Elephant’s Foot Buckling of Cylindrical Steel Storage Tanks Subjected to Earthquake Excitation

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Abstract

Thin metal cylindrical shell structures such as silos and tanks are susceptible to an elastic-plastic instability failure at the base boundary known as elephant’s foot buckling, due to its characteristic deformed shape. This form of buckling occurs under high internal pressure accompanied by axial compression in the shell structure. This work concerns with Theoretical studies on elephant's foot buckle failure of ground-supported, cylindrical liquid storage tanks under horizontal excitation. The buckling loads are obtained from finite elements models and codes and are compared. Theoretical nonlinear seismic analyses are carried out using ANSYS package. Studies are conducted on 13 models of cone roof tanks with height to diameter ratios (H/D) between 1 and 2, and a liquid level of 85% of the height of the cylinder with and without axial constraint at the point diametrically opposite the loading. The results are compared to which of the codes API 650, NZSEE guidelines and Eurocode 8. The comparisons of analytical buckling loads and those obtained by the codes reveal the following. Tanks designed by the codes API650 and Eurocode 8 tend to be unsafe due to elastic-plastic buckling occurrence of the shell. However, NZSEE guidelines have a near coherence to the analysis results. It is also obtained from the results that constraints at the base of the tank reduces sloshing height While cause the buckling capacity to rise up.

Keywords: Storage Tanks, Elephant's Foot Buckling, Seismic Analysis, Finite Element

1. INTRODUCTION

Cylindrical metal tanks are thin shell structures subject to internal pressure from stored liquid together with axial compression that can arise from roof loads, horizontal loads such as earthquake and the frictional drag of stored materials on the walls. Under earthquake loading, overturning is resisted by axial compressive stresses in the wall. The governing failure mode is usually buckling under axial compression. The internal pressure exerted by hydrostatic and hydrodynamic pressures can significantly enhance the buckling strength, but high internal pressures lead to severe local bending near the base. Local yielding then precipitates an early elastic-plastic buckling failure (Fig. 1). This failure mode is commonly known as “elephant’s foot buckling” and governs the design of many practical silo and tank structures [1].

Figure 1. Elephant’s foot buckle at the base of a storage tank.
The axial compressive stress developed at the base of tanks under seismic excitations must be less than an allowable buckling stress to preclude the occurrence of shell buckling. The allowable stress in current codes and standards is basically specified for a uni-directional stress state whereas the actual stress state at the shell bottom is bi-axial [2].

The allowable buckling stress in API (American Petroleum Institution) standards is based on the classical value of buckling stress under axial load, significantly reduced by a large knock down factor due to shell imperfections and also increased to account for the effects of internal liquid pressure [3]. Eurocode8 considers the elastic behavior of the flexible tank. The allowable stress is limited by the elastic buckling of the shell as an important factor in design of storage tanks [4,5]. Eurocode8 and NZSEE guidelines have close criteria in design, mainly based on Veletsos et.al.(1984) works. In the New Zealand guidelines, classical buckling stress in membrane compression is calculated and corrected according to imperfection amplitude and internal pressure. In addition, an elastic-plastic collapse stress is also calculated. The lower of the two stresses is used in the computation of the margin of safety [6].

Earlier studies shows that the tanks with height-to-diameter(H/D) ratios of 0.5 and below tend not to have elephant foot buckling. As this ratio increases, the propensity to elephant foot buckling increases [7]. There exists a question as to whether or not the current design criteria are conservative, reasonably accurate, or unsafe. This study was primarily intended to identify the effect of this kind of failure mechanism on design of tanks. A comparison is done between the ANSYS model results and an experimental work to verify the model behavior under horizontal excitation. Subsequently, 13 models with different H/D ratios are modeled in ANSYS 12 and analysed using nonlinear method. The results are collated with the design guidelines.

2. METHOD OF ANALYSIS

Buckling analysis, in general, may be divided into Eigen value and load-deflection analysis as shown in Figure 2. Eigen value buckling occurs when a maximum axial compressive stress becomes equal to the buckling stress. Since imperfections and nonlinearities prevent most real-world structures from achieving their theoretical elastic buckling strength, it is more accurate to use nonlinear load-deflection analysis. In general, a nonlinear buckling analysis is simply a nonlinear static analysis in which the load is increased until the solution fails to converge, indicating that the structure cannot support the applied load (or that numerical difficulties prevent solution). If the structure does not lose its ability to support additional load when it buckles, a nonlinear buckling analysis can also be used to track post-buckling behavior. We did not carry out post-buckling analysis here because of uncertainty in the validity of the results.

Herein, an appropriate numerical model including the element type and necessary number of elements in nonlinear analysis is presented. A comparison is done between the ANSYS results and an experimental study of elephant’s foot buckling by Jia and Ketter [8] to verify the numerical model. To this, both low frequency (5Hz) and high frequency (20Hz) horizontal acceleration waves with 0.2g magnitude used to shake the tank model in order to study the dynamic behavior of the model at different frequency ranges. The size of the cylinder chosen was 81cm in diameter, 106cm in height and 0.025cm in shell thickness. The design liquid
depth was 81cm (full level), which gave ratio of H/R and radius-to-thickness(R/t) equals 2.33 and 1800 respectively. The tank was modeled to be fixed at the bottom. Although most anchored tanks do not have such complete fixity, it was considered desirable for both the numerical and experimental to analyze this case.

Three-dimensional shell elements offered in ANSYS can be divided into four-node (Shell181) and eight-node (Shell182) shell elements in view of the number of nodes per element, and different degrees-of-freedom for such elements in view of the number of degrees of freedom per element. The 6-dof shell element has three displacement and three rotational components. The critical stresses obtained by numerical analyses using different element types were compared with the experimental stresses calculated [4]. When element type Shell182 was used, the error in the buckling stress was less than 5%. When elements Shell181 were used, the buckling stresses were overestimated by up to 25% even though a sufficient number of elements were used. Thus element type Shell182 for tank shell, and FLUID80 for filled liquid was selected in numerical analysis.

Buckling strength was obtained using different numbers of elements in both circumferential and axial directions. Analysis results were compared with the experimental ones. When the number of elements in the circumferential direction increased from 12 to 32, the maximum error reduced from 23% to 5%. It was preferable to keep the number of elements to more than 20 in order to obtain accurate buckling strength. Buckling stress was less sensitive to the number of elements in the axial direction than to the number in the circumferential direction. When the number of elements in the axial direction increased from 10 to 40, the maximum error reduced from 11% to 5%. When 32 elements in the circumferential direction and 40 elements in the axial direction were used, the buckling strength estimated was very accurate, within an error of 2.5%. The comparison between the numerical and experimental results is shown in Figure 3.

![Figure 3. Strain perpendicular to excitation axis, 7.5cm above the base, SN20H excitation. [8]](image)

Buckling stresses were evaluated for 13 cylindrical shells with height-to-diameter ratio from 1 to 2. The dimensions of the cylindrical shells used were diameter of 5 to 10m, height of 10m and thickness of 0.0023 to 0.003m. Material properties were E=2×10¹¹ N/m², ν=0.3 and σy=3.2×10⁸ N/m². A full model was used in ANSYS 12 to predict the buckling strength accurately. The fully anchored and the free boundary conditions were used for the bottom and top, respectively. Tank models were subjected to known base horizontal excitations. Various wave forms such as sinusoidal forms and real El Centro records were applied, and the intensity of the shaking was incrementally increased until elephant’s foot buckling occurred. Major response parameters were calculated and studied using nonlinear solver of ANSYS package.

3. RESULTS AND DISCUSSION

Buckling stresses were evaluated for cylindrical shells with different height-to-diameter ratios which has presented as the H/D ratio in Table 1. The allowable stresses of different guidelines for such tanks and boundary conditions are also presented.
Table 1- Sustainable stress of tanks with different H/D and D/t ratio.

<table>
<thead>
<tr>
<th>Model No.</th>
<th>D/t</th>
<th>H/D</th>
<th>Buckling Stress (MPa)</th>
<th>Allowable Stress (MPa)</th>
<th>API650</th>
<th>Eurocode8</th>
<th>NZSEE Guideline</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2500</td>
<td>1</td>
<td>28.2</td>
<td>80</td>
<td>45</td>
<td>32</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>2500</td>
<td>1.2</td>
<td>26.4</td>
<td>77</td>
<td>41</td>
<td>29</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>2500</td>
<td>1.4</td>
<td>23.9</td>
<td>75</td>
<td>35</td>
<td>25</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>2500</td>
<td>1.5</td>
<td>22.6</td>
<td>71</td>
<td>27</td>
<td>23</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>2500</td>
<td>1.6</td>
<td>20.2</td>
<td>65</td>
<td>22</td>
<td>19</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>2500</td>
<td>1.8</td>
<td>16.1</td>
<td>60</td>
<td>18</td>
<td>14</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>2500</td>
<td>2</td>
<td>12.0</td>
<td>53</td>
<td>13</td>
<td>11</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>2200</td>
<td>1.5</td>
<td>30.8</td>
<td>85</td>
<td>38</td>
<td>29</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>2300</td>
<td>1.5</td>
<td>27.0</td>
<td>79</td>
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<tr>
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<tr>
<td>11</td>
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<tr>
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<td>11.4</td>
<td>60</td>
<td>10</td>
<td>14</td>
<td></td>
</tr>
</tbody>
</table>

The results show that the buckling stress decreases significantly as the diameter-to-thickness ratio increases (Figure 4), while the buckling stress decreases slightly as the height-to-diameter ratio increases (Figure 5).

Table 1 shows that API criteria cannot prevent elastic-plastic buckling in the tanks generally. Eurocode8 considerations has a better approximation of allowable stress of the shell in comparison to API650. We can obviously see the close coherence of NZSEE results and the nonlinear analysis, which is in agreement with experiments.
4. CONCLUSIONS

This paper studied the elastic-plastic buckling strength of the cylindrical shell and tank subjected to horizontal excitation loads. The results were compared to some of the standard guidelines. It was seen that buckling stress decreases as the D/t increases, while the buckling stress decreases slightly as the H/D ratio increases. It was also shown that tanks designed by the codes API650 and Eurocode 8 may not be safe due to elastic-plastic buckling occurrence of the shell. However, NZSEE guidelines have a near coherence to the analysis results.

5. REFERENCES


