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Mass Transfer-I

Pressure Vessels Design

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Pressure Vessels Design

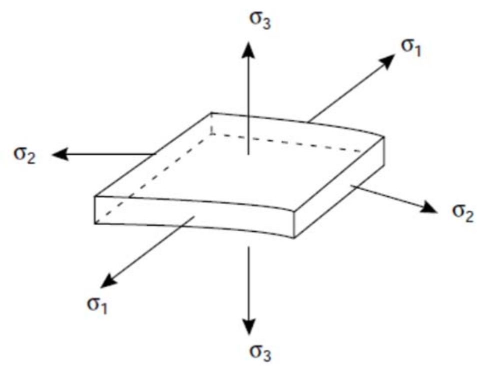
Introduction

The design of storage tanks, centrifuges and heat-exchanger tube, etc.

Classification of pressure vessels

For the purposes of design and analysis, pressure vessels are sub-divided into two classes depending on the ratio of the wall thickness to vessel diameter:

1. Thin-walled vessels, with a thickness ratio of less than 1 : 10
2. Thick-walled above this ratio.



Principal stresses in pressure-vessel wall

PRESSURE VESSEL CODES AND STANDARDS

- British Standard PD 5500
- European Standard EN 13445
- American Society of Mechanical Engineers code Section VIII (the ASME code)

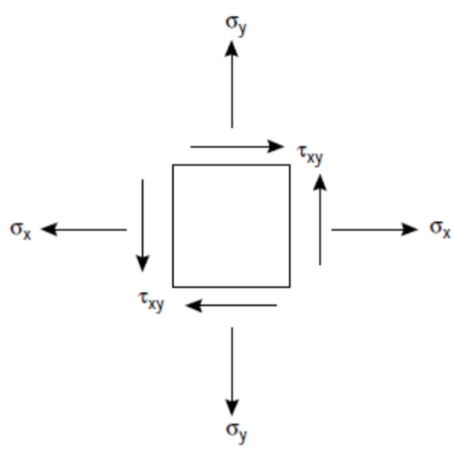
- In the European Union the design, manufacture and use of pressure systems is also covered by the Pressure Equipment Directive (Council Directive 97/23/EC) whose use became mandatory in May 2002.
- The current (2003) edition of PD 5500 covers vessels fabricated in carbon and alloy steels, and aluminium. The design of vessels constructed from reinforced plastics is covered by BS 4994. The ASME code covers steels, non-ferrous metals, and fibre-reinforced plastics.

FUNDAMENTAL PRINCIPLES AND EQUATIONS

Principal stresses

The state of stress at a point in a structural member under a complex system of loading is described by the magnitude and direction of the principal stresses. The principal stresses are the maximum values of the normal stresses at the point; which act on planes on which the shear stress is zero. In a two-dimensional stress system, Figure , the principal stresses at any point are related to the normal stresses in the ***x*** and ***y*** directions σ_x and σ_y and the shear stress ***xy*** at the point by the following equation:

$$\text{Principal stresses, } \sigma_1, \sigma_2 = \frac{1}{2}(\sigma_y + \sigma_x) \pm \frac{1}{2}\sqrt{[(\sigma_y - \sigma_x)^2 + 4\tau_{xy}^2]}$$



The maximum shear stress at the point is equal to half the algebraic difference between the principal stresses:

$$\text{Maximum shear stress } = \frac{1}{2}(\sigma_1 - \sigma_2)$$

Compressive stresses are conventionally taken as negative; tensile as positive.

Theories of failure

The failure of a simple structural element under unidirectional stress (tensile or compressive) is easy to relate to the tensile strength of the material, as determined in a standard tensile test, but for components subjected to combined stresses (normal and shear stress) the position is not so simple, and several theories of failure have been proposed. The three theories most commonly used are described below:

1. *Maximum principal stress theory*: which postulates that a member will fail when one of the principal stresses reaches the failure value in simple tension, The failure point in a simple tension is taken as the yield-point stress, or the tensile strength of the material, divided by a suitable factor of safety.

2. *Maximum shear stress theory*: which postulates that failure will occur in a complex stress system when the maximum shear stress reaches the value of the shear stress at failure in simple tension.

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For a system of combined stresses there are three shear stresses maxima:

$$\tau_1 = \frac{\sigma_1 - \sigma_2}{2}$$

$$\tau_2 = \frac{\sigma_2 - \sigma_3}{2}$$

$$\tau_3 = \frac{\sigma_3 - \sigma_1}{2}$$

In the tensile test, $$\tau_e = \frac{\sigma'_e}{2}$$

The maximum shear stress will depend on the sign of the principal stresses as well as their magnitude, and in a two-dimensional stress system, such as that in the wall of a thin-walled pressure vessel, the maximum value of the shear stress may be that given by putting $\sigma_3 = 0$ in the above equations. The maximum shear stress theory is often called Tresca's, or Guest's, theory.

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3. Maximum strain energy theory: which postulates that failure will occur in a complex stress system when the total strain energy per unit volume reaches the value at which failure occurs in simple tension. The maximum shear-stress theory has been found to be suitable for predicting the failure of ductile materials under complex loading and is the criterion normally used in the pressure-vessel design.

Elastic stability

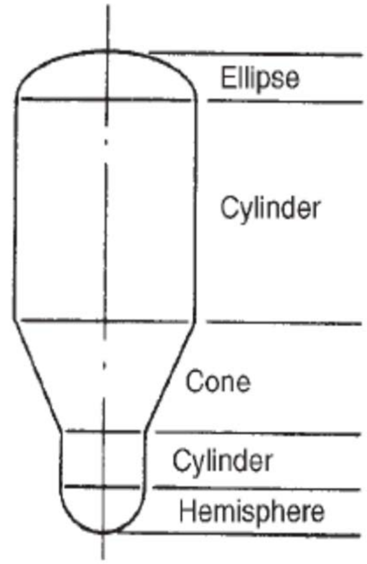
- Under certain loading conditions failure of a structure can occur not through gross yielding or plastic failure, but by buckling, or wrinkling. Buckling results in a gross and sudden change of shape of the structure; unlike failure by plastic yielding, where the structure retains the same basic shape. This mode of failure will occur when the structure is
- not elastically stable: when it lacks sufficient stiffness, or rigidity, to withstand the load. The stiffness of a structural member is dependent not on the basic strength of the material but on its elastic properties (E and v) and the cross-sectional shape of the member.
- The classic example of failure due to elastic instability is the buckling of tall thin columns (struts), which is described in any elementary text on the "Strength of Materials".
- For a structure that is likely to fail by buckling there will be a certain critical value of load below which the structure is stable; if this value is exceeded catastrophic failure through buckling can occur.
- The walls of pressure vessels are usually relatively thin compared with the other dimensions and can fail by buckling under compressive loads.
- Elastic buckling is the decisive criterion in the design of thin-walled vessels under external pressure.

Membrane stresses in shells of revolution

A shell of revolution is the form swept out by a line or curve rotated about an axis. (A solid of revolution is formed by rotating an area about an axis.) Most process vessels are made up from shells of revolution: cylindrical and conical sections; and hemispherical, ellipsoidal and torispherical heads (Figure)

The walls of thin vessels can be considered to be "membranes"; supporting loads without significant bending or shear stresses; similar to the walls of a balloon.

The analysis of the membrane stresses induced in shells of revolution by internal pressure gives a basis for determining the minimum wall thickness required for vessel shells. The actual thickness required will also depend on the stresses arising from the other loads to which the vessel is subjected.



Typical vessel shapes

Consider the shell of revolution of general shape shown in Figure, under a loading that is rotationally symmetric; that is, the load per unit area (pressure) on the shell is constant round the circumference, but not necessarily the same from top to bottom.

Let P = pressure,
t = thickness of shell,
σ_1 = the meridional (longitudinal) stress, the stress acting along a meridian,
σ_2 = the circumferential or tangential stress, the stress acting along parallel circles (often called the hoop stress),
r_1 = the meridional radius of curvature,
r_2 = circumferential radius of curvature.

Note: the vessel has a double curvature; the values of r_1 and r_2 are determined by the shape.

Consider the forces acting on the element defined by the points a, b, c, d. Then the normal component (component acting at right angles to the surface) of the pressure force on the element

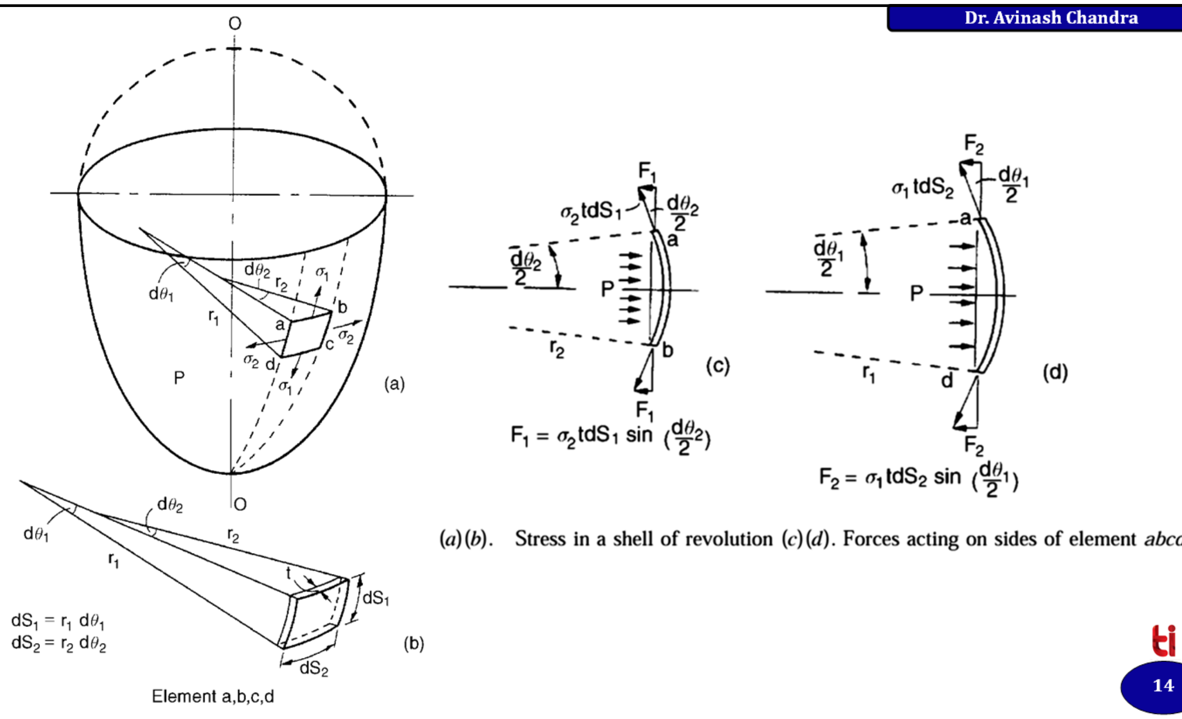
$$= P\left[2r_1 \sin\left(\frac{d\theta_1}{2}\right)\right]\left[2r_2 \sin\left(\frac{d\theta_2}{2}\right)\right]$$

This force is resisted by the normal component of the forces associated with the membrane stresses in the walls of the vessel (given by, force = stress × area)

$$= 2\sigma_2 t \mathrm{d}S_1 \sin\left(\frac{\mathrm{d}\theta_2}{2}\right) + 2\sigma_1 t\, \mathrm{d}S_2 \sin\left(\frac{\mathrm{d}\theta_1}{2}\right)$$

Equating these forces and simplifying, and noting that in the limit $\mathrm{d}\theta/2 \to \mathrm{d}S/2r$, and $\sin \mathrm{d}\theta \to \mathrm{d}\theta$, gives:

$$\frac{\sigma_1}{r_1} + \frac{\sigma_2}{r_2} = \frac{P}{t} \tag{13.5}$$



(*a*)(*b*). Stress in a shell of revolution (*c*)(*d*). Forces acting on sides of element *abcd*

An expression for the meridional stress σ_1 can be obtained by considering the equilibrium of the forces acting about any circumferential line, Figure 13.5. The vertical component of the pressure force

$$= P\pi(r_2 \sin\theta)^2$$

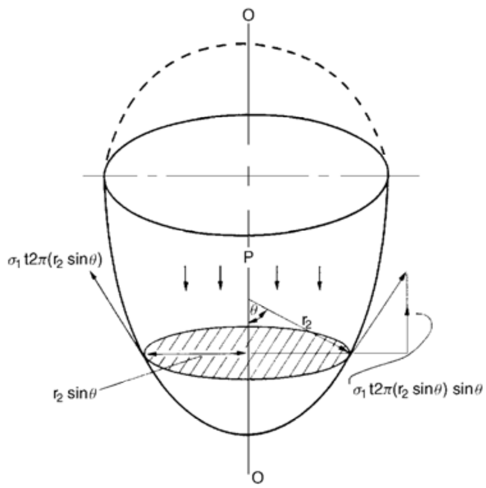


Figure 13.5. Meridional stress, force acting at a horizontal plane

This is balanced by the vertical component of the force due to the meridional stress acting in the ring of the wall of the vessel

$$= 2\sigma_1 t\pi(r_2 \sin\theta)\sin\theta$$

Equating these forces gives:

$$\boxed{\sigma_1 = \frac{Pr_2}{2t}} \tag{13.6}$$

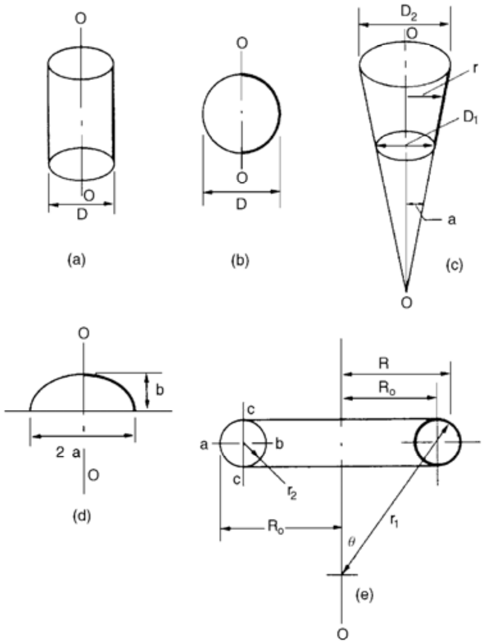


Figure 13.6. Shells of revolution

Cylinder (Figure 13.6a)

A cylinder is swept out by the rotation of a line parallel to the axis of revolution, so:

$$r_1 = \infty$$

$$r_2 = \frac{D}{2}$$

where D is the cylinder diameter.

Substitution in equations 13.5 and 13.6 gives:

$$\sigma_2 = \frac{PD}{2t}$$

$$\sigma_1 = \frac{PD}{4t}$$

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Sphere (Figure 13.6b)

$$r_1 = r_2 = \frac{D}{2}$$

hence:

$$\sigma_1 = \sigma_2 = \frac{PD}{4t}$$

Cone (Figure 13.6c)

A cone is swept out by a straight line inclined at an angle α to the axis.

$$r_1 = \infty$$

$$r_2 = \frac{r}{\cos\alpha}$$

substitution in equations 13.5 and 13.6 gives:

$$\sigma_2 = \frac{Pr}{t\cos\alpha}$$

$$\sigma_1 = \frac{Pr}{2t\cos\alpha}$$

The maximum values will occur at $r = D_2/2$.

Ellipsoid (Figure 13.6d)

For an ellipse with major axis $2a$ and minor axis $2b$, it can be shown that (see any standard geometry text):

$$r_1 = \frac{r_2^3 b^2}{a^4}$$

From equations 13.5 and 13.6

$$\sigma_1 = \frac{Pr_2}{2t} \qquad \text{(equation 13.6)}$$

$$\sigma_2 = \frac{P}{t}\left[r_2 - \frac{r_2^2}{2r_1}\right] \qquad (13.12)$$

At the crown (top)

$$r_1 = r_2 = \frac{a^2}{b}$$

$$\sigma_1 = \sigma_2 = \frac{Pa^2}{2tb} \qquad (13.13)$$

At the equator (bottom) $r_2 = a$, so $r_1 = b^2/a$

so

$$\sigma_1 = \frac{Pa}{2t} \qquad (13.13)$$

$$\sigma_2 = \frac{P}{t}\left[a - \frac{a^2}{2b^2/a}\right] = \frac{Pa}{t}\left[1 - \frac{1}{2}\frac{a^2}{b^2}\right] \qquad (13.14)$$

It should be noted that if $\frac{1}{2}(a/b)^2 > 1$, σ_2 will be negative (compressive) and the shell could fail by buckling. This consideration places a limit on the practical proportions of ellipsoidal heads.

Torus (Figure 13.6e)

A torus is formed by rotating a circle, radius r_2, about an axis.

$$\sigma_1 = \frac{Pr_2}{2t} \qquad \text{(equation 13.6)}$$

$$r_1 = \frac{R}{\sin\theta} = \frac{R_0 + r_2\sin\theta}{\sin\theta}$$

and

$$\sigma_2 = \frac{Pr_2}{t}\left[1 - \frac{r_2\sin\theta}{2(R_0 + r_2\sin\theta)}\right] \qquad (13.15)$$

On the centre line of the torus, point c, $\theta = 0$ and

$$\sigma_2 = \frac{Pr_2}{t} \qquad (13.16)$$

At the outer edge, point a, $\theta = \pi/2$, $\sin\theta = 1$ and

$$\sigma_2 = \frac{Pr_2}{2t}\left[\frac{2R_0 + r_2}{R_0 + r_2}\right] \qquad (13.17)$$

the minimum value.

At the inner edge, point b, $\theta = 3\pi/2$, $\sin\theta = -1$ and

$$\sigma_2 = \frac{Pr_2}{2t}\left[\frac{2R_0 - r_2}{R_0 - r_2}\right] \qquad (13.18)$$

the maximum value.

So σ_2 varies from a maximum at the inner edge to a minimum at the outer edge.

Torispherical heads

A torispherical shape, which is often used as the end closure of cylindrical vessels, is formed from part of a torus and part of a sphere, Figure 13.7. The shape is close to that of an ellipse but is easier and cheaper to fabricate.

In Figure 13.7 R_k is the knuckle radius (the radius of the torus) and R_c the crown radius (the radius of the sphere). For the spherical portion:

$$\sigma_1 = \sigma_2 = \frac{PR_c}{2t} \tag{13.19}$$

For the torus:

$$\sigma_1 = \frac{PR_k}{2t} \tag{13.20}$$

σ_2 depends on the location, and is a function of R_c and R_k; it can be calculated from equations 13.15 and 13.9.

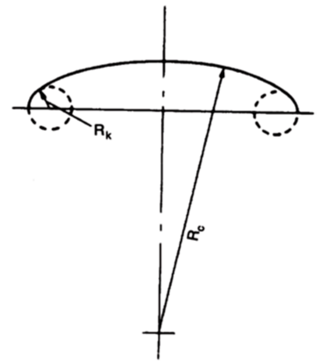

Figure 13.7. Torisphere

The ratio of the knuckle radius to crown radius should be made not less than 6/100 to avoid buckling. The stress will be higher in the torus section than the spherical section.

13.3.5. Flat plates

Flat plates are used as covers for manholes, as blind flanges, and for the ends of small diameter and low pressure vessels.

For a uniformly loaded circular plate supported at its edges, the slope ϕ at any radius x is given by:

$$\phi = -\frac{\mathrm{d}w}{\mathrm{d}x} = -\frac{1}{\mathbf{D}}\frac{Px^3}{16} + \frac{C_1 x}{2} + \frac{C_2}{x} \tag{13.21}$$

(The derivation of this equation can be found in any text on the strength of materials.)

Integration gives the deflection w:

$$w = \frac{Px^4}{64\mathbf{D}} - C_1\frac{x^2}{4} - C_2 \ln x + C_3 \tag{13.22}$$

where P = intensity of loading (pressure),
x = radial distance to point of interest,
$\mathbf{D}$ = flexual rigidity of plate = $(Et^3)/(12(1-\nu^2))$,
t = plate thickness,
ν = Poisson's ratio for the material,
E = modulus of elasticity of the material (Young's modulus).

C_1, C_2, C_3 are constants of integration which can be obtained from the boundary conditions at the edge of the plate.

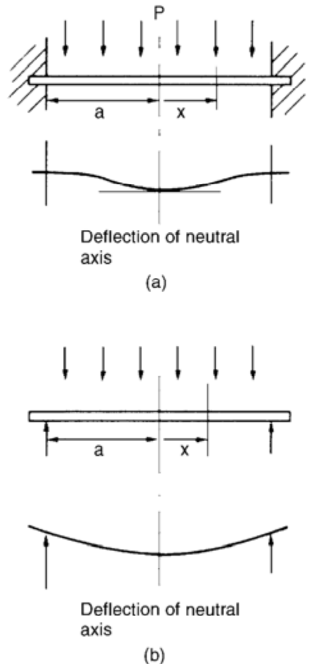

Figure 13.8. Flat circular plates (*a*) Clamped edges (*b*) Simply supported

Two limiting situations are possible:

1. When the edge of the plate is rigidly clamped, not free to rotate; which corresponds to a heavy flange, or a strong joint.

2. When the edge is free to rotate (simply supported); corresponding to a weak joint, or light flange.

1. Clamped edges (Figure 13.8a)

The edge (boundary) conditions are:

$$\phi = 0 \text{ at } x = 0$$
$$\phi = 0 \text{ at } x = a$$
$$w = 0 \text{ at } x = a$$

where a is the radius of the plate.
Which gives:

$$C_2 = 0, \quad C_1 = \frac{Pa^2}{8\mathbf{D}}, \quad \text{and} \quad C_3 = \frac{Pa^4}{64\mathbf{D}}$$

hence

$$\phi = \frac{Px}{16\mathbf{D}}(a^2 - x^2) \tag{13.23}$$

and

$$w = \frac{P}{64\mathbf{D}}(x^2 - a^2)^2 \tag{13.24}$$

The maximum deflection will occur at the centre of the plate at $x = 0$

$$\hat{w} = \frac{Pa^4}{64\mathbf{D}} \tag{13.25}$$

The bending moments per unit length due to the pressure load are related to the slope and deflection by:

$$M_1 = \mathbf{D}\left[\frac{\mathrm{d}\phi}{\mathrm{d}x} + \nu\frac{\phi}{x}\right] \tag{13.26}$$

$$M_2 = \mathbf{D}\left[\frac{\phi}{x} + \nu\frac{\mathrm{d}\phi}{\mathrm{d}x}\right] \tag{13.27}$$

Where M_1 is the moment acting along cylindrical sections, and M_2 that acting along diametrical sections.

Substituting for ϕ and $\mathrm{d}\phi/\mathrm{dx}$ in equations 13.26 and 13.27 gives:

$$M_1 = \frac{P}{16}[a^2(1+\nu) - x^2(3+\nu)] \tag{13.28}$$

$$M_2 = \frac{P}{16}[a^2(1+\nu) - x^2(1+3\nu)] \tag{13.29}$$

The maximum values will occur at the edge of the plate, $x = a$.

$$\hat{M}_1 = -\frac{Pa^2}{8}, \quad \hat{M}_2 = -\nu\frac{Pa^2}{8}$$

The bending stress is given by:

$$\sigma_b = \frac{M_1}{I'} \times \frac{t}{2}$$

where I' = second moment of area per unit length $= t^3/12$, hence

$$\hat{\sigma}_b = \frac{6\hat{M}_1}{t^2} = \tfrac{3}{4}\frac{Pa^2}{t^2} \tag{13.30}$$

2. Simply supported plate (Figure 13.8b)

The edge (boundary) conditions are:

$$\phi = 0 \text{ at } x = 0$$

$$w = 0 \text{ at } x = a$$

$$M_1 = 0 \text{ at } x = a \text{ (free to rotate)}$$

which gives C_2 and $C_3 = 0$.
Hence

$$\phi = -\frac{1}{\mathbf{D}}\frac{Px^3}{16} + \frac{C_1 x}{2}$$

and

$$\frac{d\phi}{dx} = -\frac{1}{\mathbf{D}}\left[\frac{3Px^2}{16}\right] + \frac{C_1}{2}$$

Substituting these values in equation 13.26, and equating to zero at $x = a$, gives:

$$C_1 = \frac{Pa^2}{8\mathbf{D}}\frac{(3+\nu)}{(1+\nu)}$$

and hence

$$M_1 = \frac{P}{16}(3+\nu)(a^2 - x^2) \qquad (13.31)$$

The maximum bending moment will occur at the centre, where $M_1 = M_2$

so $$\hat{M}_1 = \hat{M}_2 = \frac{P(3+\nu)a^2}{16} \qquad (13.32)$$

and $$\hat{\sigma}_b = \frac{6\hat{M}_1}{t^2} = \tfrac{3}{8}(3+\nu)\frac{Pa^2}{t^2} \qquad (13.33)$$

General equation for flat plates

A general equation for the thickness of a flat plate required to resist a given pressure load can be written in the form:

$$t = CD\sqrt{\frac{P}{f}} \tag{13.34}$$

where f = the maximum allowable stress (the design stress),
D = the effective plate diameter,
C = a constant, which depends on the edge support.

The limiting value of C can be obtained from equations 13.30 and 13.33. Taking Poisson's ratio as 0.3, a typical value for steels, then if the edge can be taken as completely rigid $C = 0.43$, and if it is essentially free to rotate $C = 0.56$.

13.3.6. Dilation of vessels

Under internal pressure a vessel will expand slightly. The radial growth can be calculated from the elastic strain in the radial direction. The principal strains in a two-dimensional system are related to the principal stresses by:

$$\varepsilon_1 = \frac{1}{E}(\sigma_1 - \nu\sigma_2) \tag{13.35}$$

$$\varepsilon_2 = \frac{1}{E}(\sigma_2 - \nu\sigma_1) \tag{13.36}$$

The radial (diametrical strain) will be the same as the circumferential strain ε_2. For any shell of revolution the dilation can be found by substituting the appropriate expressions for the circumferential and meridional stresses in equation 13.36.

The diametrical dilation $\Delta = D\varepsilon_1$.

For a cylinder

$$\sigma_1 = \frac{PD}{4t}$$

$$\sigma_2 = \frac{PD}{2t}$$

substitution in equation 13.36 gives:

$$\Delta_c = \frac{PD^2}{4tE}(2 - \nu) \qquad (13.37)$$

For a sphere (or hemisphere)

$$\sigma_1 = \sigma_2 = \frac{PD}{4t}$$

and

$$\Delta_s = \frac{PD^2}{4tE}(1 - \nu) \qquad (13.38)$$

So for a cylinder closed by a hemispherical head of the same thickness the difference in dilation of the two sections, if they were free to expand separately, would be:

$$\Delta_c - \Delta_s = \frac{PD^2}{4tE}$$

Design pressure

A vessel must be designed to withstand the maximum pressure to which it is likely to be subjected in operation.

For vessels under internal pressure, the design pressure is normally taken as the pressure at which the relief device is set. This will normally be 5 to 10 per cent above the normal working pressure, to avoid spurious operation during minor process upsets. When deciding the design pressure, the hydrostatic pressure in the base of the column should be added to the operating pressure, if significant.

Vessels subject to external pressure should be designed to resist the maximum differential pressure that is likely to occur in service. Vessels likely to be subjected to vacuum should be designed for a full negative pressure of 1 bar, unless fitted with an effective, and reliable, vacuum breaker.

Design temperature

The strength of metals decreases with increasing temperature so the maximum allowable design stress will depend on the material temperature. The design temperature at which the design stress is evaluated should be taken as the maximum working temperature of the material, with due allowance for any uncertainty involved in predicting vessel wall temperatures.

Materials

Pressure vessels are constructed from plain carbon steels, low and high alloy steels, other alloys, clad plate, and reinforced plastics.

Selection of a suitable material must take into account the suitability of the material for fabrication (particularly welding) as well as the compatibility of the material with the process environment.

The pressure vessel design codes and standards include lists of acceptable materials; in accordance with the appropriate material standards.

Design stress (nominal design strength)

For design purposes it is necessary to decide a value for the maximum allowable stress (nominal design strength) that can be accepted in the material of construction.

This is determined by applying a suitable "design stress factor" (factor of safety) to the maximum stress that the material could be expected to withstand without failure under standard test conditions. The design stress factor allows for any uncertainty in the design methods, the loading, the quality of the materials, and the workmanship.

For materials not subject to high temperatures the design stress is based on the yield stress (or proof stress), or the tensile strength (ultimate tensile stress) of the material at the design temperature.

For materials subject to conditions at which the creep is likely to be a consideration, the design stress is based on the creep characteristics of the material: the average stress to produce rupture after 10^5 hours, or the average stress to produce a 1 per cent strain after 10^5 hours, at the design temperature. Typical design stress factors for pressure components are shown in Table 13.1.

Table 13.1. Design stress factors

Property	Material		
	Carbon Carbon-manganese, low alloy steels	Austenitic stainless steels	Non-ferrous metals
Minimum yield stress or 0.2 per cent proof stress, at the design temperature	1.5	1.5	1.5
Minimum tensile strength, at room temperature	2.35	2.5	4.0
Mean stress to produce rupture at 10^5 h at the design temperature	1.5	1.5	1.0

Welded joint efficiency, and construction categories

The strength of a welded joint will depend on the type of joint and the quality of the welding.

The soundness of welds is checked by visual inspection and by non-destructive testing (radiography). The possible lower strength of a welded joint compared with the virgin plate is usually allowed for in design by multiplying the allowable design stress for the material by a "welded joint factor" J. The value of the joint factor used in design will depend on the type of joint and amount of radiography required by the design code. Typical values are shown in Table 13.3. Taking the factor as 1.0 implies that the joint is equally as strong as the virgin plate; this is achieved by radiographing the complete weld length, and cutting out and remaking any defects. The use of lower joint factors in design, though saving costs on radiography, will result in a thicker, heavier, vessel, and the designer must balance any cost savings on inspection and fabrication against the increased cost of materials.

Table 13.3. Maximum allowable joint efficiency

Type of joint	Degree of radiography		
	100 per cent	spot	none
Double-welded butt or equivalent	1.0	0.85	0.7
Single-weld butt joint with bonding strips	0.9	0.80	0.65

The standard specifies three construction categories:

Category 1: the highest class, requires 100 per cent non-destructive testing (NDT) of the welds; and allows the use of all materials covered by the standard, with no restriction on the plate thickness.

Category 2: requires less non-destructive testing but places some limitations on the materials which can be used and the maximum plate thickness.

Category 3: the lowest class, requires only visual inspection of the welds, but is restricted to carbon and carbon-manganese steels, and austenitic stainless steel; and limits are placed on the plate thickness and the nominal design stress. For carbon and carbon-manganese steels the plate thickness is restricted to less than 13 mm and the design stress is about half that allowed for categories 1 and 2. For stainless steel the thickness is restricted to less than 25 mm and the allowable design stress is around 80 per cent of that for the other categories.

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Corrosion allowance

The "corrosion allowance" is the additional thickness of metal added to allow for material lost by corrosion and erosion, or scaling. The allowance to be used should be agreed between the customer and manufacturer. Corrosion is a complex phenomenon, and it is not possible to give specific rules for the estimation of the corrosion allowance required for all circumstances. The allowance should be based on experience with the material of construction under similar service conditions to those for the proposed design.

For carbon and low-alloy steels, where severe corrosion is not expected, a minimum allowance of 2.0 mm should be used; where more severe conditions are anticipated this should be increased to 4.0 mm. Most design codes and standards specify a minimum allowance of 1.0 mm.

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Design loads

Major loads

1. Design pressure: including any significant static head of liquid.
2. Maximum weight of the vessel and contents, under operating conditions.
3. Maximum weight of the vessel and contents under the hydraulic test conditions.
4. Wind loads.
5. Earthquake (seismic) loads.
6. Loads supported by, or reacting on, the vessel.

Subsidiary loads

1. Local stresses caused by supports, internal structures and connecting pipes.
2. Shock loads caused by water hammer, or by surging of the vessel contents.
3. Bending moments caused by eccentricity of the centre of the working pressure relative to the neutral axis of the vessel.
4. Stresses due to temperature differences and differences in the coefficient expansion of materials.
5. Loads caused by fluctuations in temperature and pressure.

A vessel will not be subject to all these loads simultaneously. The designer must determine what combination of possible loads gives the worst situation, and design for that loading condition.

Minimum practical wall thickness

There will be a minimum wall thickness required to ensure that any vessel is sufficiently rigid to withstand its own weight, and any incidental loads. As a general guide the wall thickness of any vessel should not be less than the values given below; the values include a corrosion allowance of 2 mm:

Vessel diameter (m)	Minimum thickness (mm)
1	5
1 to 2	7
2 to 2.5	9
2.5 to 3.0	10
3.0 to 3.5	12

References

ETH

Mass Transfer

Theories for Mass Transfer Coefficients

Lecture 9, 15.11.2017, Dr. K. Wegner

- Lecture notes/ppt of Dr. Yahya Banat (**ybanat@qu.edu.qa**)