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105-A WASTE STORAGE TANK MODEL TEST

I. INTRODUCTION

Measurements indicate that the metal bottom of the 105-A waste storage tank, 241-A Tank Farm, has bulged upward approximately eight and one-half feet at one point. The most plausible theory of what caused the bulge is that a pressure greater than the hydrostatic tank load was exerted against the bottom of the tank resulting in deformation of the steel bottom. This pressure may have been caused by the formation of steam from the water in the cement grout and heat from the waste tank. In an effort to learn more about the effect of pressure on the bottom of the waste tank liner, a scale model of the tank liner was built and pressure was applied to the bottom until the tank liner failed. This report describes this test.

II. SUMMARY

A .096 to 1 scale model of the 105-A waste tank liner was pressurized on the bottom until it bulged and failed. The differential pressure at failure was 0.63 psi and the maximum bulge height was 4-5/8 inches, corresponding to 48 inches in the actual tank. The tank bulge resulted from the side of the tank bending in below the first stiffener. Final failure was the rupture of about two inches of the joint which joins the bottom to the side of the tank.

The actual tank should bulge at the same pressure as the model while actual tank deflection is equal to the model tank deflection divided by the scaling factor. If the actual tank bulge was caused by steam pressure, a saturated steam temperature of 251° F. would be sufficient to overcome the 15 psi of liquid head in the tank and to supply the .63 psi necessary to bulge the tank. Failure of the bottom-to-side joint in the actual tank may or may not occur.

III. DISCUSSION

The 105-A waste tank liner is constructed of 3/8" carbon steel plate in the shape of a round tank with a flat bottom and open top. Ten stiffener rings of 5" x 3-1/2" x 3/16" angle iron are welded to the inside of the liner walls. The tank liner is 75 feet in diameter and about 32 feet high and is encased in a reinforced concrete shell.

In 1958, a study of the bulge in the bottom of the 113-SX waste storage tank was made by Brownell (1) in which he postulated that it was caused by pressure underneath the tank. This pressure was most likely caused by the formation of steam from the water in the grout and heat from the waste in the tank. Borwnell further states that the asphalt layer under the grout may have flowed upward, due to heat and pressure, around the tank liner to form a vapor seal which kept the steam from escaping. This theory that pressure can cause the bottom of a waste tank to bulge appears to be the best explanation of what happened to the 105-A waste tank. Using this theory as a basis, it was decided to build a model of the waste tank liner, apply pressure to the tank bottom and see if it would fail.

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(1) HW-57274, "Instability of Steel Bottoms in Waste Storage Tanks", by LE Brownell, dated August 29, 1958.

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A. 105-A Tank Liner Model

The tank liner is a very thin membrane, and fabrication of a scaled model of the tank was difficult. The metal was thin and distortion was a problem. The model was fabricated as nearly to scale as possible with the exception of height and the number of stiffener rings. Details of the model are shown on attached drawing SK-2-21442.

1. Scaling Factor

Calculations (see Appendix A) indicate that if the same scaling factor is used for both wall thickness and dimensions, and if the mechanical properties of the metal are the same, then under identical pressurizing conditions, stresses in the model and actual tank will be the same and deflections will be proportional to the scaling factor. Twenty gage carbon steel sheet was selected as the thinnest sheet which could be welded satisfactorily which resulted in a scale factor of 0.096. A comparison of actual and model tank data are listed in the following table.

COMPARISON OF ACTUAL AND MODEL TANK

Scale: 0.096 to 1

	Actual	Model
Diameter	75'-0"	7'-2"
Metal Thickness	0.375"	.0359"
Height	32' - 6-3/4"	2' - 3" (not scaled)
Angle Stiffeners	5"x 3-1/2"x 5/16"	1/2"x 11/32"x 0.0299"
Distance Floor to 1st Stiffener	2'-0"	2-5/16"
Area of Tank Bottom	4,420 ft^2	40.6 ft^2
Steel Plate	ASTM 283 Grade B or C	ASTM 415
Yield Strength	~ 30,000 psi	~30,000 psi
Tensile Strength	~ 60,000 psi	~60,000 psi

2. Pressure Simulation

The portion of the actual tank which may be pressurized is not known. One can assume it is all of the bottom and possibly a portion of the sidewalls. Brownell postulated that the vapor seal point was at the first stiffener ring. Without going to an expensive sealing device, it was decided that an adequate pressurization of the model could be attained by using a hydrostatic head which would also be against the sides of the tank. As the external pressure required to callapse the tank walls is much greater than that required to fail the tank by pressure on the bottom, this method was considered satisfactory.

In this test it was not possible to simulate the effect of increased differential pressure on the bulge which results from a constant steam pressure underneath and a decreased liquid waste head

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on top. However, some water was added to the inside of the model tank so that as the bottom started to bulge, the differential pressure at the bulge would remain constant rather than decrease as would be the case if no water was inside the tank.

B. Test Set-Up

The test set-up is shown in Figure 1 and on attached drawing SK-2-21441. The model was placed inside of the outer tank and an angle iron structure bolted on top to keep the model tank from floating. A 3/4" diameter rod simulating the 8 inches dry well was welded to the framework and extended to within 1-17/32" of the bottom. Sight glasses were used to indicate the liquid heads.

Six inches of water were added to the model tank and then water was added to the outside tank to create the external hydrostatic head on the tank bottom. The upward deflection of the tank bottom was measured with a ruler using the angle iron framework as a reference point.

C. Test Results

Calculations (See Appendix B) indicated that the tank would fail by bending of the tank wall below the first stiffener and that yielding should occur with only three inches of water differential. However, failure had not occurred when the water level reached the top of the inner tank and a differential of about 17 inches. At this point the bottom had raised upward approximately two inches and was against the 3/4" rod simulating the dry well. The two-inch deflection indicates that yielding had occurred as one inch is all that it could move elastically. Water was then syphoned from inside the tank model and failure occurred at 17-1/2 inches differential water pressure or 0.63 psi.

Part of the tank wall bent inward at the first stiffener allowing the bottom to bulge upward over an area about three feet across. Then five minutes later it bulged slightly more and ruptured adjacent to the weld at the bottom-to-side joint. This break was about two inches long. Pictures of the bulge and joint failure are shown in Figures 2 through 6. Maximum height of bulge was 4-5/8 inches which is equivalent to 48 inches in the actual tank.

A second test was conducted by welding up the rupture in the tank liner without disturbing the bulged area and again adding water to the outer tank. Failure had not occurred when the water reached the top of the tank -- a differential water pressure of 24 inches (.87 psi). However, the maximum bulge height increased to 5-1/4 inches or equivalent to 55 inches in the actual tank.

The tank wall had pulled in at the bottom about one-fourth inch and the tank bottom was wavy around the edge. This indicates that the metal has gone beyond its yield strength.

D. Conclusions

Although failure of the tank occurred at a pressure several times greater than that calculated, it is still relatively low. Part of this difference may be explained by the restraining force of the simulated dry well, the relationship between pressure and side wall force and redistribution of forces upon yielding. However, it is reasonable to assume that these differences will be similar in the actual tank and that both tanks will bulge at about the same pressure. Saturated steam at 251° F.

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would produce the pressure necessary to bulge the actual tank containing 15 psi of liquid waste head.

The significance of the model tank joint failure is difficult to evaluate. Failure occurred in the heat affected zone adjacent to the weld where high residual stresses were present due to the welding. It was not possible to make a scale reproduction of the weld used in the actual tank because of metal thickness. However, the actual tank also had many residual stresses as a result of welding the large steel plates. It cannot be concluded that the actual tank joint would fail under similar circumstances but there is a good possibility that it would.

ACKNOWLEDGEMENT

Thanks are given to F. J. Molerus, Jr., New Reactor Department, for his help in establishing the scaling factor.

D D Wodrich

D. D. Wodrich, Engineer
General Plant & Utilities Engineering
Facilities Engineering Operation

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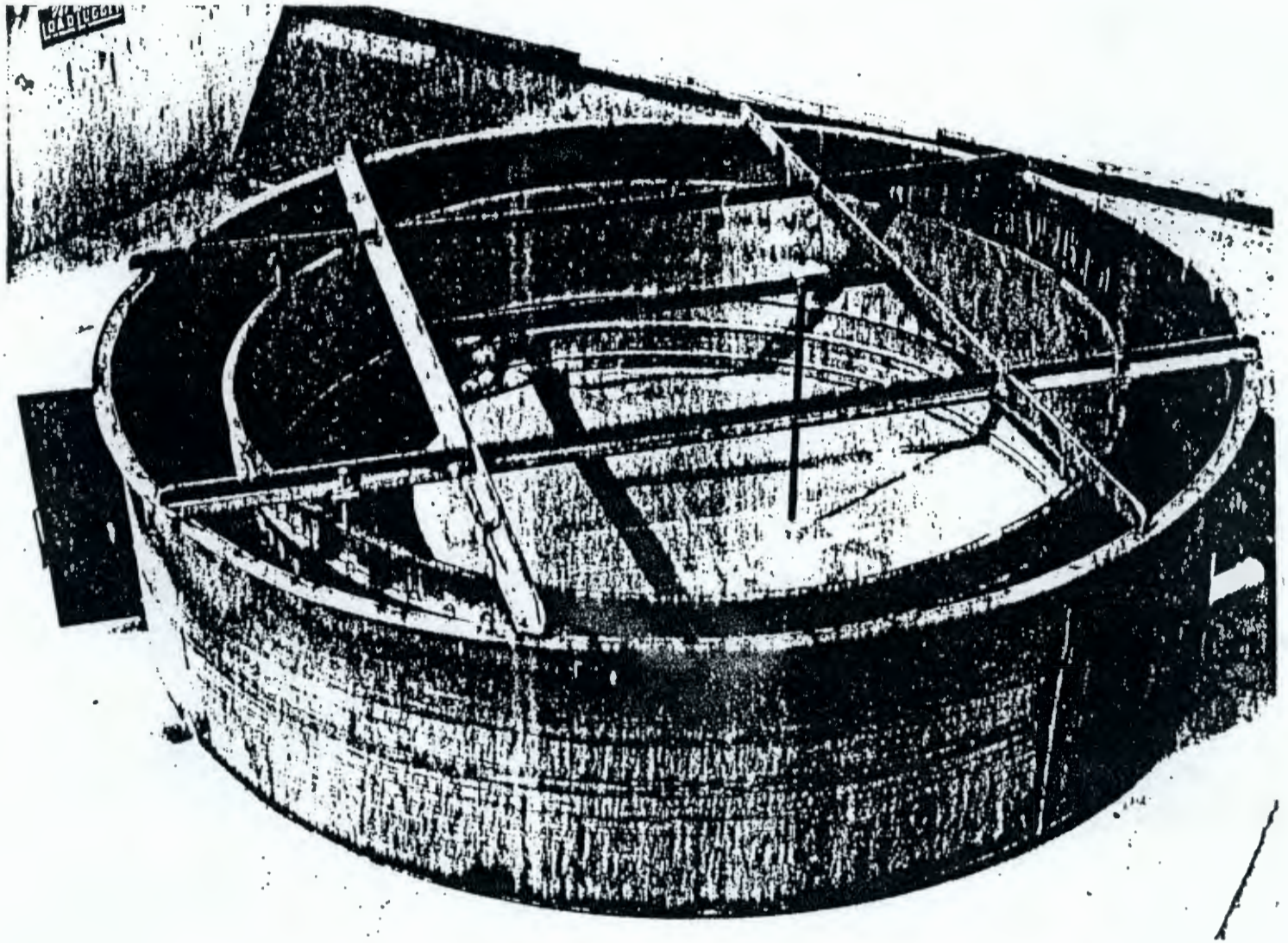

FIGURE 1 - 105-A WASTE TANK LINER MODEL TEST SET-UP

AEC-GE RICHLAND, WASH

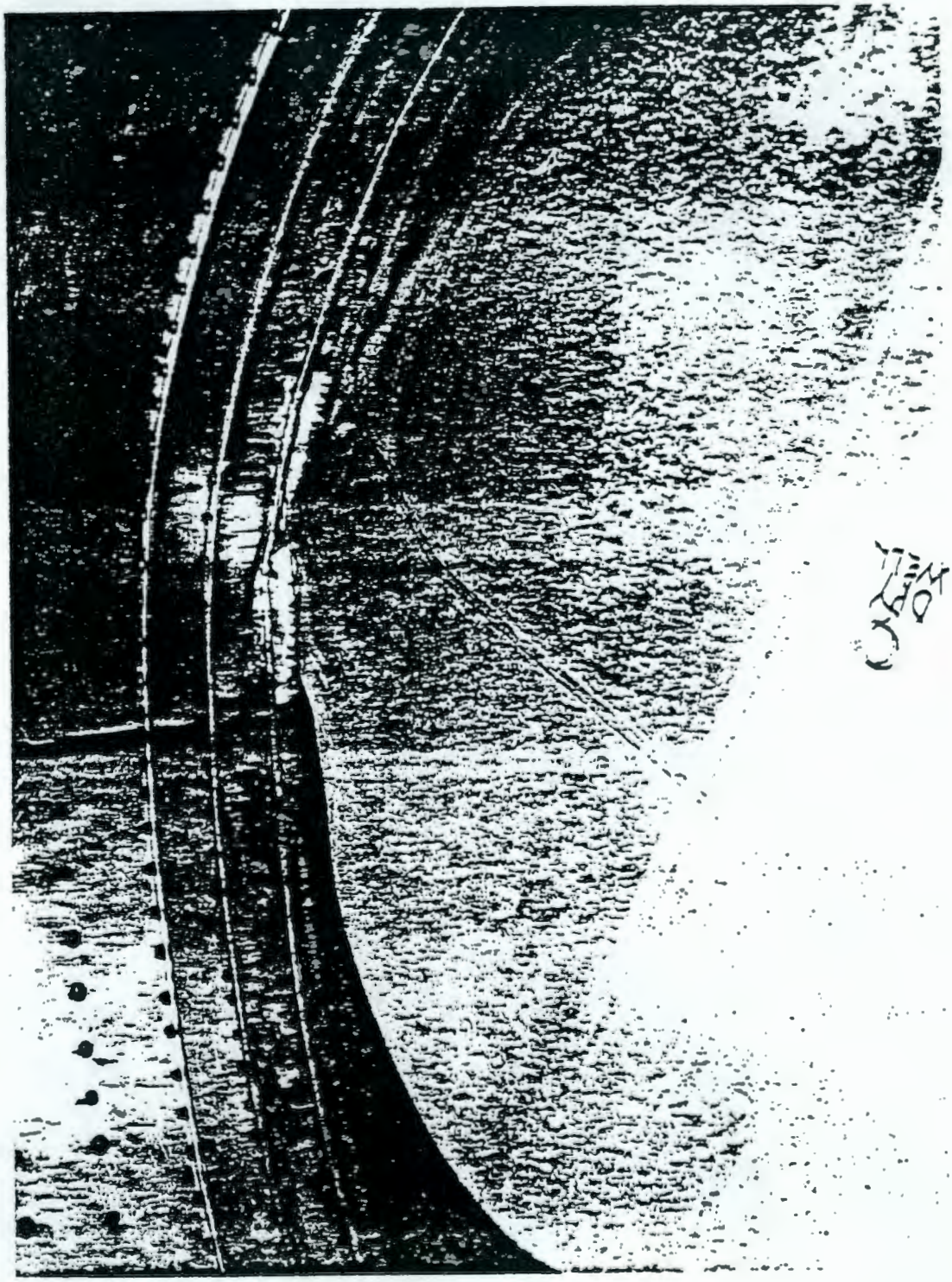

FIGURE 2 - GENERAL VIEW OF BULGED AREA

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FIGURE 3 - WELD JOINT FAILURE

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AEC-GE RICHLAND, WASH.

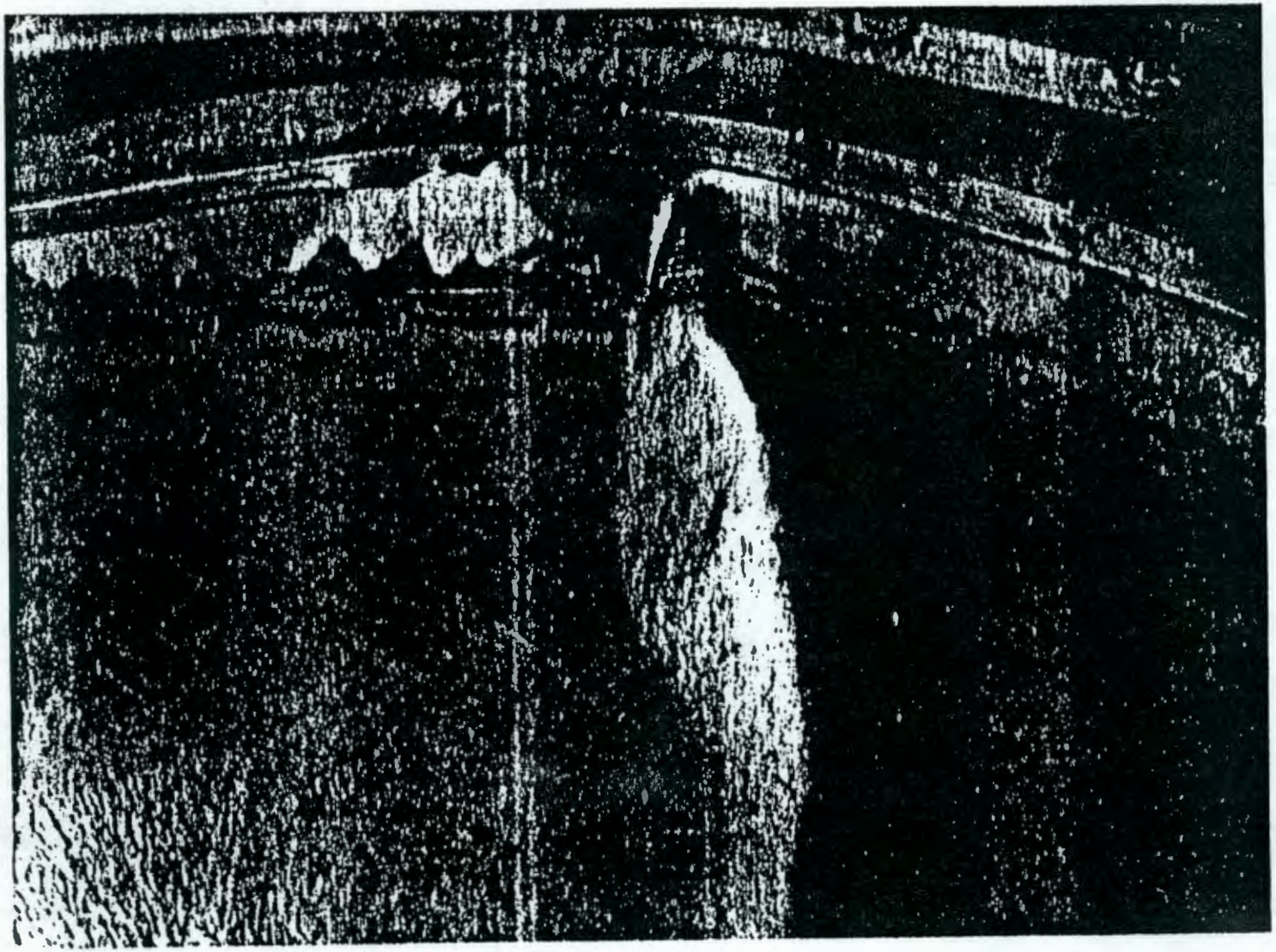


FIGURE 4 - CLOSE-UP OF JOINT FAILURE

AEC-GE RICHLAND, WASH

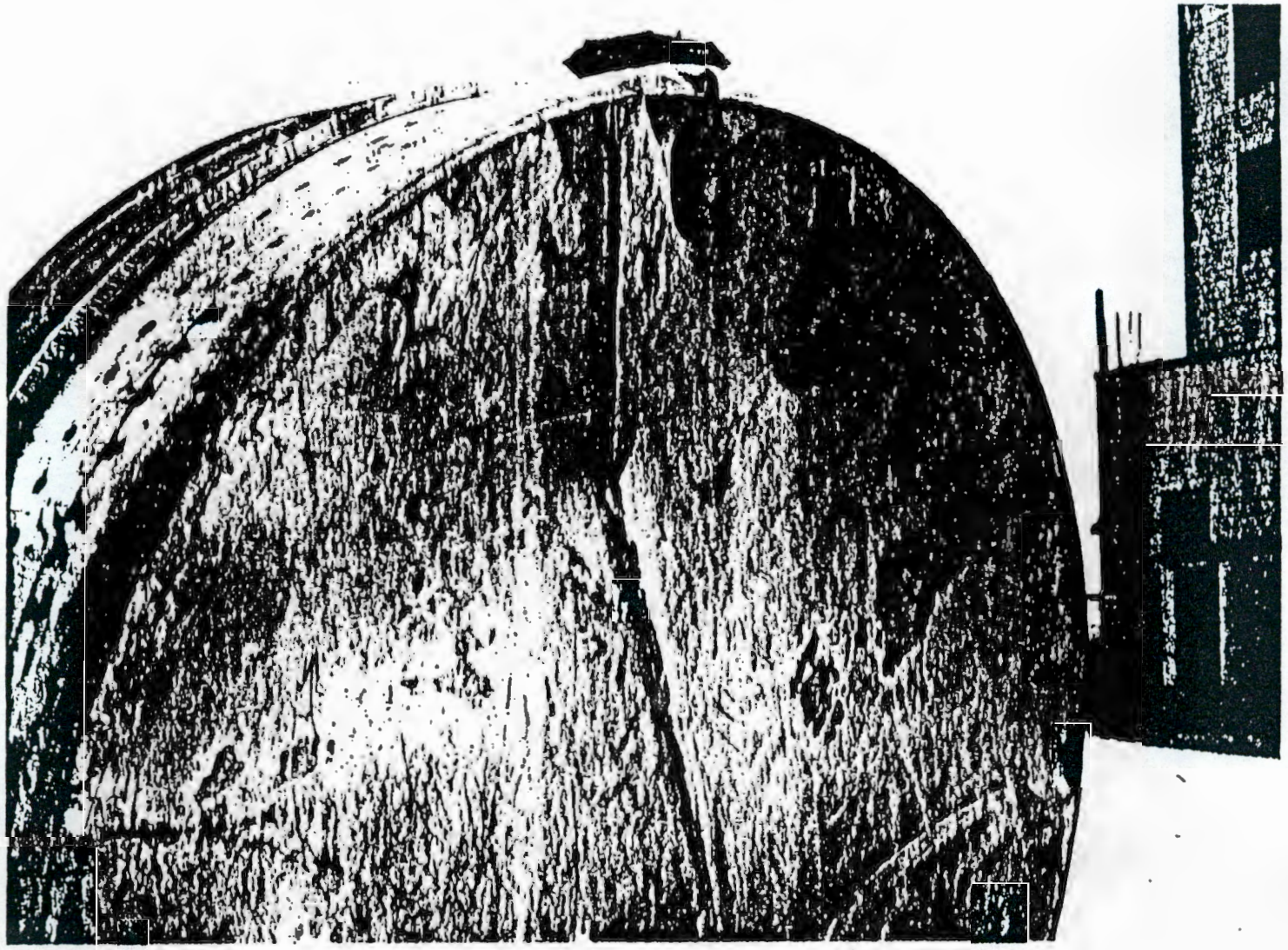


FIGURE 5 - BULGED AREA VIEWED FROM BOTTOM OF TANK

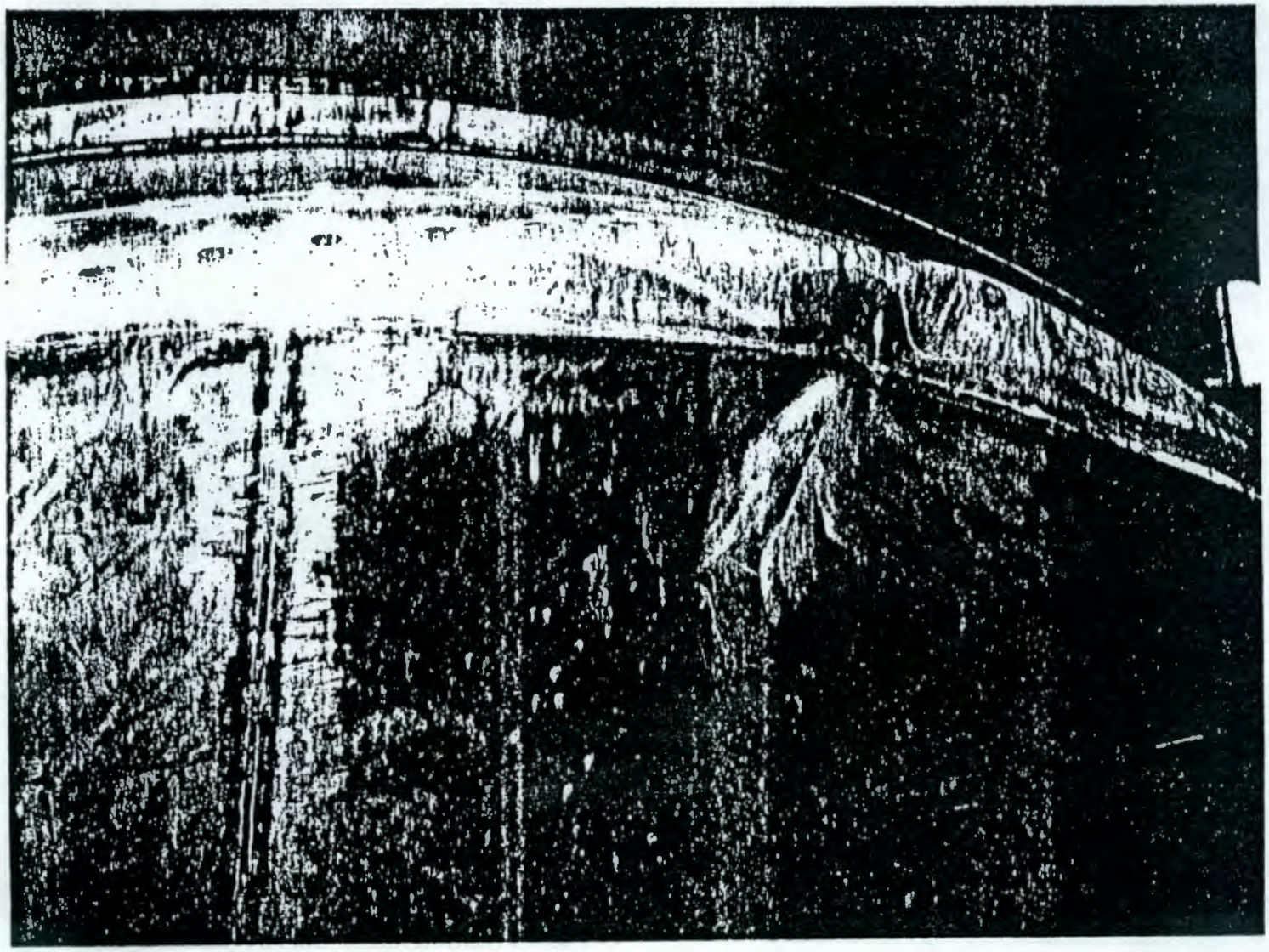
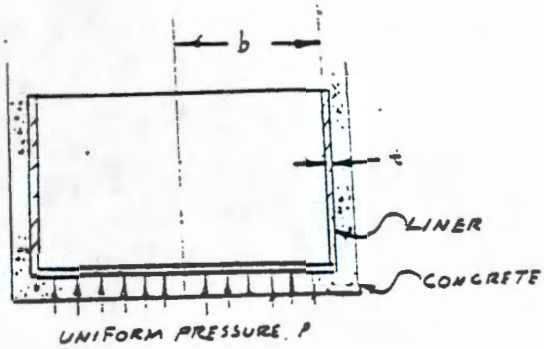


FIGURE 6 - FAILED JOINT, BOTTOM OF TANK

APPENDIX A



Waste Tank

Determining effect of scaling factor on behavior of tank.

NOMENCLATURE:

σ = Unit Stress, psi
P = Pressure, psi
E = Modulus of Elasticity, psi
t = Metal Thickness, Inches
b = Tank Radius, Inches
Δ = Deflection of Bottom, Inches
M = Poisson's Ratio
k = Model Scale Factor
m = Bending Moment, In-Lb
I = Moment of Inertia, In^4
S = Bending Stress, psi

Subscripts o = Actual
1 = Model

Assumptions: Model is fabricated of carbon steel.
Then $E_o = E_1$
$M_o = M_1$

The same scale factor is used for thickness and dimensions; therefore
$b_1 = kb_o$
$t_1 = k\,t_o$

9 1 1 2 0 5 2 1 1 2 3

APPENDIX A (cont.)

Consider the bottom a uniformly loaded plate with the edges restrained. For thin plates with large deflections apply the membrane theory.

$$\sigma = B\sqrt[3]{P^2E\,\frac{b^2}{t^2}} \qquad \Delta = A\,a\sqrt[3]{\frac{P\,b}{Et}}$$

Where A and B are shape factors and are the same in both model and actual tank.

$$\sigma_1 = B_1\sqrt[3]{\frac{P_1^2E_1b_1^2}{t_1^2}}$$

$$\sigma_o = B_o\sqrt[3]{\frac{P_o^2E_ob_o^2}{t_o^2}}$$

But $B_1=B_o$, $E_1=E_o$, $b_1=K\,b_o$, and $t_1=K\,t_o$

$$\Delta_1 = A_1b_1\sqrt[3]{\frac{P_1b_1}{E_1t_1}}$$

$$\frac{\sigma_1}{\sigma_o} = \frac{B_1\sqrt[3]{P_1^2E_o\,\frac{k^2b_o^2}{k^2t_o^2}}}{B_1\sqrt[3]{P_o^2E_o\,\frac{b_o^2}{t_o^2}}}$$

$$\Delta = A_ob_o\sqrt[3]{\frac{P_ob_o}{E_ot_o}}$$

But $A_1 = A_o$
$b_1=k\,b_o$, and $t_1=k\,t_o$

$$\frac{\sigma_1}{\sigma_o} = \sqrt[3]{\frac{P_1^2\,E_o\,\frac{k^2b_o^2}{k^2t_o^2}}{P_o^2\,E_o\,\frac{b_o^2}{t_o^2}}}$$

$$\frac{\Delta_1}{\Delta_o} = \frac{A_o\,k\,b_o\sqrt[3]{\frac{P_1\,k\,b_o}{E_o\,k\,t_o}}}{A_o\,b_o\sqrt[3]{\frac{P_o\,b_o}{E_o\,t_o}}}$$

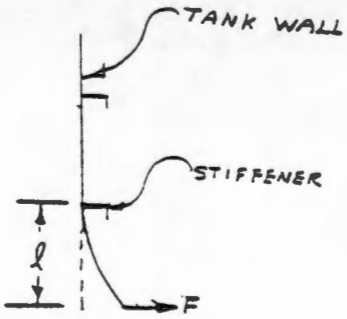
$$\frac{\sigma_1}{\sigma_o} = \sqrt[3]{\frac{P_1^2}{P_o^2}}$$

$$\frac{\Delta_1}{\Delta_o} = k\sqrt[3]{\frac{\frac{P_1kbo}{E_o.kto}}{\frac{P_ob_o}{E_ot_o}}}$$

$$\frac{\Delta_1}{\Delta_2} = k\sqrt[3]{\frac{P_1}{P_o}}$$

Therefore, at the same pressure ($P_1=P_o$) the stress in the bottoms of the actual tank and model will be the same and deflection will be proportional to the scale factor, $\Delta_o = k\,\Delta_1$.

As the weakest point appears to be the tank wall, let us consider the stress produced if the tank bottom pulls in on the tank walls and tends to bend them below the first stiffener. Taking a strip one inch wide and assuming the wall is flat, it will act as a cantilever beam with the force F being caused by the bottom pulling in as it bulges upward. The bending stress in the tank wall is:



$$S = \frac{my}{I}$$

Substituting

$$S = \frac{t\sigma\ell\frac{t}{2}}{\frac{t^3}{12}}$$

$$S = \frac{6\sigma\ell}{t}$$

Where

$m = F\ell$
$Y = 1/2\ t$
$I = \frac{(1)t^3}{12}$
$F = t(1)\ \sigma$
ℓ = Distance from stiffener to bottom, inches

$$\frac{S_1}{S_o} = \frac{\frac{6\sigma_1 \ell_1}{t_1}}{\frac{6\sigma_o \ell_o}{t_o}}$$

but $\ell_1 = k\ell_o$
$t_1 = kt_o$

$$\frac{S_1}{S_o} = \frac{\frac{6\sigma_1 k\ell_o}{kt_o}}{\frac{6\sigma_o \ell_o}{t_o}}$$

$$\frac{S_1}{S_o} = \frac{\sigma_1}{\sigma_o}$$

$$\frac{\sigma_1}{\sigma_o} = \sqrt[3]{\frac{P_1^2}{P_o^2}}$$

Then using equation for stress in bottom plate from page 12

$$\frac{S_1}{S_o} = \sqrt[3]{\frac{P_1^2}{P_o^2}}$$

Therefore, at the same pressure $P_1 = P_o$, the stress in the sidewall will be the same in both the actual and model tanks.

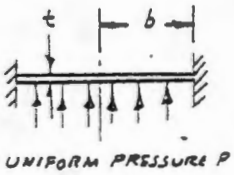
APPENDIX B

PRESSURE REQUIRED TO FAIL 105-A TANK MODEL

A. Failure of Bottom

Assumptions:

1. Failure occurs when metal yields.
2. Bottom acts like a thin plate with edges clamped.



$$\sigma = 0.423 \sqrt[3]{\frac{E\,P^2\,b^2}{t^2}}$$

Where σ = 30,000 psi yield stress
E = 30×10^6 psi
P = pressure, psi
b = tank radius = 43"
t = bottom thickness = .0359"

$$30{,}000 = 0.423\sqrt[3]{\frac{30\times10^6 P^2 (43)^2}{(.0359)^2}}$$

$$(70{,}900)^3 = \frac{30\times10^6 P^2 (1{,}850)}{.00129}$$

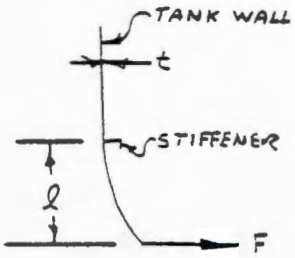
$P^2 = 3.28$

$P = 2.88$ psi Pressure on bottom required to yield bottom plate in center.

B. Failure of Tank Wall Below First Stiffener

Assumptions:

1. Failure occurs when the metal yields.
2. Wall acts similar to a cantilever beam.



Consider strip one inch wide

$$S = \frac{mY}{I}$$

$$S = \frac{F\ell Y}{I}$$

$$F = \frac{SI}{\ell Y} = \frac{30{,}000\ (3.33\times10^{-6})}{2.31\ (.018)}$$

where
S = 30,000 psi yield stress
$m = F\ell$ in - lb
$Y = \frac{t}{2} = \frac{.036}{2} = .018$
$I = \frac{(1)(.036)^3}{12} = 3.33\times10^{-6}\ in^4$

F = 2.41 pounds force required per inch width of wall to produce yielding.

Stress in bottom at edge to produce force $F = \frac{2.41}{.036} = 67$ psi.

Using membrane theory, the pressure required to produce 67 psi of stress at edge is:

$$\sigma = 0.328 \sqrt[3]{\frac{E\,P^2\,b^2}{t^2}}$$

$$67 = .328 \sqrt[3]{\frac{30\times10^6 P^2 (43)^2}{(.036)^2}}$$

$$\left(\frac{67}{.328}\right)^3 = \frac{30\times10^6 P^2 (1{,}850)}{.00129}$$

$$P^2 = 19.75 \times 10^{-4}$$

$$P = 4.36 \times 10^{-2} \text{ psi}$$

$$P = 1.2 \text{ inches } H_2O$$

This will be somewhat higher because of tank curvature so will assume 3 inches water. Therefore, the tank side wall should be the point of failure when pressure is applied to the bottom of the tank.

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