

Thermofluids and Applied Mechanics (Applied Mechanics element)



SGDM6022 - COMPLEX STATIC AND DYNAMIC SYSTEMS

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1.0 Introduction

The following paper has been used to complete five tasks, some of which are descriptive and some of which have been completed using calculations. The aim of the paper is to critically evaluate complex static and dynamic systems.

2.0 Balancing

The following task has been broken into two sections. The first section has been used to explain the importance of balancing and the difference between static and dynamic balancing in a non-coplanar system. The second section has been used to complete a mathematical example to identify a systems complete static and dynamic balance.

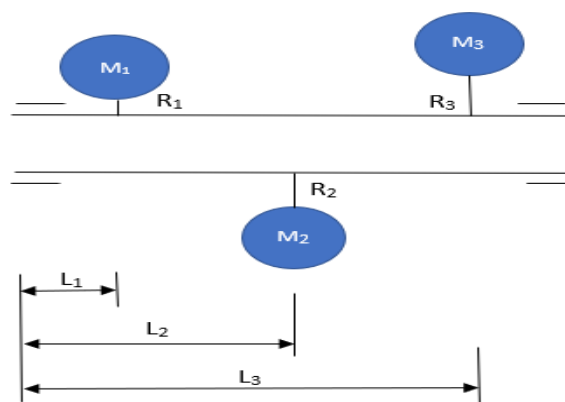
2.1 Dynamic Balancing

Explain the importance of balancing and the difference between static and dynamic balancing in a non-coplanar system. Provide relevant examples where balancing is of importance.

To gain static and dynamic balancing in a non-coplanar system, the moments and reactions must all be balanced.

Non-coplanar balancing is used when masses are positioned at different positions along a shaft.

Figure 1 - Co-Planner Balancing



For complete balance the total sum of centrifugal forces must = 0 $\sum mrw^2 = 0$

And the total sum of moments must = 0 $\sum mrw^2 \times L = 0$

For forces: $\sum F = 0$ Referred to as “static balance”

For moments: $\sum M = 0$ Referred to as “dynamic balance”

Balancing has a clear importance within the engineering world. For balancing to happen it has been identified that the center of gravity of an object must be centered over an objects base. Center of gravity refers to the center of the object's bodyweight. Research shows that (Reference*, 2020) *"balancing also plays an important role in machines. For instance, the balancing of a rotating body helps to avoid vibration that can cause failure, especially in heavy industrial machines such as electric generators and gas turbines. Machine vibrations can also cause excessive noise and discomfort."*

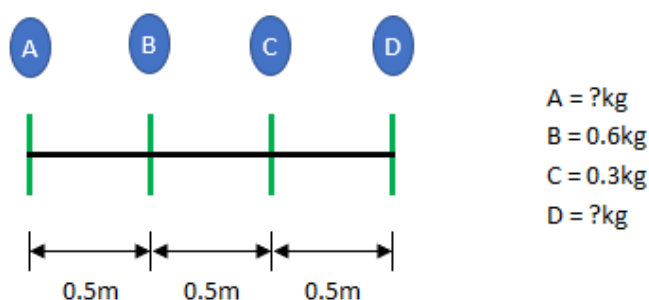


2.2 Balancing Example

'An experimental balancing rig consists of four discs A, B, C and D mounted on a rotating shaft, the linear distances between each of the discs being 0.5m. On each disc provision is made for the attachment of unbalanced masses at different radii and angular displacement. If a mass of 0.6kg at radius 120mm is attached to B at 0° and a mass of 0.3kg at radius 120mm is attached at C at 120° find the magnitude and angular position relative to B of masses placed at A and D at radii 100mm and 90mm respectively to give the system complete static and dynamic balance.'

To determine the above, a series of sketches are used in the following section.

Figure 2 - Balancing Rig



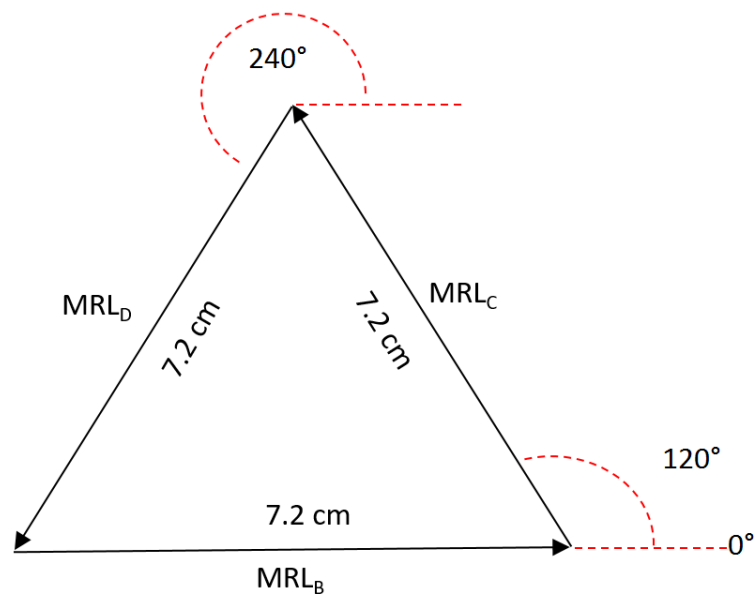
To determine the magnitude of M_A and M_D along with their angular position, the following MRL table (figure 3) must be used.

Figure 3 - MRL Table

PLANE	MASS (KG)	RADIUS (m)	M x R (kg/m)	L (m)	M x R x L (kg/m ²)	θ
A	M_A	0.1	$0.1M_A$	0	0	θ_A
B	0.6	0.12	0.072	0.5	0.036	0°
C	0.3	0.12	0.036	1	0.036	120°
D	M_D	0.09	$0.09M_D$	1.5	$0.136M_D$	θ_D

To determine M_d and θ_d produce the MRL polygon, Scale: $2\text{cm} = 0.01\text{kgm}^2$

Figure 4 - MRL Polygon

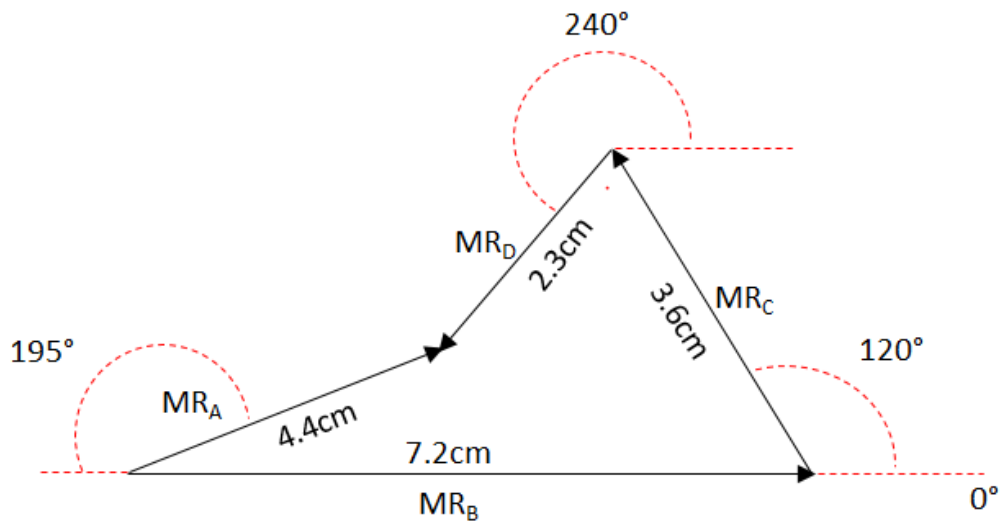


- $MRL_B = 7.2\text{cm} = 0.036\text{kgm}^2$ at 0°
- $MRL_C = 7.2\text{cm} = 0.036\text{kgm}^2$ at 120°
- $MRL_D = 7.2\text{cm} = 0.036\text{kgm}^2$ at 240°

$$0.135M_D = 0.036 \text{ so } M_D = \frac{0.036}{0.135} \therefore M_D = 0.26\text{kg at } 240^\circ$$

To determine M_A and θ_A produce the MR polygon, Scale: $1\text{cm} = 0.01\text{kgm}^2$

Figure 5 - MR Polygon



- $MR_B = 7.2\text{cm} = 0.072\text{kgm}^2$ at 0°
- $MR_C = 3.6\text{cm} = 0.036\text{kgm}^2$ at 120°
- $MR_D = 2.3\text{cm} = 0.023\text{kgm}^2$ at 240°
- $MR_A = 4.4\text{cm} = 0.044\text{kgm}^2$ at 195°

$$0.1M_A = 0.044 \quad \text{so } M_A = \frac{0.044}{0.1} \quad \therefore M_A = 0.44\text{kg at } 195^\circ$$



Figure 6 - Completed MRL Table

PLANE	MASS (KG)	RADIUS (m)	M x R (kg/m)	L (m)	M x R x L (kg/m ²)	θ
A	0.44	0.1	0.044	0	0	195°
B	0.6	0.12	0.072	0.5	0.036	0°
C	0.3	0.12	0.036	1	0.036	120°
D	0.26	0.09	0.023	1.5	0.036	240°

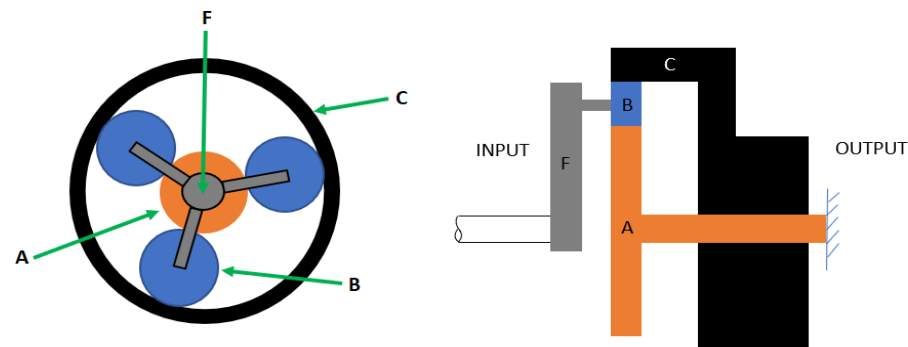
3.0 Gears

The following section identifies the components of the Epicyclic gearbox along with its key applications.

3.1 Epicyclic Gearbox

(a) Provide a sketch of a typical epicyclic gearbox and identify the main elements.

Figure 7 - Epicyclic Gearbox



(b) Provide an explanation of each of the major elements identified in (a) above.

Epicyclic gear trains always include a sun gear, a planet carrier (or link arm) and one or more planet gears. Often a ring is included which has internal teeth that mesh with the planet gear, this is known as the annulus.

- **A = Sun gear** - It is the gear with angular cut teeth and is placed in the middle of the Epicyclic gearbox. The sun gear is in constant mesh at inner point with the planetary gears and is connected with the input shaft of the Epicyclic gearbox.
- **B = Planet gears** - These are small gears used in between ring and sun gears. The teeth of the planet gears are in constant mesh with the sun and the ring gear at both the inner and outer points respectively.
- **C = Annulus Gear** - is a type of gear which looks like a ring and has angular cut teeth at its inner surface. It is placed in an outermost position in an Epicyclic gearbox. The inner teeth of Annulus gear is in constant mesh at outer point with the set of planetary gears.
- **F = Fixed Link Arm or Planet carrier** - It is a carrier attached with the axis of the planet gears and is responsible for the final transmission of the output to the output shaft.

3.2 Epicyclic Gearbox Arrangement

(c) In the case of a standard epicyclic gearbox arrangement obtain expressions for the sun, planet and ring gears when, the arm is fixed, the sun is fixed, the ring is fixed.

The three possible configurations of an Epicyclic gear are as follows,

- (1) The arm is fixed, which is known as the **star** formation where the sun and planetary gears are free to rotate. (Beardmore, n.d.)
 - *Fixed frame assembly*
 - *Input and output shaft rotate in opposite directions*
 - *Ratios up to 10:1 (Reduction or increase depending on drive direction.)*
- (2) The sun is fixed, which is known as **planetary** formation where the sun and carrier arm rotate.
 - *Fixed annulus system*
 - *Input and output shaft rotate in same directions*
 - *Ratios up to 12:1 (Reduction or increase depending on drive direction.)*
- (3) The ring is fixed, which is known as **solar formation** where the annulus ring and carrier arm rotate.
 - *Fixed sun wheel*
 - *Input and output shafts rotate in same direction*
 - *Ratios up to 2:1 (Reduction or increase depending on drive direction.)*

To determine the gear ratio of an Epicyclic gearbox, relative motion must be considered as follows.

- (1) Rotate whole train +1 rev
- (2) Fix Arm and rotate fixed gear -1 rev
- (3) Add lines 1 and 2 together

Steps 1 to 3 can be seen on the following page within figure 8 table.

Figure 8 - Fixed Epicyclic Gears

	L (link Arm)	A (Annulus)	P (Planet)	S (Sun)
Step: 1	+1	+1	+1	+1
Step: 2	0	-1	$-\left(\frac{T_a}{T_p}\right)$	$+\left(\frac{T_a}{T_s}\right)$
Step: 3	1	0	$1 - \left(\frac{T_a}{T_p}\right)$	$1 + \left(\frac{T_a}{T_s}\right)$

3.2.4 Free Rotation

(d) Again in the case of a standard epicyclic gearbox arrangement obtain expressions for the sun, planet and ring gears when all gears are free to rotate.

In cases where no wheel is fixed then the whole gear train can be rotated causing free rotation. The gear ratio is determined from the following.

- (1) Rotate whole train +m revs
- (2) Fix Arm and rotate sun +n revs
- (3) Add lines 1 and 2 together

Figure 9 - Epicyclic Gear Free Rotation

	L (link Arm)	A (Annulus)	P (Planet)	S (Sun)
Step: 1	+m	+m	+m	+m
Step: 2	0	$-n\left(\frac{T_s}{T_a}\right)$	$-n\left(\frac{T_s}{T_p}\right)$	+n
Step: 3	m	$m - n\left(\frac{T_s}{T_a}\right)$	$m - n\left(\frac{T_s}{T_p}\right)$	m + n

3.3 Epicyclical Gear Box Applications

(e) Discuss and evaluate typical applications for epicyclical gear boxes. Evaluate the use of epicyclical gear trains against standard gear trains.

The main application of an epicyclical gearbox is within an automobile. The advantages that an automobile epicyclic gearbox has is achieving high gear ratios. These types of gearboxes also act as bearings as well as the transmissions.

Other applications of Epicyclic gears are also found in back gears of lathes, hoists, pulley blocks and wrist watches. They are also found in bicycles for controlling power of pedaling automatically or manually and are used for a power train between an internal combustion engine and an electric motor.

The Epicyclic gearing is particularly useful for transmitting high velocity ratios with gears of a moderate size in comparatively lesser space than standard gears. They have much higher gear ratios than standard gears and can achieve higher transmission ratio due to their size.

3.4 Complex and Compound Epicyclic Gear Trains

(f) Discuss and evaluate the design features and functions of complex and compound epicyclic gear trains.

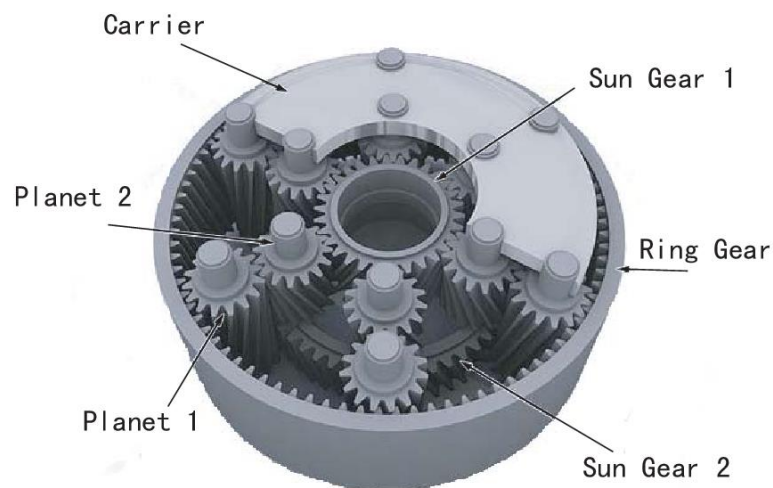
Epicyclic gear are gear systems consisting of one or more planetary gears that revolve around a central sun gear. Typically, the planet gears are mounted on a movable arm or *carrier*, which itself may rotate relative to the sun gear. The Epicyclic gear train also uses an outer ring gear or annulus, which meshes with the planet gears. Planetary gears can be classified as simple or complex and compound planetary gears. Simple planetary gears have one sun, one ring, one carrier, and one planet set. Complex and compound planetary gears have one or more of the following types of structures,

- meshed-planet, where there are at least two planets in mesh with each other in each planet train

- stepped-planet, where there is a shaft connection between two planets in each planet train
- multi-stage structures, where the system contains two or more planet sets

The more complex the compound Epicyclic gear train is the more planetary gears that will be used. Compared to simple planetary gears, compound planetary gears have the advantages of larger reduction ratio, higher torque-to-weight ratio, and have far more flexible design feature configurations. Figure 10 below features a complex and compound gear train that details a meshed planet configuration.

Figure 10 - Complex and Compound Epicyclic Gear Train



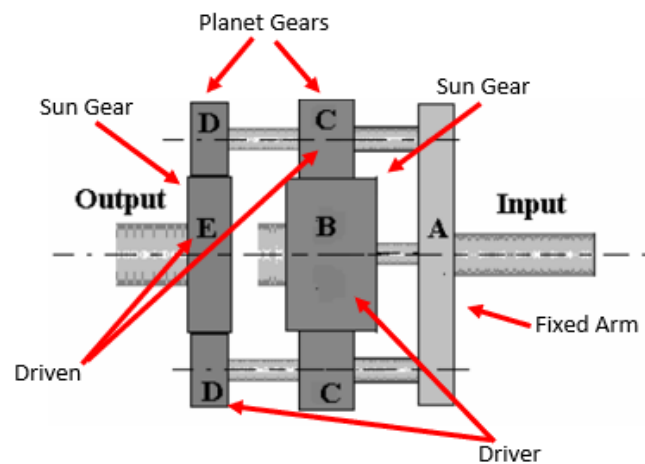
(Guo, 2011)



3.5 Gears Example

'A compound epicyclic gear box is as shown below (figure 1). Gear C has 200 teeth, B has 50, D has 50 and E has 200. Calculate the ratio of the gear box. Assume that B is fixed, and that C and D are rigidly connected.'

Figure 11 - Epicyclic Gear Box



The following assumptions have been made to determine the table below.

- Gear B is the sun gear and C/D are planetary gears not annulus
- The systems input is through B and the output of E is regarded as the required ratio

$$T_D = 50$$

$$T_C = 50$$

$$T_E = 200$$

$$T_B = 200$$

$$\text{Compound Epicyclic Gear} = \frac{\text{Driver}}{\text{Driven}} \therefore \frac{B \times D}{C \times E}$$

	A	B	C/D	E
Rotate whole train by +1 rev	+1	+1	+1	+1
Fix arm and rotate fixed gear -1 rev	0	-1	$+1 \left(\frac{50}{200} \right)$	$-1 \left(\frac{50 \times 50}{200 \times 200} \right)$
Add (1) and (2) together	1	0	1.25	0.94

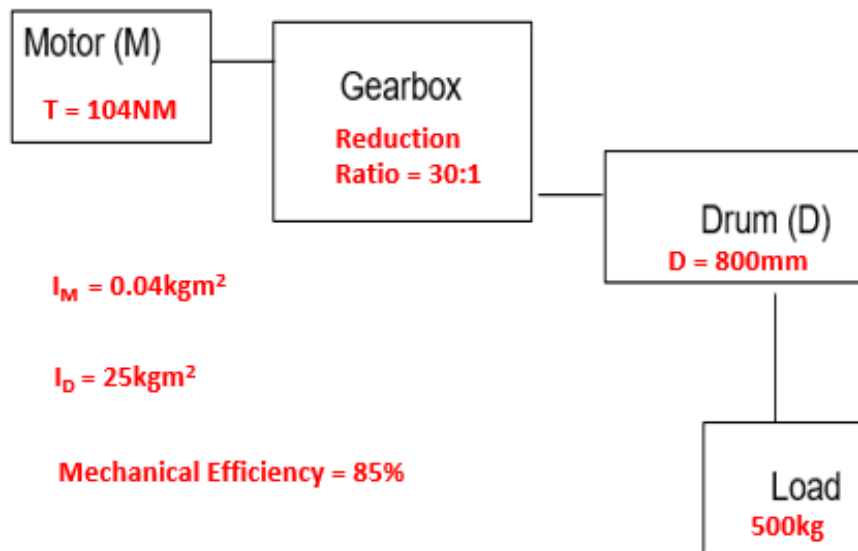
Gear Ratio = 0.94:1 at Output 'E'



4.0 Pulleys Example

'The lifting device shown in figure 2 consists of a motor M driving a drum D through a reduction gearbox. The load is lifted by a cable wrapped around the drum. Using the data provided calculate the vertical acceleration of the load neglecting the mass of the cable. Take the moment of inertia of the motor and gears on the motor shaft as 0.04Kgm^2 and the moment of inertia of the drum and cables on the drum shaft as 25Kgm^2 . The gear reduction is 30:1, the drum diameter is 800mm, the mass of the load is 500kg, the motor torque is 104Nm and the mechanical efficiency of the motor is 85%.'

Figure 12 - Lifting Device



$$P = T \cdot \omega$$

$$T = I \cdot \alpha$$

$$V = \omega \cdot r$$

$$a = \alpha \cdot r$$

$$T \cdot \omega_m = (I_m \alpha_m) \omega_m + (I_D \alpha_D) \omega_D + m(g + a)r_D \omega_D$$

Where, $T = 104 \times 0.85 = 88.4\text{Nm}$, $M = 500\text{kg}$ and $r_D = 0.4\text{m}$

When using the gear reduction ratio of 30:1 $\omega_m = 30\omega_D$ and $\alpha_m = 30\alpha_D$

To calculate the vertical acceleration (a)

$$88.4(30\cancel{\omega_D}) = 0.04(30\alpha_D)(30\cancel{\omega_D}) + 25\alpha_D\cancel{\omega_D} + 500(9.81 + a)0.4\cancel{\omega_D}$$

$$2652 = 36\alpha_D + 25\alpha_D + 1962 + 200a$$

$$\text{Where } \alpha_D = \frac{a}{r_D} \therefore \alpha_D = \frac{a}{0.4}$$

$$\text{So } 2652 - 1962 = \frac{61a}{0.4} + 200a$$

$$690 = 152.5a + 200a$$

$$\frac{690}{352.5} = a$$

$$\text{Vertical acceleration (a) = } 1.96\text{m/s}^2$$

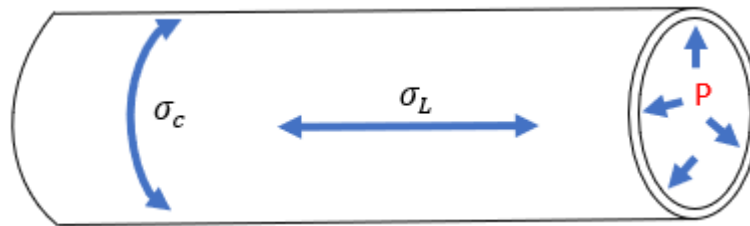
5.0 Pressure Vessels

There are three main items to consider when discussing thin and thick cylinders. These items are circumferential, longitudinal and radial stresses, the following section will discuss in more detail.

5.1 Thin Pressure vessels

A thin cylinder is when $d/t > 20$

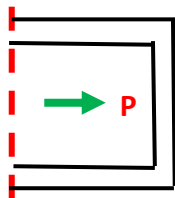
Figure 13 - Thin Pressure Vessel



With an internal pressure two stresses are considered,

- (1) Circumferential (Hoop) stress - σ_c
- (2) Longitudinal Stress - σ_L

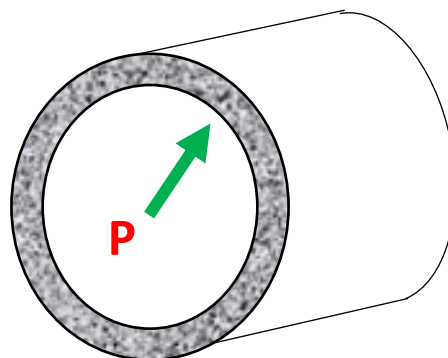
Consider the longitudinal Stress



$$\text{Where, } \sigma_L = \frac{F}{A} = \frac{PAX}{A}$$

$$\sigma_L = \frac{P \times \frac{\pi D^2}{4}}{\pi DT} = \sigma_L = \frac{Pd}{4t}$$

The internal diameter is used for thin vessels, this is also the area that is resisting force.

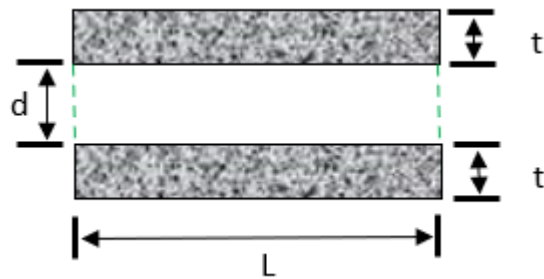


Consider the circumferential stress

Where, $\sigma_c = \frac{F}{A} = \frac{PAX}{A}$

$A = 2tL$

$\sigma_c = \frac{P(\cancel{L}d)}{2t\cancel{L}} = \sigma_c = \frac{Pd}{2t}$

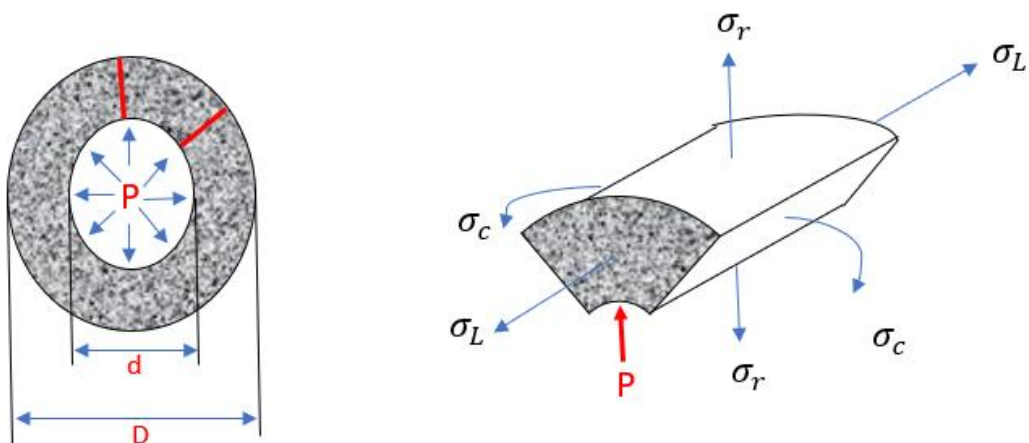


5.2 Thick Pressure Vessels

A thick cylinder is when $d/t < 20$

In the case of a thick cylinder a radial stress σ_r is also considered.

Figure 14 - Thick Pressure Vessel



To determine σ_r and σ_c the "Lame's" equation must be used.

$\sigma_r = a + \frac{b}{r^2}$

$\sigma_c = a - \frac{b}{r^2}$

In the formula above, r = radius (m) and a & b are both constants.



5.3 Pressure Vessel Example

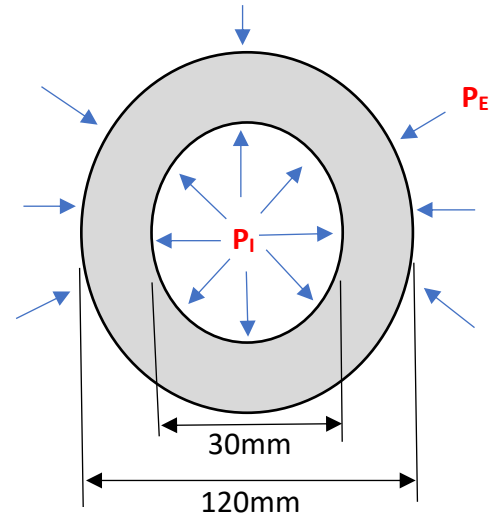
A steel pipe of internal diameter 30mm and external diameter 120mm is subject to an internal pressure of 12MPa and an external pressure of 5MPa?

Radial Stress $\sigma_r = a + \frac{b}{r^2}$

Circumferential Stress $\sigma_c = a - \frac{b}{r^2}$

(pressure internal) $P_i = 12\text{MN/m}^2$

(pressure external) $P_e = 5\text{MN/m}^2$



5.3.1 Radial and Circumferential Stresses

What are the radial and circumferential stresses at the inner and outer surfaces?

Set boundary conditions

(1) when $r = 0.015\text{m}$ and $\sigma_r = -12\text{MN/m}^2$

(2) when $r = 0.06\text{m}$ and $\sigma_r = 5\text{MN/m}^2$

So $\sigma_r = a + \frac{b}{r^2} \therefore -12 \times 10^6 = a + \frac{b}{(0.015)^2} \rightarrow \text{Eqn (1)}$

And $\sigma_r = a + \frac{b}{r^2} \therefore -5 \times 10^6 = a + \frac{b}{(0.06)^2} \rightarrow \text{Eqn (2)}$

Now from equation (2)

$$-5 \times 10^6 = a + 277.7b$$

$$\therefore a = -5 \times 10^6 - 277.7b$$

Sub into equation (1)

$$-12 \times 10^6 = (-5 \times 10^6 - 277.7) + 4444.4b$$

$$-12 \times 10^6 = -5 \times 10^6 + 4166.7b$$

$$-12 \times 10^6 + 5 \times 10^6 = 4166.7b$$

$$-7 \times 10^6 = 4166.7b$$

$$\therefore b = \frac{-7 \times 10^6}{4166.7}$$

$$\therefore b = -1679.9$$

Now determine 'a'

$$a = -5 \times 10^6 - 277.7(-1679.9)$$

$$\therefore a = -4.5 \times 10^6$$

Circumferential Stress

$$\sigma_c = -4.5 \times 10^6 - \frac{(-1679.9)}{(0.015)^2}$$

$$\therefore \sigma_c = 2.9 \times 10^6$$

$$\sigma_c = -4.5 \times 10^6 - \frac{(-1679.9)}{(0.06)^2}$$

$$\therefore \sigma_c = -4 \times 10^6$$

Therefore, the internal radial stresses and external circumferential stresses can be seen in the table below.

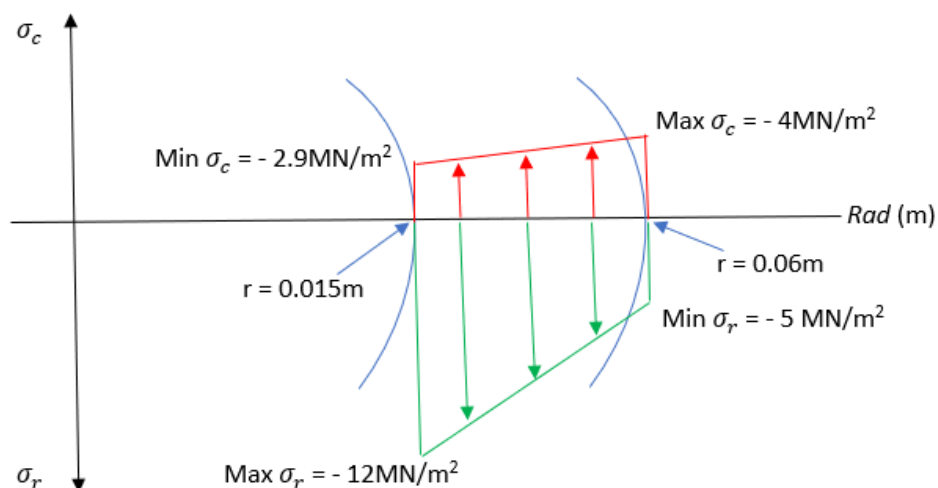
Radius (m)	Radial Stress (N/m ²)	Circumferential Stress (N/m ²)
0.015	-12x10 ⁶	2.9x10 ⁶
0.06	-5x10 ⁶	-4x10 ⁶



5.3.2 Hoop Stress

Sketch the variation of hoop stress across the material, clearly showing the maximum and minimum positions of hoop stress?

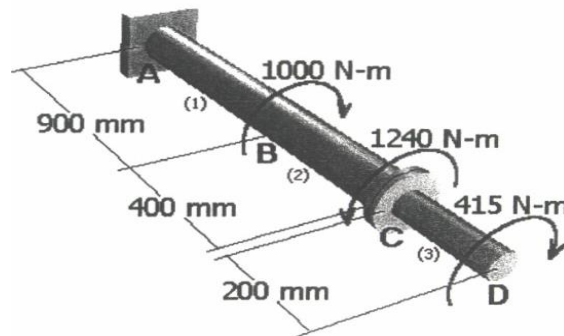
Figure 15 - Hoop Stress Curve



6.0 Torsion Example

'The shaft shown in Figure 3 consists of a 30mm diameter solid section CD and a tube AC that has an outside diameter of 45mm and an inside diameter of 35mm. The shaft is subjected to concentrated torques at B, C, and D. The shear modulus of elasticity is $G = 75 \text{ GPa}$.'

Figure 16 - Torsion



$$T_1 = -415 \text{ MN}$$

$$T_2 = -415 \text{ MN} + 1240 = 825 \text{ MN}$$

$$T_3 = -415 \text{ MN} + 1240 - 1000 = -175 \text{ MN}$$

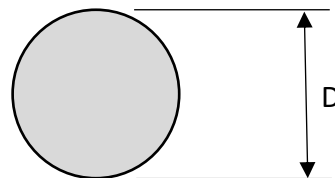
To complete the example, the 'General Torsion Equation' must be used.

$$\frac{T}{J} = \frac{\tau}{r} = \frac{G\theta}{L}$$

In addition to the General Torsion Equation, the 'Polar second moment of Area (J)' must also be considered.

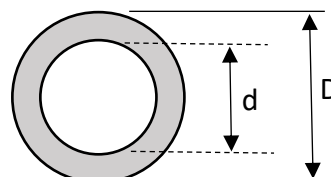
$$J = \frac{\pi D^4}{32}$$

(solid section)



$$J = \frac{\pi D^4}{32} - \frac{\pi d^4}{32}$$

(tube section)



6.1 Maximum Shear Stress

The maximum shear stress in the shaft?

$$\frac{T}{J} = \frac{\tau}{r} = \frac{G\theta}{L} \text{ using } \frac{T}{J} = \frac{\tau}{r} \rightarrow \boxed{\tau = \frac{Tr}{J}}$$

Section (1) C to D,

$$\boxed{J_1 = \frac{\pi D_1^4}{32}} = \frac{\pi \times 0.030^4}{32} = 7.952 \times 10^{-8}$$

$$\therefore \tau_1 = \frac{-415 \times 0.015}{7.952 \times 10^{-8}} = -78.3 \text{ MN/m}^2$$

Section (2) B to C,

$$\boxed{J_2 = \frac{\pi D_2^4}{32} - \frac{\pi d_2^4}{32}} = \frac{\pi \times 0.045^4}{32} - \frac{\pi \times 0.035^4}{32} = 2.552 \times 10^{-7}$$

$$\therefore \tau_2 = \frac{825 \times 0.0225}{2.552 \times 10^{-7}} = 72.7 \text{ MN/m}^2$$

Section (3) A to B,

$$\boxed{J_3 = \frac{\pi D_3^4}{32} - \frac{\pi d_3^4}{32}} = \frac{\pi \times 0.045^4}{32} - \frac{\pi \times 0.035^4}{32} = 2.552 \times 10^{-7}$$

$$\therefore \tau_3 = \frac{-175 \times 0.0225}{2.552 \times 10^{-7}} = -15.4 \text{ MN/m}^2$$

Maximum shear stress in the shaft = section 1 at 78.3 MN/m²

6.2 Angle of Rotation

The angle of rotation of each section?

$$\frac{T}{J} = \frac{\tau}{r} = \frac{G\theta}{L} \text{ using } \frac{T}{J} = \frac{G\theta}{L} \rightarrow \theta = \frac{TL}{JG}$$

Section (1) C to D,

$$\theta_1 = \frac{T_1 L_1}{J_1 G} = \frac{-415 \times 0.2}{7.952 \times 10^{-8} \times 75 \times 10^9}$$
$$\therefore \theta_1 = -0.0139 \text{ RADS}$$

Section (2) B to C,

$$\theta_2 = \frac{T_2 L_2}{J_2 G} = \frac{825 \times 0.4}{2.552 \times 10^{-7} \times 75 \times 10^9}$$
$$\therefore \theta_2 = 0.0172 \text{ RADS}$$

Section (2) B to C,

$$\theta_3 = \frac{T_3 L_3}{J_3 G} = \frac{-175 \times 0.9}{2.552 \times 10^{-7} \times 75 \times 10^9}$$
$$\therefore \theta_3 = -0.0082 \text{ RADS}$$

6.3 Angle of Rotation Cont...

The angle of rotation of D relative to A?

$$\theta_1 + \theta_2 + \theta_3$$

$$\therefore -0.0139 + 0.0172 - 0.0082 = -0.0049 \text{ RADS}$$

$$\text{Converted to degrees} = \frac{-0.0049}{2\pi} \times 360 = 0.3^\circ$$



7.0 Bibliography

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