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Prepared for the U.S. Department of Energy
Office of Environmental Restoration and
Waste Management

Westinghouse
Hanford Company Richland, Washington

Hanford Operations and Engineering Contractor for the
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ABSTRACT

This study addresses the structural analysis and evaluation of an abnormal rectangular pressure vessel. The vessel was to house equipment for drilling and collecting samples from radioactive waste stored in high-capacity waste storage tanks on the Hanford Site. The vessel had to be fabricated with a removable cover plate.

The dimensions and working pressure of the vessel classified it as a pressure vessel; therefore it had to be qualified according to the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section VIII, referred to hereafter as the Code (ASME 1989).

Section VIII of the Code provides guidelines, rules, and examples for analyzing and evaluating rectangular pressure vessels. However, the subject vessel had the cover plate bolted to the vessel along the long face of the vessel, a configuration not addressed by the Code. Applying the Code formulas for analysis would lead to very conservative stress results and excessive fabrication costs.

Finite-element methodology was used to analyze and calculate the stresses resulting from the internal pressure. The stresses were then used to evaluate and qualify the vessel. Because fatigue was shown not to be a concern, the subject vessel can be exempted from fatigue analysis. Thus, it can be built according to the ASME Code, Section VIII, Division I (ASME 1989a) instead of Section VIII, Division 2 (ASME 1989b). The result was a considerable cost savings.

Results of the stress analysis were checked against the Code. To satisfy Code requirements, the design was modified by adding a stayed plate to stiffen the long side of the vessel. The formulas of the Code for rectangular vessels, although not directly applicable, were also considered and are addressed in this study.

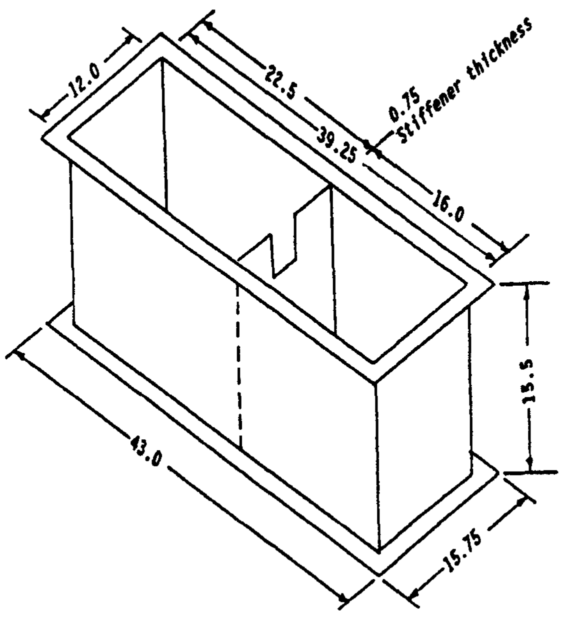
INTRODUCTION

A rectangular-shaped pressure vessel was analyzed and evaluated to the ASME Code requirements. This paper summarizes the approach used in the analysis, the example guidelines in the Code, the evaluation criteria in the Code, and the difference between the general shapes of the rectangular vessels assumed in the Code and the shape of the subject vessel.

The vessel has inside dimensions of 39.25 by 12.0 in. (99.7 by 30.5 cm) and is 15.5 in. (39.4 cm) deep. The vessel is not an integral unit. To allow maintenance and service for the equipment housed inside the vessel, the cover plate (43.0 by 15.75 in. (109.2 by 40.0 cm)) has to be removable. A stay-plate stiffener 12.0 by 15.0 in. (30.5 by 38.1 cm) was welded to the inside of the vessel at a distance of 16.0 in. (40.6 cm) from the inside edge of the vessel (22.5 in. (57.2 cm) from the other edge). Figure 1 is a sketch of the vessel with the cover plate removed.

The subject vessel will operate at 37.0 lbf/in^2 (255 KPa) internal pressure. The design pressure, as outlined by the Code, is 1.5 times the operating pressure. Hence, for the analysis, an internal pressure of 55.0 lbf/in^2 (379.2 KPa) was applied to the vessel to calculate the displacements, reactions, and stresses.

The vessel will be used as part of the machinery to collect samples from radioactive waste stored in the waste storage tanks. The vessel is expected to be pressurized 500 times per year. The life expectancy of the vessel is 10 years. Therefore, it had to be considered for a total of 5,000 pressurized cycles.



Working pressure = 37.0 lbf/in^2 = 255 KPa
Design pressure = 55.0 lbf/in^2 = 379 KPa
Wall thickness = 0.75 in. = 1.9 cm
All dimensions in inches (1 in. = 2.54 cm)

FIGURE 1. SCHEMATIC CONFIGURATION OF THE RECTANGULAR PRESSURE VESSEL.

GEOMETRY AND MATERIAL

As mentioned in the Introduction, the inside dimensions of the vessel are 39.25 by 12.0 in. (99.7 by 30.5 cm), and it is 15.5 in. (39.4 cm) deep. The side walls and the stay plate are 0.75-in. (1.9 cm) thick. The stay plate is added as a stiffener to reduce the stresses at the long side of the vessel wall. The cover plate is 1.0 in. (2.54 cm) and has outside dimensions of 43.0 by 15.75 in. (109.2 by 40.0 cm). Twenty-eight bolts secure it to the flange of the vessel at the outside edges. The bolts are 7/16-14UNC-2A and are made of ASTM A193, Grade B8 steel. A 1/16-in. (0.16 cm) neoprene gasket was used to prevent the vessel from leaking. The operating temperature of the vessel is below 200 °F (93.3 °C). The material of the side walls, the stay plate, and the cover plate are ASTM-A240 316 stainless steel.

The material properties of the materials used are listed below.

<u>ASTM-A240, 316 Stainless Steel</u>

Fy = yield strength = 30,000 lbf/in^2 (206,850 KPa)
Fu = ultimate tensile strength
= 75,000 lbf/in^2 (517, 125 KPa)
S = maximum allowable stress (ASME 1989a)
= 18,800 lbf/in^2 (129,626 KPa)
S_m = allowable stress intensity (ASME 1989b)
= 20,000 lbf/in^2 (137,900 KPa)
S_a = alternating stress (used for fatigue evaluation)

<u>ASTM - A193, Grade B8</u>

Fy = 100,000 lbf/in^2 (689,500 KPa)
Fu = 125,000 lbf/in^2 (861,875 KPa)
S = 25,000 lbf/in^2 (172,375 KPa)

APPROACH

The analysis used two different approaches. First, finite-element methodology was used to calculated the displacements, reactions, and the stresses. The stresses were evaluated by comparing them to the allowables of the Code with consideration given to the reduction factors for the welds. Using the guidelines in Appendix 13 of the ASME Code, Section VIII, Division 1 (ASME 1989a) that address vessels with noncircular cross-sections showed that the Code approach is very conservative.

Because the vessel will undergo 5,000 operational cycles in its 10-year lifetime, fatigue considerations also need to be investigated. Cyclic fatigue requires compliance with the ASME Code Section VIII, Division 2 (ASME 1989b). However, in this case, the two conditions (A and B, below) were statisfied and showed that the vessel could be exempted from fatigue analysis. Therefore, the subject vessel can be analyzed and constructed according to Section VIII, Division 1 rules (ASME 1989a).

<u>Fatigue Analysis</u>

The ASME Code Section VIII, Division 2, provides rules to determine whether fatigue evaluation is required or not. These rules include two conditions, of which one must be satisfied to exempt the structure from fatigue analysis. These two conditions were checked.

<u>Condition A</u>. Fatigue analysis is not mandatory for materials having a specified minimum tensile strength not exceeding 80,000 lbf/in^2 (551,600 KPa) when the total expected number of pressure cycles does not exceed 1,000 cycles.

The 316 SS material has a minimum tensile strength less than 80,000 lbf/in^2 (551,600 KPa), but the number of pressure cycles (5,000 cycles) exceeds the specified 1,000 cycles. Therefore, Condition A is not satisfied.

<u>Condition B</u>. When the following conditions are met, fatigue analysis is not mandatory.

(a) The expected design number of full-range pressure cycles, including startup and shutdown, does not exceed the number of cycles in the applicable fatigue curve of the Code corresponding to an S_a value of three times S_m value found in the tables of design stress intensity values.

$S_a = 3\ S_m = 60{,}000$ lbf/in^2 (413,700 KPa)

This stress value translates to a maximum of 13,000 allowable cycles (see Figure 2 reproduced from ASME 1989b), which is greater than the 5,000 cycles expected.

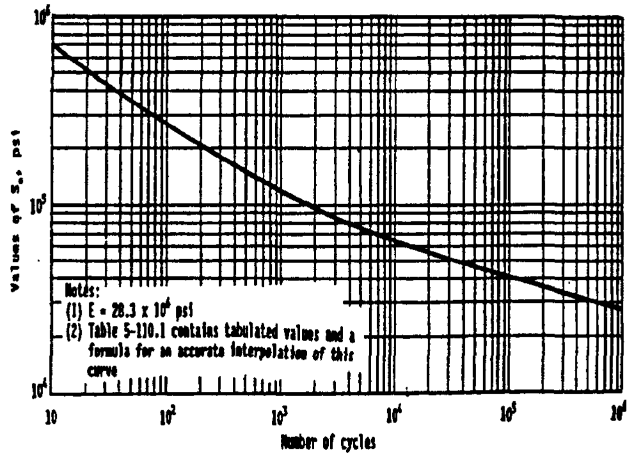


FIGURE 2. DESIGN FATIGUE CURVE FOR SERIES 3XX HIGH ALLOY STEEL.

(b) Additionally, the maximum allowable pressure calculated by the formula below should envelope the 55.0 lbf/in^2 (379.2 KPa) internal design pressure, P_d.

$$P_{max} = \frac{1}{3}\ P_d \left(\frac{Sa}{Sm}\right)$$

where

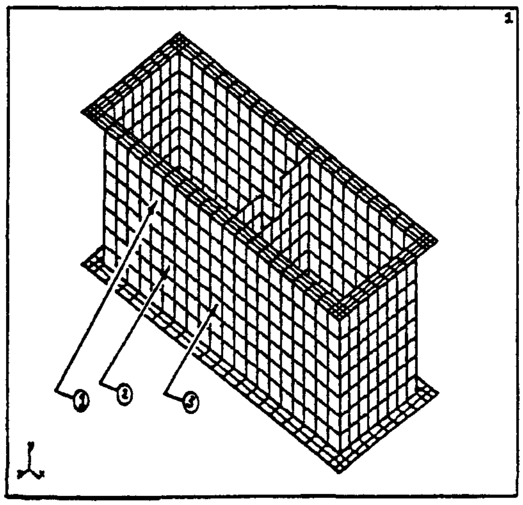
Sa = alternate stress for 5,000 cycles, lbf/in^2
= 75,000 (517,125 KPa)
Sm = maximum allowable design stress, lbf/in^2
= 20,000 (137,900 KPa)
P_{max} = 68.75 lbf/in^2 (474.0 KPa)

Therefore, the vessel could be exempted from fatigue analysis.

Finite-Element Analysis

As an alternate approach to the Code design formulas, finite-element methodology (Cook 1981) was used to calculate the stresses in the vessel. The COSMOS/M[1] software package was used to model the body of the assembly with four-noded quadrilateral shell elements having membrane and bending capabilities. The locations of the bolts on the bottom plate flange, representing the bolts mounting the vessel to the foundation, were characterized as nodal points and fixed against movement in the vertical direction. One additional node, a corner node on the bottom, was constrained against translation in both planes of the bottom plate and in the vertical direction.

The bolt locations on the top flange, where it mounts to the cover plate, were modeled with spring elements using the stiffness of the bolts. The junction where the outer boundary of the top flange meets the assembly cover plate was characterized by gap elements with 0.005-in. (0.0127-cm) openings. The gap elements were introduced to account for the prying action of the flange. Figure 3 depicts the finite-element model with locations of interest.



1— Maximum general primary membrane stress
2— Maximum membrane plus bending stress
3— Maximum local primary membrane plus primary bending stress

FIGURE 3. THE PRESSURE VESSEL FINITE-ELEMENT MODEL WITH LOCATIONS OF INTEREST.

The stress results indicate a maximum general primary membrane stress of 1,245.0 lbf/in^2 (8,584 KPa) in the middle of the long side of the vessel. The maximum membrane-plus-bending stress, a value of 7,929 lbf/in^2 (54,670 KPa), occurs in the long side of the vessel close to the top flange. The maximum local primary membrane stress plus the primary bending stress, a value of 9,048 lbf/in^2 (62,386 KPa), occurs at the location of the stay plate. The cover plate, 1-in. (2.54 cm) thick, has a maximum membrane stress of 14,700 lbf/in^2 (101,357 KPa). All these stresses are within the allowables set by the ASME Code, Section VIII, Division 1 (ASME 1989a).

ASME CODE ANALYSIS

The minimum requirements for design and analysis of rectangular cross-section vessels are covered in Appendix 13 of the ASME Code (ASME 1989a). In particular, Figure 13-2(a), Sketch 7 (reproduced as Figure 4 in this paper), addresses the rectangular vessel with a stay plate. However, the stay plate is shown at the middle of the vessel; in the subject vessel, the distance from the inside edge of the side of the vessel to the stay plate is 22.5 in. (57.2 cm), with a total overall length of 39.25 in. (99.7 cm). For a conservative analysis the length is taken as twice the distance from the edge to the stay plate, i.e., h = 22.5 in. (57.2 cm), where h is the inside length of the long side of the rectangular vessel. The length of the vessel, L_v – in this case the depth of the vessel – is equal to 15.5 in. (39.4 cm). The Code equations are developed for vessels with an aspect ratio, L_v/H or L_v/h, greater than 4.0. Thus, the equations in the Code (ASME 1989a) are not directly applicable to the subject pressure vessel, and some dimensional changes have to be assumed to perform the analysis. In this case the aspect ratio is less than 4.0 and leads to very conservative stress results for the combined stresses. However, for the purpose of comparison to the finite-element analysis results, the calculations are carried through. The equations used here are from the ASME Code Appendix 13, Subsection 13-9.

$$S_1 = S_m = \frac{Ph}{4t_1}\left(4 - \left[\frac{2 + K(5 - \alpha^2)}{1 + 2K}\right]\right) \qquad (1)$$

$$S_2 = S_m + \frac{Ph^2c}{12I_2}\left[\frac{1 + K(3 - \alpha^2)}{1 + 2K}\right] \qquad (2)$$

$$S_3 = S_m + \frac{Ph^2c}{12I_2}\left(\frac{1 + 2\alpha^2 K}{1 + 2K}\right) \qquad (3)$$

The above notations are defined below.

S_1 = general primary membrane stress, lbf/in^2 (KPa)
S_2 = general membrane plus bending stress lbf/in^2 (KPa)
S_3 = local primary membrane plus the primary bending stress, lbf/in^2 (KPa)
S_m = membrane stress, lbf/in^2 (KPa)
P = internal pressure, lbf/in^2 (KPa)
h = inside length of long side of rectangular vessel, in. (cm)
t_1 = thickness of short-side plates of vessel, in. (cm)
K = vessel parameter $(I_2/I_1)\alpha$
C = distance from neutral axis of cross section to extreme fiber, in. (cm)
I_1 = moment of inertia of strip of thickness, t_1, in^4 (cm^4)
I_2 = moment of inertia of strip of thickness, t_2, in^4 (cm^4)
α = rectangular vessel parameter, H/h
H = inside length of short side of rectangular vessel, in, cm
h = inside length of long side of rectangular vessel, in., cm.

The resulting stresses for equations (1) through (3) are listed below:

S_1 = 750 lbf/in^2 (5,171 KPa)

S_2 = 29,760 lbf/in^2 (205,195 KPa)

S_3 = 16,050 lbf/in^2 (110,665 KPa).

Except for the S_1 value, the other stresses are conservative when compared to the finite-element stress results.

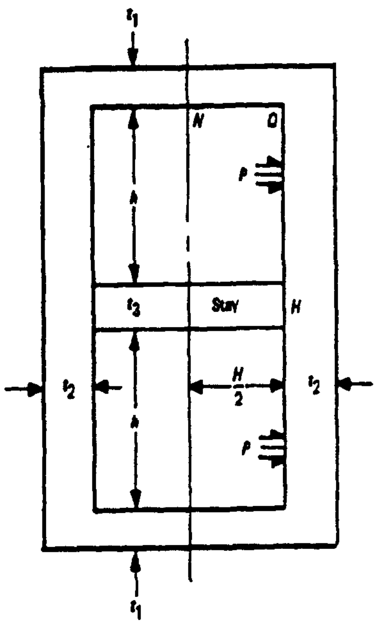


FIGURE 4. RECTANGULAR VESSEL WITH A STAY PLATE.

CONCLUSIONS

Two approaches were taken to analyze a rectangular pressure vessel with a design internal pressure of 55.0 lbf/in^2 (379 KPa). First the vessel was modeled and analyzed with finite-element methodology to establish a benchmark for calculating the minimum wall thickness required by the ASME Code. Next, the ASME Code-recommended algorithm was applied to calculate the stresses. The equations of the Code were not directly applicable. The design dimensions were modified, and then the vessel was analyzed. A comparison of the results show that, for low aspect ratios, the stresses calculated by the Code are very conservative.

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