COMPARING ANALYSIS OF A SPECIFIED PRESSURE VESSEL’S DESIGN METHODS

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Abstract: The utilization range of pressure vessels is really wide, especially in chemical, oil industry and energetic. These instruments store mostly on high pressure and temperature and often hazardous materials. For that very reason the design of these equipments is regulated by strict specifications. To these specifications belong the EN13445-1,2,3,4,5, and ASME SEC VIII DIV 1-2 standards too. The standardized calculations of pressure vessels found on the one part on the membrane stress of shell of revolution and on the other part on the stresses coming from bending the shell [1,2]. While the previous one has analytical solution for almost every type of shell the latter has solution only with significant simplifications. Originally we made the calculation for educational purpose, than we thought it is worthy to release it. In our work we analyzed the shell weakening effect of saddle in a horizontal retention tank. The calculations involve the whole strength control of the tank but considering the maximized length of the work we introduce only the supporting saddle’s stress increasing effect. We compared standardized calculations (EN 13445:2009 Unfired pressure vessels- Part 3: Design[3]/ ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 2007 Edition [4]) to Inventor software’s stress analysis

Keywords: pressure vessels, membrane stress, shell, tank, hazardous materials

1. STRENGTH CONTROL OF HORIZONTAL RETENTION TANK SUPPORTS (SADDLE)

Design data:
» \( p_i = 0.52 \text{ MPa} \) - Design Internal Pressure
» \( p_s = 0.024 \text{ MPa} \) - Static Head of the fluid fill
» \( D = 2440 \text{ mm} \) - Outside diameter of the cylinder
» \( V = 14.25 \text{ m}^3 \) - Tank estimated volume
» \( e = 8 \text{ mm} \) - Shell thickness
» Shell Material: P295GH (EN 10028:2009 [5])

Geometric dimensions from Figure 1 and 2: \( L = 3130 \text{ mm} \), \( H = 617 \text{ mm} \), \( a_1 = 343 \text{ mm} \), \( b_1 = 305 \text{ mm} \), \( b_2 = 0 \text{ mm} \) (vessels have no reinforcing plate), \( e_0 = 0 \text{ mm} \), \( a_0 = 0 \)

1.1. Control for excess stress caused by supports according to EN 13445:2009 standard [3]

The data of thank are coming from previous calculations. The loads of support are the saddle-reactions shown on Figure 3. The highest load occurs when the tank is full of water (e.g. at test pressure).
Determination of reaction forces:

\[ W_{\text{RT}} = 25,000 \text{ N (Retention Tank Weight)} \]
\[ W_f = 142,500 \text{ N (Weight of Fluid)} \]
\[ W = W_{\text{RT}} + W_f = 167,500 \text{ N (Total Flooded Shell Weight)} \]

\[ F_1 = F_2 = \frac{W}{2} = \frac{167,500}{2} \text{ N} = 83,750 \text{ N} \]

Maximum value of the reaction at the saddle support from weight and other loads as applicable

For vessels with two saddles of type A (Figure 1), the calculation is not required when the following conditions are met (the existing values in brackets):

a. there is no external pressure on the tank density of fluid \( \leq 1000 \text{ kg/m}^3 \) (density of the stored medium \( < 1000 \text{ kg/m}^3 \), density of the testing medium (water) = 1000 kg/m³)

b. permissible stress in shell material \( \geq 130 \text{ MPa} \) (\( f_s = 197 \text{ MPa} > 130 \text{ MPa} \))

c. welding factor of support steel structure \( \geq 0,8 \) (\( z = 1,0 > 0,8 \))

d. \( a_1 \leq 0,5 \times D_t \) (\( a_1 = 343 \text{ mm} < 0,5 \times 2424 \text{ mm} = 1212 \text{ mm} \))

e. \( L \leq L_{\text{max}} \), where \( L_{\text{max}} \) derived from Figure 4.

![Figure 4: L_max for horizontal vessel symmetrically on two saddles (3)](image)

Key: continuous lines: vessels without reinforcing plate dotted lines: vessels with reinforcing plate

\[ L = 3,13 \text{ m} < L_{\text{max}} = 12 \text{ m} \]

\[ b_2 = \frac{1,1 \times \sqrt{D_t \times e_t}}{L} \]

\[ b_2 = \frac{305 \text{ mm}}{1,1 \times \sqrt{2424 \times 8 \text{ mm}}} = 153,2 \text{ mm} \]

where \( e_t = 8 \text{ mm} \) (nominal thickness)

As the circumstances exist, the support is not necessary to be controlled for stress according to EN13445:2009 standard

1.2. Determination of stresses derived from excess loads according to ASMEVIII-2 [4]

Application of Rules control:

Geometry (the existing values in brackets):

\[ \delta \geq 120^\circ \) (\( \delta = 120^\circ \))

\[ a_1 \leq 0,25 \times L \) (\( a_1 = 343 \text{ mm} < 0,25 \times 3130 \text{ mm} = 782,5 \text{ mm} \))

Reinforcing Plates: vessels have no reinforcing plate

Stiffening Rings: vessels have no stiffening rings

- Moment and Shear Force

  the moment at the saddle:

\[ M = -R_2 \times a_2 \times \left( 1 - \frac{a_1}{L} + \frac{R_2^2 - R_2^2}{2 \times e_t \times L} \right) \]

where \( R_2 = \frac{D_t - e_t}{2} = \frac{2448 \text{ mm} - 8 \text{ mm}}{2} = 1216 \text{ mm} \)

\[ \text{mean radius of the cylindrical shell} \]
\[ M_1 = -83750 \times 343\, \text{mm} \times \left( 1 - \frac{\frac{343\, \text{mm}}{3130\, \text{mm}} \times \frac{1216^2\, \text{mm}^2}{2 \times 343 \times 3130\, \text{mm}^2} - \frac{617^2\, \text{mm}^2}{3 \times 3130\, \text{mm}}}{1 + \frac{4 \times 617\, \text{mm}}{3 \times 3130\, \text{mm}}} \right) \]
\[ M_1 = -2.011 \times 10^7 \, \text{Nm} \]

- the moment at the center of the vessel:

\[ M_2 = \frac{R_1 \times L}{4} \times \left( 1 + \frac{2 \times \left( \frac{R_1^2}{L^2} - \frac{R_2^2}{L^2} \right)}{1 + \frac{4 \times H_1}{3 \times L} - \frac{4 \times a_1}{L}} \right) \times \left( \frac{2 \times (1216^2 - 617^2)\, \text{mm}^2}{3130^2\, \text{mm}^2} - \frac{4 \times 343\, \text{mm}}{3130\, \text{mm}} \right) \]
\[ M_2 = \frac{63750 \times 3130\, \text{mm}}{4} \times \left( 1 + \frac{2 \times \left( \frac{3130^2\, \text{mm}^2}{3130^2\, \text{mm}^2} - \frac{4 \times 617\, \text{mm}}{3 \times 3130\, \text{mm}} \right)}{1 + \frac{4 \times 617\, \text{mm}}{3 \times 3130\, \text{mm}}} \right) \]

\[ M_2 = 4.643 \times 10^7 \, \text{Nm} \]

- the shear force at the saddle:

\[ T = \frac{R_1 \times (L - 2 \times a_1)}{L + \frac{4 \times H_1}{3}} = \frac{63750 \times (3130\, \text{mm} - 2 \times 343\, \text{mm})}{3130\, \text{mm} + \frac{4 \times 617\, \text{mm}}{3}} = 51784 \, \text{N} \]

**Longitudinal Stress**

The longitudinal membrane plus bending stresses in the cylindrical shell between the supports:

- top of shell:

\[ \sigma_1 = \frac{p \times R_m}{2 \times a} + \frac{M_2}{\pi \times R_m^2 \times a} = \frac{0.544 \, \text{MPa} \times 1216\, \text{mm}^2}{2 \times 8\, \text{mm}} - \frac{4.043 \times 10^7}{\pi \times 1216^2\, \text{mm}^2 \times 8\, \text{mm}} = 40 \, \text{MPa} \]

- bottom of shell:

\[ \sigma_2 = \frac{p \times R_m}{2 \times a} + \frac{M_2}{\pi \times R_m^2 \times a} = \frac{0.544 \, \text{MPa} \times 1216\, \text{mm}^2}{2 \times 8\, \text{mm}} + \frac{4.043 \times 10^7}{\pi \times 1216^2\, \text{mm}^2 \times 8\, \text{mm}} = 42.6 \, \text{MPa} \]

The longitudinal membrane plus bending stresses in the cylindrical shell at the support location:

- at points A and B in Figure 5. a):

\[ \sigma_3 = \frac{p \times R_m}{2 \times a} - \frac{M_4}{R_m \times \pi \times R_m^2 \times a} \]

- bottom of shell:

\[ \sigma_4 = \frac{p \times R_m}{2 \times a} - \frac{M_4}{R_m \times \pi \times R_m^2 \times a} \]

\[ K_1 = \Delta + \sin \Delta \times \cos \Delta - \frac{2 \times \sin^2 \Delta}{\pi \times \left( \frac{\sin \Delta}{\Delta} - \cos \Delta \right)} \]
\[ K_1^* = \Delta + \sin \Delta \times \cos \Delta - \frac{2 \times \sin^2 \Delta}{\pi \times \left( 1 - \sin \Delta \right)} \]
\[ K_2 = \Delta - \pi \times \left( \frac{\sin \Delta}{\Delta} \right) - \frac{16}{16 \times \frac{\pi}{6} + \frac{5 \times 2 \times \pi}{6}} \]
\[ K_2^* = \Delta - \pi \times \left( \frac{1}{\sin \Delta} \right) - \frac{1.178}{1.178 \times \left( 1 - \cos \Delta \right)} \]
\[ K_3 = \frac{1.176 \times \sin 1.178 \times \cos 1.178 - 0.06256}{1.178} = 0.0655 \]
\[ \sigma_3 = \frac{0.544 \, \text{MPa} \times 1216\, \text{mm}^2}{2 \times 8\, \text{mm}} - \frac{2.011 \times 10^7}{0.0655 \times \pi \times 1216^2\, \text{mm}^2 \times 8\, \text{mm}} = 49.6 \, \text{MPa} \]
\[ \sigma_4 = \frac{0.544 \, \text{MPa} \times 1216\, \text{mm}^2}{2 \times 8\, \text{mm}} - \frac{2.011 \times 10^7}{0.1218 \times \pi \times 1216^2\, \text{mm}^2 \times 8\, \text{mm}} = 36.9 \, \text{MPa} \]
Figure 5- Locations of Maximum Longitudinal Normal Stress and Shear Stress in the Cylinder

**Acceptance Criteria**

The absolute value of $\sigma_1, \sigma_2,$ and $\sigma_3, \sigma_4,$ as applicable shall not exceed $f_d$.

\[
\begin{align*}
\sigma_1 & = 40 \text{ MPa} < f_d = 197 \text{ MPa} \\
\sigma_2 & = 42.6 \text{ MPa} < f_d = 197 \text{ MPa} \\
\sigma_3 & = 49.6 \text{ MPa} < f_d = 197 \text{ MPa} \\
\sigma_4 & = 36.9 \text{ MPa} < f_d = 197 \text{ MPa}
\end{align*}
\]

**Shear Stresses**

As the shell is without stiffening rings and $a_1=313 \text{ mm} < 0.5*R_m=0.5*1216 \text{ mm}=608 \text{ mm},$ so.

The shear stress in the cylindrical shell is a maximum at Points E and F of Figure 5. b)

\[
\tau = \frac{K_3 \times F_1}{R_m \times \sigma_h}
\]

\[
K_3 = \frac{\sin \alpha}{\pi} \times \frac{\alpha - \sin \alpha \times \cos \alpha}{\pi - \alpha + \sin \alpha \times \cos \alpha}
\]

\[
\alpha = 0.95 \times \left( \frac{\pi}{2} - \frac{1}{2} \right) = 0.95 \times \left( \pi - \frac{2}{2} \times \frac{3}{2} \right) = 1.99
\]

\[
K_3 = \frac{\sin 1.99}{\pi} \times \frac{\pi - 1.99 + \sin 1.99 \times \cos 1.99}{1.99 - \sin 1.99 \times \cos 1.99} = 0.081
\]

\[
\tau = \frac{0.081 \times 83750 \text{ N}}{1216 \text{ mm} \times 8 \text{ mm}} = 7.58 \text{ MPa}
\]

The shear stress in the formed head is a maximum at Points E and F of Figure 5. b)

\[
\tau^* = \frac{K_3 \times F_1}{R_m \times \sigma_h} = \frac{0.081 \times 83750 \text{ N}}{1216 \text{ mm} \times 8 \text{ mm}} = 7.58 \text{ MPa}
\]

**Acceptance Criteria**

The absolute value of $\tau, \tau^*$, as applicable, shall not exceed $0.8* f_d$

\[
\tau = \tau^* = 7.58 \text{ MPa} < 0.8 \times 197 \text{ MPa} = 157.6 \text{ MPa}
\]

**Circumferential Stress**

The maximum compressive circumferential membrane stress in the cylindrical shell at the base of the saddle support:

\[
\sigma_0 = \frac{-K_2 \times F_2}{R_m \times \sigma_h} \text{ where}
\]

\[
K_2 = \frac{1 + \cos \alpha}{\pi - \alpha + \sin \alpha \times \cos \alpha} = \frac{1 + \cos 1.99}{\pi - 1.99 + \sin 1.99 \times \cos 1.99} = 0.75
\]

\[
K_2 = \frac{x_1 \times x_2}{R_m \times \sigma_h} = \frac{-0.75 \times 83750 \text{ N} \times 0.1}{1216 \text{ mm} \times 8 \text{ mm}} = -2.53 \text{ MPa}
\]

The circumferential compressive membrane plus bending stress at Points G and H of Figure 5:

\[
\sigma_B = \frac{83750 \text{ N} \times 1}{8 \times 1216 \text{ mm} = 9728 \text{ mm}}
\]

then

\[
\sigma_G = \frac{-F_2}{4 \times (2 \pi \times x_3 \times x_4)} = \frac{12 \times K_2 \times F_2}{\cos \phi} \text{ where}
\]

\[
\sigma_G = \frac{-F_2}{4 \times (2 \pi \times x_3 \times x_4)} = \frac{12 \times K_2 \times F_2}{\cos \phi}
\]
Acceptance Criteria

- The absolute value of \( \sigma_6 \) as applicable, shall not exceed \( f_a \)

- The absolute value of \( \sigma_7 \) as applicable, shall not exceed 1,25\( f_d \)

2. STRESS ANALYSIS WITH AUTODESK INVENTOR PROFESSIONAL 2012 SOFTWARE

With the given data we made a 3D model of the tank and tested it with water or pressure[6] (Figure 6). For the educational purpose we had simulations for the shell and the saddle as well, but here we introduce only the saddle part. It is obvious that the shell has higher stress at the touching areas of saddle (Figure 6. left). But the light blue colour does not sign extreme stresses, only tolerable stress-rising comes into existence. The peak stress occurs in the saddle (yellow area), but it can be solved by the construction of the saddle.

Figure 6.a, b. Stress analysis with water without pressure (a. left) and pressure without gravity (b. right)
3. CONCLUSION

In the analysis of subsidiary bending stresses of the support of horizontal retentional tanks we applied the calculation processes of the EN 13445:2009, and the ASME VIII-2 standard for a specific example. With this example our educational purpose was to introduce standard design methods to our students. The other purpose of this example is to familiarize our students the final element method. The final element calculations are meticulous, responsible tasks. The introduced models are only approaching to be able to qualify the tank’s stress conditions. But even by these models can be shown the shell stresses coming from inner overpressure (6.b.) and the subsidiary stress distribution (6.a.). Our further developing goals are to purchase and enter higher performance hardware and software into education.

Bibliography:


[3.] EN 13445-3:2009 Unfired pressure vessels- Part 3: Design

[4.] ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 2007 Edition

[5.] EN 10028:2009 Flat Products Made of Steel for Pressure Purposes Standard