Design By Analysis as an Alternative to CSA B620 and 49CFR Structural Requirements for Pressurized Highway Tanks

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ABSTRACT

Current pressurized highway tank and its support designs must be substantiated using the design by formula (DBF) methods prescribed in CSA B620 (in Canada) and 49CFR (in US). This paper proposed an alternative design by analysis (DBA) procedure using Finite Element Analysis (FEA). A typical pressurized highway tank model, a twin delivery unit, is analyzed. The FEA results are compared with the design criteria of CSA B620 and 49CFR.

Keywords: FEA, pressurized cargo transport, 49CFR, CSA-B620, stress analysis, pressurized highway tank

INTRODUCTION

Pressurized highway tanks are widely used in industry for transporting dangerous goods, especially in the oil and gas industry. They are utilized to transport pressurized liquefied gases, such as LPG, N2, NH3, and CO2. The U.S. Department of Transportation has strict regulations governing the transport of such hazardous materials. The regulations appear in the Code of Federal Regulations Title 49 (49CFR). The design of such kinds of transports shall be registered in terms of the requirements described in CSA B620 in Canada or 49CFR in the United States. Fig. 1 shows a B-Train for transporting LPG, which consists of a pressure vessel, sub-frame or saddle, piping, and a truck.

Fig. 1 Pressurized cargo transport — B-Train (Source: CP Photo/MaXfield Inc.)
In 1866, the first Federal law was passed regulating the transportation of hazardous materials, specifically shipments of explosives and flammable materials such as nitroglycerin and glynoin oil. An earlier procedure to calculate the highway tank and its accessories dates back to 1953 when the first formula-based Code of Federal Regulations Title 49, in short 49CFR, was released. Since then, the calculations of highway tank have been traditionally based on analytical methods including those found in ASME VIII-1 and CSA B620 and 49CFR. However, the method of analysis of the vessel, the support structure and accessories is chosen by the design engineer. Tanks used in highway transportation experience dynamic loads in addition to static loads. In CSA B620 and 49CFR, there is still some ambiguity in vessel design under dynamic loading, especially the criteria for classified shell stresses under normal operating and extreme loading conditions. ASME VIII-1&2 coded pressure vessels have safety factor 3.5 and 3 respectively in Design by Formula (DBF) and Design by Analysis (DBA). In CSA B620 and CFR49 the allowable stress is defined using a safety factor of 4 based on the ultimate strength of material. This allowable stress only applies to DBF method. If one uses Finite Element Method (FEM) to analyze and verify the design, what the basic stress allowable \( S_{m} \), like that in ASME VIII-2, for highway pressurized tanks, i.e. what the safety factor shall be in DBA method.

A review of the referenced literature did not reveal a clear procedure for the interpretation of FE code developed stress data with CSA B620 and 49CFR. Although there are many papers published concerning the assessment of storage vessel stress and classification [8] [11] [12], there seems to have been little information published concerning the interpretation of stress data developed by FEA of highway transport tanks and supports.

A Trailer Mounted Twin Delivery Unit, shown in Fig. 2, is selected to explain the new approach to design and safety margins.

![Fig. 2 49" ID 2,000 USWG NH3 Twin Delivery Unit](image)

The Trailer Mounted Twin Delivery Unit consists of two identical 2,000 USWG pressure vessels mounted side by side on common saddles welded to a sub-frame. The vessels are designed to contain anhydrous ammonia. The vessel saddles are rigidly attached to the vessel skid. The skid is drawn tight down to a trailer frame with three hydraulic cylinders. Vertical movement (in the event of overturn) is restricted by line-locks in the
hydraulic circuit, positioned near to the cylinders. Longitudinal and lateral movement of the vessel/sub-frame assembly is restricted by angle stops welded to the trailer mounted frame. The unit is intended to be unloaded and loaded by weight. This is accomplished by lifting the vessels with the hydraulic cylinders and weighing the vessel/skid assembly with three load cells. Liquid is withdrawn through the liquid out connections of both vessels by one rear mounted pump. Surplus pump discharge is returned through a bypass valve to the vapour space of both vessels. Vapour equalization is achieved through the vapour connections of both vessels.

FE MODEL OF TWIN DELIVERY UNIT

The model used consists of two 49” ID vessels with 0.349” shell thickness and 0.347” semi elliptical (2:1) head thickness, and a welded sub-frame. The total seam-to-seam length of the shell is 224”. The shell and SE heads are both made from SA 455. The vessel design pressure is 265 psi plus hydrostatic head. To consider the worst possible product loading case, the tank is designed to fill up to 89% volume. The weight of the loaded unit is described as following:

- Weight of 89% filling product = 18,249 lbs
- Weight of empty vessel plus attachments = 10,230 lbs
- TOTAL weight of loaded tank with attachment = 28,479 lbs

The FE code Algor is employed. The geometry is created in Superdraw. If the unit is modeled with 3D solid elements, the simulated result must be linearized to compare with code-defined stress limits. There is great deal of literature on the topic of stress linearization and its interpretation [6-11], e.g., Hus and McKinley (1990) published a description of a computer program for computing the linearized stresses. Currently, at least one of the FE codes commercially available has a linearization routine built into the package. Fortunately, if the unit is modeled with plate elements, the assumption that stress is linear through the element is implicit in the element formulation, and no further stress linearization is necessary. Therefore, the unit is modeled wholly with plate elements. The mesh size was selected based upon previous work with a similar model, whose results showed reasonable convergence. In order to obtain the loaded unit weight, the “element load multiplier” is adjusted in the FE code. The acceleration is applied according to the respective load cases in CSA B620 and 49CFR. The loads that shall be considered for MC331 and TC331 type transport tanks in CSA B620 and 49CFR are interpreted as follows:

Normal Operating Loadings:

Load Case 1: 0.35G vertical and longitudinal accelerations combined with 0.2G lateral accelerations (per 49CFR 178.337-3(c)(1), CSA B620 Section 5.3.6.3.2)

Extreme Dynamic Loadings:

Load Case 2: 0.7G longitudinal acceleration (per 49CFR 178. 337-3(c)(2), CSA B620 5.3.6.3.3 (i))

Load Case 3: 0.4G lateral acceleration (per 49CFR 178. 337-3(c)(2), CSA B620 5.3.6.3.3 (iii))
Load Case 4: 1.7G vertical acceleration of the loaded unit (per 49CFR 178. 337-3(c)(2), CSA B620 5.3.6.3.3(ii))

Load Case 5: 2G longitudinal deceleration to simulate impact in an accident (per 49CFR 178. 337-3(d), CSA B620 5.3.6.5)

The analysis for the static load case in 49CFR 178.337-3(b) and CSA B620 5.3.6.1) is not covered here since it is covered in ASME VIII-1.

RESULTS AND COMPARISON

The stress contours in normal operating loadings are shown from Figure A-1 to Figure A-8 in the Appendix. The stress results of other load cases are listed in algebraic data in Table1, Table 2 and Table 3. Figure 3 illustrates the mesh and Von Mises Stress contour of the unit under normal operating loading.

For interpreting the results developed by FEM with CSA B620 or 49CFR, the stress intensities are calculated as both Principle Stresses and 2 x Tresca Stress and listed in Table 1. Furthermore, in Table 2 the stresses calculated by the method recommended by TTMA [4], a formula procedure per equations in CSA B620 and 49CFR, are compared with those developed by FEM. The highest stresses of sub-frame in different load cases are compared in Maximum Principle Stress and in Von Mises Stress in Table 3.

By comparing the values listed in Table 2, it is evident that results obtained by FEM show good agreement with the results calculated by the recommended analytical method [4]. The smallest stress difference between FEM and formula is 0.07% in load case 4, 1.7G vertical acceleration loading, and the largest is 3.45% in load case 5, 2G longitudinal loading. The moment direction of longitudinal loading at the calculated point is contrary to the moment direction of static weight of the loaded unit. The longitudinal Principle Stress induced by longitudinal loading negatively contributes to the effective
stresses. This is the reason why the stresses calculated in load case 2 and 5 are less than calculated in other load cases, whereas there is little stress discrepancy calculated by formula. In the FE model, the structure is broken up into small pieces that are easier to analyze. In the case of vessel and sub-frame they are broken up into small plates called element. All of the elements make up the mesh. Each of these elements can be easily solved by using element transfer function for stress and strain. As the number of elements increases, their size decreases and the solution accuracy increases. Thus, in FE model the negative effect on the effective stress due to longitudinal deceleration load is more accurately reflected than the simple formulaic approach. That is why load case 5 has the largest stress difference between the formula based method and FEM. The pink point in Figure 4 indicates the point used in the formula based calculation.

![Fig. 4 Maximum Principle stress contour under load case 5](image)

It is noted that the point selected in the formulaic calculation is not the highest stress intensity point. The highest stress intensity occurs near the front area of the saddle horn with a stress of 25,484 psi, tagged in red in Figure 4. In addition, there is much area in red shown in Figure 4 that has stress intensity higher than the selected point.

Tresca Stress is defined as: 
\[ S = \max(|S_{12}|, |S_{23}|, |S_{31}|) \], where 
\[ S_{12} = \frac{1}{2}(\sigma_1 - \sigma_2); \]
\[ S_{23} = \frac{1}{2}(\sigma_2 - \sigma_3); \]
\[ S_{31} = \frac{1}{2}(\sigma_3 - \sigma_1) \] and Tresca Criteria is described as 
\[ 2 \times S < S_y \] where \( \sigma_1, \sigma_2, \sigma_3 \) are longitudinal, hoop and radial Principle Stresses, respectively, and \( S_y \) is yield strength of material. For a thin wall vessel modeled with plate elements, the radial Principle Stress \( \sigma_3 \) is zero and the Tresca Stress equation reduces to 
\[ 2 \times S = \max(|\sigma_1|, |\sigma_2|, |\sigma_1 - \sigma_2|) \]. Since longitudinal Principle Stress \( \sigma_1 \) and hoop Principle Stress \( \sigma_2 \) are both tensile stresses at any points on the shell except at pad locations, \( 2 \times \) Tresca Stress equals the Maximum Principle Stress \( |\sigma_1| \) or \( |\sigma_2| \) whichever is larger. The Maximum Principle Stresses on the head occur near the shell junction, and consist of compressive stresses in Figure 5 and tensile stresses shown in Figure 6. 2 x Tresca
Stress on head equals $|\sigma_1| + |\sigma_2|$. This is the reason that the Maximum Principle Stress and the $2 \times$ Tresca Stress have the same value on the shell for each load case as listed in Table 1, while on the heads, the value of $2 \times$ Tresca Stress is greater than that of the Maximum Principle Stress. The Tresca Criteria are more conservative compared with the Maximum Principle criteria because it contains both tensile and compressive stresses.

Fig. 5  Minimum Principle Stress contour with amplified deformation in load case 5

Fig. 6  Intermediate Principle Stress contour with amplified deformation in load case 5
### Table 1  FEM Calculated largest value of Maximum Principle Stress and 2 x Tresca Stress (psi)

<table>
<thead>
<tr>
<th>Stress Type</th>
<th>Load Case 1</th>
<th>Load Case 2</th>
<th>Load Case 3</th>
<th>Load Case 4</th>
<th>Load Case 5</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>P₁</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SHELL</td>
<td>Maximum Principle</td>
<td>21,607</td>
<td>21,857</td>
<td>21,467</td>
<td>21,047</td>
</tr>
<tr>
<td></td>
<td>Tresca x 2</td>
<td>21,607</td>
<td>21,857</td>
<td>21,467</td>
<td>21,047</td>
</tr>
<tr>
<td>HEAD</td>
<td>Maximum Principle</td>
<td>18,800</td>
<td>18,784</td>
<td>18,790</td>
<td>18,808</td>
</tr>
<tr>
<td></td>
<td>Tresca x 2</td>
<td>23,515</td>
<td>23,383</td>
<td>23,541</td>
<td>23,405</td>
</tr>
<tr>
<td><strong>P₁ + Pₐ</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SHELL</td>
<td>Maximum Principle</td>
<td>24,725</td>
<td>24,785</td>
<td>24,786</td>
<td>24,295</td>
</tr>
<tr>
<td></td>
<td>Tresca x 2</td>
<td>24,725</td>
<td>24,785</td>
<td>24,786</td>
<td>24,295</td>
</tr>
<tr>
<td>HEAD</td>
<td>Maximum Principle</td>
<td>24,145</td>
<td>24,104</td>
<td>24,062</td>
<td>23,834</td>
</tr>
<tr>
<td></td>
<td>Tresca x 2</td>
<td>31,658</td>
<td>31,495</td>
<td>31,722</td>
<td>31,395</td>
</tr>
</tbody>
</table>

### Table 2 Comparison of Maximum Principle Stresses (psi) calculated by FEM vs. by formulas at the points selected in analytical method

<table>
<thead>
<tr>
<th>Load Case 1</th>
<th>Load Case 2</th>
<th>Load Case 3</th>
<th>Load Case 4</th>
<th>Load Case 5</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>FEM</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>18,636</td>
<td>18,495</td>
<td>18,822</td>
<td>18,689</td>
<td>18,051</td>
</tr>
<tr>
<td><strong>FORMULA</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>18,674</td>
<td>18,673</td>
<td>18,674</td>
<td>18,674</td>
<td>18,673</td>
</tr>
<tr>
<td><strong>DIFFERENCE %</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.20%</td>
<td>0.96%</td>
<td>-0.78%</td>
<td>-0.07%</td>
<td>3.45%</td>
</tr>
<tr>
<td><strong>Pₘ</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>18,218</td>
<td>18,008</td>
<td>18,328</td>
<td>18,387</td>
<td>17,493</td>
</tr>
</tbody>
</table>

### Table 3  Largest value of Maximum Principle Stresses and maximum Von Mises Stresses in sub-frame (psi)

<table>
<thead>
<tr>
<th>Load Case 1</th>
<th>Load Case 2</th>
<th>Load Case 3</th>
<th>Load Case 4</th>
<th>Load Case 5</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>VON MISES</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>16,312</td>
<td>19,157</td>
<td>15,451</td>
<td>14,109</td>
<td>35,302</td>
</tr>
<tr>
<td><strong>MAX. PRINCIPLE</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>16,303</td>
<td>18,225</td>
<td>15,049</td>
<td>12,046</td>
<td>36,992</td>
</tr>
</tbody>
</table>
RESULTS INTERPRETATION TO CSA B620 AND 49CFR

For stress analysis using an accurate and verifiable FEM, the stress analysis to highway-pressurized tanks could adopt the criteria in Appendix 4 of ASME-VIII-2. This point is verified by the consistence of determination of the safety factor for accident impact load case. An accident load is simulated by a 2G longitudinal deceleration in CSA B620 and 49CFR. The ASME VIII-2 categories these stress as \( P_l + P_b + (Q) \), and the total stress should be limited to \( 3S_m \). The basic stress allowable \( S_m \) herein shall be 25% of the tensile strength of material, i.e., design shall be based on the ultimate strength of the material, using a safety factor 4. Thus, one has \( S_m = \frac{1}{4}S_u \), which becomes \( 3S_m = \frac{3}{4}S_u = S_u/f \), where \( S_u \) is ultimate strength of material; \( f \) is a safety factor 1.3. It is identical to the safety factor of 1.3 defined in 49CFR 178.337-3(d), and CSA B620 5.3.6.5 for the accident impact load case. On the other hand, section 178.337-3(b)(2) in 49CFR and section 5.3.6.2(1) in CSA B620 both indicate that stress concentration in tension, bending, and torsion that occur at vessel, pads, saddles or other supports shall be considered in accordance with Appendix G of ASME-VIII-1, i.e., vessel, pads, saddles or other supports may be designed using the stress analysis methods given in Appendix 4 of ASME-VIII-2.

For a vessel made from SA 455, the minimum ultimate strength of the material is 75 ksi and its yield strength is 38 ksi; therefore, the basic stress allowable \( S_m \) is 18,750 psi in Principle Stress. Through analytical methods, the maximum value of Maximum Principle Stresses listed in Table 2, longitudinal Principle Stress, is 18,676 psi and by using FEM, the maximum value of Maximum Principle Stresses is 18,822 psi among the five load cases at corresponding points. The stress calculated by analytical methods is apparently within the limits set forth in B620 and 49CFR in all load cases. In the FE model, the maximum value of 18,822 psi calculated is actually the combination of Primary Local Primary Membrane Stress \( P_m \) and Primary Bending Stress \( P_b \), which should be limited to \( 1.5S_m = 28,125 \) psi. The other Maximum Principle Stresses in table 2 calculated by FEM should also be restricted under the limit of \( 1.5S_m \) since all of them are the combinations of \( P_m \) and \( P_b \). For stresses \( P_l \) from load case 1 to load case 4, all are within the limit of \( 1.5S_m \). For the accident impact load case 5, a 2G loading, the stress limit is set to the lesser of yield strength or \( 3S_m \), which is ultimate strength of material, using safety factor 1.3. Thus, the stress limit for load case 5 is set to the yield strength of 38,000 psi. Figure 7 and Figure 8 illustrate the classified Maximum Principle Stress contour of \( P_m \), \( P_l \) and \( P_l + P_b \) in load case 5. The tag in red indicates the location of highest stress. It shall be noted that all stresses shall be in Maximum Principle Stress in compliance with the stresses in B620 and 49CFR.
Saddles and other tank supports are vital to vessel integrity, and, as such, are designed with the same safety factor as the tank, i.e., Primary Membrane Stress $P_m$ on supports is limited to $S_m$; $P_l$ and $P_l + P_b$, are limited to $1.5S_m$; $P_l + P_b + Q$ are limited to $3S_m$. As shown in Figure 7 and Figure 8, the maximum Von Mises stress of 35,302 psi occurs on the front saddle web in load case 5 and Maximum Principle Stress of 36,992 psi occurs on the flange of the rear saddle, which is made from material SA 455. These two stresses
are categorized as $P_1 + P_2 + (Q)$ that shall not exceed the lesser of the yield strength or $3S_m$. The yield strength of 38 ksi of material SA 455 is set as stress limit here since $3S_m$ is greater than 38 ksi. The stresses on sub-frame under 2G longitudinal impact loading still satisfy the requirements defined. By reviewing Table 3, the classified stresses in other load cases are all within the specific stress limits set forth in B620, or 49CFR.

![Fig. 7 Von Mises Stress contour of $P_1 + P_2 + F$ on front saddle in load case 5](image1)

![Fig. 8 Maximum Principle Stress contour of $P_1 + P_2 + F$ on rear saddle in load case 5](image2)

**CONCLUSION AND SUGGESTION**

Highway transport vessels, pads, saddles or other supports may be designed using the stress analysis methods adopted from Appendix 4 of ASME code VIII-2. The basic stress allowable $S_m$ shall be based on the ultimate strength of material, using safety factor 4. The $k$ factor is select to 1 for all kinds of load combinations. In the stress analysis of the vessel, Maximum Principle Stress shall be used in compliance with the stress in B620 and CFR49 and $2 \times$ Tresca Stress should be used as a reference using.
the same specific stress limits. In the case of stress analysis for supports such as sub-frame, skid and saddle, the Von Mises Stress is suggested.

The point selected in B620 and CFR49 for formulaic calculation is not the real highest stress intensity point. There are some other areas that have stress intensity higher than the selected point. Therefore, an advanced stress analysis method such as FEM is suggested to get more detail stress distribution and more accurate stresses.

The stress analysis evaluation procedure discussed here could be applied to other kinds of highway-pressurized cargo transports, such as B-Trains, truck mounted units, and single unit trailers.

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BIBLIOGRAPHY


APPENDIX:
Fig. A-1  $P_m$ and $P_l$ Maximum Principle Stress contour

Fig. A-2  $P_m$ and $P_l$ $2\times$Tresca Stress contour

Fig. A-3  $P_l + P_b$ Maximum Principle Stress contour

Fig. A-4  $P_l + P_b$ $2\times$Tresca Stress contour
Fig. A-5 Von Mises Stress contour of sub-frame

Fig. A-6 Maximum Principle Stress contour of sub-frame

Fig. A-7 Von Mises highest stress place

Fig. A-8 Maximum Principle Stress highest stress place