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External Pressure Limitations for 0-15 PSI Storage Tanks

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EXTERNAL PRESSURE LIMITATIONS FOR 0-15 PSI STORAGE TANKS

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ABSTRACT

Large cylindrical storage tanks are designed in accordance with design rules of the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section III, Subsection NC, Article NC-3900[1] or American Petroleum Institute (API) Standard 620[2]. Both of these Codes have identical requirements. These Codes provide a limit on the partial vacuum in the gas or vapor space not to exceed 1 oz/in^2 to ensure stability of cylindrical walls against collapse. This criterion seems to be too conservative for the underground double shell storage tanks to be built at Hanford for the Department of Energy.

The analysis presented herein shows that the bottom plate of the Hanford tank is the most critical component when an empty tank is subjected to partial vacuum. However, the allowable external pressures for both cylindrical walls and the bottom plate are significantly higher than 1 oz/in^2. The allowable external pressure for the bottom plate is largely dependent upon the plate uplift considerations which in turn depends on the plate thickness. The large displacement non-linear elastic analyses and the eigenvalue buckling solutions indicate that considerable wrinkling can occur before a snap-through buckling failure occurs.

INTRODUCTION

The U.S. Department of Energy plans to construct a number of underground high-level radioactive waste storage tanks at the Hanford Site near Richland, Washington. The underground tanks, as shown in Figure 1, consist of two main concentric cylindrical structures: an inner, primary steel tank and an outer, carbon steel-lined secondary reinforced-concrete structure. The primary carbon-steel tank contains the liquid waste, and the secondary concrete structure serves as a redundant barrier to confine the radioactive fluid in the event of failure of the primary tank.

The primary tank is separated from the secondary concrete structure and liner by a reinforced-concrete supporting pad. The ellipsoidal dome of the primary tank is structurally attached to the dome of the secondary concrete structure by closely-spaced stud anchors. The vertical walls of the primary tank and the secondary concrete structure are separated by a 30-in. ventilated annular space. The primary tank has mixer pumps to maintain solids in suspension. These mixer pumps have a 6-in. clearance from the bottom of the tank. This limits the maximum permissible tank-bottom uplift to less than 6-in. when the empty tank is subjected to a negative pressure. Although during normal operations, the tank walls are subjected to positive pressure, during testing of the ventilation system when the tank is empty, the pressure can decrease to as low as - 6 inches of water.

This paper evaluates the design adequacy of the cylindrical walls under the partial vacuum condition and determines the allowable negative pressure for the bottom plate. Failure by buckling is not the governing mode of failure for the dome because of its anchored configuration.

ALLOWABLE EXTERNAL PRESSURE FOR CYLINDRICAL WALLS

For cylindrical walls, other than the limit specified in NC-3932.5, NC-3900 has no detailed procedure to calculate the external pressure allowable. Therefore, NC-3133.3 rules which are known to be conservative are used for calculating the allowable negative pressure. The following are the required data to perform this evaluation.

L = length of the cylindrical tank = 489.375 in.
D_o = outside diameter = 900.70 in.
T = wall thickness = 0.70 in. (exclusive of corrosion allowance).

An uniform wall thickness is conservatively considered although the nominal wall thickness varies between 3/4 in. and 1 1/8 in.

Factor A required by the Code procedure could not be obtained from Figure G, ASME Code Section II, Part D[3] as Figure G does not include curves for D_o/T greater than 1000. Bednar[4] provides the following formula for Factor A which is applicable to intermediate cylinders:

$$Factor\ A = \frac{1.30\left(\frac{T}{D_o}\right)^{1.5}}{\frac{L}{D_o}} \qquad (1)$$

Substitution of the above geometry parameters in equation (1) gives Factor A = 5.19×10^{-5}. An extrapolation of Figure G of the Code for D_0/T = 1287, provides similar Factor A value.

The allowable pressure, p_a, is calculated using equation (2) because Factor A falls to the left of the material curve given in Figure CS-4 of Section II, Part D of the Code for SA-537 which has properties similar to SA-516.

$$p_a = \frac{2\,A\,E}{3\left(\frac{D_o}{T}\right)} \qquad (2)$$

Using E, the Young's modulus, equal to 28.55×10^6 psi and Factor A = 5.19×10^{-5}, the above equation yields an allowable negative pressure of 0.767 psi which is equivalent to 21.25 inches of water. This pressure is greater than the expected negative pressure of 6 inches of water, therefore, the cylindrical walls have adequate thickness to preclude failure in buckling mode.

ALLOWABLE EXTERNAL PRESSURE FOR THE BOTTOM PLATE

Approach

The ASME code, paragraph NC-3922.3 does not provide a methodology to determine the allowable external pressure for a shallow conical shell such as the Hanford tank bottom. The ASME code case N-284[5] , however, provides general guidelines for buckling evaluation: first determine the theoretical buckling capacity, and then reduce it by a safety factor for the applicable Service Level, a plasticity reduction factor, and a capacity reduction factor for imperfections.

Contributing to the theoretical buckling capacity of the tank bottom are the tank bottom plate weight and the bottom plate intrinsic stiffness. Accordingly, the following criterion must be satisfied:

$$w_s + \frac{R}{Safety\ Factor} = P_a \quad (3)$$

where w_s is the weight of the steel bottom per unit area in inches of water and R is the negative pressure in inches of water required to buckle an imperfect bottom plate. R represents the pressure capacity as a result of the intrinsic stiffness of the conical shell and p_a is the allowable negative pressure also in inches of water. This criterion follows from force equilibrium in the vertical direction, and states that the sum of all forces resisting uplift (left hand side) of an imperfect bottom plate must be equal to the allowable negative pressure. Since w_s represents a load which can be determined accurately, the factor of safety need to be applied to R only whose accuracy depends upon the analysis methodology, modeling, material properties etc. R is determined from finite element analyses and p_a is then calculated from equation (3).

3-D Model of the Tank Bottom

The finite element model utilized assumes the presence of at least one symmetry plane which coincides with the global 1-3 plane, with the 3-direction vertical. The wall is considered fixed at 24 inches above the knuckle. The model employs element S9R5 (nine nodes with reduced integration) which is one of the most accurate shell elements ABAQUS[6] finite element program has to offer.

The tank radius is 450 inches. The knuckle radius is 24 inches. The nominal plate thickness is 1 in. A corrosion allowance of 0.2 in. for the bottom plate and 0.05 in. for the remainder of the tank has been established. The ABAQUS model is based upon corroded wall thickness.

Material Properties

These properties used in the analysis are:

Young's modulus = 28.55 x 10^6 psi,
Poisson's ratio = 0.3, and
thermal expansion coefficient=
6.57 x 10^{-6}/ °F.

Loading

The buckling evaluation, using ABAQUS finite-element program, is performed in two steps: eigenvalue buckling analysis with perfect configuration for determining the worst case imperfection (this is based on the eigenvector associated with the smallest eigenvalue), followed by large displacement analysis, or load-deflection analysis of an imperfect model whose initial perfect geometry is perturbed by the lowest eigenvector having a maximum perturbation of 2 inches (250% of plate thickness). Thermal expansion is also included in large displacement analysis by specifying a temperature of 231°F for all bottom plate and knuckle nodes. The cylindrical wall nodes were assumed to remain at the reference temperature of 55°F.

Eigenvalue Analysis Results

The smallest eigenvalue is 0.0975 psi However, this does not mean that the bottom plate will snap-through at this pressure because the eigenvector plot in Figure 2 indicates three circumferential waves 120 degrees apart suggesting a wrinkling effect instead of a snap-through effect. Figures 3, 4, and 5 exhibit the next three higher eigenvectors and their corresponding eigenvalues. As in the lowest mode, the buckling configurations in these figures suggest instability in the circumferential direction. These results indicate that the tank bottom plate may undergo large displacements and wrinkling before it snaps through. Therefore, the snap-through pressure, determined from large displacement analysis, may be higher than this value.

Eigenvalue buckling analysis with temperature as the driving force for buckling reveals that the smallest eigenvalue is 2233.7 °F. Since this value is much higher than the design temperature of 250 °F, it is concluded that thermal expansion is expected to have a negligible impact on the base plate buckling behavior.

Large Displacement Analysis Results

In order to obtain the snap-through pressure, large displacement analysis is performed for the tank bottom. The initial configuration is perturbed by the lowest eigenvector shown in Figure 2 to introduce geometric imperfection. As indicated earlier, a maximum perturbation of two inches is applied.

In large displacement analysis, the nodal temperatures are applied in step 1. The negative pressure is then ramped from 0.0 to 0.135 psi in the second step, and to 0.15 psi in the third step. It is observed that at a pressure of 0.135 psi vertical displacements reach a maximum of 4.317 inches, while at a pressure of 0.15 psi, these displacements become quite large with a maximum of 21.00 inches. Vertical displacement vs. pressure plots (see Figure 6) for nodes where displacements are very large show gradual changes in slopes up to 0.135 psi. However, for a small increase in pressure from 0.135 psi to 0.15 psi, the vertical displacements rapidly increase to many times larger values. Therefore, 0.135 psi can be safely assumed to be the pressure at which instability occurs. Because the eigenvalue buckling pressure of 0.0975 psi is less than 0.135 psi, the bottom plate is expected to experience significant amount of wrinkling before it snaps through.

In order to obtain a design value of the buckling pressure, R, the above pressure must be reduced by the capacity reduction factor to account geometric imperfection and by a plasticity reduction factor if the analysis indicates deformations in the plastic range. Because geometric imperfection is already included in the model and no plastic action is indicated (maximum stress intensity is 13.14 ksi and material yield stress is 34.15 ksi), these factors are equal to 1.0. Therefore, R is equal to 0.135 psi. Note at this pressure, the maximum displacement is nearly four inches which is less than the clearance between the tank bottom and the mixer pump.

The weight of the bottom plate per unit area, w_s, is 0.227 psi. A safety factor of 2.0 is assumed which corresponds to Code Case N-284 safety factors of Service Levels A and B. Then from equation (3), the allowable external pressure for the bottom plate is 0.294 psi or approximately 8 inches of water. Note that this pressure is significantly higher than 1 oz/in^2 given by NC-3932.5 or the API-620.

CONCLUSION

For partial vacuum condition, the allowable external pressure for the cylindrical tank walls may be determined in accordance with the procedure outlined in NC-3133.3 because NC-3932.5 allowable is too conservative.

The bottom plates for large diameter tanks are capable of withstanding a negative pressure at least equivalent to the weight per unit area of the plates. Additional capacity may be obtained by analyses similar to those described above. Note that stiffness contribution for large diameter to thickness ratios is small but not negligible.

REFERENCES

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2. API, (1978), API Standard 620, American Petroleum Institute, Washington, DC.

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4. Bednar H. H., (1986), Pressure Vessel Design Handbook, 2nd Edition, Van Nostrand Reinhold Company, New York, NY.

5. ASME, (1980), Code Case N-284, Metal Containment Shell-Buckling Design Methods, Section III, Division 1, Class MC, American Society of Mechanical Engineers, New York, NY.

6. ABAQUS, (1989), ABAQUS User's Manual, Version 4.8 (with revisions), Hibbitt, Karlsson & Sorenson, Inc., Pawtucket, RI.

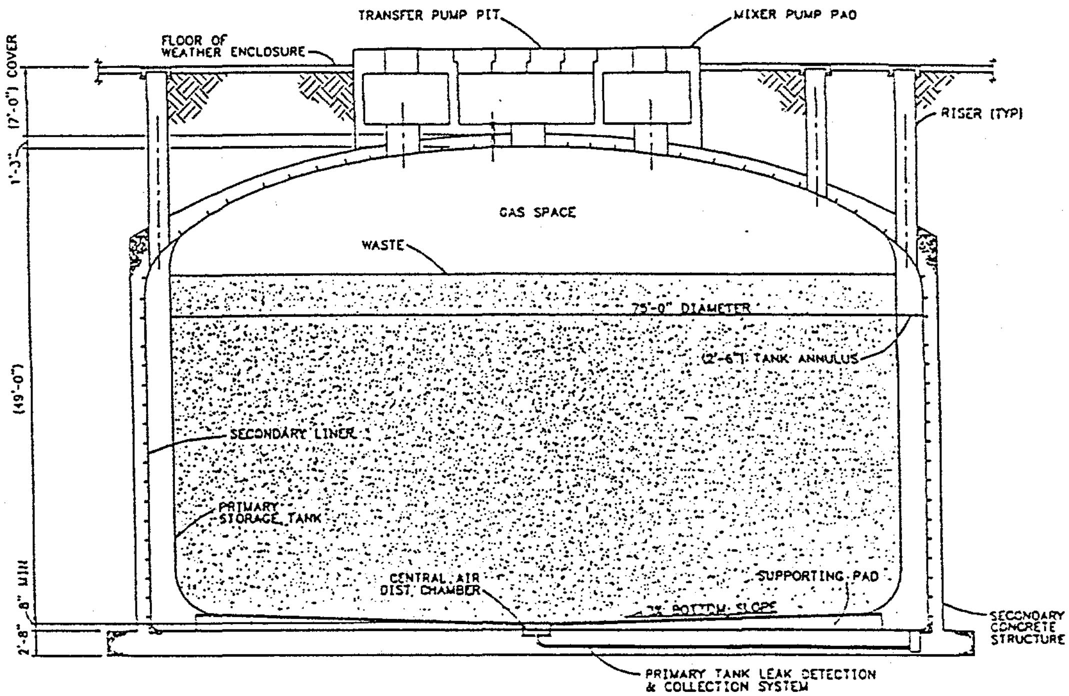

Figure 1. Tank Configuration

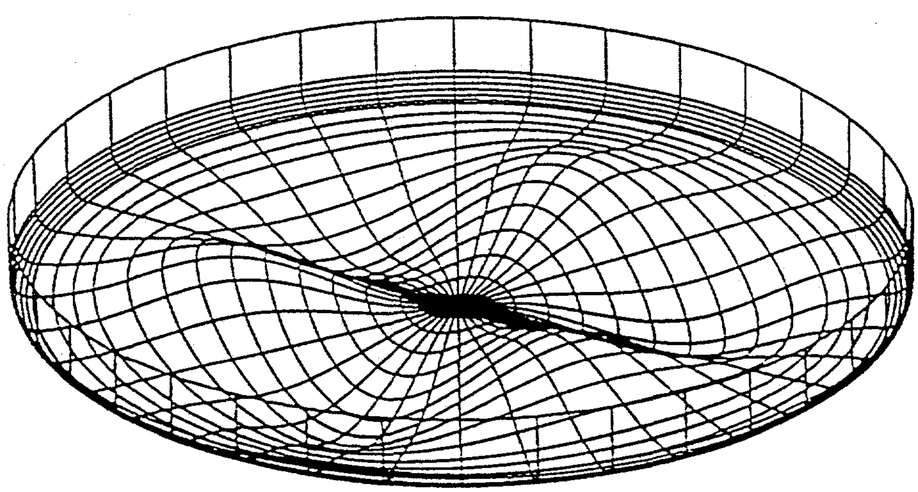

Figure 2. Buckling Mode 1, Eigenvalue = 0.0975 psi

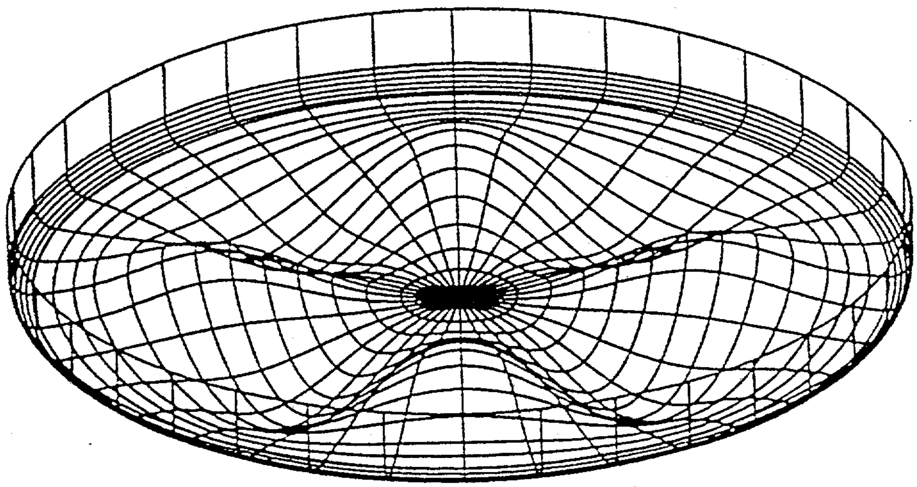

Figure 3. Buckling Mode 2, Eigenvalue = 0.106 psi

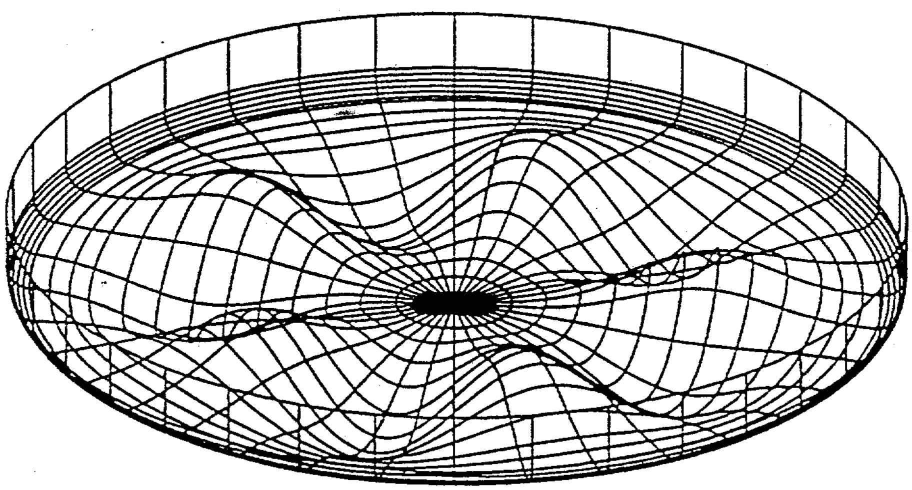

Figure 4. Buckling Mode 3, Eigenvalue = 0.13 psi

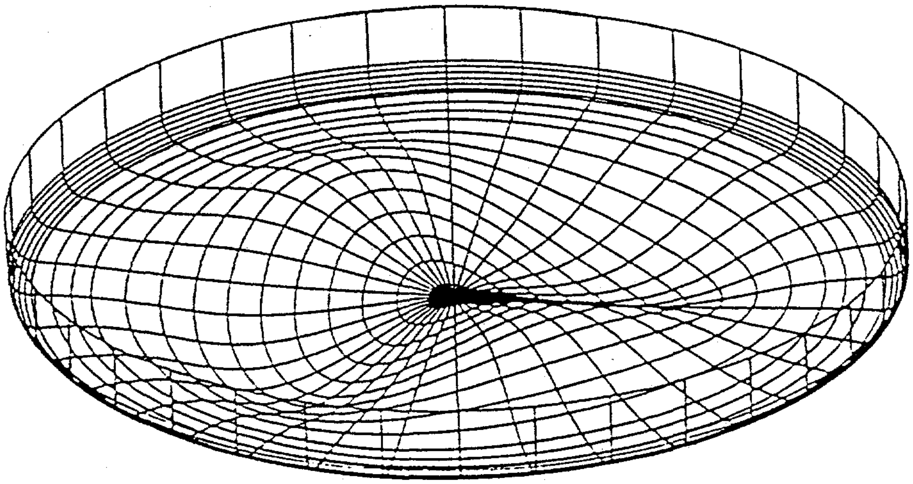

Figure 5. Buckling Mode 4, Eigenvalue = 0.152 psi

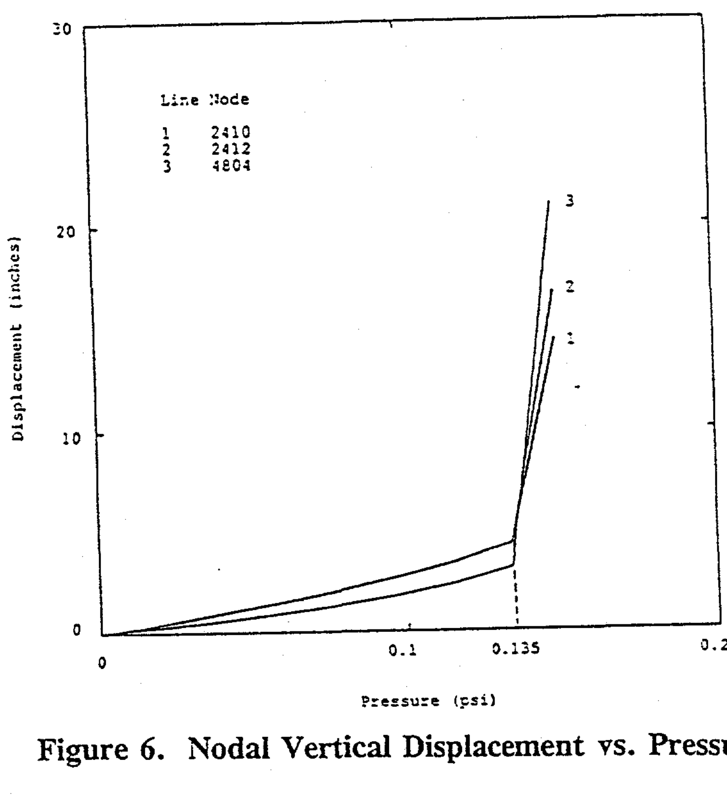

Figure 6. Nodal Vertical Displacement vs. Pressure