Failure Modes of American Petroleum Institute 12F Tanks With a Rectangular Cleanout and Stepped Shell Design

The design and fabrication of shop-welded and prefabricated relatively small tanks, when compared to field-welded tanks, used in the upstream segment of the oil and gas industry is governed by the American Petroleum Institute specification 12F (API 12F). This study explores the changing designs of API 12F tanks to include a new rectangular cleanout design with reinforcement as shell extension internally of cleanout frame and a stepped shell design. This study also investigated the introduction of two additional tank sizes in addition to existing eleven tank sizes in the current 12th edition of API 12F. The adequacy of the new design changes and proposed tank designs were verified by elastic stress analysis with nonlinear geometry, elastic–plastic stress analysis with nonlinear geometry, and elastic buckling analysis to verify their ability to operate at a design internal pressure of 16 oz/in² (6.9 kPa) and maximum pressure during emergency venting of 24 oz/in² (10.3 kPa). A vacuum pressure of 1.5 oz/in² (0.43 kPa) was also investigated using the elastic buckling analysis. The stress levels and uplift of the tanks are reported in this report to provide insights into the behavior of proposed API 12F tanks exposed to higher internal pressure and vacuum pressure. [DOI: 10.1115/1.4041340]

Keywords: internal pressure, large deformations, nonlinear geometry, nonlinear material, rupture analysis API 12F tanks

1 Introduction

Temporary storage of relatively small amounts of oil upon extraction in the upstream segment of oil and gas industry is typically accomplished by cylindrical aboveground steel storage tanks that are transportable and can withstand a range of operating pressure. American Petroleum Institute (API 12F) “specification for shop welded tanks for storage of production liquids” [1] lists tank designs that meet the operating pressure criteria and perform well within the required storage capacities. The tanks listed in the 12th edition of API 12F specification are shop-welded and transportable with sizes ranging from 90 bbl (14.3 m³) to 750 bbl (119.2 m³), where bbl is U.S. barrels of oil for flat bottom tanks. Figure 1 shows the general shape of a proposed API 12F tank design with a stepped-shell thickness where the bottom shell course (tub ring) is equal or greater in thickness compared to the upper shell course(s).

Although these API 12F tanks are relatively small in size compared with larger API 650 field erected tanks, they are produced in large numbers to accommodate for the demand of temporarily storing extracted oil during exploration and extraction processes. To ensure the reliability of the pressure vessels, tanks, and attached nozzles and appurtenances, detailed industry standards and codes have been developed [2–4].

This study seeks to expand the API 12F design options and offer more economical tank configurations. Introducing two new storage tanks and increasing the operational internal pressure and vacuum pressure capacities are also investigated in this study. The tanks investigated in this study are listed in Table 1, where the case numbering convention is adopted from the work of Rondon and Guzey [5–9].

This work is a continuation and expansion on the work done in phase 1 and phase 2 by Rondon and Guzey [5–9]. Phase 1 of this project modeled eleven existing API 12F [1] tanks in addition to two proposed new tanks for an internal pressure increase to 24 oz/
Table 1 Dimensions of studied tanks and their nominal capacities

<table>
<thead>
<tr>
<th>Tank case</th>
<th>Nominal capacity bbl (m³)</th>
<th>Outside diameter ft (m)</th>
<th>Shell height ft (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>90 (14.3)</td>
<td>7.92 (2.4)</td>
<td>10 (3)</td>
</tr>
<tr>
<td>2</td>
<td>100 (15.9)</td>
<td>9.5 (2.9)</td>
<td>8 (2.4)</td>
</tr>
<tr>
<td>3</td>
<td>150 (23.8)</td>
<td>9.5 (2.9)</td>
<td>12 (3.7)</td>
</tr>
<tr>
<td>4</td>
<td>200 (31.8)</td>
<td>12 (3.7)</td>
<td>10 (3)</td>
</tr>
<tr>
<td>5</td>
<td>210 (33.4)</td>
<td>10 (3)</td>
<td>15 (4.6)</td>
</tr>
<tr>
<td>6</td>
<td>250 (39.7)</td>
<td>11 (3.4)</td>
<td>15 (4.6)</td>
</tr>
<tr>
<td>7</td>
<td>300 (47.7)</td>
<td>12 (3.7)</td>
<td>15 (4.6)</td>
</tr>
<tr>
<td>8</td>
<td>400 (63.6)</td>
<td>12 (3.7)</td>
<td>20 (6.1)</td>
</tr>
<tr>
<td>9</td>
<td>500 (79.5)</td>
<td>12 (3.7)</td>
<td>25 (7.6)</td>
</tr>
<tr>
<td>10</td>
<td>500 (79.5)</td>
<td>15.5 (4.7)</td>
<td>16 (4.9)</td>
</tr>
<tr>
<td>11</td>
<td>750 (119.2)</td>
<td>15.5 (4.7)</td>
<td>24 (7.3)</td>
</tr>
<tr>
<td>12*</td>
<td>1000 (159)</td>
<td>15.5 (4.7)</td>
<td>30 (9.1)</td>
</tr>
<tr>
<td>14*</td>
<td>300 (47.7)</td>
<td>16.5 (5)</td>
<td>8 (2.4)</td>
</tr>
</tbody>
</table>

*aProposed tanks not in the current 12th edition of API 12F.

in² (10.3 kPa). The finite element analysis (FEA) models included a proposed cleanout with a semicircular top and repad to determine failure pressure and tank performance at increased pressure. Phase 2 of the project performed fatigue and brittle fracture evaluation. Although, the proposed cleanout with a semicircular top and repad detail was a good detail to reduce stress concentrations, it was later decided that a new rectangular cleanout detail, a more economical detail, would be more suitable for the needs of the upstream segment of the oil and gas industry. Therefore, in this phase of the study (phase 3), tanks with a new rectangular cleanout detail were investigated.

A computational approach has been used in this study by modeling the tanks in ABAQUS [10], a finite element analysis software, to implement various design parameters and obtain deformations and stresses in the tanks. As tanks with conical roof can fail in modes of buckling [11–18], plastic collapse [16–25], and overstressing due to roof-to-shell intersection [26–28], different types of analyses have been employed to investigate the different failure criteria of the tanks and assert their safety.

1.1 Types of Failure Modes. This study investigates the possibility of increasing the internal design pressure for some of the API 12F tanks to 16 oz/in² (6.9 kPa) and introducing a maximum internal pressure during emergency venting of 24 oz/in² (10.3 kPa) for all the tanks. Increasing the design vacuum pressure from 1/2 oz/in² (0.14 kPa) to 1.5 oz/in² (0.43 kPa) was also investigated. BPVC 2017, Section VIII, division 2 (ASME VIII-2), part 5 design by analysis approach [2] was followed by incorporating the suitable allowable stress and joint efficiency factors of API 650 [3].

The first type of analysis is an elastic stress analysis, with linear elastic material properties and nonlinear large deformations that was performed to obtain the stress levels at three locations of interest in the tank at the two design pressures of 16 oz/in² (6.9 kPa) and 24 oz/in² (10.3 kPa). The elastic stress analysis was also used to obtain the uplift magnitude of the tanks at these two pressure levels of interest.

The second type of analysis is the elastic-plastic stress analysis that was performed to obtain the plastic strain experienced by the tank at internal pressure levels of 16 oz/in² (6.9 kPa) and 24 oz/in² (10.3 kPa). This illustrates the ability of the material to withstand the suggested internal pressure causing localized stress concentrations near geometric discontinuities.

The third type of analysis is the elastic buckling analysis, which was performed to ensure the tank and its components do not experience instability within the operating and emergency pressures. This analysis investigated the performance of the tanks operating with positive internal pressure and vacuum pressure.

Fig. 2. Typical configuration of rafters in tanks with diameters equal to or greater than 15.5 ft (4.7 m). Shown in the figure tank case 10 with a diameter of 15.5 ft (4.7 m) and shell height of 16 ft (4.9 m).

1.2 Tank Design Parameters. This investigation looks into the possibility of changing the design parameters of the current 12th edition of API 12F [1] tanks to be able to withstand higher design pressures. The current 12th edition of API 12F tank designs requires the shell and the roof to have the same thickness, and requires a minimum shell thickness of 3/16 in (4.8 mm) or 1/4 in (6.4 mm) but does not give recommendations on how to use both thicknesses in the same tank. This study investigates the viability of introducing a stepped-shell design with a new combination of 1/4 in (6.4 mm) thick bottom course (tub ring) and a 3/16 in (4.8 mm) thick top shell. The investigation studied the following cases:

1) 3/16 in (4.8 mm) thick shell the entire height,
2) 1/4 in (6.4 mm) thick shell the entire height,
3) 1/4 in (6.4 mm) thick bottom 4 ft (1.2 m) high course, first shell course (tub ring), and 3/16 in (4.8 mm) thick remaining shell courses.

The effect of the thickness of the bottom plate was investigated by modeling all tanks with a 1/4 in (6.4 mm) thick and 3/8 in (9.5 mm) thick bottom plate. The roof has been modeled with a 3/16 in (4.8 mm) thickness for all tanks, and was also modeled with a thickness of 1/4 in (6.4 mm) for tanks with a diameter of 15.5 ft (4.7 m) and 16.5 ft (5 m).

Rafter support was provided for tanks with a diameter of 15.5 ft (4.7 m) and 16.5 ft (5 m) when the roof plate was modeled with a thickness of 3/16 in (4.8 mm) in tank cases 10–12 and 14. Figure 2 shows a typical rafter support configuration in tank case 10.

Current provisions of API 12F suggest an extended neck cleanout in the design, which is replaced in this study with a cleanout reinforced with shell extension internally of frame as shown in Fig. 3. The cleanout investigated in phase 1 [5] and phase 2 [7] of this project had a semicircular top of radius of 12 in (0.3 m) and its neck was extended into the shell for reinforcement. Also, that semicircular cleanout detail had repad with same thickness as its shell plate.

1.3 Background Information

1.3.1 American Petroleum Institute 12F Specifications for Shop-Welded Tanks for Storage of Production Liquids. The design of shop-welded tanks suitable to be transported to the oil extraction site is provided by API 12F [1]. These tanks are shop-
Regardless of the form of the failure mode the storage tanks may take, the stored product is at risk of being leaked outside. This can lead to severe environmental disasters and financial burdens [30,31], which encouraged research in the field of tank failure. One of the approaches to address the failure of the tank under uncontrolled increase in internal pressure is to have a design that would fail in a favorable mode. A favorable failure mode is one that results in no content leak as to not affect the adjacent structures or contaminate the ground with the product. Thus, the failure of the top junction before the failure of the bottom junction is favorable as it will lead to venting the excessive internal pressure without allowing the product to escape. This frangible roof design philosophy was adopted by API 650 [3] by providing provisions facilitating the calculation of the failure pressure of the top junction. However, Lu et al. [32] showed with experiments that the approach provided by API 650 is relatively conservative. The same study has been presented as API 937 report [4] in which the analysis of frangible roof design is derived.

Swenson et al. [4] investigated the behavior of roof-to-shell junctions in tanks using a finite element analysis approach to evaluate the API 650 provisions for estimating the failure pressure. Swenson et al. [4] concluded that the provisions provided by API 650 underestimate the failure pressure significantly.

Swenson et al. [4] showed the derivation of the maximum design pressure results in the formulation shown in Eq. (1), and the failure pressure in Eq. (2) from API 650

\[
P = \frac{8A\sigma_{\text{yield}}\tan \theta}{hD^2} + 8\rho_{\text{water}}t_f
\]

\[
P_f = 1.6P - 4.8\rho_{\text{water}}t_f
\]

where \( P \) is the internal design pressure in psi, \( A \) is the area resisting the compressive force in square inches, \( \sigma_{\text{yield}} \) is the material yield strength in psi, \( \tan \theta \) is the slope of the roof, \( n \) is the safety factor of 1.6, \( D \) is the nominal shell diameter in inches, \( \rho_{\text{water}} \) is the density of water in \( \text{psi}/\text{in} \), \( t_f \) is the thickness of the roof in inches, \( P_f \) is the calculated failure pressure in psi. The current 12th edition of API 650 [3] contains the same equations formatted with the safety constant and unit conversion factors pre-applied and taken into consideration. This study adopted the procedure provided by the current 12th edition of API 650 [3] in calculating the area resisting compression, \( A \).

Swenson et al.’s findings as summarized in report API 937 [4] indicate that the estimated failure pressure as calculated by Eq. (2) is significantly lower than the pressure obtained through the FEA approach. It is expected that the internal pressure causing failure at the top junction found in this study using FEA will be higher than the estimated failure pressure calculated using Eq. (2).

### 2 Methodology

This study investigated the failure modes of vertical, cylindrical, closed-top, flat bottom, aboveground shop-welded storage tanks due to internal pressure and vacuum pressure. The modeled tanks are based on the current 12th edition of API 12F [1] tanks in addition to two more proposed tanks to be potentially included in the revised 13th edition of API Standard 12F. Table 1 lists the tanks that were under consideration in this study. Tank cases 1–11 are provided in the current edition of API 12F tank designs; tank cases 12 and 14 are proposed designs that are under consideration to be included in the new 13th edition of API 12F.

The roofs of the tanks were designed to have a conical shape with a 1/12 roof slope. The roof plate had a projection extending outside the shell of 3/8 in (9.5 mm) measured from the outside surface of the shell, as shown in Fig. 5. The thickness of the roof was 3/16 in (4.8 mm) for all tanks, and the tanks with a diameter of 15.5 ft (4.7 m) or larger (i.e., cases 10–12 and 14) were modeled with rafters to support the roof. Moreover, these larger tanks are subjected to additional loads, such as wind and seismic loads, which can affect the design of the top junctions.

![Diagram of New proposed cleanout opening where it is reinforced with a shell extension internally of frame (to convert in to mm multiply values by 25.4)](Image)

**Fig. 3** New proposed cleanout opening where it is reinforced with a shell extension internally of frame (to convert in to mm multiply values by 25.4)
diameter tanks were also modeled with a 1/4 in (6.4 mm) roof and no rafters and with an entire shell thickness of 1/4 in (6.4 mm).

Figure 2 shows a typical rafter support configuration in tank case 10. The rafters were attached to the shell and followed the slope of the roof, but were not attached to the roof. Eight C6x8.2 beam-shaped rafters were modeled resting on top of a 24 in (0.61 m) diameter, 1/4 in (6.4 mm) thick center circular ASTM A36 plate. The center circular plate rested on top of a 6 in schedule 40 pipe, 6.625 in (168.3 mm) outside diameter, 0.28 in (7.1 mm) thick central pipe column of ASTM A53 Gr. B material specification. In actual production API 12F tanks, the roof support structure does not typically have a center support column. However, this minor difference may not grossly change the behavior of the tanks because rafters are not nodally attached to the roof plates.

The bottom plate was modeled with a flat circular plate profile, and a chime projection of 3/8 in (9.5 mm) measured from the outside surface of the shell, which is shown in Fig. 5. The bottom plate was modeled with two different thicknesses: 1/4 in (6.4 mm) and 3/8 in (9.5 mm).

The welds connecting the roof to the shell and the bottom plate to the shell are full filet-welds with a 45 deg slope. The full-filet weld is defined in the 12th edition of API 12F as a filet weld

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**Fig. 4** A Rectangular cleanout design (a) the actual cleanout geometry and dimension as proposed in the draft version 13th edition API 12F dated Nov 9/2017 [29] (b) schematic view of the modeled cleanout in FEA without the bolts, bolting flange, or gasket (to convert in to mm multiply values by 25.4, to convert ft to m multiply values by 0.3)

**Fig. 5** Top and bottom junctions with a chime projection of 3/8 in (9.5 mm)

**Fig. 6** Shell-to-bottom plate fillet weld geometry and FEA idealization
whose leg size is equal to the thickness of the thinner member joined. The full-filet welds at the top and bottom junctions with the roof and bottom plate, respectively, have been idealized in the FEA model using the methodology described by Niemi et al. [33]. This Niemi et al. weld idealization models the welds as shell elements with a thickness equal to the thickness of the weld throat, \( t_w \). A typical shell-to-bottom or roof-to-shell joint weld detail as modeled in this study is shown in Fig. 6.

In the elastic stress analysis, all parts of the model, including the welds, were assumed to correspond to the elastic ASTM A36 steel material [34] having an isotropic carbon steel material with a modulus of elasticity of 29,000 ksi (200 GPa), Poisson’s ratio of 0.3, density of 490 lb/ft\(^3\) (7800 kg/m\(^3\)), a yield strength, \( F_y \), of 36 ksi (250 MPa) and ultimate tensile strength, \( F_{\mu} \), of 58 ksi (400 MPa). In the elastic-plastic stress analysis, the true stress–strain curve describing the ASTM A36 steel was modeled using the true stress–strain curve provided by ASME VIII-2 with a yield strength, \( F_y \), of 36 ksi (250 MPa) and ultimate tensile strength, \( F_{\mu} \), of 58 ksi (400 MPa). The true stress–strain curve is shown in Fig. 7. Consideration for higher temperatures was not taken because this work is intended to investigate the behavior of the API 12F tanks under operating temperatures. Additionally, internal pressure would increase faster than the temperature of the metal if internal combustion occurs [4]; thus, the behavior of the metal is expected not to change significantly at the early stages of the combustion.

The influence of the level of the stored liquid on the stress and bottom uplift values for each internal pressure magnitude has been investigated. A liquid with a specific gravity of 1.0 was modeled by introducing hydrostatic pressure acting on the shell and bottom plate to the model. The water density was taken as 62.4 lb/ft\(^3\) (1000 kg/m\(^3\)). The tanks have been studied with two levels of bottom plate to the model. A liquid with a specific gravity of 1.0 was modeled in this study is shown in Fig. 6.

Finite element analysis was performed to simulate the stresses and deformations in the tanks as the internal pressure increased. ABAQUS/CAE version 2017 [10], a general purpose FEA program, was used to model the geometry and run the analyses for all the tank configurations. The shell elements used were four node quadrilateral bi-linear S4R dominated, with some three node triangular S3R elements, which were used to optimize the computational time. The S4R elements are reduced integration elements without hourglass control and finite membrane strain. The S3R elements are reduced integration elements without hourglass control, and finite membrane strain. The mesh size decreased from the center of the bottom plate and roof to the top and bottom junctions, respectively. The mesh size also decreased from the middle of the shell to the cleanout and bottom and top junctions. The mesh size ranged from 5 in (127 mm) near the center of the roof and mid-height of the tank, to 1/2 in (12.7 mm) near the edges of the top, bottom, and cleanout junctions; Fig. 8 shows a typical FEA mesh for tank case 3, 9.5 ft (2.9 m) diameter, 8 ft (3.7 m) height tank. A convergence analysis was performed to select the appropriate mesh size for each tank.

The tank foundation was considered as compacted soil to have a subgrade modulus of 0.250 ksi/in (68 GPa/mm) and was idealized as spring elements acting in the vertical direction in accordance with the Winkler model [35]. All the element nodes at the bottom surface were attached to springs to simulate the soil tank interaction. To estimate the uplift, the soil was modeled as compression only springs attached to the bottom face of the tank.

The cleanout shape is illustrated in Figs. 3 and 4 as modeled in the tanks. The rectangular cleanout has a neck and cover plate thickness of 1/4 in (6.4 mm). The actual cleanout neck and bolting flange is made using A36 angle iron L3×2×1/4 (L76×76×6.4). In the FEA model bolting flange, gasket and bolts were not modeled. FEA nodes of cover plate were directly attached to the cleanout neck (see Figs. 1, 2, and 4). This study assumes the structural contribution of the bolting flange and gaskets and bolts to be insignificant and negligible. The thickness of the neck of the cleanout is 1/4 in (6.35 mm); the depth of the cleanout is a maximum of 3 in (76.2 mm). The cleanout has shell extension internally of frame as reinforcement as shown in Fig. 3. Note that the shell elements in this detail were modeled at the

### Fig. 7
True stress–strain relationship for mild steel ASTM A36 based on ASME VIII-2 (multiply by conversion factor of 6.89 to convert ksi results to MPa)

### Fig. 8
Tank case 3 with a finer mesh near the top, bottom, and cleanout junctions

centerline of each shell and plate to account for the actual space the material takes.

There were three types of analyses performed: (1) elastic stress analysis, (2) elastic–plastic stress analysis, and (3) elastic buckling analysis. The elastic stress analysis was used to find the internal pressure at which the top and bottom junctions reach yielding. It was also used to find the maximum stresses and bottom uplift of the tanks at the design internal pressures of 16 oz/in² (6.9 kPa) and 24 oz/in² (10.3 kPa), and design vacuum pressure of 1.5 oz/in² (0.43 kPa). The elastic–plastic analysis utilized a nonlinear stress–strain profile in accordance with ASME VIII-2 part 5, and was used to find the internal pressure value at which rupture occurs. Also, we used the elastic-plastic analysis for the further evaluation of localized stresses at the cleanout junction at the internal design pressure of 16 oz/in² (6.9 kPa) and maximum emergency venting pressure of 24 oz/in² (10.3 kPa). The elastic buckling analysis used elastic material to find the internal pressure and vacuum that will cause the tank to buckle.

2.1 Elastic Stress Analysis. An elastic stress analysis was performed to estimate the approximate internal pressure at which the top and bottom junctions fail by yielding. The elastic analysis assumed a perfectly linear elastic ASTM A36 steel [34], yielding at 36 ksi (250 MPa). Nonlinear large deformations were considered in the analysis by using a modified Riks’ method [36] in ABAQUS [10] to incrementally increase the internal pressure and capture the nonlinear progression of deformation and stress level. The effects of the self-weight and hydrostatic pressure were captured by applying these loads first as an initial step and only increasing the internal or external pressure. Compression-only springs with constant stiffness were used to simulate the soil–tank interaction where the soil had a subgrade modulus of 0.250 ksi/in (68 GPa/mm).

The provisions provided by ASME VIII-2 paragraph 5.2.2 were followed in this analysis. The bottom uplift and stress levels at the bottom junction were evaluated at internal design pressures of 16 oz/in² (6.9 kPa) and 24 oz/in² (10.3 kPa), and vacuum pressure of 1.5 oz/in² (0.43 kPa).

The stress level value was evaluated as the von Mises equivalent stress at the midsurface of the shell element at the integration point. The following equation is used to calculate the von Mises equivalent stress:

\[
\sigma_v = \frac{1}{\sqrt{2}} \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}
\]

where \(\sigma_v\) is the von Mises equivalent stress, and \(\sigma_1, \sigma_2,\) and \(\sigma_3\) are the principle stresses at the evaluation point.

The global failure in the tank has been judged to occur away from the cleanout junction when the membrane stress is larger than \(S_m = 23,200\) psi (160 MPa) and the membrane plus bending stress is \(S_m = \) larger of \(1.5S_m\) or \(F_m\). Because \(1.5S_m = 34,800\) psi (240 MPa) and \(F_m = 36,000\) psi (250 MPa), \(S_m = \) equal to \(F_m = 36,000\) psi (250 MPa). This is the yielding stress of the A36 steel and the failure criteria for ASME VIII-2.

The protection against local failure criteria in accordance with ASME VIII-2 is investigated near the cleanout junction with the provisions using the elastic stress analysis and is confirmed with the elastic-plastic stress analysis. The local failure was determined to occur when the membrane stress near discontinuities, \(\sigma_v\), is compared to \(S_m = \) larger of \(1.5S_m\) or \(F_m\). Because \(1.5S_m = 34,800\) psi (240 MPa) and \(F_m = 36,000\) psi (250 MPa), \(S_m = \) equal to \(F_m = 36,000\) psi (250 MPa). Similarly, the primary and secondary membrane plus bending stresses, \(\sigma_v\), are compared to \(S_m = \) larger of \(3S_m\) or \(2F_m\). Thus, \(S_m = 2F_m = 72,000\) psi (500 MPa) in accordance with ASME VIII-2 paragraph 5.2.2.4 for load case combinations and load factors for elastic stress analysis evaluated near the discontinuity junction.

2.2 Elastic–Plastic Stress Analysis. Another mode of failure investigated is one due to the plastic behavior of the material and nonlinear deformation of the tank. To include the effects of the plastic behavior of ASTM A36 steel, a stress–strain curve from ASME VIII-2 was used and is shown in Fig. 7.

The nonlinear deformation behavior of the structure was captured using the modified Riks’ method [36] in ABAQUS [10]. Initially, the self-weight and hydrostatic pressure were applied to the tank in a static step; then they were propagated to the Riks’ analysis step, in which the internal pressure load was applied and increased incrementally until failure.

This analysis was used to investigate the rupture failure mode due to the nonlinearity in the material stress–strain relationship. To ensure that protection against the local failure near the cleanout junction, the provisions of ASME VIII-2 were followed. The limiting triaxial strain, \(\varepsilon_{tot}\), and plastic strains, \(\varepsilon_{plast}\), have been assessed for each tank at the internal pressure of 24 oz/in² (10.3 kPa) with a safety factor of 2.5 and joint efficiency of 0.7.

The safety factor of 2.5 is the API 650 [3] safety factor against ultimate stress: \(F_s = 58\) ksi/(23.2 ksi = 2.5); \(S_m\) is the maximum allowable product design stress as permitted by API 650 for ASTM A36 steel. The joint efficiency of 0.7 is assigned to joints that do not undergo spot radiography inspection following the rules of API 650 Annex A. Note that API 12F tanks do not receive any radiography inspection. The safety factor and joint efficiency produce a design factor, \(\Phi\), of \(\Phi = (2.5)/(0.7) = 3.57\). The internal pressure corresponding to 24 oz/in² (10.3 kPa) with the above-mentioned considerations is (24 oz/in²) (2.5)(0.7) = 85.7 oz/in² (36.9 kPa). The self-weight and the hydrostatic pressure in the analysis were kept constant while the internal pressure was increased from zero to the level of interest of, in this case, \(\Phi P = 85.7\) oz/in² (36.9 kPa).

The maximum plastic strain was recorded at the same point as the minimum triaxial strain was found. Both top and bottom faces of the shell finite element under consideration were assessed separately, and the minimum limiting triaxial strain was compared to the maximum plastic strain at the integration point of the elements. The limiting triaxial strain can be calculated using the following equation, while the plastic strain is reported in the finite element analysis results:

\[
\varepsilon_{plast} = \frac{\sqrt{2}}{2} \sqrt{\frac{3}{2}} \left[ \frac{\sigma_1 + \sigma_2 + \sigma_3}{3\sigma_v} - \frac{1}{3} \right] (4)
\]

where \(\varepsilon_{plast} = m_2\), \(\sigma_{plast} = 2.2\), and \(m_2 = 0.6(1 - F_s/F_m) = 0.2276\) as defined in ASME VIII-2 Table 5.7; \(\sigma_1, \sigma_2,\) and \(\sigma_3\) are the principle stresses, and \(\sigma_v\) is the von Mises stress.

For the design to be acceptable and meet the local failure criteria, the plastic strain has to be less than or equal to the limiting triaxial strain: \(\varepsilon_{plast} \leq \varepsilon_{plast}\).

The global failure behavior of tanks occurs when the tank can no longer take additional internal pressure. This is observed when the internal pressure has to decrease to maintain the stability of the structure. In the Riks’ analysis approach, the decrease in the applied internal pressure is captured and the pressure value is then divided by the safety factor of 2.5 and multiplied by the joint efficiency of 0.7.

2.3 Elastic Buckling Analysis. Buckling is investigated in this study with a perfectly elastic ASTM A36 steel. The buckling eigenvalue investigation was carried out by applying internal pressure, and included both positive and negative internal pressure scenarios. Negative internal pressure indicates vacuum in the tank. The Lanczos method in ABAQUS [10] was used to extract the buckling values of the tanks. This investigation included the self-weight of the tank and neglected the hydrostatic pressure to produce more conservative results.

As required by ASME VIII-2 for bifurcation buckling analysis using an elastic stress analysis without geometric imperfections.
and nonlinearities, a design factor of $\Phi = 2/\beta_c$, was applied to the internal pressure and vacuum pressure magnitudes to protect components with a compressive stress field from collapsing from buckling under the applied design loads. The value of $\beta_c = 0.80$ is provided by ASME VIII-2 for unstiffened and ring stiffened cylindrical cones under external pressure. With the value of $\beta_c$ known, $\Phi = 2/\beta_c = 2/0.8 = 2.5$; where 2 is the safety factor against buckling for a perfect geometry, while $\beta_c$ is a knockdown factor to account for geometric imperfections.

3 Results and Discussion

3.1 Elastic Analysis. The elastic analysis was performed to investigate the behavior of the tanks with linear elastic material properties and nonlinear large deformation of the geometry and compression-only linear springs representing the tank foundation. The stress level was evaluated at the midsurface and the top and bottom surfaces of each shell element.

The elastic analysis provides the simplest form of evaluation but limits the ability to judge the results to the material properties; the nonlinearity of the material and strain hardening are not considered. The geometry nonlinearity was considered in the analysis by using the Riks’ method, which propagates the deformation of the structure incrementally. This analysis was used to determine the relative ratio of the failure pressure of the bottom junction to the failure pressure of the top junction, the stress level in key tank components at design pressures, and the maximum uplift of the tank bottom at design pressures.

Two considerations have been investigated for the elastic failure of the tank, the first one is the failure of the tank due to internal pressure, and the second is the stress levels and deformations in the tank due to the design pressure of interest. The following subsections will show and discuss the results of both investigations. The legend convention used in the figures of the following sections is shown in Fig. 9.

3.1.1 Failure of the Tank Due to Yielding of the Top and Bottom Junctions. The yielding of the top or bottom junctions was considered to cause failure to the junctions. The internal pressure was increased until the membrane stress in the roof-to-shell and shell-to-bottom junctions reached the yielding stress of $F_y = 36,000$ psi (250 MPa). This maximum membrane stress was always found in the inside filet welds connecting roof to shell and shell to bottom at the internal pressure causing yield failure. Yielding was judged to occur when the von Mises stress reached $F_y = 36,000$ psi (250 MPa). A typical tank subject to an internal pressure causing the top junction to fail is shown in Fig. 10; the stresses near the cleanout junction are local discontinuity stresses, thus, ignored in this analysis but shall be closely investigated in the elastic–plastic analysis.

The internal pressure levels causing failure in the top and bottom junctions are shown Figs. 11 and 12, respectively. The horizontal dashed line in these figures is the proposed maximum pressure during emergency venting of 24 oz/in² (10.3 kPa), which is provided for reference. For both Figs. 11 and 12, the x-axis represents the tank case number, and the y-axis represents the internal pressure in oz/in² (multiply by conversion factor of 0.43 to obtain results in kPa). For Fig. 11, the results based on API 937 frangible roof compression area at the shell-to-roof junction denoted as API 937 in the figure legend as API 937 [4] and marked with an X are the calculated failure pressure for each tank.

The tanks can be split into three categories based on diameter as listed in Table 2: (1) tanks with a diameter less than 12 ft (3.7 m), (2) tanks with a diameter of 12 ft (3.7 m), and (3) tanks with a diameter larger than 12 ft (3.7 m). The first category of tanks exhibit a top junction failure at a pressure 3.8–6.8 times the emergency venting pressure of 24 oz/in² (10.3 kPa), while tanks in the second category exhibit a top junction failure at a pressure about 3.3 times the emergency venting pressure. The tanks in the third category exhibit a failure at a pressure about 3.1 times the emergency venting pressure. This shows that the failure of the top junction due to internal pressure is largely influenced by the diameter rather than the height of the tank or any other parameter.

The API 937 [4] results as calculated by Eq. (2) are more conservative than the FEA results obtained in this study, where the

<table>
<thead>
<tr>
<th>Bottom Thickness $b_t = 1/4$ in. (6.4 mm)</th>
<th>Bottom Thickness $b_t = 3/8$ in. (9.5 mm)</th>
</tr>
</thead>
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<tr>
<td>18 in. (0.46 m) Product Level</td>
<td>18 in. (0.46 m) Product Level</td>
</tr>
<tr>
<td>Half Full Product Level</td>
<td>Half Full Product Level</td>
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</table>

Fig. 10 Finite element analysis stress plot for a tank case 3 with shell and roof thickness of 3/16 in (4.8 mm). The legend shows units of psi. The bottom plate is 1/4 in (6.4 mm) thick with a product level of 18 in (0.46 m). The internal pressure is 114 oz/in² (49.1 kPa). The localized stresses near the cleanout are ignored (multiply by conversion factor of 6.89 to convert psi results to kPa).
expected failure pressure of the roof-to-shell junction using API 937 is almost 1/3 the expected failure pressure using FEA. The API 937 results shown in Fig. 11 range from 0.6 to 2.2 times the emergency venting pressure of 24 oz/in² (10.3 kPa). This is expected as the API 937 [4] estimation assumes the roof fails at the internal pressure causing the bottom plate to uplift, and established the strength of the roof based on the undeformed geometry. This study investigated the failure of the roof using the nonlinear geometry assumption while increasing the internal pressure in increments. This explains the discrepancy between the FEA results and the API 937 results, where for the category 3 tanks, the formulation expects the top junction to fail at an internal pressure lower than 24 oz/in² (10.3 kPa). However, the findings of the API 937 FEA investigation confirm that the results obtained by the formulation in Eq. (2) are overly conservative.

The diameter significantly influenced the failure pressure at the top junction (see Fig. 11) with all other factors like shell thickness, shell height, and liquid level having a minimal influence. To demonstrate the influence of the diameter on the yielding pressure of the roof-to-shell junction, Fig. 13 groups the tanks based on diameter and plots the internal pressure causing yielding in the top junction. Note that many tank cases have the same nominal diameter but different heights; the heights of the tanks are not shown in Fig. 13. The pressure causing yielding in the top junction is demonstrated to decrease as the nominal diameter of the tanks increase.

Figure 12 shows the internal pressure causing yielding failure in the bottom junction for all the tanks. The tanks in the first category show failure in the bottom plate at an internal pressure 4.1 to 12.5 times the emergency venting pressure of 24 oz/in² (10.3 kPa). The tanks in the second category show failure at a pressure range 4.9–8.1 times the emergency venting pressure of 24 oz/in² (10.3 kPa), while the tanks in the third category show failure at a pressure range 2.9–6.1 times the emergency venting pressure of 24 oz/in² (10.3 kPa). The bottom junction has a wider range of pressures causing failure as the bottom sell thickness and liquid level both have an effect on the failure pressure. The thicker the bottom plate is the higher the bottom junction failure pressure is, and the higher the stored product level is higher the failure pressure is.

It is observed that the tank height for a given diameter tank plays little role in the failure of the top junction, but the failure of the bottom junction happens at a higher internal pressure as the tank height increases. This is probably due to tanks with greater heights having larger weights counterbalancing the effects of internal pressure on the bottom junction. The more liquid there is, the more pressure is needed to cause the bottom weld to yield, which can be explained by the liquid hindering the ballooning effect due to internal pressure at the bottom plate, reducing the stresses at the bottom junction.

The ratio of the internal pressure causing failure to the shell-to-bottom junction relative to the internal pressure causing failure in the roof-to-shell junction is shown in Fig. 14. The bottom to top failure pressure ratio is preferred to be greater than unity to indicate that the roof junction fails before the bottom junction. This behavior is favorable in case of emergency excessive internal pressure above operating pressure, to have the roof act as an emergency venting mechanism. Out of 140 tanks tested, 6 tank cases had a relative failure ratio of less than 1. These tanks were category 3 tanks with a bottom plate thickness of 1/4 in (6.4 mm), and product level of 18 in (0.46 m), except for one case (new proposed case 14) in which the tank was half full as well.

Figure 14 has an x-axis representing the tank case number, and a y-axis representing the bottom failure pressure divided by the top failure pressure. Within each of the three diameter-based categories, the greater the height of the tank is, the larger the relative failure ratio is. While the desired bottom to top failure ratio is greater than unity, there is uncertainty in the material strength and workmanship that may not be captured by the FEA. However, considering the reasonably sufficient safety margin of roof-to-shell and bottom-to-shell yielding failures for the proposed design pressures, the slightly low relative strength ratio of bottom to top joints is not a great concern for these tanks comparing with API 650 tanks.

3.1.2 Tank Performance and Stress Levels Due to the Design Pressures. The stress levels were evaluated at two internal pressure levels: 16 oz/in² (6.9 kPa) and 24 oz/in² (10.3 kPa). The evaluation of the stress level was taken at the integration point holding the highest stress value in the junction of interest (i.e., top junction, bottom junction, or cleanout junction). The top and bottom junctions are highlighted in Fig. 15. These junctions are important because they experience the maximum stress in the tank due to the effect of the ballooning on shell deformation and plate
bending; the junctions provide movement restriction and geometry discontinuities that sustain higher stresses.

### 3.1.2.1 Top Junction

The top junction is comprised of the outer edges of the roof, the upper part of the shell, and the top fillet welds connecting the roof to the shell. The maximum membrane von Mises stress in this junction for all the tank cases modeled is shown in Figs. 16 and 17 for internal pressure of 16 oz/in² (6.9 kPa) and 24 oz/in² (10.3 kPa), respectively. Similarly, the membrane plus bending stress at the top junction for internal pressure of 16 oz/in² (6.9 kPa) and 24 oz/in² (10.3 kPa) is shown in Figs. 18 and 19, respectively.

The top junction is largely influenced by the effects of the internal pressure, where the stored liquid and shell weight are not directly acting on the junction and have minimal influence. The location of the maximum membrane stress was typically the weld, and is compared to the allowable membrane stress from API 650 with a joint efficiency of 1.0 as the joint experiences compressive circumferential stress, $S_{m} = 23,200$ psi (160 MPa); this value is indicated in Figs. 16 and 17 as a dashed line. If the membrane stress exceeds this $S_{m}$ in a general location around the circumference of the top junction, then the tank does not meet the design criteria of API 650 [3]. All the tested tanks meet the criteria with none of the membrane stresses at the top junction exceeding $S_{m}$; the stress levels in tanks of category 1 ranged from 0.19 to 0.43 times $S_{m}$, the stress levels for tank category 2 ranged from 0.34 to 0.62 times $S_{m}$, and the stress levels for tank category 3 ranged from 0.35 to 0.63 times $S_{m}$.

The maximum membrane plus bending stress at the top junction is also reported in Figs. 18 and 19 for the internal pressures of 16 oz/in² (6.9 kPa), 24 oz/in² (10.3 kPa), respectively. The horizontal dashed line is the limiting primary membrane plus bending stress $S_{PL}$ = larger of $1.5S_{m}$ or $F_{y}$. The membrane plus bending stress in the top junction for all the tanks does not exceed the yielding limit; the stress levels in tanks of category 1 ranged from 0.27 to 0.75 times $S_{PL}$, the stress levels for tank category 2 ranged from 0.52 to 0.91 times $S_{PL}$, and the stress levels for tank category 3 ranged from 0.34 to 0.93 times $S_{PL}$.

### 3.1.2.2 Bottom Junction

The bottom junction is made up of the outer edges of the bottom plate, the lower part of the shell, and the bottom welds connecting the bottom plate to the shell. The

<table>
<thead>
<tr>
<th>Table 2</th>
<th>Tank cases separated into groups based on diameter</th>
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<tbody>
<tr>
<td>Category number</td>
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![Fig. 13](https://pressurevesseltech.asmedigitalcollection.asme.org), ![Fig. 14](https://pressurevesseltech.asmedigitalcollection.asme.org), ![Fig. 15](https://pressurevesseltech.asmedigitalcollection.asme.org)
maximum membrane von Mises stress in the bottom junction for all the tank cases modeled is shown in Figs. 20 and 21 for the internal pressure of 16 oz/in² (6.9 kPa) and 24 oz/in² (10.3 kPa), respectively. Similarly, the membrane plus bending stress at the top junction for internal pressure of 16 oz/in² (6.9 kPa) and 24 oz/in² (10.3 kPa) is shown in Figs. 22 and 23, respectively.

The bottom junction is affected by the hydrostatic pressure due to the stored product, the self-weight of the tank, and the internal pressure. Because of the hydrostatic pressure and self-weight of the tank, it is expected that the bottom junction will experience larger bending stress and a lower membrane stress than the top junction at the same internal pressure value.

The typical location of the maximum membrane stress in the bottom junction is the shell, and was never at the weld at the design internal pressure. Figure 24 shows the maximum membrane location at the shell for tank case 14. The obtained stresses are reported in Figs. 20 and 21 and are compared to the allowable membrane stress from API 650 with a joint efficiency of 1.0 as the joint experiences compressive circumferential stress, \( S_{wm} = 23,200 \) psi (160 MPa). All modeled tanks pass the API 650 criteria as none of the membrane stresses exceed 23,200 psi (160 MPa); the stress levels in tanks of category 1 ranged from 0.03 to 0.31 times \( S_{wm} \), the stress levels for tank category 2 ranged from 0.03 to 0.25 times \( S_{wm} \), and the stress levels for tank category 3 ranged from 0.04 to 0.47 times \( S_{wm} \).
The maximum membrane plus bending von Mises stress at the bottom junction is also reported for all tank cases in Figs. 22 and 23 for the internal pressures of 16 oz/in² (6.9 kPa), 24 oz/in² (10.3 kPa), respectively. The horizontal dashed line represents the limiting criteria for ASME VIII-2 and is $S_{PL} = \text{larger of } 1.5S_m \text{ or } F_y$, $S_{PL} = F_y = 36,000 \text{ psi (250 MPa)}$, and only tank case 14 shows yielding at the bottom junction for an internal load level of 24 oz/in² (10.3 kPa). A typical location of the maximum membrane plus bending stress is either the shell or bottom plate. All the tanks except case 14 clear the criteria, where the membrane and bending stress does not exceed the yielding stress; the stress levels in tanks of category 1 ranged from 0.11 to 0.76 times $S_{PL}$.

The stress levels for tank category 2 ranged from 0.10 to 0.72 times $S_{PL}$, and the stress levels for tank category 3, except tank 14, ranged from 0.07 to 0.94 times $S_{PL}$. Tank case 14 stress levels ranged from 0.21 to 1.10 times $S_{PL}$. This does not necessarily mean that tank case 14 will fail at the design pressure, but it means that the outer surfaces of the shell will experience yielding, which can be investigated more with an elastic–plastic analysis.

3.1.2.3 Cleanout Junction. The cleanout junction is made up of the cleanout, the outer edges of the bottom plate, the lower part of the shell near the cleanout, and the cleanout and the shell-to-bottom junction welds as shown in Fig. 25. The cleanout junction...
contains the geometric discontinuities that will create an over-stress in nearby components. The maximum stress in the cleanout junction occurred near the change in opening’s geometry, at the bottom plate near the point where the cleanout necks meets the shell and bottom plate. Figure 26 shows the typical location of the maximum membrane stress in the cleanout junction, and Fig. 27 shows the typical location of the maximum membrane plus bending stress.

The membrane stress near discontinuities, \(P_m\), is compared to \(S_{PL} = \text{larger of } 1.5S_m \) or \(F_y\). Because \(1.5S_m = 34,800\text{ psi} (240\text{ MPa})\) and \(F_y = 36,000\text{ psi} (250\text{ MPa})\), \(S_{PL}\) is equal to \(F_y = 36,000\text{ psi} (250\text{ MPa})\). Similarly, the primary and secondary membrane plus bending stress, \(F_y + F_P + Q\), is compared to \(S_{PS} = \text{larger of } 3S_m \) or \(2F_y\). Thus, 
\[ S_{PS} = 2F_y = 72,000\text{ psi} (500\text{ MPa}) \] in accordance with ASME VIII-2 paragraph 5.2.2.4 for load case combinations and load factors for elastic stress analysis evaluated near the discontinuity junction. The maximum membrane stress in the cleanout junction for all the tank cases modeled is shown in Fig. 28 for internal pressure of 16 oz/in\(^2\) (6.9 kPa) and in Fig. 29 for internal pressure of 24 oz/in\(^2\) (10.3 kPa). Similarly, the membrane plus bending stress at the cleanout junction for 16 oz/in\(^2\) (6.9 kPa) of internal pressure is shown in Fig. 30, and for 24 oz/in\(^2\) (10.3 kPa) in Fig. 31.

Figures 28 and 29 show the maximum membrane stress at the cleanout junction for all tank cases, and only tank case 14 experiences stresses larger than \(S_{ML}\) when the internal pressure is 24 oz/in\(^2\) (10.3 kPa). The limiting stress of \(S_{PL} = F_y\) is plotted in both figures as a horizontal dashed line. All modeled tanks pass the ASME VIII-2 criteria as none of the membrane stresses exceed \(36,000\text{ psi} (250\text{ MPa})\); the stress levels in tanks of category 1 ranged from 0.08 to 0.55 times \(S_{PL}\), the stress levels for tank category 2 ranged from 0.09 to 0.60 times \(S_{PL}\), and the stress levels for tank category 3 ranged from 0.10 to 0.81 times \(S_{PL}\), with the exception of tank case 14 where some of the configurations reached 1.04 times \(S_{PL}\). Figures 30 and 31 show the maximum membrane plus bending stress for all tank cases, with no stress level in any tank exceeding the \(S_{PS}\) limit. The limiting stress of \(S_{PS} = 2F_y\) is plotted in both figures as a horizontal dashed line. All modeled tanks pass the ASME VIII-2 criteria as none of the membrane stresses exceed \(72,000\text{ psi} (250\text{ MPa})\); the stress levels in tanks of category 1 ranged from 0.07 to 0.55 times \(S_{PS}\), the stress levels for tank category 2 ranged from 0.08 to 0.63 times \(S_{PS}\), and the stress levels for tank category 3 ranged from 0.05 to 1.00 times \(S_{PS}\).

Although the safety requirements of the ASME VIII-2 provisions are met in the elastic analysis for tank cases 1–12, protection against local failure was further investigated with an elastic–plastic analysis to ensure the operational safety of the tanks.

### 3.1.3 Tank Uplift

The maximum uplift of the tank bottom of each tank has been evaluated to take into consideration the deformation tolerance for the attached piping to the tank shell. The ballooning effect resulting from the increased internal pressure causes the edges of the tank bottom to rise. The maximum uplift for the tank bottom at design pressure of 16 oz/in\(^2\) (6.9 kPa) is shown in Fig. 32 for all the tank models, and at the design pressure of 24 oz/in\(^2\) (10.3 kPa) in Fig. 33.

The uplift is reduced with the increase in stored product level as can be observed by the difference in uplift magnitude between tanks with 18 in (0.46 m) of product and the tanks half full. With the exception of tank case 14, the uplift of the half full tanks under an internal pressure of 16 oz/in\(^2\) (6.9 kPa) did not exceed 0.1 in (2.54 mm), and under an internal pressure of 24 oz/in\(^2\) (10.3 kPa) did not exceed 0.25 in (6.35 mm). For each tank case, the variability in the uplift magnitude in the half full cases due to differences in bottom plate, shell, and roof plate thicknesses is not significant. This indicates that the stored product level is a detrimental factor with more influence on the uplift than any other factor for a specific diameter. For tanks with 18 in (0.46 m) of stored product, the uplift is influenced by the shell thickness and height and the bottom plate thickness. The larger the thickness of each component is and the taller the tank is, the heavier the tank is; this results in less uplift for taller tanks with thicker components.

### 3.2 Elastic–Plastic Analysis

The elastic–plastic analysis was performed to allow for taking advantage of the strain...
hardening of the steel after yielding. The elastic analysis indicated that the steel near the cleanout junction is yielded at the higher design pressure of 24 oz/in² (10.3 kPa), which can be problematic if the plastic strain is high. To allow for protection against local failure near the cleanout, the results of the elastic–plastic analysis were evaluated in accordance with the provisions of ASME VIII-2.

The elastic–plastic analysis was performed to ensure the local behavior of the tank structure near the cleanout is acceptable under operating pressures. The localities near the cleanout experience higher stresses than the rest of the tank due to the geometric discontinuities that lead to stress concentrations. The use of the elastic analysis showed that all tanks are performing relatively well in general except for tank case 14. The elastic–plastic analysis should confirm the findings of the elastic stress analysis, and either confirm the inadequacy of the tank case 14 or show that it is acceptable.

The plastic strain in the localities near the cleanout is assessed in accordance with ASME VIII-2 and is compared to the limiting triaxial strain, \( \varepsilon_L \), as calculated using Eq. (4). The plastic strain, \( \varepsilon_{peq} \), and the limiting triaxial strain, \( \varepsilon_L \), are compared in Fig. 34. The limiting triaxial stress seems to have an almost constant profile ranging between 24% and 30% throughout all the tank cases, which indicates that the ratio of the average of principle stresses to the von Mises stress is almost constant; thus, all the limiting triaxial strain values were represented with a horizontal dashed line. The global minimum limiting triaxial strain is shown in Fig. 34 as a dashed line that corresponds to 24% strain. This minimum limiting triaxial strain occurred at tank case 14 with all the cases containing a bottom plate thickness of 3/8 in (9.5 mm) regardless of other parameters in the shell thickness or product level. All the tanks with a bottom thickness of 3/8 in (9.5 mm) had a limiting triaxial strain range falling within 24–26%. The rest of the tanks

Fig. 26 Typical location of maximum membrane von Mises stress near the meeting point of bottom plate and cleanout weld. The legend shows the stress magnitude in psi (multiply by conversion factor of 6.89 to convert psi results to kPa).

Fig. 27 Typical location of maximum membrane plus bending von Mises stress near the meeting point of bottom plate and cleanout weld. The legend shows the stress magnitude in psi (multiply by conversion factor of 6.89 to convert psi results to kPa).
with a smaller bottom plate thickness had a higher limiting triaxial strain.

The plastic strain, \( \varepsilon_{\text{peq}} \), of all the tanks with a diameter smaller than 15.5 ft (4.7 m) has a plastic strain not significantly affected by the stored product level or shell and bottom plate thicknesses within the studied combinations. The maximum plastic strains, \( \varepsilon_{\text{peq}} \), for these tanks ranged from 0.4% to 2.2%. For tanks with a diameter of 15.5 ft (4.7 m) or larger, the cases with the smaller thickness of the bottom plate of 1/4 in (6.4 mm) experienced higher plastic strains. Also, the heavier the tank due to its shell thicknesses and height, the less plastic strain it experienced near the cleanout junction. The results show that the plastic strain in all the tanks is within the minimum allowable triaxial strain limits.

The global failure due to rupture was investigated, and was judged to occur when the tank was not able to sustain any increased internal pressure in the Riks' analysis step. The global rupture failure pressure magnitude divided by the design factor for each tank case is shown in Fig. 35. The design factor of \( \Phi \) is the safety factor of 2.5 divided by the joint efficiency of 0.7: \( \Phi = 2.5 / 0.7 = 3.57 \). Some modeled tanks did not show the decrease in the internal pressure expected for rupture failure, but these tanks are the ones with thicker components, and are expected to perform better than the tanks in which global rupture failure was captured.

The global rupture failure behavior depended largely on the diameter of the tank, and increased slightly with the increase of the height for tanks with the same diameter. This is demonstrated in tanks with the same diameter like tanks in category 2 (i.e., tank...
cases 4 and 7–9). Tanks with the same height experience a decrease in the global rupture failure pressure as the diameter increase, which can be seen in tank cases 5–7. The global rupture failure pressure increased as the stored product level increased, which suggests that the bottom segment of the tank is responsible for the global rupture failure.

The horizontal dashed line in Fig. 35 represents an internal pressure of 24 oz/in² (10.3 kPa), which is the proposed maximum pressure during emergency venting. This is provided as a reference to show that the tanks do not fail due to rupture before this desired pressure.

Figure 36 shows tank case 3 at an internal pressure of 112 oz/in² (48.3 kPa) as modeled with FEA compared to an actual tank of most probably same size that over-pressurized [37]. The tank case 3 selected for the comparison has diameter of 9.5 ft (2.9 m), shell height of 12 ft (3.7 m), bottom plate thickness of 1/4 in (6.4 mm), shell and roof plate thicknesses of 3/16 in (4.8 mm), and a product level of 18 in (0.46 m). The comparison of FEA simulation and actual tank that experienced over-pressurization shows remarkably similar bulging behavior of the bottom plate near the cleanout, and shows the roof plate to experience wave-like deformations. Unfortunately, pressure level in the over-pressurized tank is not available. It appears from the photos that the roof-to-shell and shell-to-bottom joints did not ruptured. Only, the tank experienced large ballooning deformations at the roof and bottom

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**Fig. 32** Maximum uplift at tank bottom reported for internal pressure level of 16 oz/in² (6.9 kPa) (to convert in to mm multiply values by 25.4)

**Fig. 33** Maximum uplift at tank bottom reported for internal pressure level of 24 oz/in² (10.3 kPa) (to convert in to mm multiply values by 25.4)

**Fig. 34** Plastic strain, $\varepsilon_{PEQ}$, as compared to the limiting triaxial strain, $\varepsilon_L$, measured near the cleanout junction, the shown dashed line represents the global minimum triaxial strain from all the tank cases

**Fig. 35** Internal pressure divided by design factor that causes global rupture failure in each tank (multiply by conversion factor of 0.43 to convert oz/in² results to kPa)

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and corresponding buckling at the roof-to-shell joint. Tank clean-
out junction also performed relatively well with no apparent rup-
tures for this tank size.

3.3 Elastic Buckling Analysis. The buckling analysis was
performed to ensure that tanks do not fail due to buckling at a
pressure lower than the design pressures. The linear buckling
analysis without imperfections was performed for both internal
pressure and vacuum pressure. The factored internal buckling
pressure is shown in Fig. 37, where the horizontal dashed line is
the design pressure of 24 oz/in² (10.3 kPa). The factored internal
vacuum pressure causing buckling is shown in Fig. 38, where the
horizontal dashed line represents the design vacuum pressure of
1.5 oz/in² (0.43 kPa). In both figures, the y-axis represents the
buckling load divided by the design factor $U = 2.5$ to allow for
uncertainties in the material, construction, and geometric imper-
fections. The buckling only occurs in the roof plate for both inter-
nal pressure and vacuum pressure; thus, only the thickness of the
roof plate and the thickness of the top shell influence the buckling
behavior in the first mode.

Figures 37 and 38 show that tanks with the same diameter and
roof thickness buckle at approximately the same internal pressure
and vacuum pressure. This is demonstrated with tank cases 4 and
7–9 having the same diameter and same buckling load; similarly
are tank cases 10–12 buckling at the same internal pressure and
vacuum pressure.

Fig. 36 Tank with diameter of 9.5 ft (2.9 m) and height of 12 ft (3.7 m), case 3, behavior with
nonlinear material at internal pressure of 112 oz/in² (48.3 kPa) compared to an over pressur-
ized tank in Wyoming [37] of most probably the same diameter and height. The von Mises
stress magnitude in psi is used to contour the deformed shape of the tank. The displacement
magnification factor for the deformed shape is 1.0. The bottom plate thickness of the modeled
tank was 1/4 in (6.4 mm), the shell and roof plate thicknesses were 3/16 in (4.8 mm), and the
product level was 18 in (0.46 m) (multiply by conversion factor of 6.89 to convert psi results to
kPa).

Fig. 37 Buckling of the tanks due to internal pressure, $P$ (mul-
tiply by conversion factor of 0.43 to convert oz/in² results to
kPa).
The first buckling mode occurs in the roof, and a typical first buckling shape for positive internal pressure is shown in Fig. 39. The buckling waves occur near the roof-to-shell junction away from the center of the roof. Similarly, the first buckling mode shape for vacuum pressure is shown in Fig. 40. It is observed that the vacuum pressure buckling mode affects a larger area of the roof, and has fewer waves than the internal pressure buckling mode. The vacuum pressure causes the cone-shaped roof to deform into an unstable shape with compression acting in the meridional direction, which causes it to buckle at a lower pressure magnitude than the buckling mode induced by internal pressure. The instability of the structure of the tank under increased internal pressure can lead to a sudden failure with rapid progression, unlike the yielding behavior where the tank can still take more load before rupture occurs. This study investigated the buckling behavior to ensure the tanks do not experience instability before or at the design pressure of 24 oz/in\(^2\) (10.3 kPa). Figures 37 and 38 show the results of the buckling analysis and compare them to the design pressures of interest. Both buckling due to positive internal pressure and vacuum pressure affected the roof part without disturbing the shell too much. The bottom plate is where the tank meets the foundation, and buckling will not occur there. The shell is cylindrical and will not buckle under internal pressure. Thus, because the roof is almost flat, wide, and thin, it is the most vulnerable to show the buckling behavior. The results of the buckling analysis indicate that the tanks maintain stability within the design pressure, and a preferential buckling failure in the roof occurs before buckling in the bottom plate or shell. This illustrates that in case of over-pressurization of the tank, the buckling mode failure will occur at the roof, while maintaining the rest of the tank elements performing as described in the elastic–plastic analysis.

4 Conclusion

This study investigated many design parameters to assert the validity of these desired tank designs with a total of 140 models tested for each type of analysis. Three types of analyses have been performed: (1) elastic stress analysis, (2) elastic–plastic stress analysis, and (3) elastic buckling analysis.

The elastic stress analysis showed that the top junction will yield before the bottom junction for most tank models. This result leads to the conclusion that the tanks do not have a preferential failure at the bottom junction before failure at the top, which is beneficial if the roof was needed to act as a frangible roof for emergency venting for internal pressure larger than the emergency design pressure. However, by allowing for the variability and uncertainty in the material strength and workmanship, results denoting a bottom to top failure ratio close to 1.0 cannot be conclusive.
The elastic–plastic analysis showed that the plastic strain in the cleanout junction is within the allowed limits of triaxial strain required by ASME VIII-2. The results of the elastic analysis and the elastic–plastic analysis show that the tanks can withstand the design pressure and the emergency design pressure within acceptable stresses and strains.

The elastic analysis was also used to investigate the uplift experienced by the bottom plate of the tank. As the internal pressure increases, the tanks experience ballooning and inflation leading to uplift in the bottom plate and shell. This uplift behavior should be considered for the types of piping connections to be used with the tank and their movement tolerance. The results showed that uplift is not a concern when the tanks contain liquid at more than half the capacity, and when there is 18 in (0.46 m) of product, the maximum uplift is less than 1.6 in (40.6 mm).

The elastic buckling analysis shows that the tanks do not buckle within the desired design pressure loads and vacuum pressure. This allows for normal operation in the tanks without the sudden failure of the tank structure due to instability leading to buckling. Although extensive FEA simulations confirm the proposed changes, physical testing should be performed to validate FEA results. In addition, this current phase 3 work did not consider fatigue, brittle fracture, and seismic actions.

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References


