corrosion of carbon steel liner bottom is considered.
6. Potential thermal aging effects on elastic modulus and yield strength are considered.

4.0 METHOD OF ANALYSIS

Apply empirical buckling correlations available from open literature and ASME Code Case N-284-1 (ASME 1998b) for shallow spherical cap under uniform external differential pressure load to assess the allowable vacuum in excess of self-weight to prevent buckling (uplift) of the steel liner bottom. As the cylindrical wall portion of the steel liner is also subject to the vacuum load, buckling of the liner wall is also evaluated.

5.0 CONCLUSIONS

Internal vacuum from active ventilation should not exceed net pressure from weight of bottom liner and hydrostatic pressure from contained waste. As waste is removed and the tank bottom is uncovered the internal vacuum should not exceed values summarized in Table 0 to prevent buckling of the bottom steel liner depending on the level of corrosion. A factor of safety of 2.0 against buckling was applied to the net vacuum pressure in excess of the self-weight of the tank bottom. Estimates for the general corrosion of the bottom steel liner for tank C-106 are given in Appendix. The amount of corrosion loss in the wall thickness turns out to be the primary consideration. The allowable vacuum is mainly controlled by the pressure to over come the self-weight of the bottom liner. The additional vacuum in excess of the self-weight to cause buckling of the bottom liner is very small, particularly as the wall thickness decreases due to corrosion (see Figure 7). The buckling of the steel liner due to internal vacuum is elastic and is controlled by the buckling strength of the bottom portion of the liner assuming that the general corrosion rate of the side wall is not significantly greater than that of the bottom portion of the liner.

Buckling of the bottom portion of the steel liner does not necessarily result in a liner breach (McCall 1994). Previous analysis (McCall 1994) have shown that the bottom of the steel liner is extremely flexible. The weak point is at the circumferential weld where the nominal 1/4-inch thick bottom joins with the 5/16-inch thick knuckle. Calculations for nominal wall thicknesses (no corrosion considered) show that a net differential pressure of approximately 3.2 lbf/in^2 (88 in. w.g. vacuum) would be required to yield the liner at this location and create a post buckled bubbled shaped volume (approximately 1,400 ft^3) between the bottom liner and the concrete foundation. However, this vacuum level also would cause the cylindrical wall of the steel liner to buckle.

Table 0. Tank C-106 Allowable Vacuum Pressure for Empty Tank.

Corrosion	Case	Wall Thickness (in.)	Self-weight pressure (in. w.g.)	Allowable Vacuum (in. w.g)
No	Nominal + max. mill tolerance	0.28	2.2	2.31
	Nominal	0.25	1.96	2.05
	Nominal + min. mill tolerance	0.24	1.88	1.97
Yes	Min. corrosion (max. wall)	0.23	1.82	1.90
	Average	0.16	1.28	1.32
	Max. corrision (min. wall)	0.09	0.74	0.75

6.0 OPEN ITEMS

Although this analysis was specific to the steel tank bottom of C-106 the conclusions are expected to be generally applicable to C-104 as well as to other 100-Series C, B, T and U Tank Farms, which are identical in design. However, estimates of the general corrosion of the bottom of the steel liners based on stored tank waste and temperature history should be obtained on a case-by-case bases. Results given in Figure 7 can then be applied to the estimated remaining steel plate thickness to obtain the allowable vacuum pressure as tank nears empty condition.

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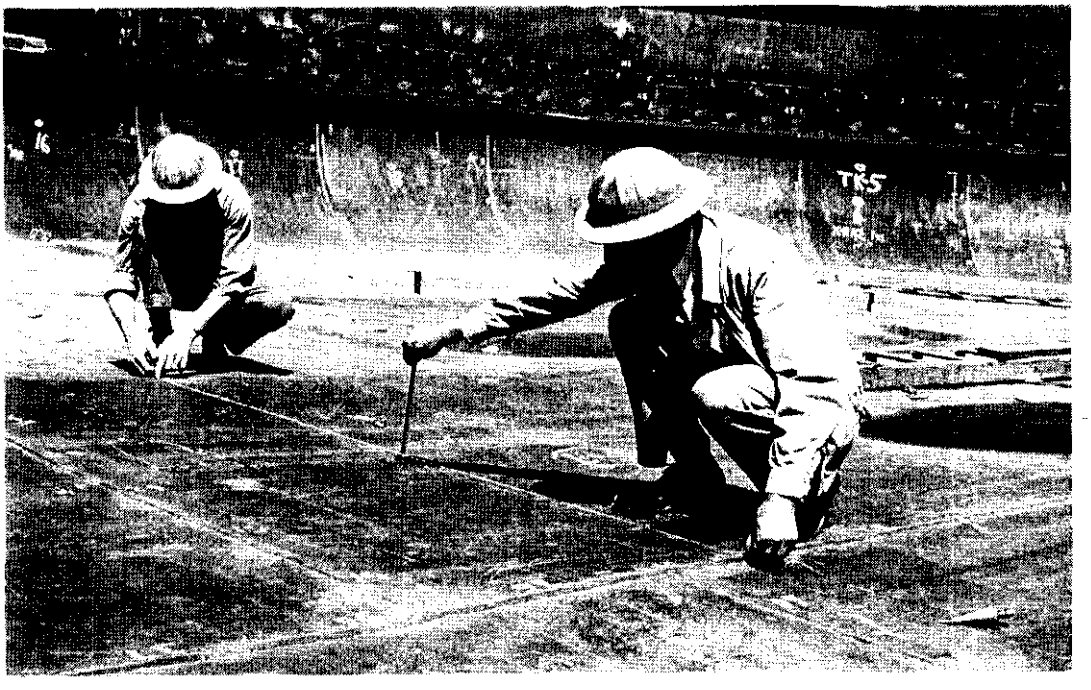
8.0 BUCKLING ANALYSIS

As the SSTs are emptied during waste retrieval operations, compressive membrane stresses in the steel liner bottom could cause the bottom liner to buckle. The sources for the compressive induced stresses include: internal vacuum from active ventilation, restrained radial differential thermal expansion of the bottom liner relative to the concrete outer wall, and residual weld induced stresses (*construction welds were not stress* relieved). However, as the tank waste level is reduced during waste retrieval the heat load is also reduced. Hence, differential thermal expansion induced compressive stresses in the tank bottom are reduced and should not be a significant consideration as the tank approaches empty. However, residual stresses and construction geometric distortions can significantly reduce the buckling strength. Due to local heating during welding, complex thermal stresses occur during welding; and residual stress and distortion result after welding. High tensile residual stresses in areas near the weld may cause stress corrosion cracking under certain conditions; distortion and compressive residual stress in the base metal plate may reduce the buckling strength of the structural component. Weld induced distortions are particularly detrimental for structures that are imperfection sensitive such as the shallow spherical cap that forms the steel liner bottom of these SSTs. Figure 2 shows potential distortions in the bottom liner during construction (BX Farm tank with 3/8-inch bottom plate thickness). Note the puddle areas of standing water that clearly indicate distortions in the tank bottom on the order of the tank thickness or greater.

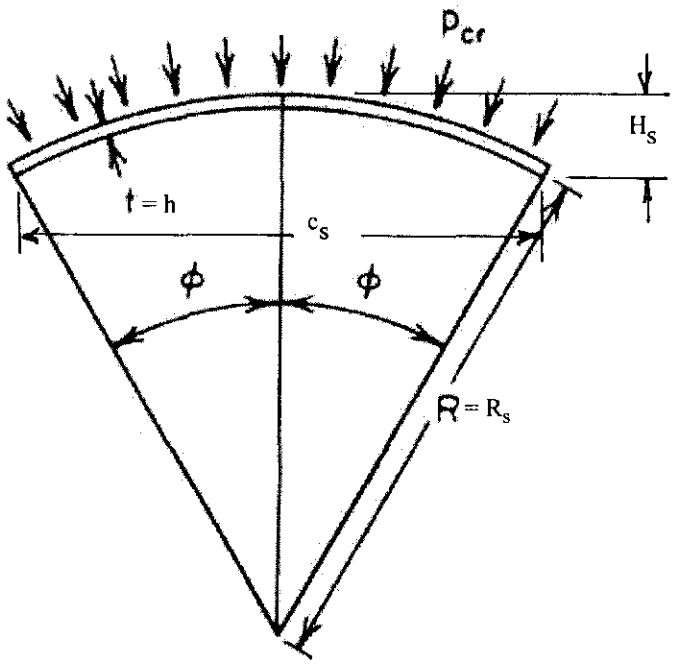
Figure 2. Typical Construction Geometric Imperfections in Single-Shell Tank Steel Liner Tank Bottom . (241-BX Tank Farm, 1309-Negative 1947)



Water puddlle

The tank bottom is modeled as an inverted shallow spherical shell of radius, R_s, thickness, t (symbolically taken as h herein), with rise, H_s, over cord length, c_s, within included angle, $2\phi_s$, and under net uniform external pressure, p (see Figure 3).

Figure 3. Geometry of Spherical Cap Under Uniform External Pressure. (Baker et al. 1972)



$mil := \frac{in}{1000}$

$rad = 57.3\,deg$

$in\_wg := 62.4 \cdot \frac{lbf}{ft^3} \cdot in$

$in\_wg = 0.0361\,psi$

$ksi := 10^3 \cdot psi$

For the C Farm tanks

$H_s := 1 \cdot ft$ — rise of spherical cap over chord length, c, between bottom knuckle tangent point to bottom liner as shown in Figure 1 from Drawing D-3 (1944)

$c_s := 2 \cdot \left[33 \cdot ft + \left(8 + \frac{7}{8}\right) \cdot in \right]$ $\quad c_s = 67.48\,ft$ — length of chord, Drawing D-3 (1944)

$R_s := \frac{1}{2} \cdot \left(H_s + \frac{c_s^2}{4 \cdot H_s} \right)$ $\quad R_s = 569.68\,ft$ — radius of shallow spherical cap (see Figure 3)

$\phi_s := asin\left(\frac{c_s}{2 \cdot R_s}\right)$ $\quad \phi_s = 3.4\,deg$ — included half angle (see Figure 3)

$h_{nom} := \frac{1}{4} \cdot in$ nominal design wall thickness, Drawing D-3 (1944)

$mill_{tol\_min} := -10 \cdot mil$

$mill_{tol\_max} := 30 \cdot mil$

standard plate mill tolerance (ASTM A20)

Material ASTM A7-39 (no longer produced) equivalent to current A283, Grade D (WHC-SD-W320-ANAL-001, 1994)

$S_{y\_min} := 33 \cdot ksi$ minimum specified yield strength (RT)

$\gamma := 490 \cdot \frac{lbf}{ft^3}$ carbon steel unit weight $\gamma = 0.284 \frac{lbf}{in^3}$

$S_{u\_min} := 60 \cdot ksi$ minimum specified tensile strength (RT)

modulus of elasticity for carbon steel with C< 0.30%, Table TM-1 of ASME Section II (1998a)

Yield strength (ksi), Table Y-1 of ASME Section II (1998a)

$i := 0..2$ $vT_i :=$ $vE_i :=$ $vS_{y_i} :=$

Temp. (°F)	Elastic modulus (ksi)	
70	29.5	33
200	28.8	30.1
300	28.3	29.3

$E(T) := 0.9 \cdot linterp(vT, vE, T) \cdot 10^6 \cdot psi$ $S_y(T) := linterp(vT, vS_y, T) \cdot ksi$

$T := 100$ °F $E(T) = 26.405 \times 10^6$ psi $\nu := 0.3$ Poisson's ratio for carbon steel

$S_y(T) = 32.33$ ksi

A 10% reduction in elastic modulus was applied to conservatively account for potential thermal aging effects as discussed in Appendix.

A conservative analysis of potential thickness reduction of the C-106 bottom steel liner due to general corrosion is given in Appendix. The estimated range of corrosion induced thickness loss is 48 to 146 mils from initial construction to present year 2001, i.e.,

$h_{corrosion\_low} := 48 \cdot mil$

$h_{corrosion\_high} := 146 \cdot mil$

Hence, the range of thickness for the bottom liner of C-106 is

$h_{min} := h_{nom} + mill_{tol\_min} - h_{corrosion\_high}$ $h_{min} = 0.09$ in

$h_{max} := h_{nom} + mill_{tol\_max} - h_{corrosion\_low}$ $h_{max} = 0.23$ in

$h_{aver} := \frac{h_{max} + h_{min}}{2}$ $h_{aver} = 0.16$ in

These results are expected to bound results for tank C-104. This will be verified when corrosion analysis specific to C-104 is completed.

8.1 OPEN LITERATURE CORRELATIONS

The classical buckling pressure for a shallow spherical cap under uniform external pressure is given by (Baker et al. 1972)

$$C := \sqrt{3\cdot\left(1-\nu^2\right)}$$

$$p_{cl}(R,h,E) := \frac{2}{C}\cdot E\cdot\left(\frac{h}{R}\right)^2$$

classical buckling pressure for complete spherical shell first developed by Zoelly in 1915 (Baker et al. 1972)

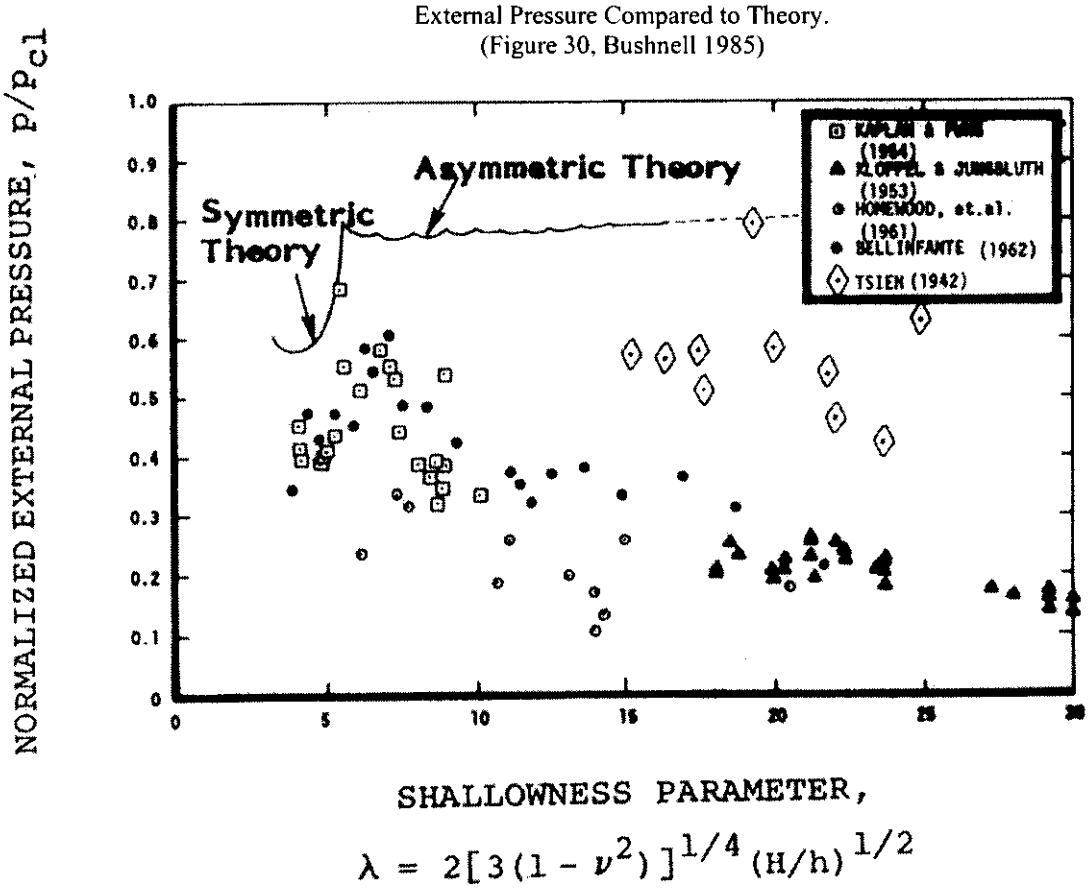
8.1.1 Effect of Imperfections

Early experimental results for clamped spherical caps under external pressure are shown in Figure 4 for critical buckling pressure to classical buckling pressure ratio versus shallowness parameter, λ, given by

$$\lambda(R,h,\phi) := \left[12\cdot\left(1-\nu^2\right)\right]^{\frac{1}{4}}\cdot\left(\frac{R}{h}\right)^{\frac{1}{2}}\cdot 2\cdot\sin\left(\frac{\phi}{2}\right) = 2\cdot\left[3\cdot\left(1-\nu^2\right)\right]^{\frac{1}{4}}\cdot\left(\frac{H}{h}\right)^{\frac{1}{2}} = 2\cdot\sqrt{C\cdot\frac{H}{h}}$$

where H is the rise of the apex of the spherical cap above its base plate.

Figure 4. Experimental Results for Clamped Spherical Caps Under External Pressure Compared to Theory. (Figure 30, Bushnell 1985)



The effects of weld induced residual stress and distortion as well as variations in boundary conditions and load are assumed to be accounted for in the above bounding correlations but it is instructive to quantify their effects. It has generally been accepted that initial imperfections of the test specimens was the principal cause for the wide scatter in experimental data shown in Figure 4. The initial imperfection can be from shape, thickness, boundary conditions, or loads. Each of these quantities has a different effect on the maximum load-carring capacity of the shell. For spherical shells under external pressure, it is well known that the shell is highly imperfection-sensitive to unstable compound (two or more modes) branching (Citerley 1982). An expansion of the governing nonlinear equilibrium equations in terms of potential symmetric and asymmetric buckling modes leads to the following relations (Hutchinson 1967 and Bushnell 1985) for $\alpha = p_{cr}/p_{cl}$ and $\xi = w_o/h$ (normalized imperfection shape amplitude):

2-mode case (symmetric), $\xi_1 > 0,\ \xi_2 = 0$

$$\left(1-\alpha_1\right)^2 = \frac{9\cdot C}{8}\cdot\xi_1\cdot\alpha_1 \quad \Rightarrow \quad \alpha_1\left(\xi_1\right) := 1 + \frac{9}{16}\cdot C\cdot\xi_1 - \frac{3}{16}\cdot\sqrt{32\cdot C\cdot\xi_1 + 9\cdot C^2\cdot\xi_1^{\,2}}$$

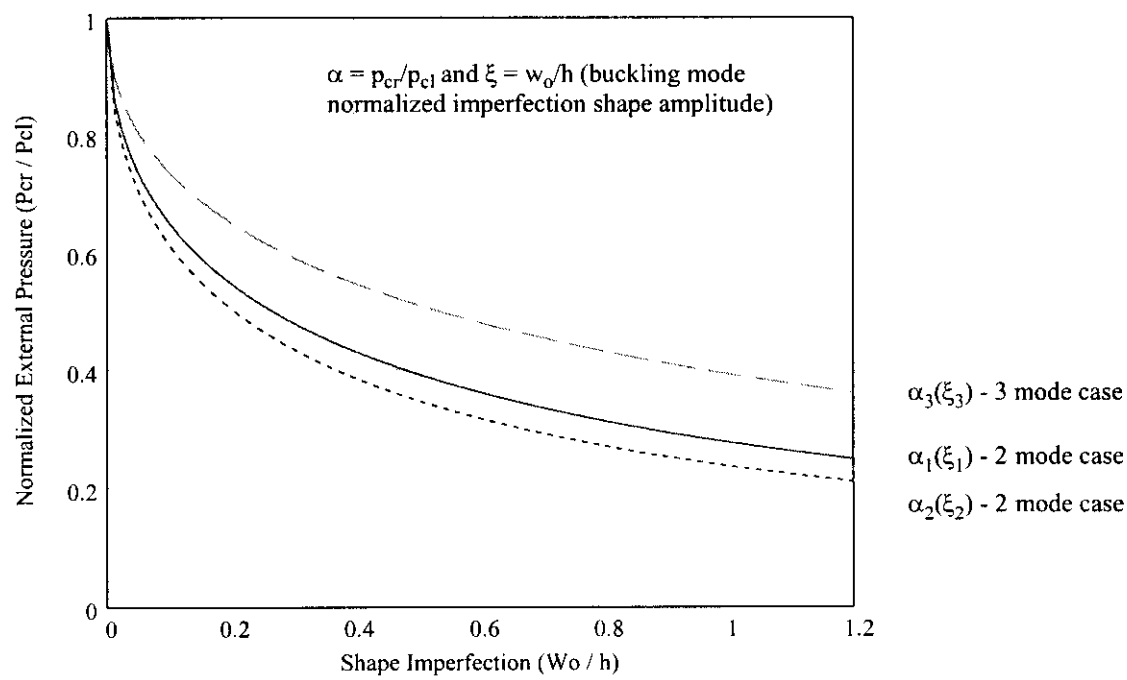
2-mode case (asymmetric), $\xi_1 = 0,\ \xi_2 \neq 0$

$$\left(1-\alpha_2\right)^2 = \frac{27\cdot\sqrt{3}\cdot C}{32}\cdot\left|\xi_2\right|\cdot\alpha_2 \quad \Rightarrow \quad \alpha_2\left(\xi_2\right) := 1 + \frac{27}{64}\cdot\sqrt{3}\cdot C\cdot\left|\xi_2\right| - \frac{3}{64}\cdot\sqrt{384\cdot\sqrt{3}\cdot C\cdot\left|\xi_2\right| + 243\cdot C^2\cdot\left(\left|\xi_2\right|\right)^2}$$

3 mode case

$$\left(1-\alpha_3\right)^2 = \frac{9\cdot C}{16}\cdot\xi_3\cdot\alpha_3 \quad \Rightarrow \quad \alpha_3\left(\xi_3\right) := 1 + \frac{9}{32}\cdot C\cdot\xi_3 - \frac{3}{32}\cdot\sqrt{64\cdot C\cdot\xi_3 + 9\cdot C^2\cdot\xi_3^{\,2}}$$

Figure 5. Effect of Shape Imperfection on Buckling of Spherical Shell under Uniform External Pressure.



Although the results shown in Figure 5 are strictly valid for $\xi << 1$ due to the numerical expansion technique applied in the solution, the results do indicate the effect of initial shape imperfections on reducing the classical buckling load of the spherical shell cap under external uniform pressure. These results are not used here directly but help to explain the experimentally observed reductions in buckling pressure shown in Figure 4.

8.1.2 Effect of Residual Stress

The fabrication process of forming and welding create residual stresses. Residual compressive stresses can reduce the buckling strength of a structure. Residual stresses are difficult to measure but have been correlated with experimental collapse pressure. Elaborate analytical modeling simulations of the welding process to quantitatively predict the effect on buckling have been relatively unsuccessful or impractical. Experimental studies of cylindrical and spherical shells with welded seams have shown that the residual compressive stresses created by the welding process reduce the critical buckling stress less than 5 percent. Whereas, the associated distortion (weld shrink, etc.) create an imperfection that is more significant in reducing the collapse pressure (Harvey 1991). The SST liners were not stress relieved. However, the liner bottoms have been exposed to high temperatures over an extended period. In the case of Tank 241-C-106, the bottom liner was exposed to approximately 300 °F for approximately 10 years (WHC-SD-W320-ANAL-001). Although this combination of time at temperature may cause some reduction in the residual stress, the analysis given in Appendix concluded that any reduction in residual stresses would be insignificant. For analysis purposes any effect of residual stress on reducing the buckling pressure is assumed to be accounted for in experimentally derived "knockdown" factors applied to theoretical buckling pressure.

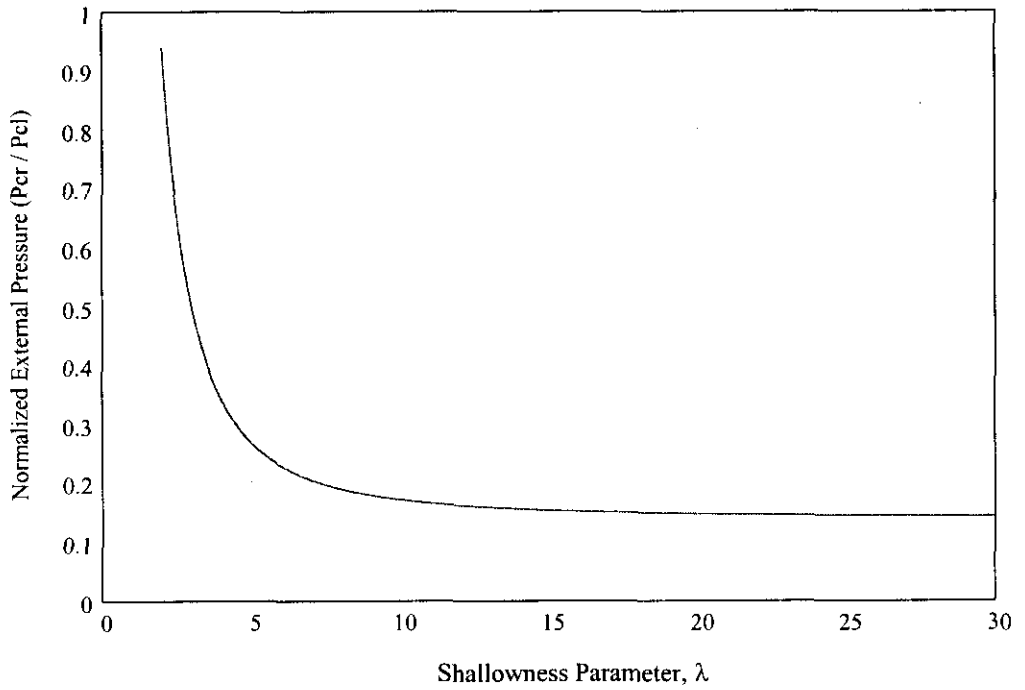
8.1.3 Spherical Shell Buckling Knockdown Factor

The following "knockdown" factor is given by Baker et al. 1972 as a lower bound to the experimental results given in Figure 4

$$\alpha(\lambda) := 0.14 + \frac{3.2}{\lambda^2} \qquad \lambda > 2$$

This equation is plotted in Figure 6 for comparison to experimental data in Figure 4 above.

Figure 6. Recommended Design Buckling Pressure for Spherical Caps Under Uniform External Loading. (Baker et al. 1972)



The resulting net critical buckling pressure from vacuum accounting for self-weight of liner bottom and corresponding critical stress is given as

$p_w(h) := \gamma \cdot h$ pressure from self-weight of liner bottom

$p_{cr}(R,h,\phi,T,FS) := \frac{\alpha(\lambda(R,h,\phi))}{FS} \cdot p_{cl}(R,h,E(T)) + p_w(h)$ buckling net critical vacuum pressure

$\sigma_{cr}(R,h,\phi,T,FS) := p_{cr}(R,h,\phi,T,FS) \cdot \frac{R}{2 \cdot h}$ buckling critical stress

including appropriate factor of safety against buckling on net vacuum pressure in excess of self-weight of liner bottom, i.e.,

$FS_i :=$

1	
1.5	
2	ASME Code Case N-284-1 required factor of safety for local buckling
2.4	ASME Code Case N-284-1 required factor of safety for global buckling

Corresponding results for the C-106 bottom liner are shown in Table 1 below for both with and without corrosion.

Table 1a. Allowable Vacuum of Bottom Spherical Cap Portion of Steel Liner of Tank C-106 without Corrosion for Empty Tank Condition.

$R := R_s$ $\phi := \phi_s$ $T = 100$ $S_y(T) = 32.33\,ksi$

$R = 569.68\,ft$ $\phi = 3.4\,deg$ $E(T) = 26.405 \times 10^6\,psi$

$h := h_{nom} + mill_{tol\_max}$ $h = 0.28\,in$ $\lambda(R,h,\phi) = 16.8$ $\alpha(\lambda(R,h,\phi)) = 0.15$ $\frac{R}{h} = 24415$

$p_{cl}(R,h,E(T)) = 1.48\,in\_wg$ $p_w(h) = 2.2\,in\_wg$

FS =

	0
0	1
1	1.5
2	2
3	2.4

$\sigma_{cr}(R,h,\phi,T,FS) =$ ksi

	0
0	1.068
1	1.035
2	1.019
3	1.011

$p_{cr}(R,h,\phi,T,FS) =$ in_wg

	0
0	2.42
1	2.35
2	2.31
3	2.29

$h := h_{nom}$ $h = 0.25\,in$ $\lambda(R,h,\phi) = 17.8$ $\alpha(\lambda(R,h,\phi)) = 0.15$ $\frac{R}{h} = 27345$

$p_{cl}(R,h,E(T)) = 1.18\,in\_wg$ $p_w(h) = 1.96\,in\_wg$

FS =

	0
0	1
1	1.5
2	2
3	2.4

$\sigma_{cr}(R,h,\phi,T,FS) =$ ksi

	0
0	1.057
1	1.028
2	1.013
3	1.006

$p_{cr}(R,h,\phi,T,FS) =$ in_wg

	0
0	2.14
1	2.08
2	2.05
3	2.04

$h := h_{nom} + mill_{tol\_min}$ $h = 0.24\,in$ $\lambda(R,h,\phi) = 18.2$ $\alpha(\lambda(R,h,\phi)) = 0.15$ $\frac{R}{h} = 28484$

$p_{cl}(R,h,E(T)) = 1.09\,in\_wg$ $p_w(h) = 1.88\,in\_wg$

FS =

	0
0	1
1	1.5
2	2
3	2.4

$\sigma_{cr}(R,h,\phi,T,FS) =$ ksi

	0
0	1.053
1	1.025
2	1.011
3	1.004

$p_{cr}(R,h,\phi,T,FS) =$ in_wg

	0
0	2.05
1	1.99
2	1.97
3	1.95

Table 1b. Allowable Vacuum of Bottom Spherical Cap Portion of Steel Liner of Tank C-106 with Corrosion for Empty Tank Condition.

$h := h_{max}$ $\quad h = 0.23\,in$ $\quad \lambda(R,h,\phi) = 18.5$ $\quad \alpha(\lambda(R,h,\phi)) = 0.15$ $\quad \frac{R}{h} = 29466$

$p_{cl}(R,h,E(T)) = 1.02\,in\_wg$ $\quad p_w(h) = 1.82\,in\_wg$

FS =	$\sigma_{cr}(R,h,\phi,T,FS) =$ ksi	$p_{cr}(R,h,\phi,T,FS) =$ in_wg
1	1.05	1.97
1.5	1.023	1.92
2	1.01	1.9
2.4	1.003	1.89

$h := h_{aver}$ $\quad h = 0.16\,in$ $\quad \lambda(R,h,\phi) = 22.1$ $\quad \alpha(\lambda(R,h,\phi)) = 0.15$ $\quad \frac{R}{h} = 41940$

$p_{cl}(R,h,E(T)) = 0.5\,in\_wg$ $\quad p_w(h) = 1.28\,in\_wg$

FS =	$\sigma_{cr}(R,h,\phi,T,FS) =$ ksi	$p_{cr}(R,h,\phi,T,FS) =$ in_wg
1	1.025	1.35
1.5	1.006	1.33
2	0.997	1.32
2.4	0.993	1.31

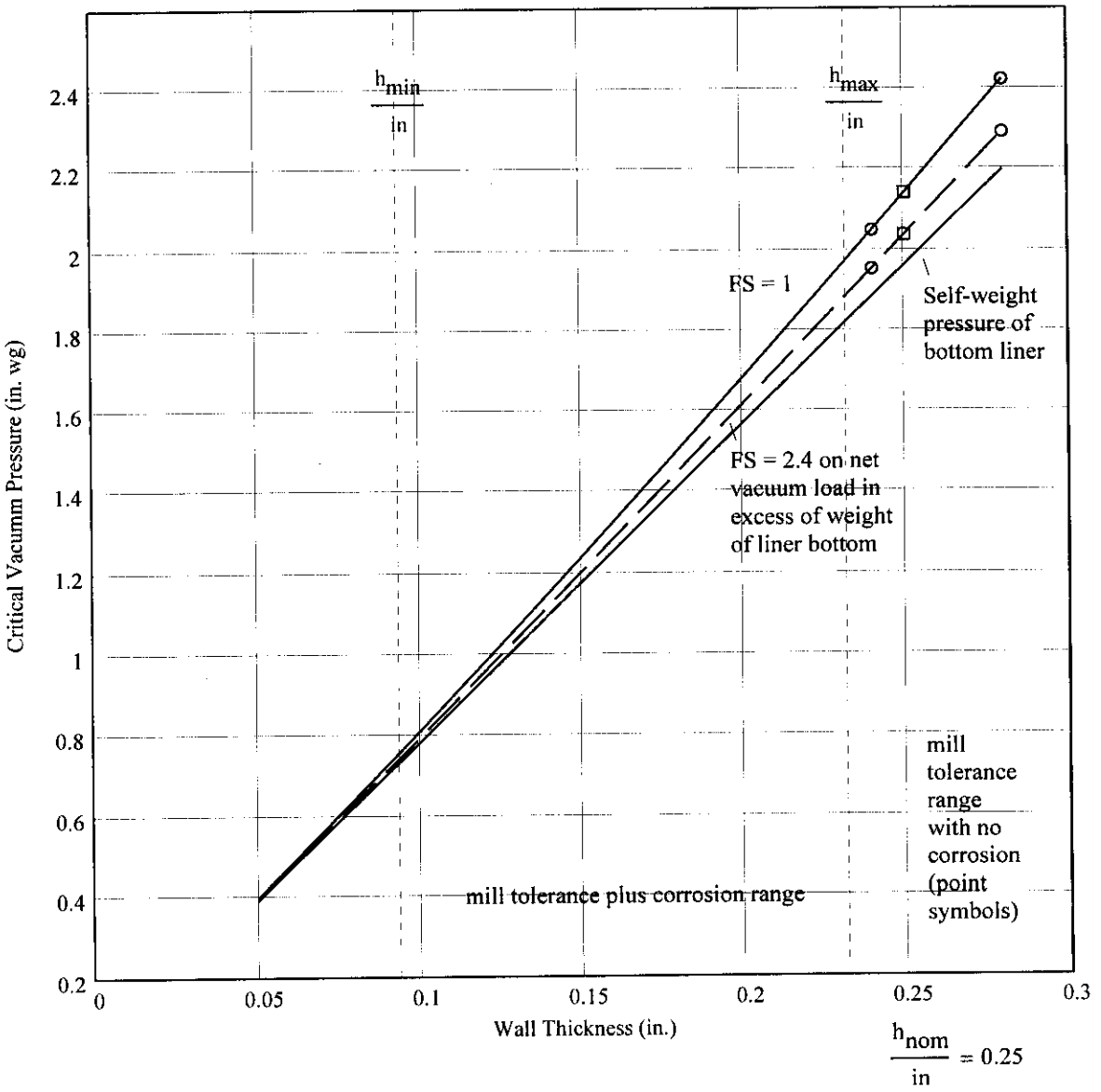
$h := h_{min}$ $\quad h = 0.09\,in$ $\quad \lambda(R,h,\phi) = 29$ $\quad \alpha(\lambda(R,h,\phi)) = 0.14$ $\quad \frac{R}{h} = 72725$

$p_{cl}(R,h,E(T)) = 0.17\,in\_wg$ $\quad p_w(h) = 0.74\,in\_wg$

FS =	$\sigma_{cr}(R,h,\phi,T,FS) =$ ksi	$p_{cr}(R,h,\phi,T,FS) =$ in_wg
1	1.001	0.76
1.5	0.99	0.75
2	0.985	0.75
2.4	0.982	0.75

Results are summarized in Figure 7 for FS = 1 and 2.4.

Figure 7. Allowable Vacuum of Bottom Spherical Cap Portion of Steel Liner of Tank C-106 with and without Corrosion for Empty Tank Condition.



Note that the additional vacuum in excess of the self-weight to cause buckling of the bottom liner is very small, particularly as the wall thickness decreases due to corrosion..

8.2 ASME CODE APPROACH (Alternate Calculation)

According to HW-1946 (1943), *Specification for Composite Storage Tanks - Bldg. #241 at Hanford Engineer Works, Project 9536*, the steel liners for the B, C, T, and U Tank Farms were designed, fabricated, and erected in accordance with the then current *Standard Specifications for Elevated Steel Water Tanks, Standpipes, and Reservoirs* as promulgated by the American Waterworks Association in conjunction with The American Welding Society where applicable and not at variance with any portion of the HW-1946 specification. Although the steel liners were not designed to the ASME, Section III, Division 1, requirements for the design of metal containment shell structures, the ASME (1998b) Code Case N-284-1, *Metal Containment Shell Buckling Design Method, Class MC, Section III, Division 1*, provides an acceptable method for determining the buckling capacity of shells that are fabricated from metal plates where the plates are cold or hot formed and joined by welding. The stability criteria is based upon classical linear bifurcation analysis which has been reduced by capacity reduction factors which account for the effects of imperfections and non linearity in geometry and boundary conditions and by plasticity reduction factors which account for non linearity in material properties. The reduction factors are determined from lower-bound buckling values from available test data. The rules from Case N-284-1 are strictly valid for shells with radius-to-thickness ratios up to 1000, shell thickness of 1/4 inch or greater, and built to fabrication requirements of ASME NE-4222.

8.2.1 Spherical Shell

Capacity Reduction Factors

equal biaxial compression

$$M(R,h,\phi) := \frac{R\cdot 2\cdot\phi}{\sqrt{R\cdot h}} \quad \text{(spherical cap)}$$

$$\alpha_{2L}(R,h,\phi) := \begin{cases} m \leftarrow M(R,h,\phi) \\ 0.627 \ \text{ if } m < 1.5 \\ 0.837 - 0.14\cdot m \ \text{ if } 1.5 \le m < 1.73 \\ \dfrac{0.826}{m^{0.6}} \ \text{ if } 1.73 \le m < 23.6 \\ 0.124 \ \text{ otherwise} \end{cases}$$

Plasticity Reduction Factors

$$\Delta(\alpha_{ij},\sigma_{iej},S_y) := \frac{\alpha_{ij}\cdot\sigma_{iej}}{S_y}$$

meridional and/or hoop compression

$$\eta_\phi(\Delta) := \begin{cases} 1 \ \text{ if } \Delta \le 0.55 \\ \dfrac{0.45}{\Delta} + 0.18 \ \text{ if } 0.55 < \Delta \le 1.6 \\ \dfrac{1.31}{1 + 1.15\cdot\Delta} \ \text{ if } 1.6 < \Delta < 6.25 \\ \dfrac{1}{\Delta} \ \text{ otherwise} \end{cases}$$

Theoretical Elastic Instability Stress

equal biaxial compression

$$C'(R,h,\phi) := \begin{cases} m \leftarrow M(R,h,\phi) \\ 0.630 \ \text{ if } m < 1.5 \\ \dfrac{0.904}{m^2} + 0.1013\cdot m^2 \ \text{ if } 1.5 \le m < 1.73 \\ 0.605 \ \text{ otherwise} \end{cases}$$

$$\sigma_{eL}(R,h,\phi,E) := C'(R,h,\phi)\cdot E\cdot\frac{h}{R}$$

Allowable Factored Stress

$$\sigma_{2a}(R,h,\phi,E,S_y,FS) := \frac{\eta_\phi\left(\Delta\left(\alpha_{2L}(R,h,\phi),\sigma_{eL}(R,h,\phi,E),S_y\right)\right)}{FS}\cdot\alpha_{2L}(R,h,\phi)\cdot\sigma_{eL}(R,h,\phi,E)$$

allowable net negative internal pressure and stress

$$p'_{2a}(R,h,\phi,E,S_y,FS) := 2\cdot\sigma_{2a}(R,h,\phi,E,S_y,FS)\cdot\frac{h}{R}$$

$$p'_{cra}(R,h,\phi,T,FS) := p'_{2a}(R,h,\phi,E(T),S_y(T),FS) + p_w(h)$$

$$\sigma'_{cr}(R,h,\phi,T,FS) := p'_{cra}(R,h,\phi,T,FS)\cdot\frac{R}{2\cdot h}$$

$R := R_s$ $\phi := \phi_s$ $T = 100$ $S_y(T) = 32.33\,ksi$

$R = 569.68\,ft$ $\phi = 3.4\,deg$ $E(T) = 26.405\times10^6\,psi$

$h := h_{max}$ $h = 0.23\,in$ $M(R,h,\phi) = 20.3$ $\alpha_{2L}(R,h,\phi) = 0.14$ $C'(R,h,\phi) = 0.61$ $\frac{R}{h} = 29466$

FS =

	0
0	1
1	1.5
2	2
3	2.4

$\sigma'_{cr}(R,h,\phi,T,FS) =$ ksi

	0
0	1.043
1	1.018
2	1.006
3	1

$p'_{cra}(R,h,\phi,T,FS) =$ in_wg

	0
0	1.96
1	1.91
2	1.89
3	1.88

$h := h_{aver}$ $h = 0.16\,in$ $M(R,h,\phi) = 24.3$ $\alpha_{2L}(R,h,\phi) = 0.12$ $C'(R,h,\phi) = 0.61$ $\frac{R}{h} = 41940$

FS =

	0
0	1
1	1.5
2	2
3	2.4

$\sigma'_{cr}(R,h,\phi,T,FS) =$ ksi

	0
0	1.016
1	1.001
2	0.993
3	0.989

$p'_{cra}(R,h,\phi,T,FS) =$ in_wg

	0
0	1.34
1	1.32
2	1.31
3	1.31

$h := h_{min}$ $h = 0.09\,in$ $M(R,h,\phi) = 32$ $\alpha_{2L}(R,h,\phi) = 0.12$ $C'(R,h,\phi) = 0.61$ $\frac{R}{h} = 72725$

FS =

	0
0	1
1	1.5
2	2
3	2.4

$\sigma'_{cr}(R,h,\phi,T,FS) =$ ksi

	0
0	0.996
1	0.987
2	0.983
3	0.981

$p'_{cra}(R,h,\phi,T,FS) =$ in_wg

	0
0	0.76
1	0.75
2	0.75
3	0.75

Results compare favorably with results in Table 1b above.

8.2.2 Buckling Check for Cylindrical Wall of Steel Liner per ASME Code Case N-284-1

Cylindrical Shell

Capacity Reduction Factors

hoop compression

$\alpha_{\theta L} := 0.8$

Plasticity Reduction Factors

hoop compression

$$\eta_\theta(\Delta) := \begin{cases} 1 & \text{if } \Delta \le 0.67 \\ \dfrac{2.53}{1 + 2.29\cdot\Delta} & \text{if } 0.67 < \Delta < 4.2 \\ \dfrac{1}{\Delta} & \text{otherwise} \end{cases}$$

Theoretical Elastic Instability Stress

hoop compression *no end pressure, σ_ϕ (p) = 0*

$$C_{\theta r}(R,h,L_\phi) := \left| \begin{array}{l} M_\phi \leftarrow \dfrac{L_\phi}{\sqrt{R\cdot h}} \\ 1.616 \ \text{ if } M_\phi \le 1.5 \\ \dfrac{2.41}{M_\phi^{1.49} - 0.338} \ \text{ if } 1.5 < M_\phi < 3.0 \\ \dfrac{0.92}{M_\phi - 1.17} \ \text{ if } 3.0 \le M_\phi < 1.65\cdot\dfrac{R}{h} \\ 0.275\cdot\dfrac{h}{R} + \dfrac{2.1}{M_\phi^{\,4}}\cdot\left(\dfrac{R}{h}\right)^3 \ \text{ otherwise} \end{array} \right.$$

$$\sigma_{\theta reL}(R,h,L_\phi,E) := C_{\theta r}(R,h,L_\phi)\cdot E\cdot\frac{h}{R}$$

Allowable Factored Stress

$$\sigma_{\theta ra}(R,h,L_\phi,E,S_y,FS) := \frac{\eta_\theta\left(\Delta\left(\alpha_{\theta L},\sigma_{\theta reL}(R,h,L_\phi,E),S_y\right)\right)}{FS}\cdot\alpha_{\theta L}\cdot\sigma_{\theta reL}(R,h,L_\phi,E)$$

allowable net negative internal pressure (no axial compressive stress)

$$\Delta p_{ra}(R,h,L_\phi,E,S_y,FS) := \sigma_{\theta ra}(R,h,L_\phi,E,S_y,FS)\cdot\frac{h}{R}$$

$R_{cyl} := \frac{75}{2}\cdot ft$ $\quad R := R_{cyl}$ radius of steel liner wall

$L_{cyl} := 4.5\cdot ft$ $\quad L_\phi := L_{cyl}$ axial length between circumferential stiffeners

Assuming same wall thickness reduction due to general corrosion for cylindrical steel liner portion with equally spaced circumferential stiffeners, results in following allowable vacuum.

Table 2. Allowable Vacuum for Cylindrical Steel Liner.

$R = 450\,in$ $L_\phi = 54\,in$ $T = 100$ $S_y(T) = 32.33\,ksi$

$E(T) = 26.405 \times 10^6\,psi$

$h := h_{max}$ $h = 0.23\,in$ $C_{\theta r}(R, h, L_\phi) = 0.22$ $\frac{R}{h} = 1940$

$\eta_\theta\left(\Delta\left(\alpha_{\theta L}, \sigma_{\theta reL}(R, h, L_\phi, E(T)), S_y(T)\right)\right) = 1$ => elastic buckling

FS =

	0
0	1
1	1.5
2	2
3	2.4

$\Delta p_{ra}(R, h, L_\phi, E(T), S_y(T), FS) =$ in_wg

	0
0	34.76
1	23.17
2	17.38
3	14.48

$h := h_{aver}$ $h = 0.16\,in$ $C_{\theta r}(R, h, L_\phi) = 0.18$ $\frac{R}{h} = 2761$

$\eta_\theta\left(\Delta\left(\alpha_{\theta L}, \sigma_{\theta reL}(R, h, L_\phi, E(T)), S_y(T)\right)\right) = 1$ => elastic buckling

FS =

	0
0	1
1	1.5
2	2
3	2.4

$\Delta p_{ra}(R, h, L_\phi, E(T), S_y(T), FS) =$ in_wg

	0
0	13.75
1	9.17
2	6.88
3	5.73

$h := h_{min}$ $h = 0.09\,in$ $C_{\theta r}(R, h, L_\phi) = 0.13$ $\frac{R}{h} = 4787$

$\eta_\theta\left(\Delta\left(\alpha_{\theta L}, \sigma_{\theta reL}(R, h, L_\phi, E(T)), S_y(T)\right)\right) = 1$ => elastic buckling

FS =

	0
0	1
1	1.5
2	2
3	2.4

$\Delta p_{ra}(R, h, L_\phi, E(T), S_y(T), FS) =$ in_wg

	0
0	3.29
1	2.19
2	1.65
3	1.37

Alternate Calculation for Cylindrical Wall of Steel Liner

Per Young, 1989, "Roak's Formulas for Stress and Strain," Table 35-19b

$$q'(R,h,L_\phi,E,FS) := \frac{0.807}{FS}\cdot\frac{E\cdot h^2}{L_\phi\cdot R}\cdot\left[\left(\frac{1}{1-\nu^2}\right)^3\cdot\frac{h^2}{R^2}\right]^{\frac{1}{4}}$$ approximate formula, R/h > 10

$R = 450\,in$ $L_\phi = 54\,in$ $T = 100$ $S_y(T) = 32.33\,ksi$

$E(T) = 26.405 \times 10^6\,psi$ $\nu = 0.3$

$h := h_{aver}$ $h = 0.16\,in$ $C_{\theta r}(R,h,L_\phi) = 0.18$ $\frac{R}{h} = 2761$

FS =	$q'(R,h,L_\phi,E(T),FS) =$ in_wg
1	13.18
1.5	8.79
2	6.59
2.4	5.49

Results compare favorably with results in Table 2 above.

APPENDIX

Interoffice Memo 7G500-01-RPA-047

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INTEROFFICE MEMO

CH2MHILL
Hanford Group, Inc.

From:	Maintenance/Reliability Engineering	7G500-01-RPA-047
Phone:	373-0785	
Date:	July 18, 2001	
Subject:	WALL THINNING OF TANK 241-C-106 BOTTOM BY GENERAL CORROSION	
To:	A. L. Friberg	R3-83
Copies:	L. J. Julyk	R3-83
	D. H. Shuford	R3-83

Per your request, I performed an evaluation of thickness loss experienced by the bottom of tank 241-C-106 from general corrosion during the period from the start of operations till the recent sluicing and retrieval of waste from tank. The evaluation also includes the effect of thermal ageing on residual stresses and elastic modulus of tank steel. The attached report describes the results of the evaluation.

Based on the report results, the wall thinning of tank 241-C-106 bottom is estimated to range from a low of 48 mils to a high of 146 mils. Thermal ageing at a maximum temperature of 310^0F for 10 years is not expected to relieve the residual stresses or significantly decrease the room temperature elastic modulus. However, the tank steel is expected to lose some of its ductility due to strain age embrittlement.

Should you have any questions concerning the report, please feel free to contact me at 373-0785.

Signed copy on file

R. P. Anantatmula
Principal Engineer

fam

Attachment

7G500-01-RPA-047

Attachment

WALL THINNING OF TANK 241-C-106 BOTTOM BY GENERAL CORROSION

Consisting of 6 pages
including cover page

WALL THINNING OF TANK 241-C-106 BOTTOM BY GENERAL CORROSION

An evaluation was performed of thickness loss experienced by the bottom of tank 241-C-106 from general corrosion during the period from the start of operations till the recent sluicing and retrieval of waste from tank. The evaluation also includes the effect of thermal ageing on residual stresses and elastic modulus of tank steel. The results of the evaluation are described in detail in the following.

According to the tank characterization report (Schreiber 1996), the first waste transferred to the tank was the metal waste (MW) from B-Plant and the tank remained full until 1953, when waste was transferred to U-Plant for uranium recovery. The tank received uranium recovery waste (UR) in 1954, followed by PUREX coating waste (CWP1) in 1958, B-Plant AR vault sludge (AR) in 1969-72, and B-Plant low level waste (BL) in 1974-76. After 1976, the transfers included B-Plant strontium recovery waste (B), supernatant and complexed evaporator wastes from tank 241-A-102. Water was also periodically added to the tank to control the waste temperature. Table 1, reproduced from Place (1998), shows sludge layers and predicted current inventory in tank C-106 according to Agnew et al. (1997). The tank was sluiced and the waste in the tank was transferred to tank 102-AY very recently.

Table 1. Predicted Current Inventory in Tank 241-C-106 (Agnew et al. 1997).

Waste type	Waste volume m³ (kgal)	Sludge elevation[a] (cm)	Percent of sludge layer
UR	56.7 (15)	0 – 13.7	0 – 7.6%
CWP1	128.7(34)	13.7 – 45.2	7.6 – 24.9%
AR	242.2 (64)	45.2 – 104.1	24.9 – 57.3%
BL	75.7 (20)	104.1 – 122.7	57.3 – 67.6%
Unknown (BL)	121.1 (32)	122.7 – 152.1	67.6 – 83.8%
Unknown (AR)	121.1 (32)	152.1 – 181.6	83.8 – 100%
Supernatant	121.1 (32)		

AR = B Plant AR vault sludge
BL = B Plant low-level waste
CWP1 = PUREX coating waste (1956 – 1960)
PUREX = Plutonium-uranium extraction
UR = Uranium recovery

[a]Referenced to the bottom edge of the tank assuming a flat bottom configuration. The bottom centerline of tank 241-C-106 is actually 30.5 cm (12 in.) below the bottom edge of the tank due to a dished bottom configuration. Since the top surface of the sludge is known to be indented or pancaked in the middle, a flat bottom tank configuration will be assumed in this analysis. Actually, the dished bottom contains about 12,000 gal of sludge, which corresponds to about 6 percent of the total sludge inventory.

Based on the composition listed in Hill et al. (1995) and the corrosivity factor defined by Anantatmula et al. (1994), MW is benign to tank steel from a corrosion perspective. The Best Basis Inventory (BBI) estimate report (Place 1998) indicates that the UR waste contains a high concentration of nitrate compared to the nitrite, while the CWP1 waste contains almost equal amounts of each component. However, the AR and BL wastes contain a very high nitrite/nitrate

ratio, which is extremely favorable for inhibiting tank corrosion. Based on the total inventories of nitrate, nitrite and hydroxide reported in Place (1998), the waste in tank 241-C-106, as it existed in recent years, was quite benign for tank corrosion (Anantatmula et al. 1994). Therefore, any tank bottom corrosion by waste would have occurred during the early stages of operation after UR waste was introduced into the tank. Aqueous corrosion of tank bottom also occurred prior to waste storage when the tank was filled with water during construction.

The high nitrate concentration of UR waste could have caused stress corrosion cracking (SCC) of the tank but the tank temperature (Julyk et al. 1994) was low enough ($\leq 50^0$C) to preclude SCC (Ondrejcin 1978). However, the tank must have experienced some pitting corrosion under these conditions. Because nitrate ions are not as aggressive as chloride ions, the pitting corrosion rates will not be linear and decrease with time. Initial general corrosion rates at these high nitrate concentrations can be as high as 15 mils per year (mpy) in nitrate containing solutions of neutral to 9.5 pH (NACE 1985, Revie 2000). However, if the pH > 9.5 or if there is not enough oxygen available for the corrosion reaction, the general corrosion rate is expected to decrease to lower and possibly negligible values. Lack of oxygen near the tank bottom is expected to decrease both the general corrosion and pitting corrosion. On this basis, a conservative linear general corrosion rate of 8 mpy was assumed for the time period covering the UR and CWP1 waste transfers. It is well known that radiolysis of waste solutions decreases the concentration of nitrate while simultaneously increasing the concentration of nitrite (Walker et al. 1992). However, because cesium and strontium concentrations are low in UR waste, this process is expected to be somewhat slow. On the other hand, although the cesium concentration is low for CWP1 waste, the strontium concentration is higher compared to the UR waste. The strontium concentration is very high for the AR sludge, which was responsible for the high heat load in the tank. BL waste also has high strontium concentration with no cesium. It should be pointed out that although this reversal of nitrite and nitrate concentrations by radiolysis is a slow process during the period covering the UR and CWP1 waste transfers, when combined with low oxygen availability this would have a definite reducing effect on tank bottom corrosion.

Table 2 summarizes the results of the evaluation of general wall thinning of tank 241-C-106 bottom. The results include a high and a low value for general corrosion in each case. The general corrosion depth by aqueous corrosion from 1943-47 was evaluated using aqueous corrosion model available in the literature (Lee at al. 1996). Because the MW waste is benign toward carbon steel, a high value of 1 mpy and a low value of 0.25 mpy were assumed during the period 1947-54. For period covering 1954-69, which includes UR and CWP1 wastes, a high value of 6 mpy and a low value of 2 mpy were assumed. Although wastes stored after 1969 were extremely benign toward carbon steel, conservatism is already built into the estimates for the period 1954-69, and ultrasonic testing data on double-shell tanks containing wastes compliant with waste specifications indicated little or no wall thinning, a low value of general corrosion rate of 0.25 mpy and a high value of 1 mpy were assumed for period covering 1969 to sluicing and retrieval, similar to the period covering the MW waste. As can be seen from Table 2, the wall thinning of tank 241-C-106 bottom ranges from a low of 48 mils to a high of 146 mils.

Table 2. Evaluation of General Corrosion Depth for 241-C-106 Tank Bottom.

Time period	Waste stored	Corrosion Depth (mils) Low	High	Assumptions/Comments
1943-47	Water	8	17	Aqueous corrosion[a]
1947-54	Metal waste	2	7	Benign waste. Pumped out in 1954. Assumed 1 mpy for high and 0.25 mpy for low.
1954-58	UR waste	8	24	High nitrate in UR, pH=9.8, lack of oxygen and CWP1 more compatible with carbon steel. Assumed 6 mpy for high and 2 mpy for low.
1958-69	CWP1 waste	22	66	
1969-74	AR sludge	1	5	Lack of oxygen, high nitrite and pH=9.8. Composite benign waste.
1974-76	BL waste	1	2	
1976-Retrieval	Composite benign waste.	6	25	Lack of oxygen, high nitrite and pH=9.8. Benign waste. 1 mpy for high and 0.25 mpy for low
Total corrosion depth		48	146	

[a] Used aqueous corrosion model equation developed by Lee et al., (1996) based on long-term corrosion data on cast iron and carbon steel available in the literature.

The following paragraphs discuss the effect of thermal ageing on residual stresses, elastic modulus and ductility.

Relief of residual stresses is a time-temperature related phenomenon. The thermal effect on residual stresses is usually expressed as the Larson-Miller (L-M) parameter (ASM 1985) as

$$P = 1.8T(20+\log t)10^{-3}$$

where P is the L-M parameter, T is temperature in ^{0}K and t is time in hours. Residual stresses in carbon steel are generally relieved by heating the steel to 1100^{0}F and maintaining the temperature for at least 1 hour. In order to obtain the same effect at 310^{0}F, which is the maximum temperature seen by tank 241-C-106 (Julyk et al. 1994), the time required from the above equation is 3.38 x 10^{20} hours, which is several orders of magnitude larger than 60 years, which is the approximate age of tank 241-C-106. Therefore, it can be very easily concluded that the high temperatures experienced by tank 241-C-106 will not relieve the residual stresses.

As far as the elastic modulus is concerned, raising the temperature of carbon steel lowers its elastic modulus (ASM 1985). If we assume that the thermal effect on Young's modulus can also be expressed as a time-temperature related phenomenon, then the effect of ageing for 10 years at 310^{0}F can easily be calculated using the L-M parameter equation above. Based on the above equation, the thermal effect of ageing the carbon steel at 310^{0}F for 10 years is equivalent to performing a modulus test for 1 hour at 500^{0}F. The net thermal effect would be an approximate 7% reduction in elastic modulus (ASM 1985) compared to the instantaneous value at 310^{0}F, which is not significant. However, if the tank cools to room temperature because of retrieval of heat generating waste, the modulus value is presumed to increase back up to within 10% of the original room temperature value.

Thermal ageing of tank 241-C-106 at 310^{0}F for 10 years is expected to embrittle the non-stress-relieved carbon steel liner by a loss in ductility; this phenomenon is called strain aging or strain age embrittlement.

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