Design of Pressure vessel Mechanical Aspect

Mechanical Design of Process Equipment

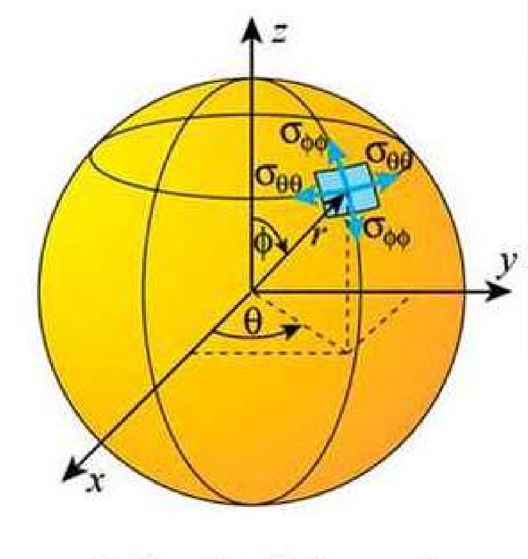
- This chapter covers those aspects of the **mechanical design** of **chemical plant** that are of particular interest to **chemical engineers**.
- The design of storage tanks, kettles, distillation columns, centrifuges, evaporators, autoclaves and heat-exchanger are part of process vessel.
- The chemical engineer will not usually be called on to undertake the **detailed mechanical design of a pressure vessel.**
- Vessel design is a specialized subject, and will be carried out by **mechanical engineers** who are conversant with the current design codes and practices, and methods of stress analysis.
- However, the chemical engineer will be responsible for developing and specifying the basic design information for a particular vessel, and needs to have a general appreciation of pressure vessel design to work effectively with the specialist designer.

- The basic data needed by the specialist designer will be:
    1. Vessel function.
    2. Process materials and services.
    3. Operating and design temperature and pressure.
    4. Materials of construction.
    5. Vessel dimensions and orientation.
    6. Type of vessel heads to be used.
    7. Openings and connections required.
    8. Specification of heating and cooling jackets or coils.
    9. Type of agitator.
    10. Specification of internal fittings.

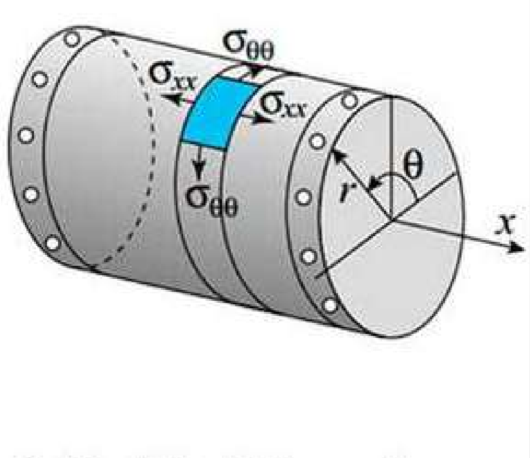
Classification of pressure vessels

- Pressure vessels are compressed gas storage tanks designed to hold gases or liquids at a pressure substantially different from the ambient pressure.
- Pressure vessels can be **any shape**, but shapes made of sections of **spheres** and **cylinders** are usually employed.
- A common design is a cylinder with end caps called **heads.**
- **Head shapes** are frequently **hemispherical.**
- Cracked or damaged vessels can result in leakage or rupture failures.
- ***Potential health and safety hazards*** of leaking vessels include poisonings, suffocations, fires, and explosion hazards.
- There is no strict definition of what constitutes a pressure vessel, but it is generally accepted that any closed vessel over 150 mm diameter subject to a pressure difference of more than 0.5 bar should be designed as a pressure vessel.

- For the purposes of **design and analysis**, pressure vessels are subdivided into **two classes** depending on the **ratio of the wall thickness** to **vessel diameter**: **thin-walled vessels**, with a thickness ratio of less than 1 : 10; and thick-walled above this ratio.
- **Rupture failures** can be much more disastrous and can cause considerable damage to **life** and **property.**
- The safe design, installation, operation, and maintenance of pressure vessels are in accordance with codes such as American Society of Mechanical Engineers (ASME) boiler and pressure vessel code.
- Pressure vessels are widely used in industry for storage and transportation of liquids and gases when configured as tanks.
- For most applications, pressure vessels are either spherical or cylindrical.



Spherical Vessel

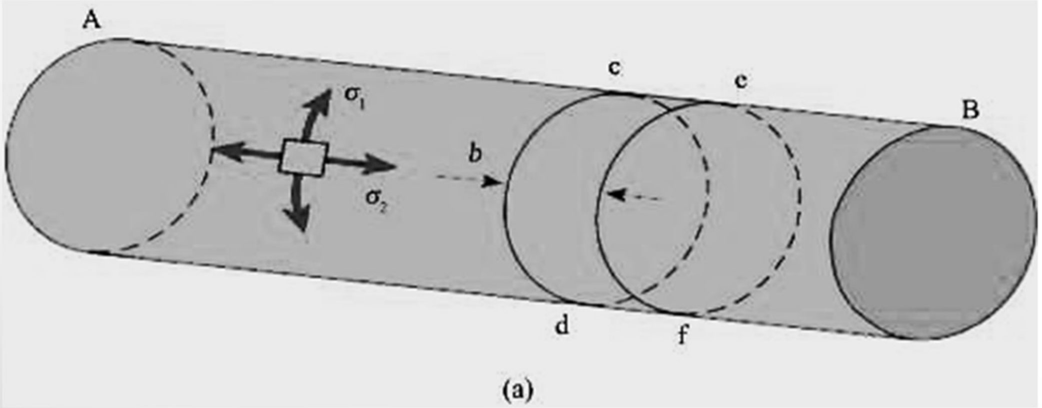


Cylindrical Vessel

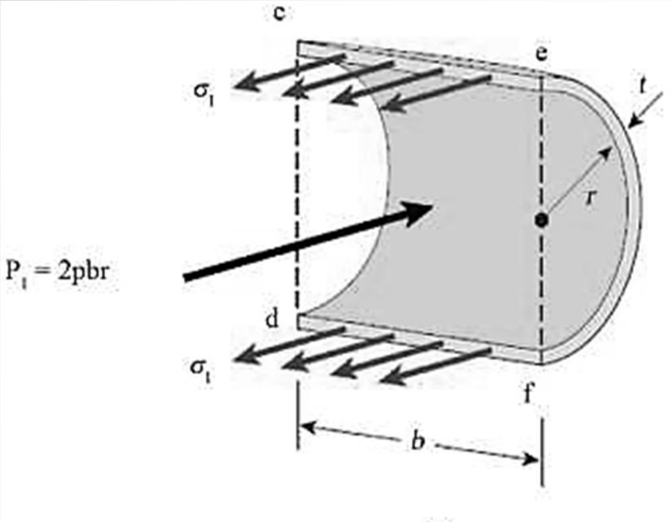
There is no strict definition of what constitutes a pressure vessel, but it is generally accepted that any closed vessel over 150 mm diameter subject to a pressure difference of more than 0.5 bar should be designed as a pressure vessel.

Thin-Walled-Cylindrical Pressure Vessel

A thin-walled circular tank AB subjected to internal pressure shown in **Figure.** A stress element with its faces parallel and perpendicular to the axis of the tank is shown on the wall of the tank. The normal stresses σ_1 and σ_2 acting on the side faces of this element. These stresses are known as Principal Stresses.



(a)



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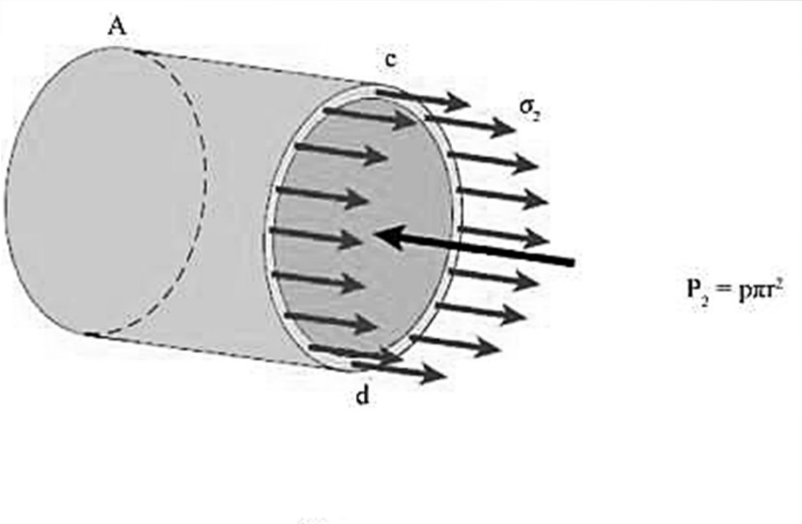


(c)

Because of their directions, the Stress σ_1 is called the circumferential stress or the hoop stress, and the stress σ_2 is called the longitudinal stress or the axial stress.

Pressure Vessel Codes And Standards

In all the major industrialized countries **the design** and **fabrication** of **thin-walled pressure vessels** is covered by national standards and codes of practice. Bureau of Indian Standards given a code for process vessel IS:2825.

The codes and standards cover design, materials of construction, fabrication (manufacture and workmanship), and inspection and testing.

They form a basis of agreement between the manufacturer and customer, and the customer’s insurance company.

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.

- Where national codes are not available, the British, European or American codes would be used.

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.

Design temperature

- The strength of metals decreases with increasing temperature, 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.
- 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 (Al, Cu, Ni), 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 after10^5 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.

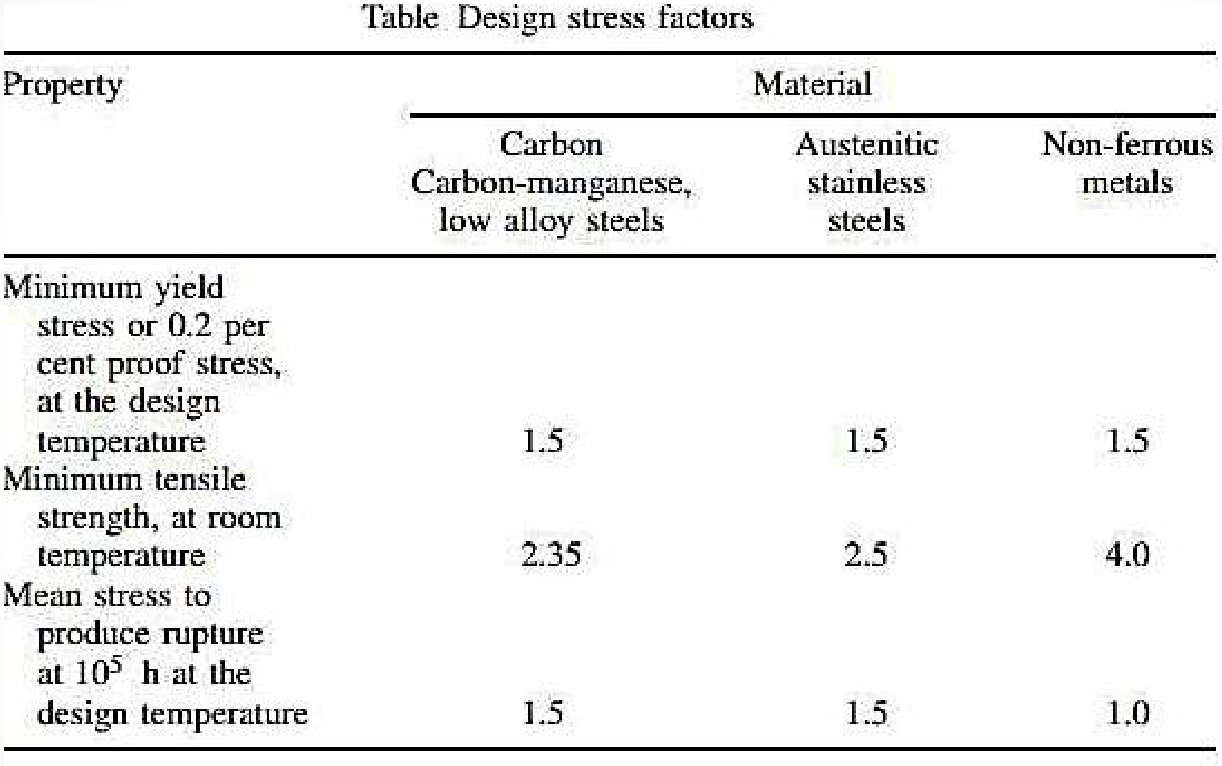


Table Design stress factors

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

Typical design stress values for some common materials are shown in Table.

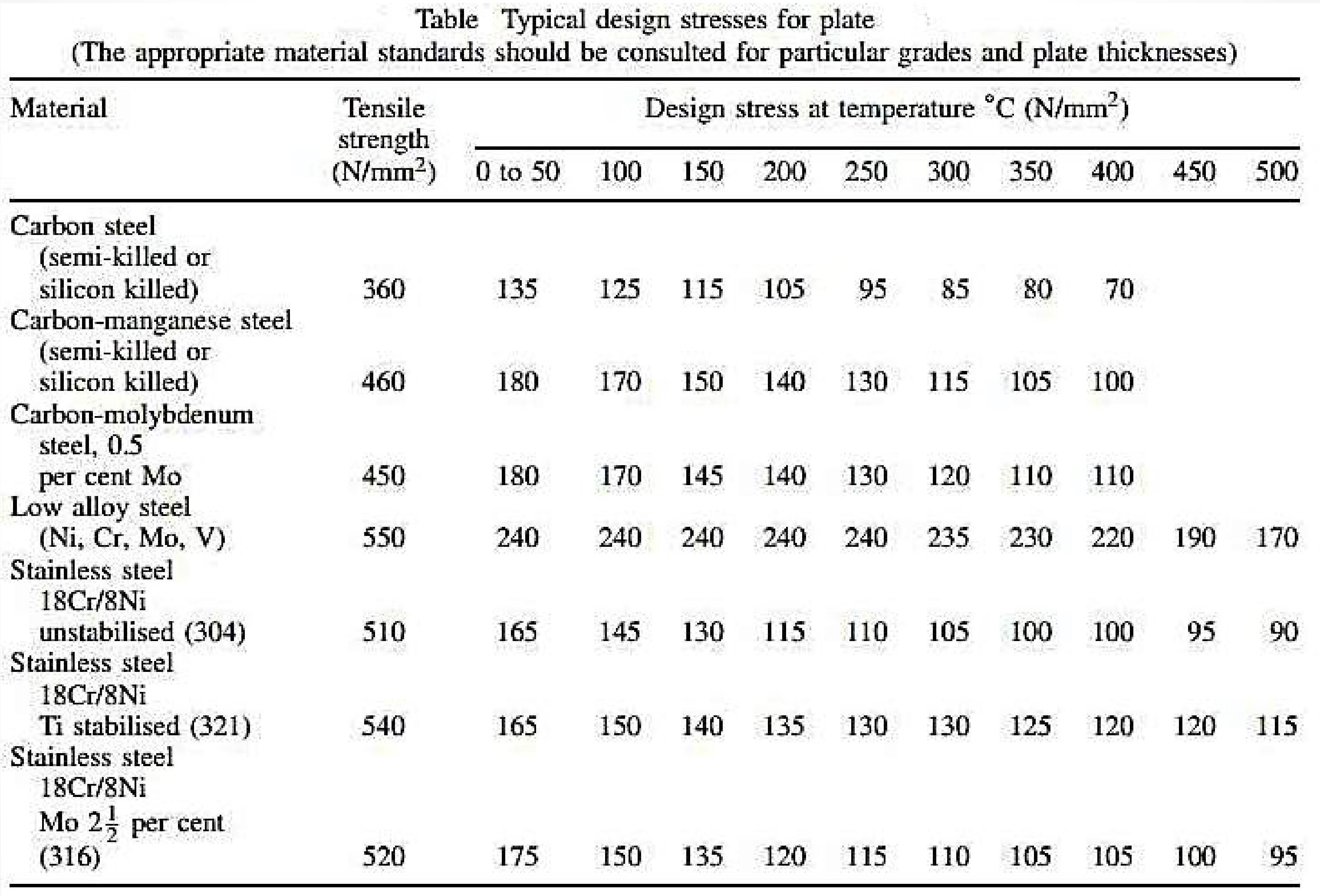


Table Typical design stresses for plate
(The appropriate material standards should be consulted for particular grades and plate thicknesses)

Material	Tensile strength (N/mm^2)	Design stress at temperature °C (N/mm^2)									
		0 to 50	100	150	200	250	300	350	400	450	500
Carbon steel (semi-killed or silicon killed)	360	135	125	115	105	95	85	80	70		
Carbon-manganese steel (semi-killed or silicon killed)	460	180	170	150	140	130	115	105	100		
Carbon-molybdenum steel, 0.5 per cent Mo	450	180	170	145	140	130	120	110	110		
Low alloy steel (Ni, Cr, Mo, V)	550	240	240	240	240	240	235	230	220	190	170
Stainless steel 18Cr/8Ni unstabilised (304)	510	165	145	130	115	110	105	100	100	95	90
Stainless steel 18Cr/8Ni Ti stabilised (321)	540	165	150	140	135	130	130	125	120	120	115
Stainless steel 18Cr/8Ni Mo $2\frac{1}{2}$ per cent (316)	520	175	150	135	120	115	110	105	105	100	95

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.

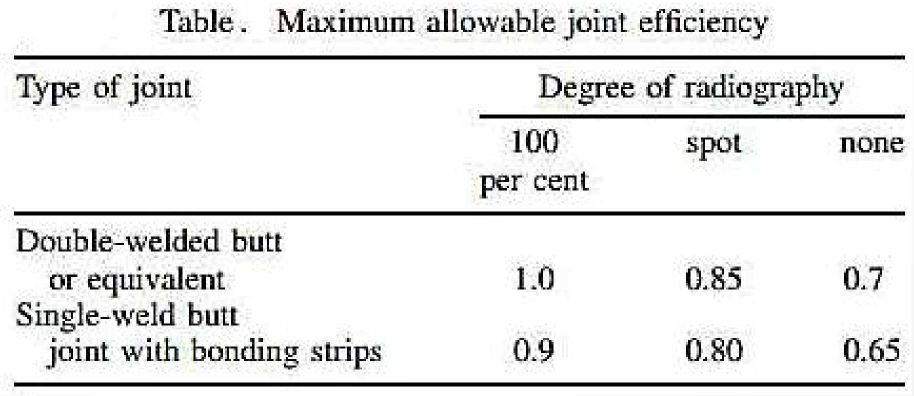


Table. Maximum allowable joint efficiency

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

- 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.

Corrosion Allowance

- Corrosion of chemical equipment due to environmental condition is a common feature. In pressure vessel corrosion rates may be predictable, unpredictable or negligible.
- For carbon steel and cast iron parts, the corrosion allowance is 1.5 mm on all parts except tubes. for severe conditions it may be 3 mm.
- For alloy steel and non-ferrous parts under pressure no corrosion allowance is necessary.
- Similarly when thickness is more than 30 mm corrosion allowance is not necessary

THE DESIGN OF THIN-WALLED VESSELS

UNDER INTERNAL PRESSURE

- The simplest pressure vessel having following parts: Shell, head or cover, nozzle, flanged joint and supports.
- 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

Cylinders and spherical shells

- For a cylindrical shell the minimum thickness required to resist internal pressure can be determined from equation; the cylindrical stress will be the greater of the two principal stresses.
- If D_i is internal diameter and t the minimum thickness required, the mean diameter will be D = (D_i + t); substituting this into stress equation:

Circumferential stress

$$f_p = \frac{P_i D}{2t}$$

where f_p is the design stress/permissible stress, and Pi is the design internal pressure.

Longitudinal or Axial stress

$$f_a = \frac{P_i D}{4t}$$

Both the above stress are tensile. Since the circumferential stress is greater, this is taken as design stress. The shell is generally formed y a joint in the longitudinal direction, which is considered in terms of joint efficiency. Therefore, thickness of the shell is given by

$$t = \frac{P_i D}{2fJ} = \frac{P_i D_i}{2fJ - P_i}$$

$$= \frac{P_i D_o}{2fJ + P_i}$$

Spherical Cell

In this case oth stress are equal to the longitudinal stress

$$t = \frac{P_i D}{4fJ} = \frac{P_i D_i}{4fJ - P_i}$$

$$= \frac{P_i D_o}{4fJ + P_i}$$

Cylindrical vessel under combined loading

- In addition to internal pressure, the other loadings are the weight of the vessel with its contents and the wind.
- The effect of piping can also e taken consideration. The stress created due to each of the above loading can e stated as:

1. Stress in the circumferential direction due to internal pressure, this is known as tangential or hoop stress

$$f_t = \frac{P_i(D_i + t)}{2t}\ (tensile)$$

2. Stress in the longitudinal or axial direction

Due to internal pressure

$$f_1 = \frac{P_i D_i}{4t}(tensile)$$

Due to weight of vessel and its content

$$f_2 = \frac{W}{\pi t(D_i + t)}(compressive)$$

Due to wind or piping in the case of vertical vessel or due to weight of vessel in case of horizontal vessel

$$f_3 = \frac{M}{Z} = \frac{M}{\pi {D_i}^2\, t} (tensile\ or\ compressive)$$

Where

M Bending moment, due to loads normal to the vessel axis

Z modulus of section of the cylindrical vessel

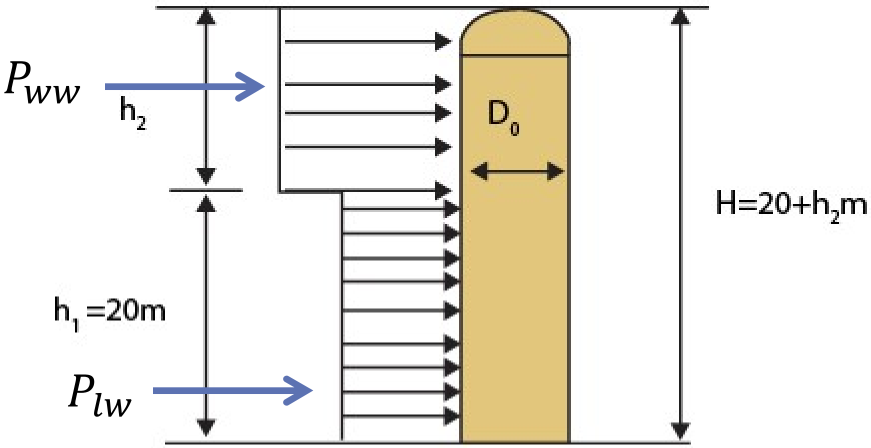
$M = P_{lw}\frac{H}{2}$ (upto H ≤ 20 m)

$= P_{lw}\frac{h_1}{2} + P_{ww}(h_1 + \frac{h_2}{2})$ (H > 20 m)

The force due to wind load acting on the lower and upper parts of the Vessel are shown in figure. Determined as follows:

$P_{lw} = k\, p_1\, h_1\, D_0$ upto 20 m height

$P_{ww} = k\, p_2\, h_2\, D_0$ above 20 m height



p1 = wind pressure for the lower part of the vessel (40 to 100 kg/cm^2)

p2 = wind pressure for the upper part of the vessel (upto 200 kg/cm^2)

k = coefficient depending on the shape factor (0.7 for cylindrical surface)

D_0 = outside diameter of the vessel

Total stress in the longitudinal or axial direction.

$$f_a = f_1 + f_2 + f_3 \text{ (tensile or compressive)}$$

3. Stress due to offset piping or wind

$$f_s = \frac{T}{\pi D_i t (D_i + t)}$$

Where

T-torque about the vessel axis

Combining the above stress on the basis of shear strain energy theory criterion

The equivalent stress

$$f_R = \sqrt{[f_t{}^2 - f_t f_a + f_a{}^2 + 3f_s{}^2}$$

For satisfactory design, the following conditions must e satisfied.

f_R (tensile) ≤ f_t (permissible)

fa (tensile) ≤ f_t (permissible)

fa (compressive) ≤ f_t (permissible)