UNDERSTANDING MAWP AND HIGH-PRESSURE VESSEL DESIGN

How the ASME VIII Division 1 Code Formulas for Calculation of the Thickness of the Various Components Have Been Developed

An ioMosaic White Paper

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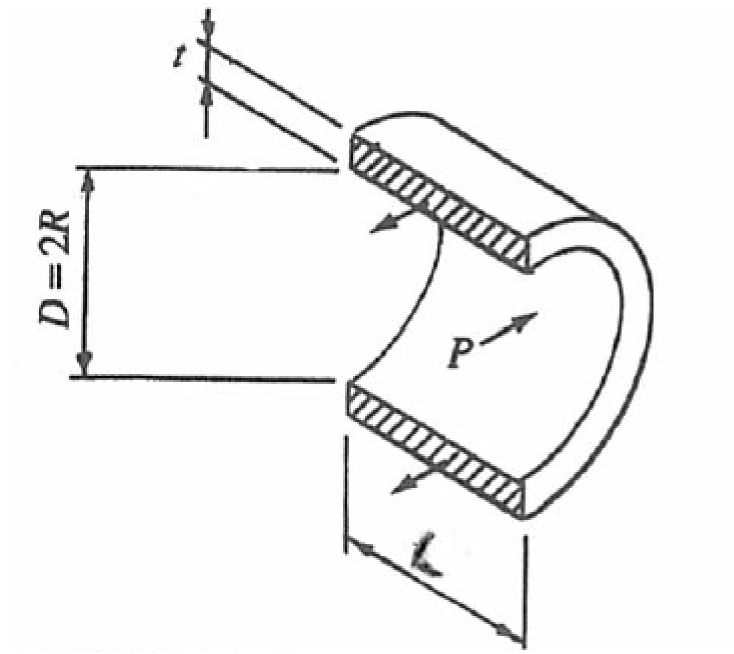
Introduction

Consider how the circumferential forces in a thin cylindrical shell due to internal pressure affects the thickness of the shell.

Figure 1. Circumferential Stress or Hoop Stress (longitudinal Joint) in a thin cylindrical shell due to internal pressure



In the above figure, the internal pressure acts on the projected area formed by the diameter, D, and the length, L, of the cylinder. The force generated as a result of this pressure acts circumferentially and tends to increase radius and tear apart the cylinder along the axis. This is how a cylinder filled with liquid fails when subjected to thermal expansion. We know that pressure is force per unit area. Therefore, this force is thus calculated by:

$$\text{Tearing Force} = \text{Pressure x Projected Area}$$
$$= P \text{ x } D \text{ x } L = 2PRL \qquad (1)$$

The allowable stress, S, of the material of construction of the vessel at the design temperature that opposes the tearing by the pressure acts on the two longitudinal areas, each Lt units, of the vessel. The opposing force:

$$\text{Opposing Force} = \text{Area x Allowable Stress (Force/area)}$$
$$= 2Lt \text{ x } S = 2LtS$$

If the vessel is made by rolling and welding of a plate, and the welding strength has an efficiency of E, the effective opposing force:

$$\text{Effective opposing force} = 2ELtS \qquad (2)$$

From equations (1) & (2):

$$2PRL = 2ELtS$$

$$\text{or, } t = \frac{PR}{SE} \qquad (3)$$

The above formula holds good for thin walls. In order to use it for thicker walls, the formula was modified by ASME VIII committee in 1942 as:

$$t = \frac{PR}{SE - 0.6P} \qquad (4)$$

Or,

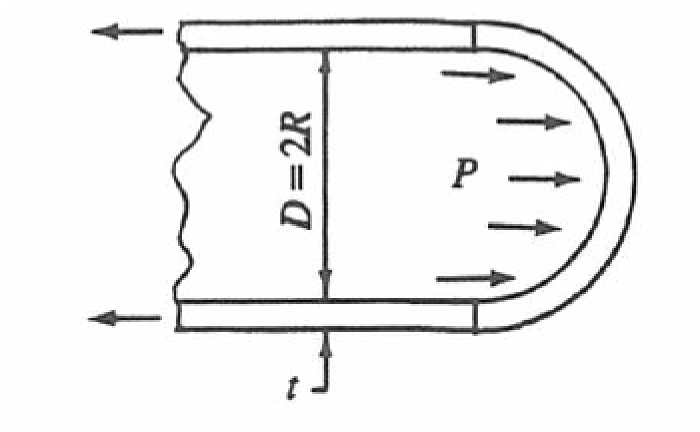
$$P = \frac{SEt}{R + 0.6t} \qquad (5)$$

Similarly, the thickness of the wall necessary to oppose the tearing of the vessel circumferentially can be developed by considering the pressure in the axial direction on the projected area that equals the cross section of the vessel.

$$\text{Tearing Force} = \text{Pressure x Projected Area}$$

$$= \text{P x } \frac{\pi}{4}\text{D}^2 \qquad (6)$$

Figure 2. Longitudinal Stress (Circumferential Joint) On a Thin Cylindrical Shell Due To Internal Pressure



Effective opposing Force

= Area x Allowable Stress (Force/area)

$$= \pi Dt \ x \ SE \qquad (7)$$

By equating equations (6) & (7) one gets for longitudinal force, the required thickness:

$$t = \frac{PD}{4SE} = \frac{PR}{2SE} \qquad (8)$$

Thus a cylindrical vessel weld breaks at the longitudinal joint weld because this joint is weaker requiring higher plate thickness compared with circumferential joint.

The longitudinal force formula was also modified in 1942 for the same reason as the circumferential force formula [ASME VIII Division 1 (2015), UG-27, p18]:

$$t = \frac{PR}{2SE + 0.4P} \qquad (8)$$

$$\text{or, } P = \frac{2SEt}{R - 0.4t} \qquad (9)$$

For spherical shells,

$$t = \frac{PR}{2SE - 0.2P} \qquad (10A)$$

$$P = \frac{2SEt}{R + 0.2t} \qquad (10B)$$

Formed heads*

*ASME VIII Division 1 (2015)[UG-32, p28]

Required thickness is calculated by:

$$t = \begin{cases} \begin{cases} \dfrac{0.88PL}{SE - 0.1P}, \ \mathrm{t_s/L \geq 0.002, Flanged \ \& \ Dished \ or \ Torrispherical \ head(11a)} \\ \dfrac{PL}{SE - 0.1P}, \ \mathrm{t_s/L \geq 0.002, Falnged \ \& \ Dished \ or \ Torrispherical \ head(11b)} \end{cases} \\ \dfrac{PR}{SE - 0.1P}, \ \mathrm{t_s/L \geq 0.002, 2{:}1 \ Ellipsoidal \ head \ (12)} \\ \dfrac{PR}{2SE - 0.2P}, \ \mathrm{Hemispherical \ head \ or \ sphere \ (13)} \\ \dfrac{PR}{Cos\alpha\left(SE - 0.6P\right)}, \ \alpha \ \mathrm{(=half \ apex \ angle) \leq 30 \ degree, Conical \ head \ or \ cone(14)} \end{cases}$$

Allowable Internal Pressure of formed head is calculated by:

$$P = \begin{cases} \begin{cases} \dfrac{SEt}{0.885L + 0.1\text{t}}, & \text{L/r} = 16\tfrac{2}{3}, \text{ Flanged \& Dished head} \quad (15A) \\ \dfrac{2SEt}{LM + 0.2t}, & \text{L/ r} < 16\tfrac{2}{3}, \text{ Flanged \& Dished head} \quad (15B) \end{cases} \\ \dfrac{SEt}{R + 0.1t}, \text{ 2:1 Ellipsoidal head} & (16) \\ \dfrac{2SEt}{R + 0.2t}, \text{ Sphere and Hemispherical head} & (17) \\ \dfrac{SEtCos\alpha}{R + 0.6tCos\alpha}, \text{ Cone or conical section} & (18) \end{cases}$$

Designing for external pressure

When a pressure vessel is subjected to vacuum, it is required to design the vessel for 15 psi external pressure. If the same vessel is also jacketed, then the outside wall of the jacket is designed for the design pressure of the jacket, but the shell wall covered by the jacket must be designed for external pressure that equals the internal design pressure in psig of the jacket plus 15 psig. The wall of the vessel not covered by the jacket may be designed for the specified internal design pressure and 15 psig external.

The design of a pressure vessel for external pressure involves trial-and-error including the consideration of length to external diameter ratio and external diameter to shell thickness ratio. Please see ASME VIII Division 1 for details.

Calculation of Maximum Allowable Working Pressure (MAWP)

The MAWP is not the same as the Design Pressure. It is almost always higher than the Design Pressure. A vessel may have multiple design pressures with coincident design temperatures. All data should be transmitted to the fabricator. The Design Pressure and the coincident Design Temperature, fluid properties such as density, applicable hydrostatic head together with dimensions, material of construction, applicable corrosion allowance, joint efficiency and extent of radiography, shape of the ends, number and nominal size of the fittings and their facing (whether flanged or weldable), vertical or horizontal orientation, geographical location of equipment are supplied to the fabricator. The fabricator, in turn, calculates the theoretical thickness, adds forming allowance and corrosion allowance, then picks up the nearest higher commercially available plate thickness for fabrication. He/she then subtracts the corrosion allowance from the selected commercial plate thickness and plugs the reduced thickness in the code formula, and back-calculates the allowable pressure. He/she repeats the calculation for all the component parts of the vessel. The least allowable pressure of all the components rounded to nearest lower integer is labeled as the MAWP of the vessel.

A few things to remember about MAWP:

1) The commercial plate thickness chosen, as described under paragraph **Calculation of Maximum Allowable Working Pressure (MAWP),** may not be the final selection of the plate thickness because of the influence of the following aspects:

    a. Thickness required for tall towers

        i. Wind load including deflection effect
        ii. Seismic load (Earthquake load)
        iii. Eccentric load
        iv. Elastic stability
        v. Cyclic load (heating and cooling in cycles)
        vi. Combination Stresses such as combination of wind load or earthquake load, internal pressure and weight of vessel with fluid.

    b. The lack of space on a component to install reinforcement pad around an opening created for process requirement

2) So, the calculated MAWP based on selected plate thickness may be devalued.

Stamping MAWP

After the fabricator decided on the selection of MAWP, the value of MAWP along with coincident design temperature is stamped on the name plate and shown on the U-1 form.

Hydrostatic Pressure (UG-99)

Hydrostatic Test Pressure (HTP) for vessels designed for internal pressure is determined by:

$$HTP \geq 1.3MAWP\left(\frac{\text{Material Stress Value at Test Temperature}}{\text{Material Stress Value at Design Temperature}}\right) \quad (19)$$

Test fluid is water unless water is not acceptable for the vessel. Any nonhazardous liquid below its boiling point may be used as the test fluid. If no liquid can be used, a pneumatic test as shown in equation (19) would be used by replacing 1.3 with 1.1.

Test fluid temperature must be maintained at least 30 F above minimum design metal temperature, but the fluid temperature must not be more than 120 F.

High Pressure Design for Pressure Vessel

The sections of ASME which cover the mechanical design of pressure vessels depending on the design pressure are as follows.

1. 15 psig to 3,000 psig: ASME VIII Division 1
2. 3,000 psig to 10.000 psig: ASME VIII Division 2
3. 10,000 psig to 100,000 psig or higher: ASME VIII Division 3

The choice of choosing ASME VIII Division 2 above 3,000 psig is typically the choice of the user. The use of ASME VIII Division 1 beyond 3,000 psig results in relatively higher plate thickness, and it is the combination equipment cost plus the more rigorous engineering cost involving finite element analysis decides the choice between Division I and Division II. Division 2 also handles pressure vessel design involved in cyclic load where fatigue analysis is required.

In the analysis, axial stress, shearing stress, the corresponding strains in the x, y, and z directions, modulus elasticity of material, and Poisson's are considered. The Poisson's ratio is the ratio of relative lateral contraction to the relative longitudinal extension, dimensionless.

Properties of Structural Materials Affecting High Pressure Design

Elasticity

This is a property by which a structural material deforms under externally applied force and returns to original shape if the deformation does not exceed a certain limit called elastic limit. It is assumed that bodies undergoing such limited deformation are perfectly elastic.

Homogeneity

The structural materials are not pure substances. It is assumed that the elastic body is homogeneous in terms of molecular structure which is theoretically wrong.

Isotropy

It is assumed that the elastic property is same in all directions of an elastic body, which is also theoretically wrong. The lack of isotropic property results in anisotropy which must be considered in high pressure design where thin plates are not usable. Anisotropy appears as a result of fabrication processes, such as rolling, whereby the elastic properties becomes different in different directions.

To appreciate why we need Division III, let is revisit the equation for calculating plate thickness for Division 1 formula Equation (8):

$$t = \frac{PR}{SE - 0.6P} \tag{8}$$

Now, at high design pressure, the quantity (SE-0.6P) approaches zero as pressure increases. In other words, beyond a certain pressure, the allowable stress for the material of construction must be increased beyond 0.6P.

The limitations of Equation (8) for high pressure design are usually overcome by using different equations using the theory of plasticity. For thin vessels, ASME assumes that failure occurs when the yield point is reached. This is a convenient criterion and is called the maximum-principal-stress theory.

Historical Background of ASME & API

In the beginning of American Institutions of Codes and Standards, the ASME and API were joint institutions. In fact, the 1951 API – ASME Code mentions about the layered vessel construction method of high-pressure vessel design. In later years, the layered vessel construction was deleted when the API-ASME marriage was dissolved. Then in 1979, ASME VIII-3 Code included the layered vessel construction.

The mathematics of thick plate tension analysis is dominantly algebraic although some differential equations are involved. Nevertheless, the methods are lot more complex and are not presented here.

There are three types of high-pressure vessels: Pre-stressed vessel, Layered vessel, and wire-wound vessel.

Pre-stressed Vessel (or Autofrettaged Vessel)

In thick vessels, the criterion is based on the energy of distortion theory. This theory states that “The inelastic action at any point in body under any combination of stresses begins when the strain energy of distortion per unit volume absorbed at the point is equal to the strain energy of distortion absorbed per unit volume in a bar stressed to the elastic limit under the state of uniaxial stress as occurs in simple tension test." This theory applies only to ductile materials. The equation that expresses the theory is given by:

$$W = \left(\frac{1+\mu}{6E}\right)\left[\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2\right] \tag{9}$$

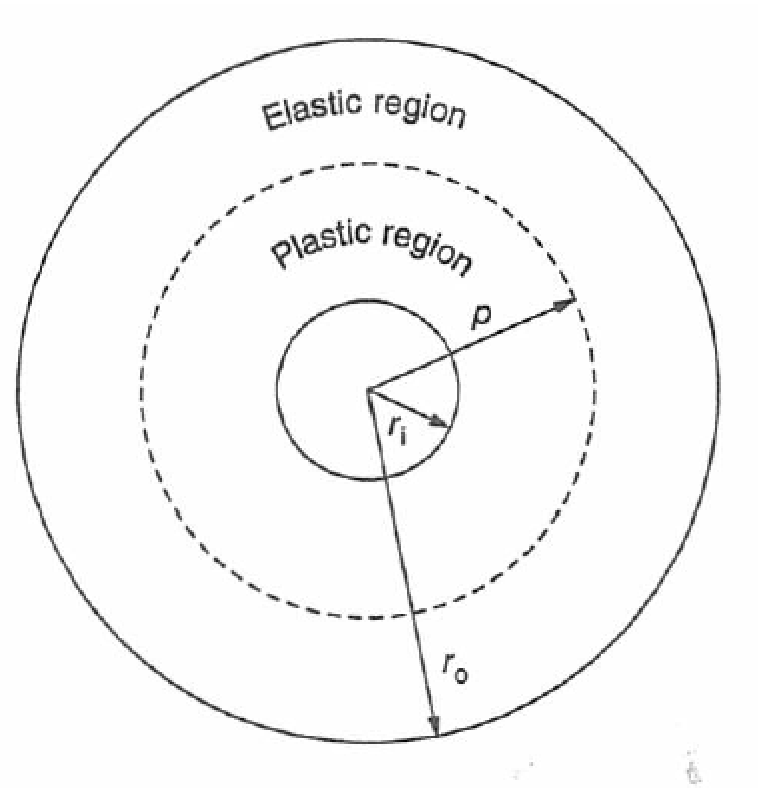
W = strain energy

μ = Poisson's ratio =ratio of lateral contraction to longitudinal strain

E = Modulus of elasticity

$\sigma_1, \sigma_2, \sigma_3 =$ Principal stresses: hoop, radial, and longitudinal

Figure 3: Cylindrical Shell with Elastic and Plastic Regions



$r_i =$ inside radius of shell

$\rho =$ Elastic-plastic interface radius

$r_o =$ outside radius of shell

The maximum pressure, P^*, at which yield is assumed to occur is given by:

$$P^* = \frac{\sigma_y}{\sqrt{3}} \frac{r_0^2 - r_i^2}{r_i^2} \left(\frac{r_i}{r_0} \right)^2 \tag{10}$$

Where: σ_y = yield stress of the material, psi

When the pressure in equation (10) is exceeded, the inner part of the shell becomes plastic while the outer part remains elastic as shown in Figure 3. The relationship between the design pressure P and the elastic-plastic interface is given by:

$$P = \frac{\sigma_y}{\sqrt{3}} \left(1 - \frac{\rho^2}{r_0^2} - 2 \ln \frac{r_1}{\rho} \right) \tag{11}$$

Layered Vessel

During World War II, the design of high-pressure vessels known as “layered vessels” was developed both in USA and Germany to manufacture synthetic chemicals which had favorable shift of equilibrium at high pressure. One such process was synthesis of ammonia for the production of explosives. Much improvement has been made since then and now the layered vessels are used in many high-pressure applications including heat-exchangers, reactors and autoclaves.

The layered vessels are made of a multitude of layers of vessels wrapped around an inner vessel with vent holes in each layer drilled radially starting from the inner layer to the outermost layer. The venting is to prevent pressure build-up in the layers.

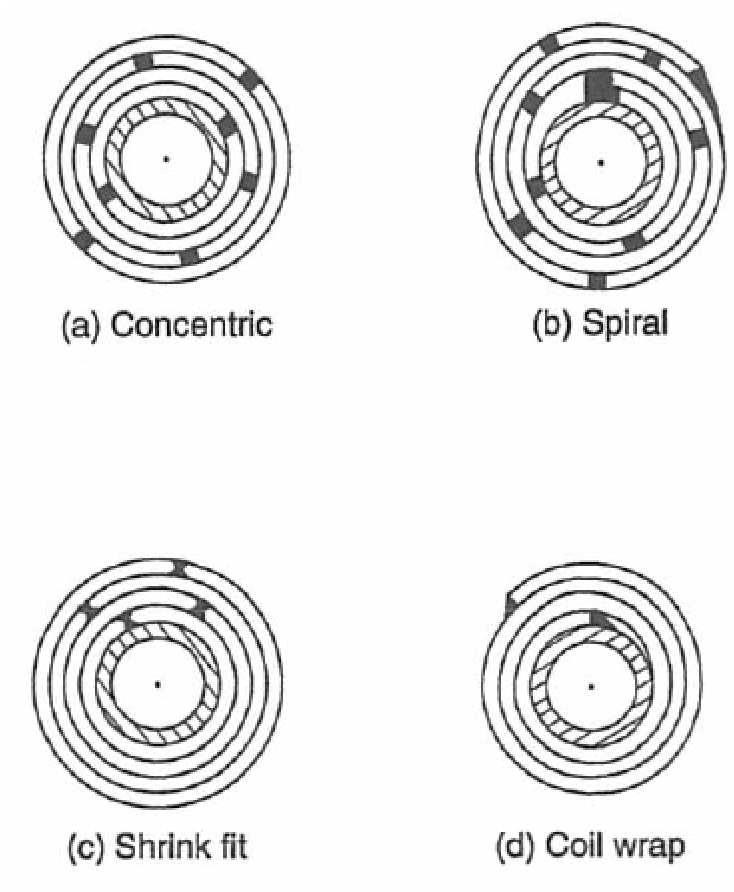
There are three methods of fabricating layered vessels.

1) The concentric or spiral wound method: Segments of vessel are welded in concentric (Figure 4(a)) or spiral fashion (Figure 4(b)) to form the required thickness.

2) The shrink-fit method: The layers of cylinders are individually fabricated and shrunk on each other to form the required total thickness (Figure 4(c)).

3) The coil-wrapped method: A spiral of cylinder is fabricated around the inner shell by a continuous sheet or strip wound around in a spiral or helical manner to form a cylinder (Figure 4(d)).

Figure 4: Various Types of Layered Cylindrical Vessels



Conclusion

The most important aspect of the article is to introduce to the ERS designers as to how MAWP is determined and limitations of the final value of the MAWP that will be stamped.

The possibility that the ERS designer will be involved in the ERS design for a very high-pressure vessel is remote, and therefore, the presentation made herein for the latter case is deliberately cursory.