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Introduction

Definition:
Pressure vessels are containers for fluids that are under pressure.

Function:
* Pressure vessels convert crude oil or petrochemical feedstock's into useful products, such as gasoline, diesel fuel, or jet fuel.
* Some pressure vessels used to storage raw materials
* Others used in the separation process.

Pressure vessel types according to its application:

- Horizontal Drum on Saddle Supports
- Vertical Drum on Leg Supports
- Tall Vertical Tower
- Vertical Reactor
- Spherical Pressurized Storage Vessel
The main components of the pressure vessels:

* Shell
* Head
* Nozzle
* Support
The shell:
The shell is the primary component that contains the pressure. Pressure vessel shells are welded together to form a structure that has a common rotational axis. Most pressure vessel shells are either cylindrical, spherical, or conical in shape.

Head:
* All pressure vessel shells must be closed at the ends by heads.
* Heads are typically curved rather than flat. Curved configurations are stronger and allow the heads to be thinner, lighter, and less expensive than are heads with a flat shape.
* The shape of the curve is usually semi-elliptical or hemispherical.
* The semi-elliptical shape is more common.

Nozzle:
* A nozzle is a cylindrical component that penetrates the shell and/or heads of a pressure vessel.
* **Nozzles may be used for the following applications:**
  * Attaching piping systems that are used for flow into or out of the vessel.
  * Attaching instrument connections, such as level gauges, thermo wells, or pressure gauges.
  * Providing access to the vessel interior at man ways.
  * Providing for direct attachment of other equipment items, such as a heat exchanger.
Internal pressure calculation procedure

- Divided to two calculation:
  - Head
  - Shell

First the Head:
At the beginning we calculate the corroded factor (calculation based on the worst condition)
And it equal

$$\text{Factor } K, \text{ corroded condition } [K_{cor}]:$$

$$= \left( 2 + \frac{\text{Inside Diameter}}{2 \times \text{Inside Head Depth}} \right)^2 / 6$$

Then the procedure done as the following:

1. Required Thickness due to Internal Pressure
   $$t = \frac{PDK}{2SE - 0.2P} \quad K = \frac{1}{6} \left[ 2 + \left( \frac{D}{2h} \right)^2 \right]$$

2. [MAWP]

3. [MAPNC]

4. Actual stress at given pressure and thickness, corroded

5. Straight Flange Required Thickness

6. Straight Flange Maximum Allowable Working Pressure
Second : the shell calculation:
The procedure is the same as the head procedures Except the equations which is:

1- The thickness \[ t = \frac{PR}{SE - 0.6P} \]
2- The Pressure \[ P = \frac{SEt}{R + 0.6t} \]

Sample of the calculation:

According to ASME VII Div.1 (UG-27), after calculate the thickness we add the Corrosion allowance, then find the reasonable commercial thickness.

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

\[ \text{Less Operating Hydrostatic Head Pressure of } 0.015 \text{ MPa} \]
\[ = \frac{(S*E*t)}{(R+0.6*t)} \text{ per UG-27 (c)(1)} \]
\[ = \frac{(137.90*1.00*6.8000)}{(753.2000+0.6*6.8000)} \]
\[ = 1.238 - 0.015 = 1.224 \text{ MPa} \]

Maximum Allowable Pressure, New and Cold [MAPNC]:

\[ = \frac{(S*E*t)}{(R+0.6*t)} \text{ per UG-27 (c)(1)} \]
\[ = \frac{(137.90*1.00*10.0000)}{(750.0000+0.6*10.0000)} \]
\[ = 1.824 \text{ MPa} \]

Actual stress at given pressure and thickness, corroded [Sact]:

\[ = \frac{P*(R+0.6*t)}{(E*t)} \]
\[ = \frac{(0.365*(753.2000+0.6*6.8000))}{(1.00*6.8000)} \]
\[ = 40.619 \text{ MPa} \]
External pressure calculation

The External pressure calculation divided into two types of calculation due to the vessel’s part:

1. The head
2. The shell

First: the head:

Assume a thickness and calculate Factor “A.”

\[ A = \frac{0.125t}{R_o} \]

Note: \( R_o = 0.9 \times D_o \)

Find Factor “B” from applicable material chart

The tables of the materials found in ASME II part D subpart 3

Compute \( P_a \).

\[ P_a = \frac{0.0625E}{(R_o/t)^2} \]

\[ P_a = \frac{Bt}{R_o} \]

Then we make the Requirement check according to the ASME VII Div.1 (UG-33a) due to the internal pressure

\[ P = 1.67 \times \text{External Design pressure for this head.} \]

An Example for the calculation:

Results for Maximum Allowable External Pressure (MAEP):

<table>
<thead>
<tr>
<th>Tca</th>
<th>OD</th>
<th>D/t</th>
<th>Factor A</th>
<th>Factor B</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.800</td>
<td>1510.00</td>
<td>261.72</td>
<td>0.0005307</td>
<td>53.05</td>
</tr>
</tbody>
</table>

\[ \text{EMAP = } B/(R_o \times D/t) = 53.0550/(0.9000 \times 261.7241) = 0.2252 \text{ MPa} \]

Required Thickness due to Internal Pressure [tr]:

\[ = (P \times D \times K_o)/((2 \times S \times E - 0.2 \times F) \text{ Appendix 1-4(c)}) \]

\[ = (0.172 \times 1506.4000 \times 0.994)/(2 \times 137.90 \times 1.00 - 0.2 \times 0.172) \]

\[ = 0.9343 + 3.2000 = 4.1343 \text{ mm} \]

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

\[ = ((2 \times S \times E \times t)/(K_o \times D + 0.2 \times t))/1.67 \text{ per Appendix 1-4 (c)} \]

\[ = ((2 \times 137.90 \times 1.00 \times 5.8000)/(0.994 \times 1506.4000 + 0.2 \times 5.8000))/1.67 \]

\[ = 0.639 \text{ MPa} \]
Second: the Shell

Assume a thickness (t) (we can use the thickness requirement for the internal pressure)

Calculate L/D, and Do/t ratios

If the allowable external pressure is less than the design external pressure, then a decision must be made on how to proceed. Either

(b) Select to use stiffening rings to reduce the “L’ dimension.

(a) select a new thickness

Hemi-heads.... \( L = LT - T + 0.333 \)

2:1 Semi-elliptical heads.... \( L = LT - T + 0.166 \)

The tables of the materials found in ASME II part D subpart 3

Assume a thickness and calculate Factor “A.”

Find Factor “B” from applicable material chart

If Factor “A” falls to the left of the material line,

Compute \( P_a \).

If Factor “A” falls to the right of the material line,

Compute \( P_a \).

Example for calculation

Results for Maximum Allowable External Pressure (MAEP):

<table>
<thead>
<tr>
<th>Tca</th>
<th>OD</th>
<th>SLEN</th>
<th>D/t</th>
<th>L/D</th>
<th>Factor A</th>
<th>Factor B</th>
</tr>
</thead>
<tbody>
<tr>
<td>6.800</td>
<td>1520.00</td>
<td>1625.00</td>
<td>223.53</td>
<td>1.0691</td>
<td>0.0003742</td>
<td>37.42</td>
</tr>
</tbody>
</table>

\( P_a = \frac{2AE}{3(D_o/t)} \)

\( P_a = \frac{4B}{3(D_o/t)} \)

EMAP = \( \frac{4B}{3(D_o/t)} \) = \( \frac{4 \times 37.4161}{3 \times 223.5294} \) = 0.2232 MPa
Minimum Design Metal Temperature
Min. Design Metal Temperature:

MDMT is the lowest temperature which the vessel’s material bear with it. Can get it as the following steps:

1. From the USC-66 Table @ the ASME VII Div.1 we use the thickness and the material type (Curve A, B, C, D).

2. Then know that the vessel need an Impact test or not by the UCS-66M we enter the figure with thickness and MDMT.
3- we also use the UCS-66.1 to know the reduction in MDMT without impact test

This factor can get by this equation:

\[
\frac{(Tr \times E')}{(Tg - C)}
\]

An Example for The MDMT calculation

Minimum Design Metal Temperature Results:

Govrn. thk, \( tg = 10.000 \), \( tr = 5.864 \), \( c = 3.2000 \) mm, \( E^* = 1.00 \)
Stress Ratio = \( tr \times \frac{(E^*)}{(tg - c)} = 0.862 \), Temp. Reduction = 8 C

Min Metal Temp. w/o impact per UCS-66 = -48 C
The Nozzle calculation:
First: The calculation of the nozzle required thickness:
According to the ASME VII Div.1 (UG-45)

\[ t_{UG-45} = \max(t_a, t_b) \]

\[ t_b = \min[t_{b3}, \max(t_{b1}, t_{b2})] \]

Ta = minimum neck thickness using UG 27 (internal pressure) and UG 28 (External pressure) plus Corrosion allowance

\( T_{b1} \) = for vessels under internal pressure plus corrosion allowance
\( T_{b2} \) = for vessels under external pressure plus corrosion allowance
\( T_{b3} \) = thickness given in UG 45 plus corrosion allowance
\( Tr_{16b} \) = thickness given due to ASME (UG-16-b) the min. thickness for head and shell (1.5mm + C.A.)

Example for NA1 nozzle (inlet nozzle):

**UG-45 Minimum Nozzle Neck Thickness Requirement: [Int. Press.]**
- Wall Thickness for Internal/External pressures \( ta = 3.8491 \) mm
- Wall Thickness per UG16(b), \( tr_{16b} = 4.7000 \) mm
- Wall Thickness, shell/head, internal pressure \( tr_{b1} = 9.0555 \) mm
- Wall Thickness \( tb1 = \max(tr_{b1}, tr_{16b}) = 9.0555 \) mm
- Wall Thickness, shell/head, external pressure \( tr_{b2} = 3.7628 \) mm
- Wall Thickness \( tb2 = \max(tr_{b2}, tr_{16b}) = 4.7000 \) mm
- Wall Thickness per table UG-45 \( tb3 = 9.4200 \) mm
The Calculation of nozzle reinforcement:

1. \( F \): correction factor that substitute the variation in internal pressure stresses on different planes with respect to the axis of a vessel, and can get it from this curve.

2. \( F_{r1} = \frac{S_n}{S_v} \)

3. \( F_{r2} = \frac{S_n}{S_v} \)

4. \( F_{r3} = \frac{(S_n \text{ or } S_v \text{ the lesser})}{S_v} \)

5. \( F_{r4} = \frac{S_p}{S_v} \)

6. \( S_n \): the allowable stress for the nozzle material.

7. \( S_v \): the allowable stress for the vessel material.

8. \( S_p \): the allowable stress for the pad material.

   Any allowable stress can get from ASME II part D – Subpart 1

The Diminutions:

The thickness:

1. \( t_n \): the nozzle diameter (can get from schedule by get \((t_{min} \times 0.875)\)-corrosion allowance.

2. \( t_{rn} \): the nozzle required thickness due to the internal pressure.

3. \( t \): the shell total thickness \((t_r + \text{the thickness required for the reinforcement area})\)

4. \( t_r \): the shell required thickness due to the internal pressure.

5. \( t_e \): the pad thickness.

The area:

\( A \): the required area.

\( A_1 \): the reinforcement area in the shell.

\( A_2 \): the reinforcement area in the nozzle.

\( A_3 \): the reinforcement area in the inside wall thickness of the nozzle.

\( A_4 \): the reinforcement area in the welding.

\( A_5 \): the reinforcement area in the pad.

Without Reinforcing Element:

\[
A = A_1 + A_2 + A_3 + A_4_1 + A_4_3
\]

Area required

Area available in shell; use larger value

Area available in nozzle projecting outward; use smaller value

Area available in inward nozzle; use smallest value

Area available in outward weld

Area available in inward weld

Opening is adequately reinforced

Opening is not adequately reinforced so reinforcing elements must be added and/or thicknesses must be increased
For the using a hub for the nozzle an additional area will add to the reinforcement calculation (A6):

\[
\text{Height value from sketch (e-1) [te]:} \\
= \frac{(\text{Hub Thickness} - \text{Neck Thickness})}{\cos(30)} \\
= 65.399/0.5773 \\
= 113.2741 \text{ mm}
\]

Note: Hub Height was < 2.5 times Hub Thickness, use sketch UG-40 (e-1).

**UG-40, Limits of Reinforcement: [Internal Pressure]**

- Parallel to Vessel Wall (Diameter Limit) $D_l$ 310.9452 mm
- Parallel to Vessel Wall, opening length $d$ 155.4726 mm
- Normal to Vessel Wall (Thickness Limit), no pad $T_{lnp}$ 17.0000 mm

So the result of the reinforcement of the nozzle NA1:

<table>
<thead>
<tr>
<th>AREA AVAILABLE, A1 to A5</th>
<th>MAWP</th>
<th>External</th>
<th>Mapnc</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area Required Ar</td>
<td>910.370</td>
<td>390.033</td>
<td>NA mm²</td>
</tr>
<tr>
<td>Area in Shell A1</td>
<td>146.843</td>
<td>277.147</td>
<td>NA mm²</td>
</tr>
<tr>
<td>Area in Nozzle Wall A2</td>
<td>195.573</td>
<td>197.935</td>
<td>NA mm²</td>
</tr>
<tr>
<td>Area in Inward Nozzle A3</td>
<td>0.000</td>
<td>0.000</td>
<td>NA mm²</td>
</tr>
<tr>
<td>Area in Welds A41+A42+A43</td>
<td>83.486</td>
<td>83.486</td>
<td>NA mm²</td>
</tr>
<tr>
<td>Area in Element A5</td>
<td>0.000</td>
<td>0.000</td>
<td>NA mm²</td>
</tr>
<tr>
<td>Area in Hub A6</td>
<td>2223.559</td>
<td>2223.559</td>
<td>NA mm²</td>
</tr>
<tr>
<td>TOTAL AREA AVAILABLE Atot</td>
<td>2649.461</td>
<td>2782.128</td>
<td>NA mm²</td>
</tr>
</tbody>
</table>
The Hydro test Calculation

According to ASME VII Div.1 (UG -99)

**Hydrostatic Test Pressure Results:**

- Pressure per UG99b = 1.3 * M.A.W.P. * Sa/S = 1.370 MPa
- Pressure per UG99b[34] = 1.3 * Design Pres * Sa/S = 0.455 MPa
- Pressure per UG99c = 1.3 * M.A.P. - Head(Hyd) = 2.128 MPa
The Wind Calculation:

According to UBC code we use this equation:

\[ P = C_e C_q q_s I \]

For

\[ P = \text{design wind pressure, PSF} \]
\[ C_e = \text{combined height, exposure, and gust factor. See Table 3-6.} \]
\[ C_q = \text{pressure coefficient. See Table 3-7. Use 0.8 for most vessels} \]
\[ q_s = \text{wind stagnation pressure. See Table 3-5.} \]
\[ I = \text{importance factor, 1.15 for most vessels. See Table 3-8.} \]
Example for the wind calculation:

The parameters

The pressure of the wind equation

The combined height

P(height) = Ce(height,Exp) * Cq * qs * Imp Fact. [18-1](1994) or [20-1](1997)

The values of Ce are shown as the in the table below:

<table>
<thead>
<tr>
<th>Element</th>
<th>Ce</th>
</tr>
</thead>
<tbody>
<tr>
<td>From: 10</td>
<td>1.0600</td>
</tr>
<tr>
<td>From: 20</td>
<td>1.0600</td>
</tr>
<tr>
<td>From: 30</td>
<td>1.0600</td>
</tr>
<tr>
<td>From: 40</td>
<td>1.0600</td>
</tr>
<tr>
<td>From: 50</td>
<td>1.0600</td>
</tr>
</tbody>
</table>

Wind Load Calculation

<table>
<thead>
<tr>
<th>From</th>
<th>To</th>
<th>Height</th>
<th>Diameter</th>
<th>Wind Area</th>
<th>Wind Pressure</th>
<th>Wind Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>20</td>
<td>1500.00</td>
<td>1941.60</td>
<td>649596.00</td>
<td>1.18266</td>
<td>767.047</td>
</tr>
<tr>
<td>20</td>
<td>30</td>
<td>1500.00</td>
<td>1944.00</td>
<td>2.430E+06</td>
<td>1.18266</td>
<td>2873.78</td>
</tr>
<tr>
<td>30</td>
<td>40</td>
<td>1500.00</td>
<td>1944.00</td>
<td>3.694E+06</td>
<td>1.18266</td>
<td>4368.18</td>
</tr>
<tr>
<td>40</td>
<td>50</td>
<td>1500.00</td>
<td>1944.00</td>
<td>2.430E+06</td>
<td>1.18266</td>
<td>2873.78</td>
</tr>
<tr>
<td>50</td>
<td>60</td>
<td>1500.00</td>
<td>1941.60</td>
<td>649596.00</td>
<td>1.18266</td>
<td>767.047</td>
</tr>
</tbody>
</table>

The force of the wind acting upon the certain section
Saddle supports calculations procedures:

The saddle design depends on two main forces:
1- the horizontal force ($F_h$)
2- the total load force acting on the saddle ($Q$)

The total force per saddle calculated from three forces:
A- the weight of the vessel
B- the tangential forces
C- the longitudinal forces

**Longitudinal force**

- **Seismic**
  $$ F_L = C_w * W_o $$

- **Wind**
  $$ F_L = A_f * C_f * q_z * G_0 $$

- $C_f = 0.8$
- $G = 0.85$
- $q = 0.00256KzV^2I$
- $K_z$ = from Table 3-23
- $I = 1.15$
- $V$ = basic wind speed

<table>
<thead>
<tr>
<th>Height (ft)</th>
<th>$K_z$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-15</td>
<td>0.85</td>
</tr>
<tr>
<td>20</td>
<td>0.9</td>
</tr>
<tr>
<td>25</td>
<td>0.94</td>
</tr>
<tr>
<td>30</td>
<td>0.98</td>
</tr>
<tr>
<td>40</td>
<td>1.04</td>
</tr>
<tr>
<td>50</td>
<td>1.09</td>
</tr>
<tr>
<td>60</td>
<td>1.13</td>
</tr>
</tbody>
</table>
Transverse forces $F_t$ per saddle.

Seismic

$F_t = (C_h W_0)^{0.5}$

Wind

$F_t = (A_f c_f G_d q_z)^{0.5}$

$A_f = D_e (L + 2H)$

So the total saddle reaction forces $(Q)$ equal:

$Q = \text{greater of } Q_1 \text{ or } Q_2$

$Q_1 = \frac{W_0}{2} + \frac{F_L B}{L_s}$

$Q_2 = \frac{W_0}{2} + \frac{3F_t B}{L_s}$

Second the horizontal force:

$F_h = Q \left( \frac{1 + \cos \beta - 0.5 \sin^2 \beta}{\pi - \beta + \sin \beta \cdot \cos \beta} \right)$

For:

$\beta = \pi - \frac{\theta}{2}$

Then we design the saddle according to the stresses from these two main forces.
Example for calculation:

Load Combination Results for Q + Wind or Seismic [Q]:

\[ = \text{Saddle Load} + \max( Fw1, Fwt, Fsl, Fst) \]

\[ = 34027 + \max( 1188, 20002, 21879, 81391 ) \]

\[ = 115419.5 \text{ N} \]

- Longitudinal wind force
- Tangential wind force
- Longitudinal seismic force
- Tangential seismic force
Then The types of stress which acting on the saddle:

**Horizontal Vessel Analysis Results:**

<table>
<thead>
<tr>
<th>Stress Type</th>
<th>Actual</th>
<th>Allowable</th>
</tr>
</thead>
<tbody>
<tr>
<td>Long. Stress at Top of Midspan</td>
<td>15.67</td>
<td>137.90 MPa</td>
</tr>
<tr>
<td>Long. Stress at Bottom of Midspan</td>
<td>24.09</td>
<td>137.90 MPa</td>
</tr>
<tr>
<td>Long. Stress at Top of Saddles</td>
<td>27.66</td>
<td>137.90 MPa</td>
</tr>
<tr>
<td>Long. Stress at Bottom of Saddles</td>
<td>15.57</td>
<td>137.90 MPa</td>
</tr>
<tr>
<td>Tangential Shear in Shell</td>
<td>17.00</td>
<td>110.32 MPa</td>
</tr>
<tr>
<td>Circ. Stress at Horn of Saddle</td>
<td>31.93</td>
<td>172.37 MPa</td>
</tr>
<tr>
<td>Circ. Compressive Stress in Shell</td>
<td>4.06</td>
<td>137.90 MPa</td>
</tr>
</tbody>
</table>
The stresses:

1– the Longitudinal stress:

- Tension stress
- Compression stress

Max. value of these stresses:

- Stress at the saddle top area
- Stress at the saddle bottom area
- Stress at the mid span (top or bottom)

Max. value of these stresses:

- @ mid span (Top)
- @ Saddle (bottom)
2— the Tangential stress:

- A > (R/2)
  - Max. value of these stresses
  - Shell
    - Equation with (K2 factor)
    - Equation with (K3 factor)

- A <= (R/2)
  - Max. value of these stresses
  - Shell
  - Head
    - additional stress @ head

3— the Circumferential stress:

- Max. value of these stresses
- at saddle horn
- Length >= (8R)
- Length < (8R)
- Stresses at bottom of shell
Lifting Lugs:

The design bases at the empty weight of the vessel.

Making the different combined stress:

1. Total Shear Stress for Combined Loads
   Primary Shear Stress in the Welds due to Shear Loads + Shear Stress in the Welds due to Bending Loads

2. Total Combined Stress at the base of the lug
   Bending stress at the base of the lug + Tensile stress at the base of the lug
## Check the results:

1. **Allowable Shear Stress for Combined Loads**
   - 96.90 MPa
   - Total Shear Stress for Combined Loads
     - 78.91 MPa

2. **Allowable Shear Stress in Lug above Hole**
   - 96.90 MPa
   - Shear Stress in Lug above Hole
     - 35.52 MPa

3. **Allowable Bearing Stress**
   - 181.69 MPa
   - Pin Hole Bearing Stress
     - 70.72 MPa

4. **Lug Allowable Stress for Bending and Tension**
   - 159.9 MPa
   - Total Combined Stress at the base of the lug
     - 61.1 MPa