Pressure Vessels

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(https://www.cometto.com/stories/record-pressure-vessels-moved-on-cometto-mspe/; https://hotlinetradingllc.com/product/asme-pressure-vessels-lined-and-unlined-storage-tanks/; https://m.indiamart.com/proddetail/pressure-vessels-21628811362.html?pos=2&pla=n)

Pressure Vessels

- Any closed vessel over 150 mm diameter subject to a pressure difference of more than 0.5 bar may be considered as a pressure vessel.

- Data required for design:
  - Vessel function
  - Process materials and services
  - Operating and design temperature and pressure
  - Materials of construction
  - Vessel dimensions and orientation
  - Types of vessel heads
  - Openings, connections required
  - Specifications of heating and cooling jackets or coils
  - Type of agitator and specification of internal fittings

Design Codes – Pressure Vessel

- ASME Boiler & Pressure Vessel Code (BPVC) in 1907.
- EU Pressure Equipment Directive (PED 2014/68/EU)
  - Standard: EN 13445 pressure vessel inspection and testing
- ISO 16528-1:2007
  - Performance requirements for boilers and pressure vessels
- IS 2825:1969 Code for unfired pressure vessel by BIS

SECTIONS

I	Rules for construction of power boilers
II	Materials
	Part A Ferrous metal specifications
	Part B Nonferrous metal specifications
	Part C Specifications for welding rods, electrodes, and filler metals
	Part D Properties (customary or metric versions)
III	Nuclear power plant components
	NCA General requirements
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	Division 2 Code for concrete containments
	Division 3 Containments for transport and storage of spent nuclear fuel and high-level radioactive material and waste
IV	Rules for construction of heating boilers
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VI	Recommended rules for the care and operation of heating boilers
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VIII	Rules for the construction of pressure vessels
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IX	Welding and brazing qualifications
X	Fiber-reinforced plastic vessels
XI	Rules for in-service inspection of nuclear power plant components
XII	Rules for construction and continued service of transport tanks

(Towler, G., Sinnott, R. (2008). Chemical Engineering Design.)

Utilization of Design Codes

- Codes provide formulas for thickness and stress of basic components.
  - States maximum primary stress, safety factor.

- Designer needs to:
  - Select suitable analytical procedures for determining stress.
  - Select the most probably combination of simultaneous loads for an economical and safe design.

- Does not require detailed evaluation of stress, but need to identify all loadings and maximum stress.

Terminologies (1)

- Maximum Allowable Working Pressure (MAWP or MWP)
  - Max gauge pressure permissible under operating conditions, at the top of the vessel in its normal position, at a specific temperature (design temperature).

- Design Pressure
  - Involves both internal and external pressures (including static head).
  - Used to determine the minimum thickness of component parts.
  - Specified at specific temperature (design temperature)

Design Pressure

- 5% safety factor is to be added to the MAWP to calculate Design Pressure

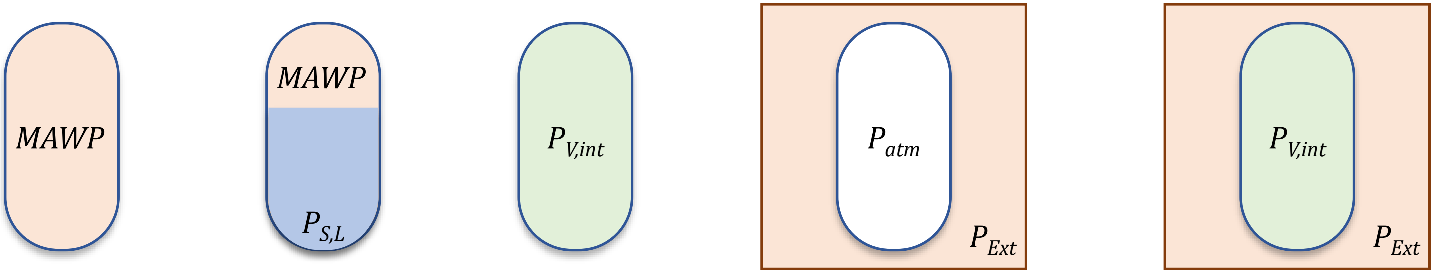
  For vessel under internal pressure:

  If hydrostatic pressure inside the vessel exceeds 5% of MAWP:

  If the vessel is subjected to vacuum:

  If the vessel is subjected to external pressure:

  If the vessel has vacuum pressure and subjected to external pressure:



Terminologies (2)

- Design Temperature
  - The design temperature at which the design stress is evaluated should be taken as the maximum working temperature of the material.
    - Allowable stress value of the material of construction is temperature dependent.
  - Determining Design Temperature:
    - For unheated parts – highest temperature of the stored material.
    - Parts heated by steam, hot water, or other media – highest temperature of the heating media or 10 °C higher than the temperature that is likely to be attained in course of operation.
    - Vessels undergoing direct heating (internal or external by means of fire, flue-gas, electricity, reaction)
      - Shielded parts – highest temp. of inside material + 20 °C.
      - Unshielded parts – highest temp. of inside material + 50 °C.
    - As per IS 2825:1969, $T_{Des} \geq 250$ °C.

Design Temperature

- Maximum permissible operating fluid temperatures for pressure parts of different materials should follow specifications:

Material	Max. permissible temperature (°C)
Carbon steel	540
C-Mo steel	590
Cr-Mo steel	650
Low allow steel (< 6% Cr)	590
Alloy steel (< 17% Cr)	590
Austenitic Cr-Ni steel	650
Cast iron	200
Brass	200

Terminologies (3)

- Design Stress / Working Stress / Allowable Stress
  - The maximum allowable stress that can be accepted in the material of construction.
  - Must be less than damaging stress.
  - Design stress is calculated by applying **Design Stress Factor** (**Factor of Safety**) to the maximum stress that the material is expected to withstand without failure under standard test conditions.

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

(Sinnott, R., Towler, G. (2005). Coulson & Richardon's Chemical Engineering Vol. 6, Chemical Engineering Design. Elsevier Butterworth-Heinemann.)

Allowable Stress

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

(Sinnott, R. (2005). Coulson & Richardon's Chemical Engineering Vol. 6)

Terminologies (4)

- Design Wall Thickness
  - Minimum thickness calculated solely based on stress analysis.
  - Disregards rigidity of equipment, fabricational feasibility, or availability.

- Minimum Actual/Practical Wall Thickness
  - Minimum thickness required for rigid construction depends on the size of the vessel, material of construction, type of reinforcement, etc.
  - Minimum actual wall thickness is a standard available sheet metal which satisfies the design thickness requirement and considers factors like rigidity, weldability, corrosion-erosion allowances, etc.

Vessel Dia. (m)	Min. Thickness (mm)
1.0	5
1.0-2.0	7
2.0-2.5	9
2.5-3.0	10
3.0-3.5	12

(Sinnott, R. (2005). Coulson & Richardon's Chemical Engineering Vol. 6)

Terminologies (5)

- Corrosion Allowance
  - Additional thickness of metal added to compensate for the material loss by corrosion.
  - Corrosion includes chemical corrosion (oxidation), erosion, abrasion.

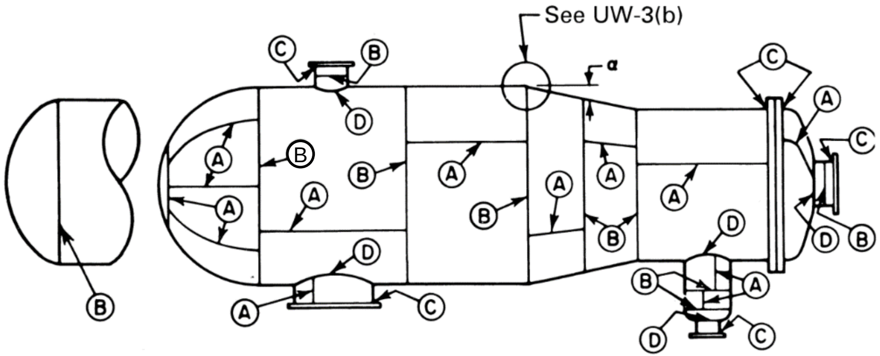
Case description	Corrosion allowance
Carbon steel and cast iron pressure parts (for chemical industries where severe operating conditions not expected)	1.5 mm
Carbon steel and cast iron pressure parts (for petrochemical or other industries where severe operating conditions is expected)	3.0 mm
For stainless steel and non-ferrous parts	No corrosion allowance
If wall thickness is greater than 30 mm	Corrosion allowance maybe neglected

Terminologies (6)

- Weld Joint Efficiency Factor (J)
  - Joints are considered weaker than plate metal.
  - Thicker wall required in and around joint section to improve strength.
  - However, sheets come in uniform thickness, hence, the overall wall thickness has to be increased to compensate for joints.
  - Additional thickness of the material depends on the quality of weld joint.
  - *J* is the ratio of the strength of welded joint to the strength of the virgin metal plates.
  - The soundness of welds is checked by visual inspection and by non-destructive testing (radiography).
  - Allowable design stress is to be multiplied by the '*J*'.
  - *J* may range from 0.5 to 1.0; typically used range is 0.85-1.0.

Weld Joint Efficiency Factor (J)

- Types of weld joints as per ASME BPVC 2017 VIII Div. 1:
  - Category A – Longitudinal welded joints.
  - Category B – Circumferential welded joints.
  - Category C – Welded joints connecting flanges, flat heads, etc.
  - Category D – Welded joints connecting nozzles to main shells, transitions in diameter, to spheres, to curved heads, etc.

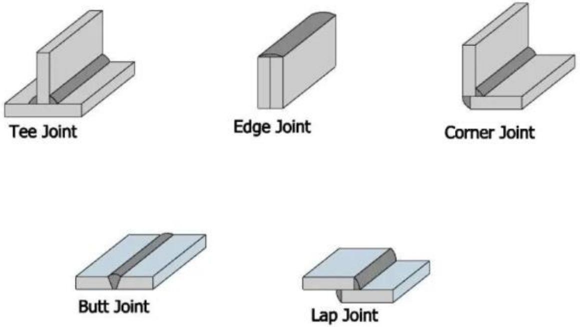


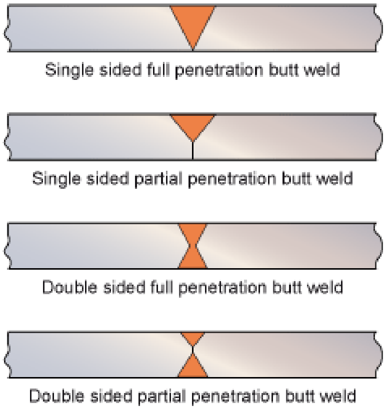
Weld Joint Efficiency Factor (J)

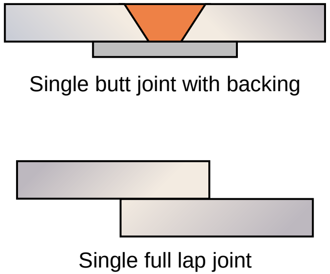
- As per IS 2825-1969, welded pressure vessels are divided into three classes:
  - Class I
    - Vessels contain lethal or toxic substances or operated under severe conditions.
    - Fully radiographed joints required; J can be 1; Even minor defects unacceptable.
  - Class II
    - Vessels for medium duty operation; most chemical process equipment.
    - J can be 0.80-0.85.
    - Maximum wall thickness (including corrosion allowance) is 38 mm.
  - Class III
    - Light duty vessels with plate thickness less than 16 mm (excluding corrosion allowance).
    - Not recommended to operate below 0 °C.

Weld Joint Efficiency Factor (J)

Type of Welding	Class I vessel	Class II vessel	Class III vessel
Double welded butt joint with full penetration	1.00	0.85	0.70
Single welded butt joints with backing strip	0.90	0.80	0.65
Single welded butt joints without backing strip	NA	NA	0.55
Single full lap joints	NA	NA	0.50







(https://welderslab.com/what-are-the-5-basic-types-of-welding-joints; https://www.twi-global.com/technical-knowledge/job-knowledge/design-part-3-092; https://www.thefabricator.com/thefabricator/article/testingmeasuring/upfront-welding-inspection-considerations)

Design Loads

- Major Loads
  - Design pressure: including static head of liquid.
  - Max. weight of the vessel and contents, under operating and test conditions.
  - Wind and earth quake loads.
  - Any other loads directly supported by the vessel.
- Subsidiary Loads
  - Local stresses caused by supports, internal structures, and connecting pipes.
  - Shock loads caused by surging of vessel components, water hammer, etc.
  - Bending moments caused by eccentricity of the centre of the working pressure relative to the neutral axis of the vessel.
  - Stresses due to temperature differences and differences in the expansion coefficients of materials.
  - Loads caused by fluctuations in temperature and pressure.

Stresses in Pressure Vessels

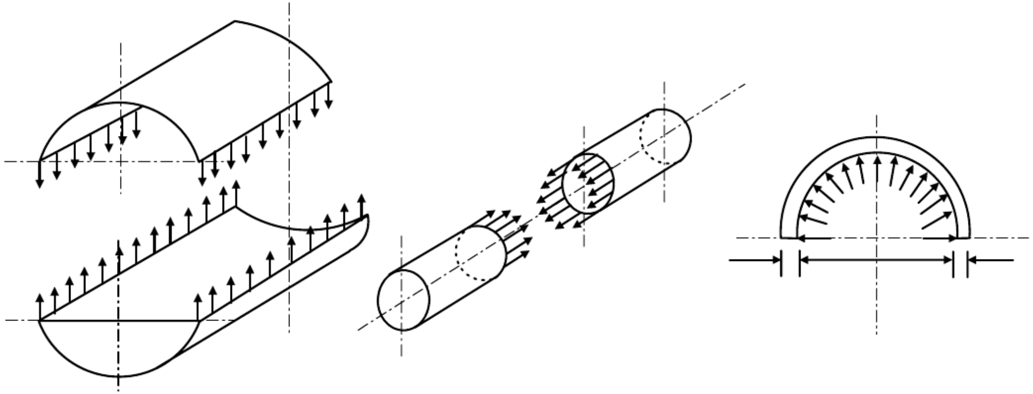
- Types of vessels
  - Thin-walled vessels – thickness:diameter ratio less than 1:10
  - Thick-walled vessels – thickness:diameter ratio greater than 1:10

- Membrane stresses
  - Vessels/shells formed of thin plate offers little resistance to bending perpendicular to their surface; implies negligible radial stress.
  - Such plates are called Membranes.
  - Stresses calculated by neglecting bending are called membrane stresses.

Membrane Stresses in Cylindrical Shell

Consider a cylindrical vessel under internal pressure.
It has length (L), internal diameter (d), thickness (t), and internal gauge pressure (p).
Assuming thin-walled vessel:

Hoop Stress Axial Stress Radial Stress



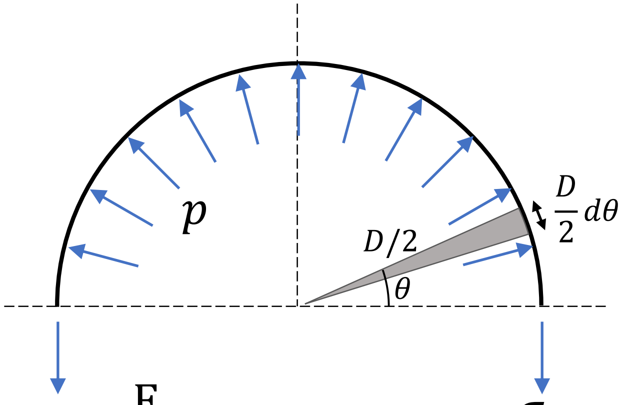
Hoop Stress or Circumferential Stress or Tangential Stress, $\sigma_\theta = \sigma_C = \sigma_2 = \sigma_H$

Axial Stress or Longitudinal Stress, $\sigma_l = \sigma_A = \sigma_1$

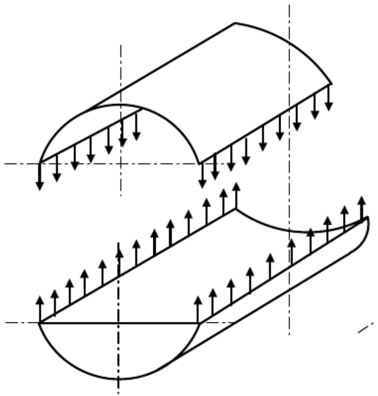
Radial Stress, $\sigma_r = \sigma_3 = 0$

Hoop Stress

$$F = \int_0^{\pi/2} p\left(L\frac{D_i}{2}d\theta\right)\sin\theta$$



$$\sigma_\theta = \frac{\mathrm{F}}{\mathrm{A}}$$

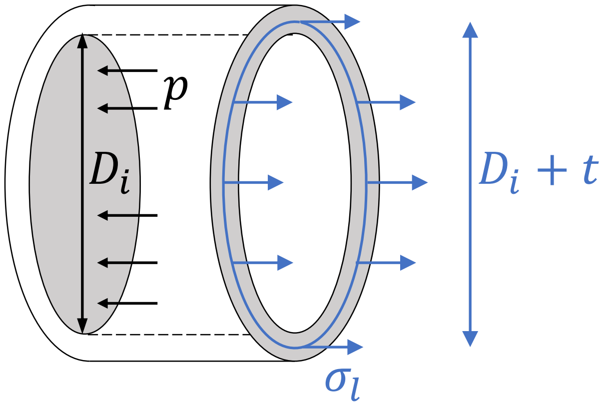


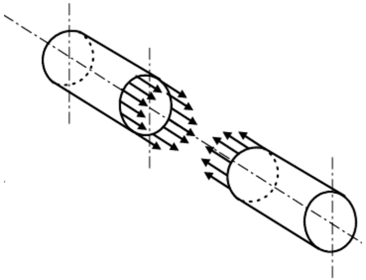
$$\sigma_\theta = \frac{F}{tL} = \frac{pD_i}{2t}$$

Longitudinal Stress

For thin-walled cylindrical vessel,

$$D = D_i + t$$



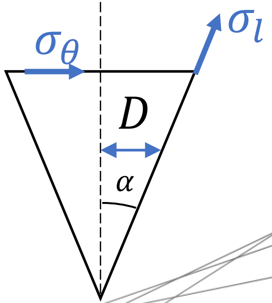


For thin-walled spherical vessel,

$$\sigma_\theta = \sigma_L = \frac{pD_i}{4t}$$

For thin-walled conical vessel,

$$\sigma_\theta = \frac{pD_i}{tcos\alpha}; \; \sigma_L = \frac{pD_i}{4tcos\alpha}$$



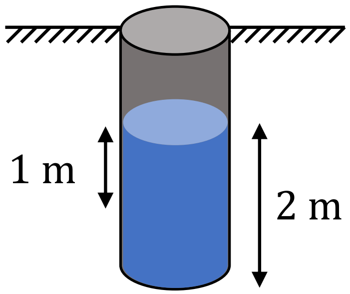
Numerical 3.1

A thin cylindrical shell having 3 m inner diameter and 12 m length is being operated at 3 MN/m^2 (gauge) pressure. Compute the thickness of shell if allowable stress of material is 200 MN/m^2.

Numerical 3.2

A cylindrical container of radius R = 1 m, wall thickness 1 mm is filled with water up to a depth of 2 m and suspended along its upper rim. The density of water is 1000 kg/m^3 and g = 10 m/s^2.

Calculate axial and circumferential stress at mid point (1 m)?



Dilation of Pressure Vessel

- Dilation – radial growth, or change in the shell thickness.

Let radial growth of a vessel with inner radius '*r*' and internal pressure '*p*' be '*δ*'.

$$\delta = \varepsilon r$$ ε is the strain.

$$\varepsilon = \frac{\sigma_\theta}{E} - \nu \frac{\sigma_l}{E}$$

For cylindrical vessel:

$$\delta = \left[\frac{pr}{tE} - \nu \frac{pr}{2tE}\right] r$$

$$\delta = \frac{pr^2}{2tE}(2 - \nu)$$

For spherical vessel:

$$\delta = \left[\frac{pr}{2tE} - \nu \frac{pr}{2tE}\right] r$$

$$\delta = \frac{pr^2}{2tE}(1 - \nu)$$

For conical vessel:

$$\delta = \frac{pr^2(2 - \nu)}{2tE cos\alpha}$$

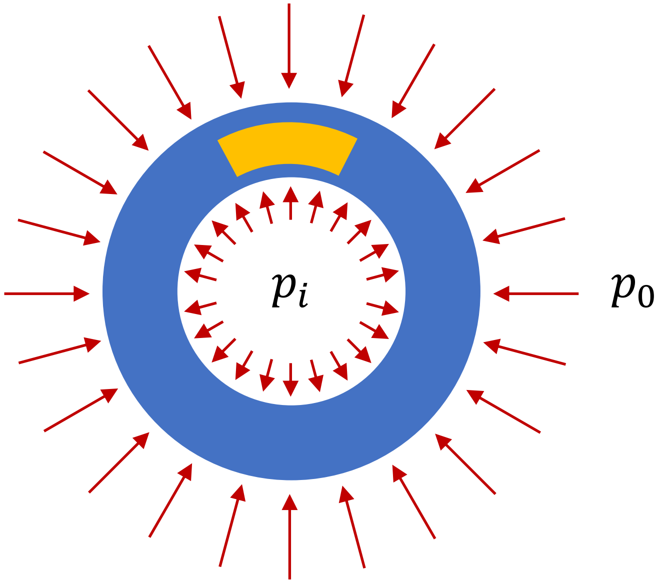
Thick-Walled Pressure Vessel

- High pressure & high temperature reactions
  - Haber process, hydrogenation of petrochemicals and coal, urea synthesis, etc.

- When the shell thickness is large, radial stress varies across the shell thickness.

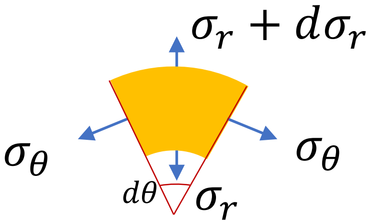
- Lame's Stress Analysis

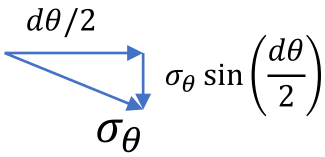
Lame's Stress Analysis



Consider a thick-walled vessel subjected to internal pressure p_i and external pressure p_0.

Assumption: deformation will be symmetrical about the axis and constant along the length.





Considering vertical force balance,

$$(\sigma_r + d\sigma_r)(r + dr)(d\theta)l - \sigma_r(rd\theta)l = 2\sigma_\theta \sin\left(\frac{d\theta}{2}\right)(dr \times l)$$

Lame's Stress Analysis (2)

For small angles, $sin\theta \approx \theta$

$$\Rightarrow \sigma_r + r\frac{d\sigma_r}{dr} = \sigma_\theta \qquad (1)$$

Since longitudinal stress is constant along with length of the shell, longitudinal strain should also be constant.

$$\varepsilon_l = \frac{1}{E}[\sigma_l - \nu(\sigma_r + \sigma_\theta)]$$

$$\Rightarrow \sigma_r + \sigma_\theta = constant = 2A \qquad (2)$$

Lame's Stress Analysis (3)

Substituting (2) in (1),

Hence, $\sigma_r r^2 - Ar^2 = constant = -B$

$$\boxed{\begin{aligned}\sigma_r &= A - \frac{B}{r^2} \\ \sigma_\theta &= A + \frac{B}{r^2}\end{aligned}}$$

Lame's Equations

Lame's Stress Analysis (4)

Considering σ_r at r_i and r_o,

$$r = r_i; \quad \sigma_r = -P_i$$

$$r = r_o; \quad \sigma_r = -P_0$$

$$\therefore B = \frac{(P_o - P_i) r_i^2 r_o^2}{r_i^2 - r_o^2}$$

$$\therefore A = \frac{\left(P_o r_o^2 - P_i r_i^2\right)}{r_i^2 - r_o^2}$$

Lame's Stress Analysis (5)

Hence, Lame's equations become:

$$\sigma_r = \frac{\left(P_o r_o^2 - P_i r_i^2\right)}{r_i^2 - r_o^2} - \frac{(P_o - P_i) r_i^2 r_o^2}{r_i^2 - r_o^2} \frac{1}{r^2} \qquad \sigma_\theta = \frac{\left(P_o r_o^2 - P_i r_i^2\right)}{r_i^2 - r_o^2} + \frac{(P_o - P_i) r_i^2 r_o^2}{r_i^2 - r_o^2} \frac{1}{r^2}$$

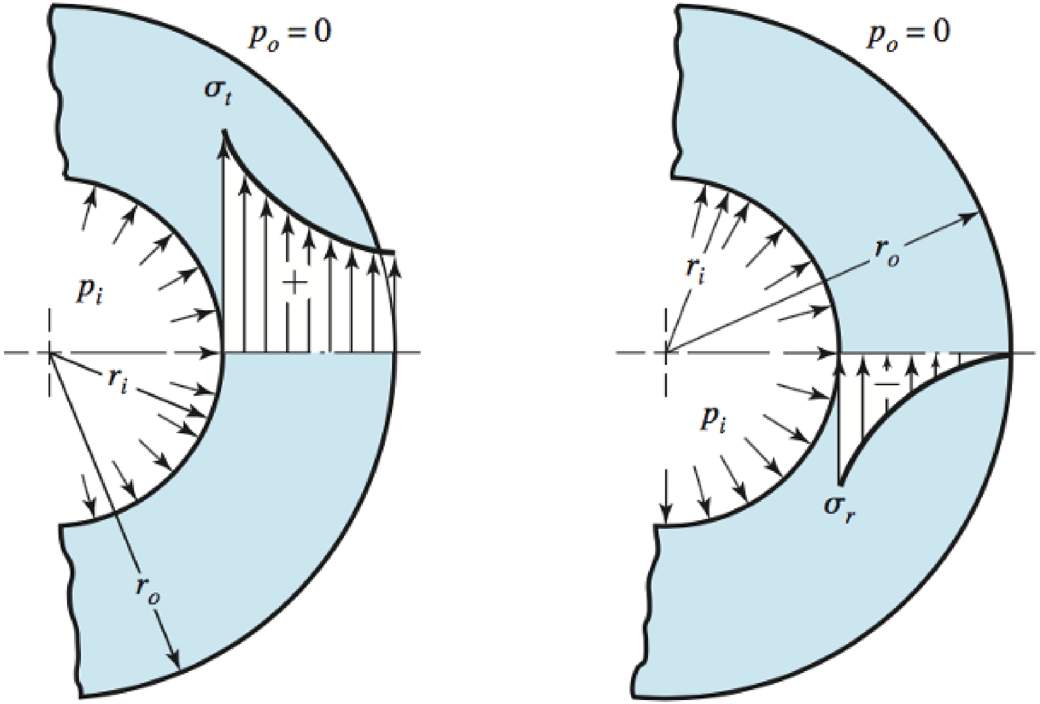
For internal pressure vessel,

Max. stress will be for σ_θ when $r = r_i$.

$$\sigma_{max} = (\sigma_\theta)_{r=r_i} =$$

If allowable stress of a material is f and J is the weld-efficiency factor, $fJ = \sigma_{max}$

Stress Distribution



(a) Tangential stress distribution

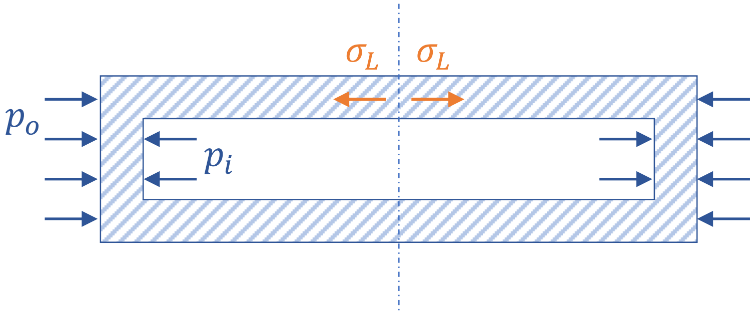
(b) Radial stress distribution

Lame's Stress Analysis (6)

Max. shearing stress at any point is given by,

$$\tau = \frac{\sigma_\theta - \sigma_r}{2}$$

Longitudinal stress:



For internal press. vessel,

Thickness of the shell

Given,

$$fJ = \sigma_{max} = \frac{P\left(r_i^2 + r_o^2\right)}{r_o^2 - r_i^2}$$

$$P = fJ\,\frac{(r_o^2 - r_i^2)}{\left(r_i^2 + r_o^2\right)}$$

If ‘t’ is the min. wall thickness,

$$P = fJ\,\frac{(r_i + t)^2 - r_i^2}{(r_i + t) + r_i^2}$$

$$\boxed{P = \frac{fJt}{\left[r_i + \dfrac{t(r_i + t)}{2r_i + t}\right]} = \frac{fJt}{\left[r_i + t\left(\dfrac{1 + \dfrac{t}{r_i}}{2 + \dfrac{t}{r_i}}\right)\right]}}$$

Design equation for shell thickness as per IS:2825-1969 when $D_o/D_i \leq 1.5$.

Thickness of the shell (2)

When $D_o/D_i = 1.5$, $\left(\frac{t}{D_i}\right)_{max} = 0.25$ $\quad \therefore 0 \leq t/D_i \leq 0.25$

$$P = \frac{2fJt}{\left[D_i + t\left(\dfrac{1+\frac{2t}{D_i}}{1+\frac{t}{D_i}}\right)\right]}$$

$= 1.2$ IS: 2825-1969 approximated this to 1.

$$\therefore P = \frac{2fJt}{D_i + t} = \frac{2fJt}{D_o - t} \qquad \Rightarrow \boxed{t = \frac{PD_i}{2fJ - P} = \frac{PD_o}{2fJ + P}}$$

Minimum thickness of shell plate for the design pressure (excluding allowances).

Numerical 3.3

- A process vessel is to be designed for maximum operating pressure of 800 kN/m^2 (abs). The vessel has the nominal diameter of 1.2 m and tangent to tangent length of 2.4 m. The vessel is made of IS:2002-1962 Grade 2B steel having allowable design stress value of 118 MN/m^2 at working temperature. 2 mm is the recommended corrosion allowance for the expected life span of the vessel. This is a class 2 vessel as per Indian Standards, stipulating weld joint efficiency of 0.85. Calculate the standard plate thickness to fabricate this vessel.

Note: as per Indian Standard, nominal diameter = outer diameter of the vessel.

Theories of Elastic Failure

Indicates max. stress or strain before permanent deformation occurs.

- Maximum Stress Theory (Rankine Theory)

$$\sigma_{max} = (\sigma_\theta)_{r=r_i} = \frac{P\left(r_i^2 + r_o^2\right)}{r_o^2 - r_i^2} = P\,\frac{(K^2+1)}{K^2-1} \qquad \left[K = \frac{D_i}{D_o}\right]$$

- Maximum Strain Theory

$$\sigma_{max} = P\,\frac{(K^2(1+\nu) + (1-\nu))}{K^2-1}$$

- Maximum Strain Energy Theory

$$\sigma_{max} = P\,\frac{\sqrt{6+10K^4}}{2(K^2-1)}$$

- Maximum Shear Theory

$$\sigma_{max} = P\,\frac{2K^2}{(K^2-1)} = 2\tau_{max}$$

Theories of Elastic Failure (2)

- For design purpose, value of calculated stress must be divided by a suitable factor of safety.

$$Design\ stress = \frac{Yield\ stress}{Factor\ of\ Safety\ (FOS)}$$

Numerical 3.4

A cylinder is 350 mm ID and 450 mm OD. The internal pressure is 160 MPa and the external pressure is 80 MPa. Find the minimum and maximum radial and tangential stresses and the maximum shear stress. The ends are closed.

Numerical 3.5

A cylinder has an ID of 100 mm and an internal pressure of 50 MPa. Find the needed wall thickness if the factor of safety is 2.0 and the yield stress is 250 MPa.

a) Use Maximum Stress theory (as per IS 2825-1969)

b) Use the maximum shear stress theory, i.e. maximum shear stress = yield strength/(2xFOS).