

Study to Establish Relations for the Relative Strength of API 650 Cone Roof Roof-to-Shell and Shell-to-Bottom Joints

API PUBLICATION 937-A
AUGUST 2005



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Prepared by:
Thunderhead Engineering Consultants, Incorporated
1006 Poyntz Ave.
Manhattan, KS 66502-5459
785-770-8511
www.thunderheadeng.com



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1. Introduction

This report documents an evaluation of the relative strengths of the roof-to-shell and shell-to-bottom joints in API 650 cone roof tanks. This information is supplied to the American Petroleum Institute as background material for development of design rules that govern frangible roof joints for API 650 tanks.

API 650 (American Petroleum Institute, 2001) provides design criteria for fluid storage tanks used to store flammable products. Due to filling and emptying of the tanks, the vapor above the product surface inside the tank may be within its flammability limits. Ignition of this vapor can cause sudden over-pressurization and can lead to the catastrophic loss of tank integrity. To prevent shell or bottom failure, the rules in API 650 are intended to ensure that the frangible roof-to-shell joint fails before failure occurs in the tank shell or the shell-to-bottom joint. Failure of the frangible roof-to-shell joint provides a large venting area and reduces the pressure in the tank.

Although the criteria in API 650 function well for large tanks, small tanks designed to the API 650 rules have not always functioned as intended. Morgenegg, 1978, provides a description of a 20 foot diameter by 20 foot tall tank in which the shell-to-bottom failed. Other such failures have been noted by API, providing the incentive for this study.

As presently written, the API 650 rules do not address the strength of the shell-to-bottom joint directly. Instead, the present rule is intended to ensure that the roof-to-shell joint fails at a pressure lower than that required to lift the weight of tank. It is assumed that with no uplift, the shell-to-bottom joint will not have significant additional loads and that failure of the shell-to-bottom will be avoided.

A study of roof-to-shell joint failure (Swenson, et al., 1996) showed that for large tanks, the roof-to-shell joint did indeed fail before tank uplift, but that for smaller tanks uplift would occur before roof-to-shell joint failure. Since uplift occurs for small tanks, this increases the possibility of shell-to-bottom joint failure.

The purpose of this study is to investigate the relative strengths of the roof-to-shell and shell-to-bottom joints, with the goal of providing suggestions for frangible roof design criteria applicable to smaller tanks.

2. SafeRoof

The calculations in this report were made using the SafeRoof computer program (Lu and Swenson, 1994). SafeRoof was developed to design and analyze storage tanks with frangible roof joints. The program is the result of a research program into frangible joint design, sponsored by the American Petroleum Institute and the Pressure Vessel Research Council.

SafeRoof includes design, analysis, and post-processing modules. In the design module, the user can input tank parameters and SafeRoof will develop a design following API 650 guidelines. This design can either be accepted or modified. The user can then analyze the stresses and displacements in the tank at pressures corresponding to selected tank failure modes. The analysis can be coupled to a combustion/joint failure analysis. The pressures at each failure mode can be used to help evaluate safety of the tank due to overload pressures.

The original version of SafeRoof used a static, large displacement, elastic finite element model. As part of this project, version 2.0 was extended to incorporate the capability to perform dynamic, large displacement, elastic-plastic analyses of tank response. This capability is based on the FMA-3D code (FMA, 2004).

Version 2.1 includes the capability to approximate circumferential buckling in the roof and floor. Buckling is approximated by reducing the circumferential stiffness of the roof (or floor) finite elements by a factor of 10 in the elements in which compressive circumferential stresses are detected. Based on beam flange buckling practice, buckling effects are not included within a distance of 32 times the roof (or floor) thickness from the joint. In addition, for buckling of the floor, the floor must have uplifted from the supporting foundation.

3. Tank Response to Over-Pressurization

Before discussing the general results for the study, it is important to examine in detail the response of an oil storage tank to over-pressurization, based on previous work (Swenson et al., 1996). A tank with a 30 foot diameter and a 32 foot height will be discussed as a representative tank. The tank parameters are given in Figure 3-1.

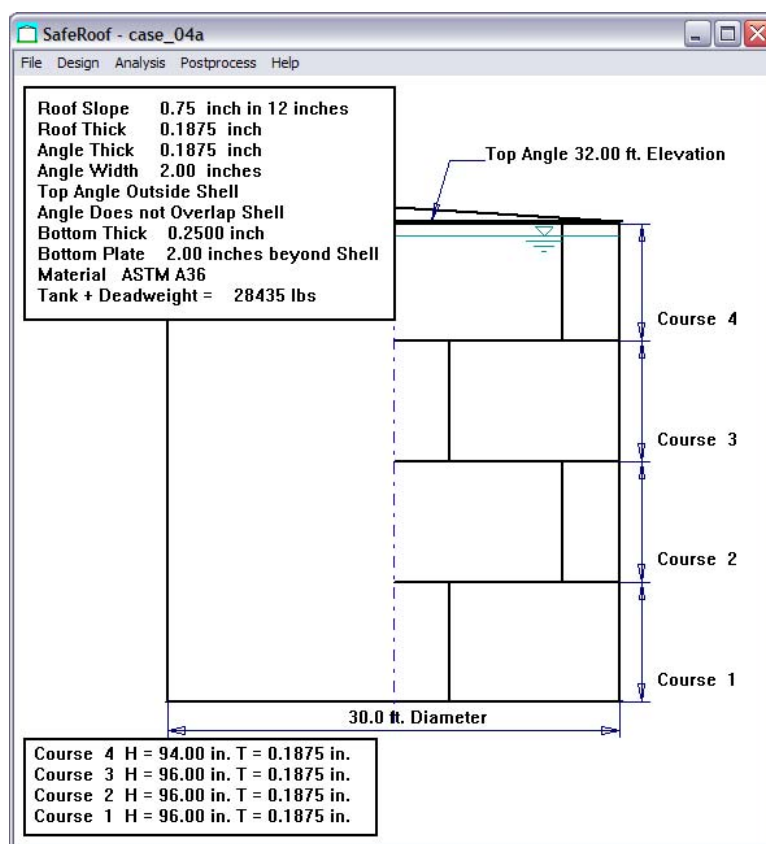


Figure 3-1: Design of representative 30 foot diameter tank

This design was done using the SafeRoof program (Lu and Swenson, 1994). This program follows API 650 rules to design the tank. The maximum fluid level is assumed to be 31 feet, with a specific gravity of 0.95. The material is ASTM A36, with a minimum yield strength of 36,000 psi, a modulus of 30E6 psi, and a Poisson's ratio of 0.25. In this example, the minimum yield strength was used, however, the typical yield strength should be used for design calculations.

The design has four courses with a thickness of 0.1875 inch. The top angle faces radially outward, with an angle width of 2 inches and a thickness of 0.1875 inches. The roof is welded to the top angle at a distance of 1 inch outside the radius of the tank. The slope of the roof is 0.75 inches in 12 inches. The bottom thickness is 0.25 inches. The tank is assumed to rest on sand, with a ringwall foundation. The stiffness of the sand is assumed to be 250 lb/sq. in/in and the stiffness of the foundation is assumed to be 1,000 lb/sq. in/in. The inner radius of the ringwall is

14.5 ft. The weight of the roof and tank shell is calculated to be 28,400 lbs. This does not include any deadweight due to stairways or other attachments.

As will be discussed, the roof-to-shell and shell-to-bottom joints act in circumferential compression at their respective failure pressures. This can lead to circumferential buckling of the roof near the roof-to-shell joint. The same buckling can occur at the shell-to-bottom joint, although to a lesser extent. If buckling occurs, it reduces the participation of the roof in carrying the compressive load at the joint. This leads to a lower calculated failure pressure than if buckling is not taken into account. This will be discussed for the case of an empty tank.

3.1 Empty Tank (no buckling)

We will first examine the response of the empty tank to four cases:

- Zero internal gauge pressure
- The pressure required to just cause uplift of the tank
- The pressure at failure of the roof-to-shell joint
- The pressure at failure of the shell-to-bottom joint

These results are based on the elastic, large deformation, static finite element analysis in SafeRoof. Results for inelastic, large deformation, dynamic analyses are similar and are presented later in this report.

3.1.1 Zero Internal Gauge Pressure

At zero internal gauge pressure and for an empty tank, the only load is the weight of the tank. As shown in Figure 3-2, there is little displacement except at the foundation. Figure 3-3 shows a detail of this displacement, which has a value of -0.005 inch directly under the tank shell. A plot of the equivalent stress (which can be used to predict onset of yielding), is shown in Figure 3-4. The stress is largest slightly above the shell-to-bottom, however the maximum stress is only 280 psi, so it is very low.

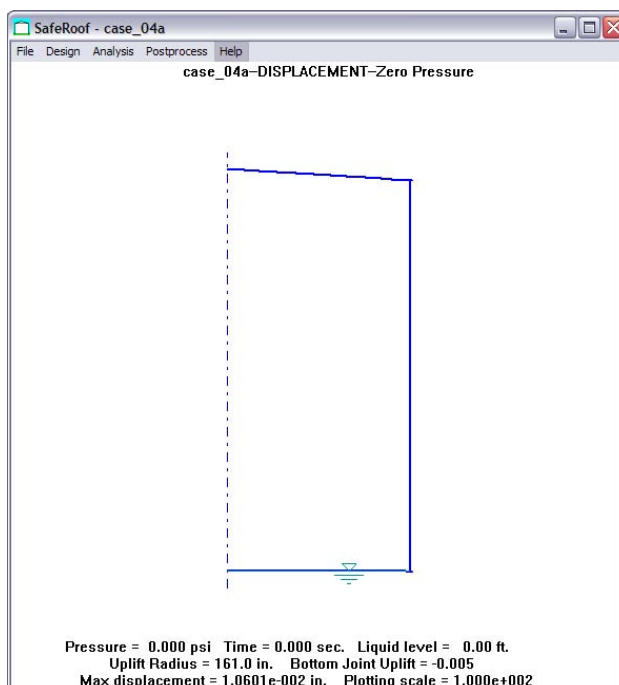


Figure 3-2: Tank displacement at zero internal gauge pressure (magnification=100x)

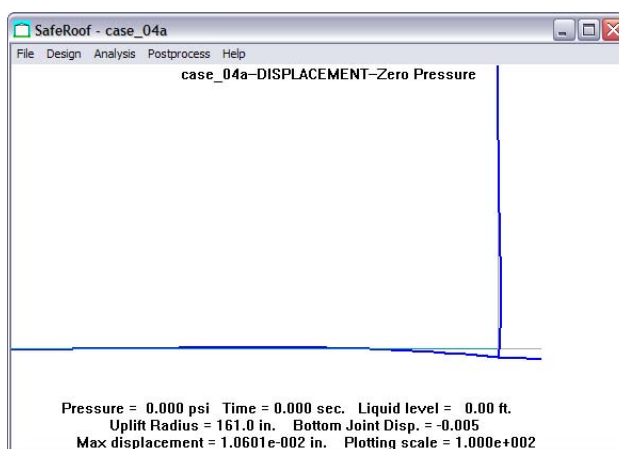


Figure 3-3: Detail of displacement of empty tank at foundation (magnification=100x)

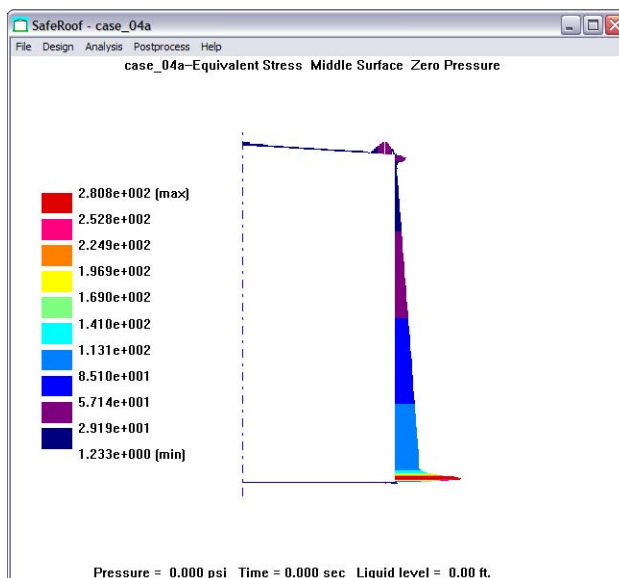


Figure 3-4: Middle surface equivalent stress contours in empty tank (min=0 psi, max=280 psi)

3.1.2 Balanced Uplift Pressure

Using the SafeRoof program, the pressure needed to just cause uplift of the empty tank (the “balanced uplift pressure” is calculated to be 0.295 psi. The deformed tank shape at this pressure is shown in Figure 3-5. The roof has lifted off the rafters and the displacement at the bottom of the shell is zero. The equivalent stresses shown in Figure 3-6 show that the peak stress is now at the roof-to-shell joint. However, the maximum equivalent stress is 10,370 psi, still below the yield stress of 36,000 psi. Therefore, no failure has occurred in the tank.

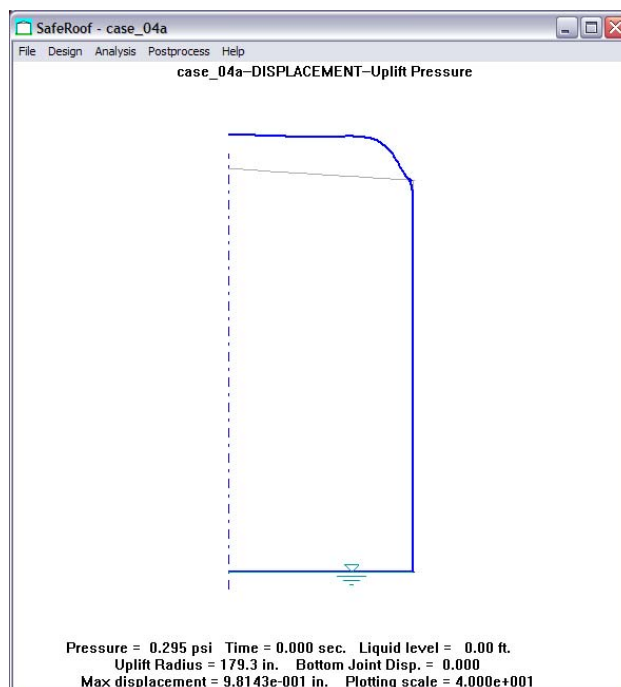


Figure 3-5: Displacement of empty tank at balanced uplift pressure (magnification=40x)

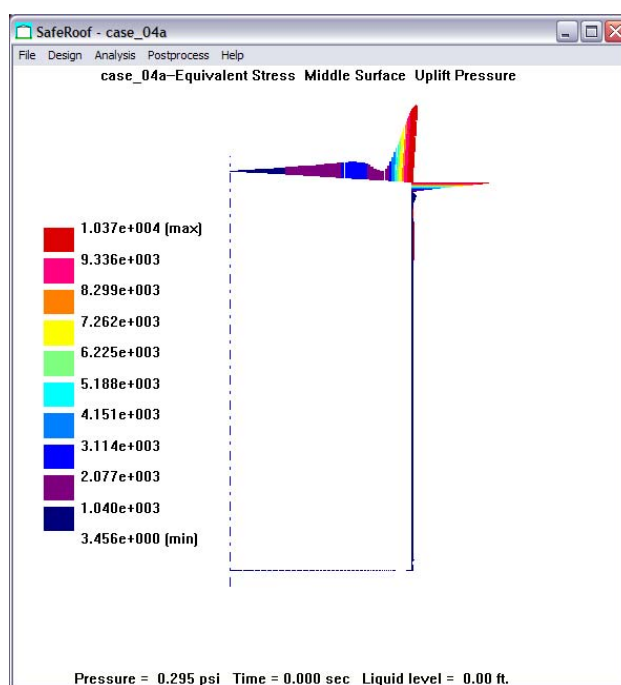


Figure 3-6: Middle surface equivalent stress at balanced uplift pressure (min=0 psi, max=10,370 psi)

3.1.3 Roof-to-Shell Joint Failure Pressure

Using the SafeRoof program, the pressure for failure of the roof-to-shell joint was calculated to be 1.04 psi. The criterion for failure of the roof-to-shell is yielding at the roof-to-shell joint in compression. This yielding then results in local buckling and kinking of the angle which causes the roof attachment weld to fail and to initiate detachment of the roof. Since the roof-to-shell failure pressure is greater than the balanced uplift pressure of 0.295 psi, significant uplift occurs before the roof-to-shell fails.

Both the roof-to-shell and the shell-to-bottom joints are in compression, as shown in the details of Figure 3-8 and Figure 3-9. This is due to the “doming” that has occurred in the roof (where it has lifted from the rafters) and the “bowling” of the tank bottom (which has resulted in a concave bottom). The deformations of both the roof and bottom result in inward radial displacements at the roof-to-shell and shell-to-bottom joints and a corresponding compressive circumferential stress.

At the bottom, the radius at which uplift starts is 98 inches (8.16 feet), so that the bottom has uplifted for a radial distance of 6.83 feet from the tank wall. The uplift displacement of the tank shell is 4.6 inches.

Equivalent stresses for the middle surface are plotted in Figure 3-10. These show that the top angle is at yielding (approximately 36,000 psi), while the stresses at the shell-to-bottom joint are large (approximately 26,000 psi), but not yet at yielding. At this load, the circumferential stresses in the bottom near the shell are in compression, Figure 3-11. The meridional stresses are in tension, with the largest (approximately 5,300 psi) values in the center of the bottom. However, in all cases, the meridional tension stresses in the bottom are much smaller than the circumferential stresses near the shell-to-bottom joint. Therefore, they are not expected to cause failure.

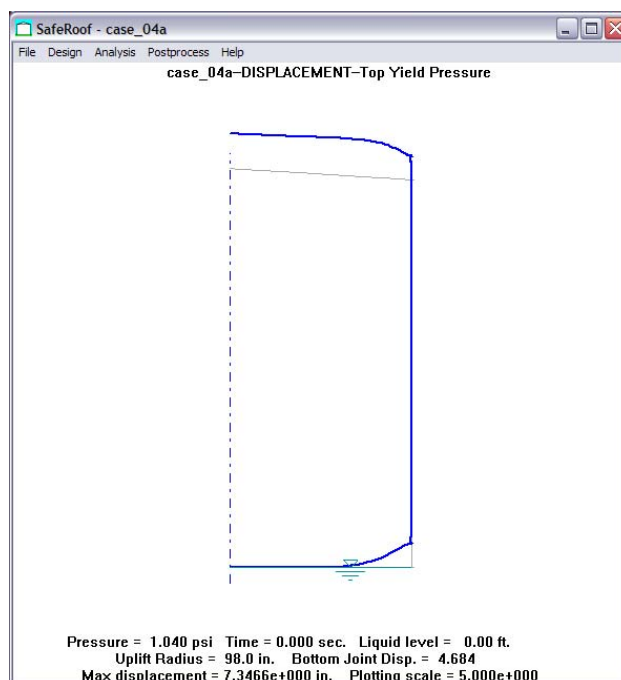


Figure 3-7: Displacement at roof-to-shell joint failure pressure (magnification=5x)

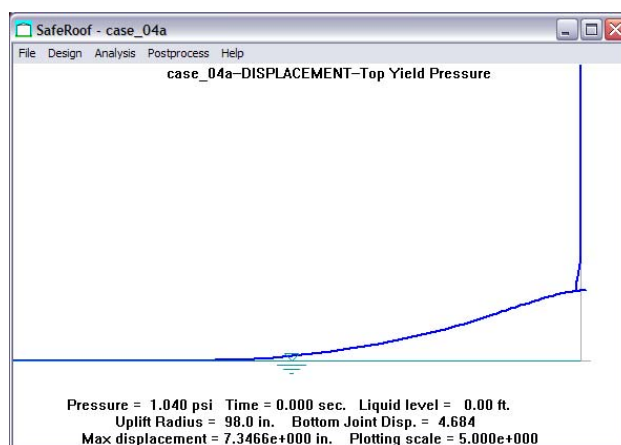


Figure 3-8: Detail of bottom displacement at roof-to-shell joint failure pressure (magnification=5x)

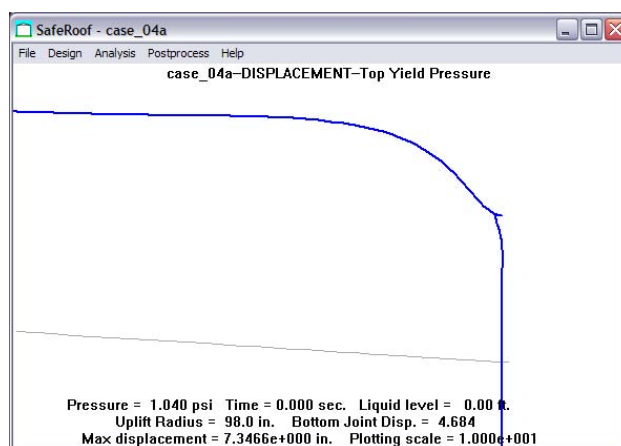


Figure 3-9: Detail of top displacement at roof-to-shell joint failure pressure (magnification = 10x)

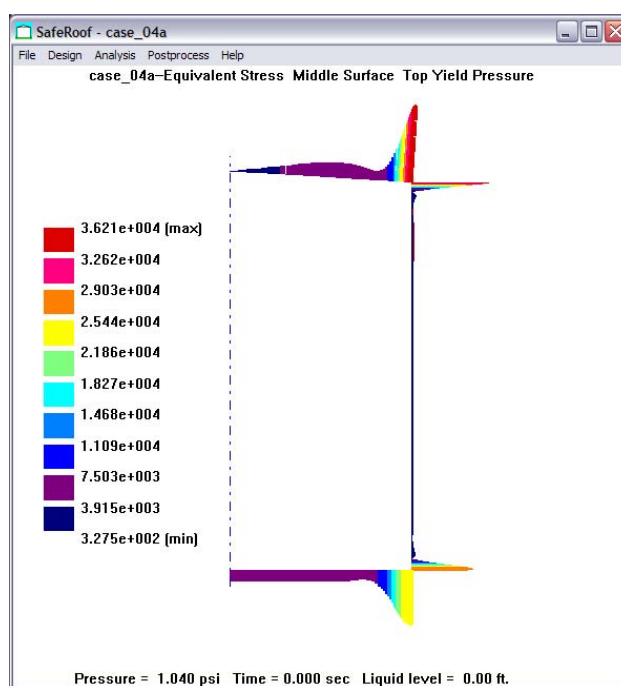


Figure 3-10: Middle surface equivalent stress at roof-to-shell joint failure pressure (min=330 psi, max=36,210 psi)

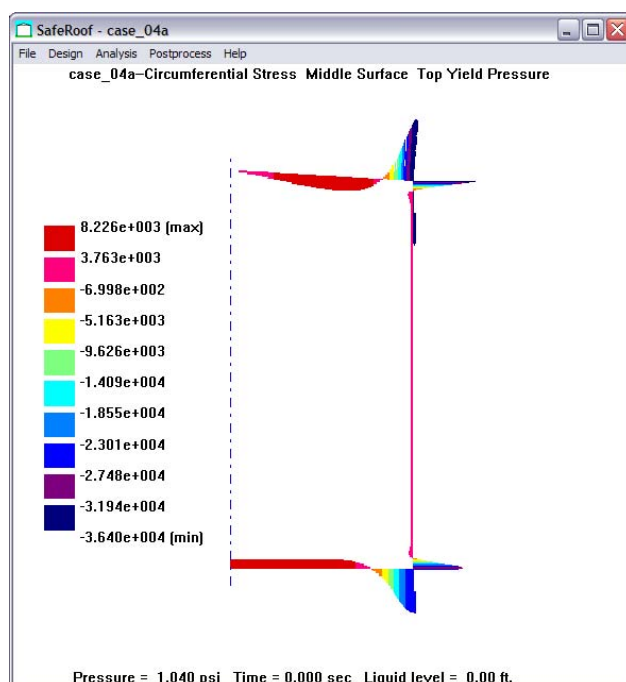


Figure 3-11: Middle surface circumferential stress at roof-to-shell joint failure pressure (min=-36,400 psi, max=8,230 psi)

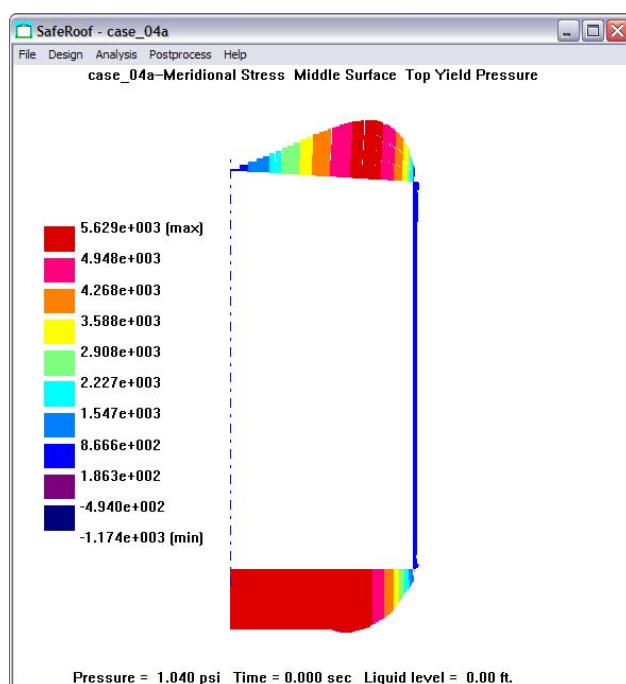


Figure 3-12: Middle surface meridional stress at roof-to-shell joint failure pressure (min=-11,740, max=5,630 psi)

3.1.4 Shell-to-Bottom Joint Failure Pressure

The calculated shell-to-bottom joint failure pressure is 1.27 psi for the empty tank. Bottom failure pressure is defined to be the pressure at which the shell-to-bottom joint reaches yielding. The largest stress component is the circumferential compressive stress that results from the “bowling” of the bottom as the shell uplifts. This “bowling” draws the outer radius of the bottom (and of the bottom of the shell) radially inward, causing the large compressive stress. As for the roof-to-shell joint, the failure mode is local buckling at the shell-to-bottom joint.

The tank deformed at this pressure is shown in Figure 3-13. At this higher pressure, the uplift radius is reduced to 90 inches and the magnitude of the uplift is increased to 5.5 inches.

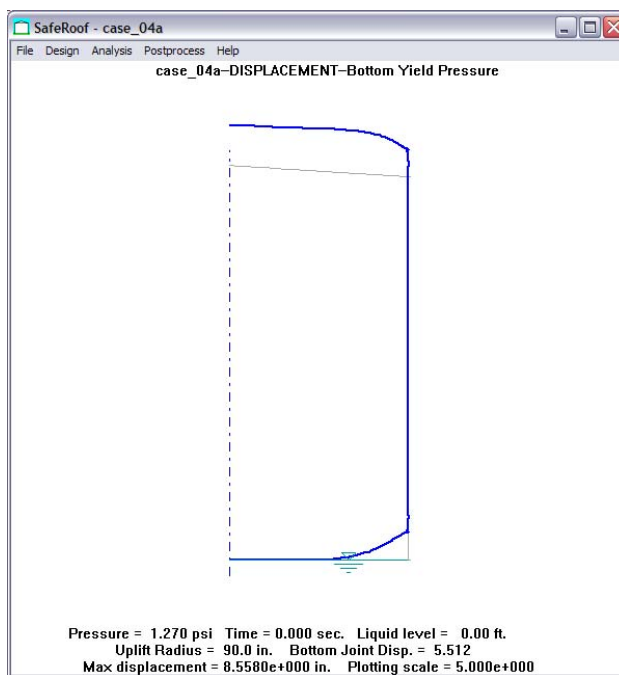


Figure 3-13: Deformation at shell-to-bottom joint failure pressure (magnification=5x)

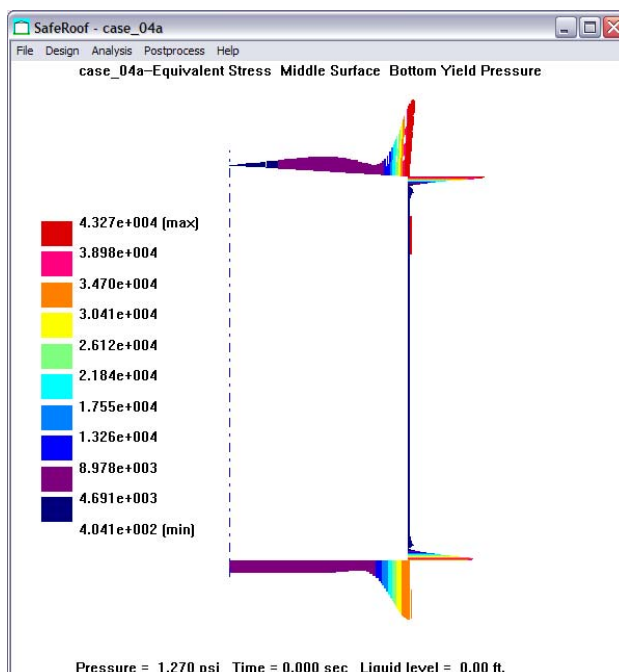


Figure 3-14: Middle surface equivalent stresses at shell-to-bottom joint failure pressure (min=400 psi, max=43,270 psi)

3.2 Full Tank (no buckling)

The response of a full tank is different at the shell-to-bottom joint. Because the product level does not affect the roof-to-shell joint, the failure pressure of the roof-to-shell joint will remain the same for both empty and full tanks. As for the empty tank, we will examine the response of the full tank to four cases:

- Zero internal gauge pressure
- The pressure required to just cause uplift of the tank
- The pressure at failure of the roof-to-shell joint
- The pressure at failure of the shell-to-bottom joint

These results are based on the elastic, large deformation, static finite element analysis in SafeRoof. Results for inelastic, large deformation, dynamic analyses are similar and are presented later in this report.

3.2.1 Zero Internal Gauge Pressure

The displacements for a full tank at zero internal gauge pressure are shown in Figure 3-15 and Figure 3-16. Figure 3-16 clearly shows the downward displacement of the bottom due to the pressure load of the product. The product also causes the circumferential stress to increase approximately linearly with depth, Figure 3-17. However, at the shell-to-bottom joint, the bottom (which is relatively stiff in tension) constrains the radial displacement of the shell, decreasing the circumferential stresses near the joint.

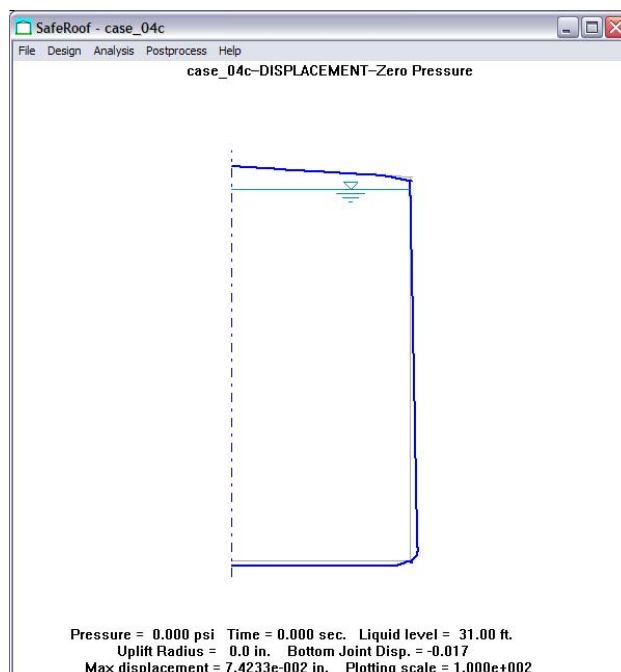


Figure 3-15: Displacement for a full tank at zero internal gauge pressure (magnification=100x)

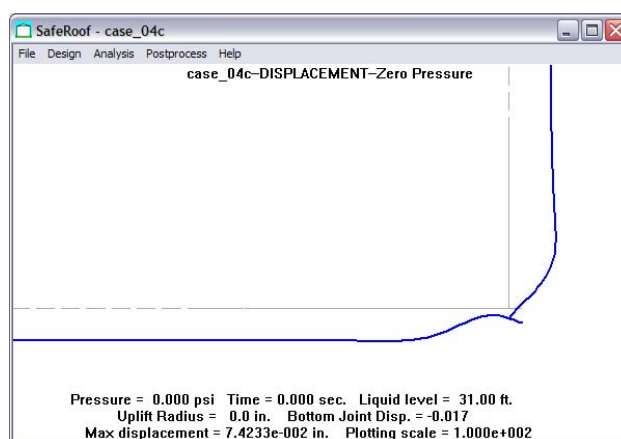


Figure 3-16: Detail of displacement for full tank at zero internal gauge pressure (magnification=100x)

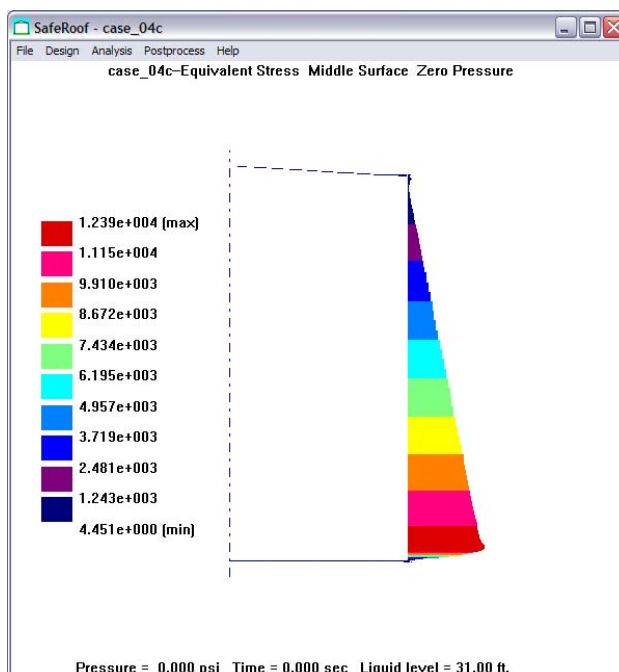


Figure 3-17: Middle surface equivalent stress for a full tank at zero internal gauge pressure (min=0 psi, max=12,390 psi)

3.2.2 Balanced Uplift Pressure

The full tank balanced uplift pressure is calculated to be 0.80 psi. A full tank has a higher balanced uplift pressure because the tank is resting on an elastic foundation. The force to uplift the tank must not only be greater than the tank weight, but it must also be sufficient to compensate for the reduced support of the elastic foundation as the bottom is lifted, Figure 3-18.

Even though the balanced uplift pressure for a full tank is greater than for an empty tank, the balanced uplift pressure is still smaller than the failure pressure of the roof-to-shell joint, so some uplift will occur before the frangible joint fails and relieves the internal pressure.

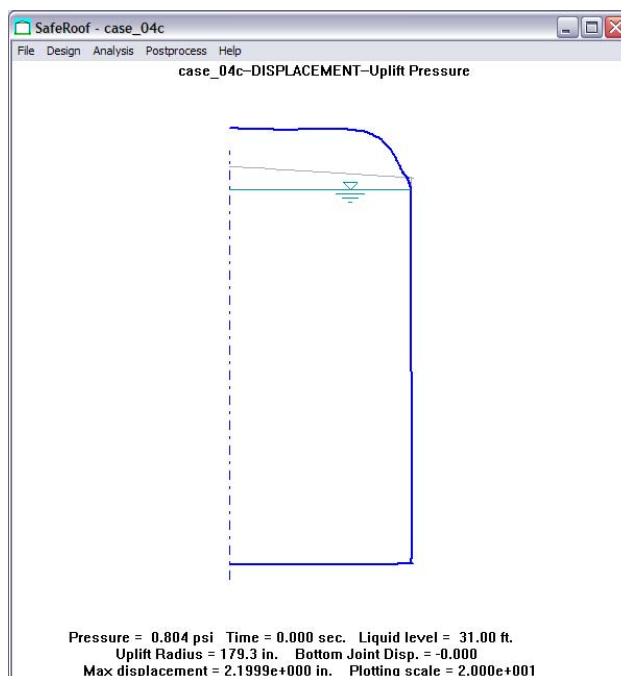


Figure 3-18: Displacement of full tank at balanced uplift pressure (magnification=20x)

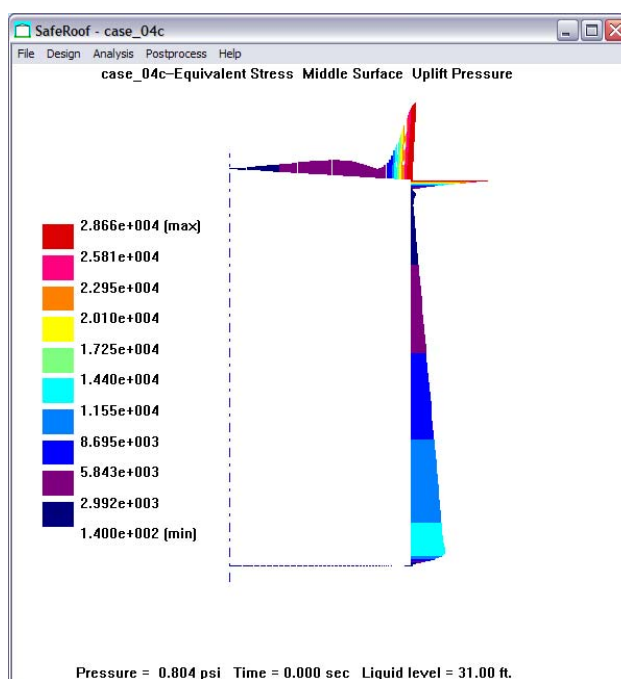


Figure 3-19: Middle surface equivalent stress at balanced uplift pressure (min=140 psi, max=28,660 psi)

3.2.3 Roof-to-Shell Joint Failure Pressure

The failure pressure of the roof-to-shell joint remains the same as the empty tank (1.04 psi), however the displacements at the shell-to-bottom joint are very different than for the empty case, Figure 3-20 and Figure 3-21. Although there is some uplift, the radius of first uplift is 176 inches (14.6 feet), nearly equal to the tank radius of 15 feet. The magnitude of the bottom uplift is also much smaller, at 0.027 inch.

Because the uplift is less, the stresses at the shell-to-bottom joint for a full tank at the roof-to-shell joint failure pressure are also less than an empty tank. For the empty tank the stress was approximately 26,000 psi, while for the full tank the shell-to-bottom joint stress is approximately 13,000 psi. Thus, a tank full of product has the effect of actually reducing the stress at the shell-to-bottom joint at the roof-to-shell joint failure pressure.

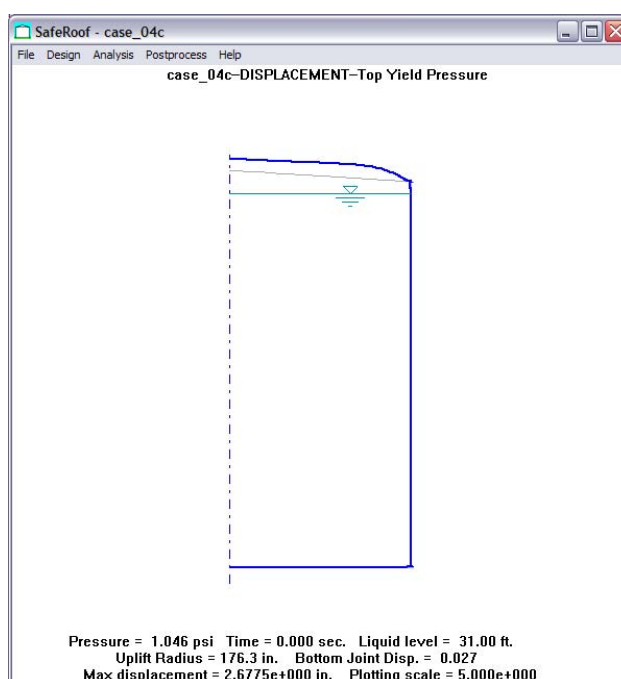


Figure 3-20: Displacement of full tank at roof-to-shell failure pressure (magnification=5x)

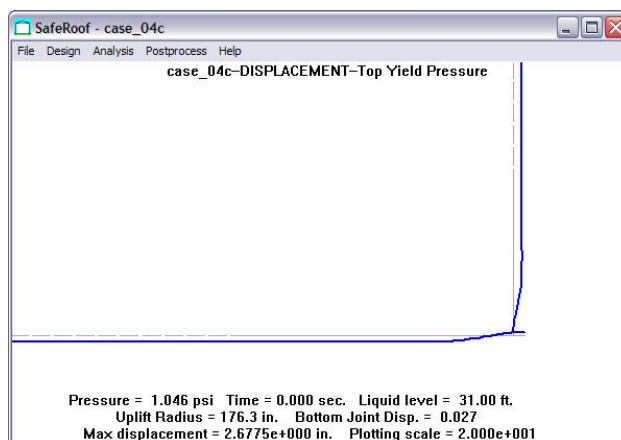


Figure 3-21: Detail of displacement for full tank at top failure pressure (magnification=20x)

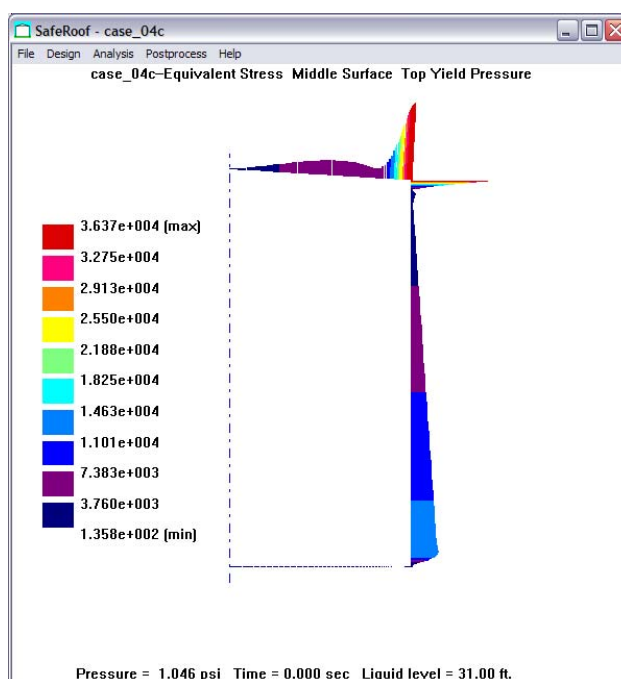


Figure 3-22: Middle surface equivalent stress at roof-to-shell joint failure pressure (min=130 psi, max=36,370 psi)

3.2.4 Shell-to-Bottom Joint Failure Pressure

The shell-to-bottom joint failure pressure for a full tank is calculated to be 3.25 psi. The displacements at this pressure are shown in Figure 3-23 and Figure 3-24. At this pressure the radius of first uplift is 161 inches (13.4 feet). The uplift magnitude is 2.35 inches.

Stresses are plotted in Figure 3-25. At this pressure, the stresses in the shell-to-bottom are just at yielding (the elastic roof-to-shell joint stresses have far exceeded yielding).